Mechanical advantaged single drum vibratory roller earth compactor design for rural road construction, maintenance & upgrading within S.A.

by

Author: Terence Miller
Student Number: 809789273

a dissertation submitted in fulfillment of the degree of

Magister of Technologiae
in
Mechanical Engineering

in the

Faculty of Engineering and the Built Environment
(The Department of Mechanical Engineering Technology)

ABSTRACT

“Rural South Africa is characterised by poor infrastructure, large distances, dispersed demand and low incomes. There are also historical backlogs in service delivery, rural people also have poor access to basic social services and the economic mainstream [1].”

Rural areas in South Africa are in need of better services regarding their existing infrastructure and maintenance. Some services includes the building, upgrading and maintenance of roads [1]. These roads have to be cost effective and yet efficient enough to carry the loads and traffic of vehicles that will use these roads. Roads are the backbone of rural well-being and advances rural economy in many ways. Without proper roads, accessibility becomes a huge problem and thus the rural economy suffers.

Road building is subjective to many types of road construction machinery. These machines are used at various stages of the road construction process.

Due to budget constraints that local municipalities encounter in rural areas, expensive machinery to build and maintain these roads are difficult or in some cases almost impossible to purchase. To alleviate the constraints rural budgets are facing, a fresh look has to be taken on the cost of road construction machinery. In this dissertation, focus is placed on the compaction part of the earth road building process and the ultimate design of a towed road roller.

A different approach is taken in the design of single drum vibratory road roller’s (SDVRR). This approach is to move away from a self-propelled SDVRR toward a towed SDVRR that is operated by mechanical advantage only.

The design of a SDVRR, towed by a tractor with fewer parts will drastically reduce the manufacturing and operating costs.

The advantages of this type of SDVRR are lower manufacturing and maintenance costs due to the fact that in general, most self-propelled SDVRR’s today, come with a cabin, expensive hydraulics, hydraulic pump, cabin instrumentation, electronics and a set of extra tyres that can be excluded in the towed SDVRR.

The main marketed self-propelled SDVRR machine brands such as Caterpillar, Bomag, Sakai etc., all is dependent on the mentioned hardware and parts (SDVRR compactor model catalogues, Appendix 15 – 20).

Municipalities in rural South Africa make use of tractors for most tasks such as grass cutting, general cleaning and farming applications. Mostly semi-skilled to unskilled labour is utilized in performing these tasks on a day to day basis. By taking advantage of the tractor as a means of puling and powering the machine, costs can be minimized and skilled labour that is needed to operate a self-propelled SDVRR with its fancy cabin instrumentation can be made redundant as well, which will result in further costs reduction. All that is needed is the knowledge of how to operate the tractor itself at the required machine operating speed by the unskilled laborer, thus no extra costly training is needed since most rural laborers are familiar with the operation of a standard tractor fitted with a power take off (PTO). A PTO is a device that draws power from a tractor engine of which this power is transferred to be used for machine operation.

QUOTATIONS

English

“The horse is made ready for the day of battle, but victory rests with the Lord” – Proverbs 21:31

Afrikaans

“Die perd word reggemaak vir die dag van die geveg, maar die oorwinning is deur die Here”- Spreuke 21:31

"It is not the wealth of nations that builds roads, but the roads that build the wealth of nations" J. F. Kennedy

ACKNOWLEDGEMENTS

Thanks to my Heavenly Father through His Son, Jesus Christ, the most High, that is and always was a great help in all the ways He chose for me and this dissertation is one of His ways. Thank you Lord, I love you.

Thanks to my beautiful wife, Farida, for her valuable patience during this dissertation and our children, Tefaree and Terence. I love you.

Thanks to my supervisor, Dorina Ionescu, for all her excellent guidance through the years and especially regarding this design dissertation where her excellent knowledge in the subject of design and research helped me to reach my goal. Thanks Dorina.

Lastly, thanks to my colleague Frank Martinez who also gave me some useful design guidance.

Table of Contents

List of Figures

List of Tables

Nomenclature

a	Acceleration
A	Area
C	Basic load rating
C_o	Static load rating
d	Shaft diameter or Pulley diameter
E	Modulus of Elasticity
F	Pulling force
F_a	Accelerating force
fos	Factor of safety
g	Gravitational acceleration
HB	Brinell hardness
HRC	Rockwell C-Scale hardness
I	Mass Moment of Inertia
J	Polar Second Moment of Area
L	Length
L_{10h}	Bearing life
M	Moment
m	Mass
N	Revolutions per minute
p	pressure
P	Power
$P_{drawbar}$	Draw bar pulling force
R	Reaction force
R_r	Rolling resistance
T	Torque
T_1	Maximum belt tension
T_2	Maximum belt tension
u	Initial velocity
v	Final operating velocity

Symbols

ε	Strain
μ	Coefficient of friction
ν	Poisson's Ratio
ω	Angular velocity
σ	Stress
τ	Shear stress
θ	Angle
ρ	Density
δ	Radial Clearance
^{o}	Degrees
η	Efficiency

List of Abbreviations

AASHTO	American Association of State Highway and Transportation Officials
AMIPD	Actual Minimum Pulley Diameter
ARP	Actual Ratio Possible
BMD	Bending Moment Diagram
CBR	California Bearing Ratio
EWAL	Eccentric Weight Arch Lever
FS	First Shaft
ID	Ideal Diameter
IDD	Ideal Diameter Diagram
IS	Intermediate Shaft
LHS	Left-hand side
LL	Liquid Limit
MIPD	Minimum Pulley Diameter
MIPR	Minimum Possible Ratio
MOV	Maximum Operating Velocity
MS	Main Shaft
OMC	Optimum Moisture Content
PI	Plastic Index
PL	Plastic Limit
PTO	Power Take Off
RHS	Right-hand side
RMV	Roller Measurement Values
SDVRR	Single Drum Vibratory Road Roller
SF	Service factor
SFD	Shear Force Diagram
SL	Shrinkage Limit
SPIF	Speed Increase factor
TMH	Technical Methods for Highways
TRH	Technical Recommendations for Highways

1. Summary

Rural areas are experiencing ever increasing poverty and sometimes neglect from provincial or national governments. There are many areas in which rural people have to endure the lack of proper roads, medical services, food, jobs, etc. The backbone of all these services is the building of economical viable roads that can handle sufficient traffic to advance service delivery in rural areas [1].

Rural roads are low volume roads which consist mainly of earth and gravel roads where the in-situ soil is lightly compacted [2].Very few bitumen sealed roads are used in South African rural situations, because the initial construction costs is very high and puts great strain on rural budgets, although the benefit is greater at a later stage. This is why the majority of rural roads are gravel or dirt. There is a lot of work to be done when it comes to un-surfaced roads in rural South Africa. South Africa has a very large road network, consisting of 155 000 km of surfaced roads and 600 000 km of un-surfaced roads [3]. There is also later research done by CSIR in 2009 [4] indicating that the total road network consists of 741,172 km of roads, where 489,623 km is provincial access rural roads of which many of them are un-surfaced.

Rural municipalities have constrained budgets and cannot afford modern day compaction and road construction machinery and thus cannot cater for their un-surfaced roads. These machines forms part of the initial construction costs for road building.
Cheaper alternatives need to be considered to alleviate the obstacles rural budgets are facing in conjunction with road construction, maintenance and upgrading.

The vibratory road roller machine documented in this dissertation will be developed into a working prototype by Terragrader (Pty) Ltd. The machine will be cheap, affordable, easy to operate (tractor towed) and yet will provide efficient road maintenance and construction standards for rural road building, maintenance and upgrading.

The Terragrader (Pty) Ltd vibratory road roller will be cheaper than conventional road rollers on the market, since it will be designed with fewer parts. The tractor that is part of the everyday rural working environment will eliminate the need for the cabin, expensive hydraulics, hydraulic pump, cabin instrumentation, electronics and a set of extra tyres.

The machine operation will be performed by available semi-skilled to unskilled labour which will further reduce initial road constructions costs.

Designing of any machine needs to take into consideration the environment where the machine will be operated and the tasks the machine needs to accomplish in its environment. The towed SDVRR will be operated on rural South African earth roads which comprises mainly of soil and gravel.

The task of the machine is to compact the soil into a denser state than the in-situ soil (soil found on site) and then if need be, compact a layer of gravel on the soil base. In order to achieve the required goals and objectives to get a sophisticated design, more research had to be done in fields that relate to the rural situation and its environment, where the machine is going to operate in, different sciences, relevant technologies and design simulation software pertaining to the design.

Stipulated below are all complimentary research done related to this dissertation;

- Compaction entails the history, the meaning, lab testing, road failure and how to get maximum compaction for certain types of soil under optimum moisture content.

- Soil Mechanics, to get an understanding of the load bearing capacity of a compacted surface under load and before load. This includes many parameters that need to be looked at and the actual science of how soil behaves after compaction. An in depth understanding of the content of soil and how it is broken into different particles according the unified soil classification system.

- Road design & Compaction according to the official Technical Recommendations for Highways (TRH) and Technical Methods for Highways (TMH) documentation that is use for road construction within rural South Africa.

- Company Product Catalogues for selecting power transmission parts and equipment such as the Spicer & Hardy drive shaft, Bonfiglioli gearbox, Fenner tyre couplings, Fenner belt drives, Fenner chain drives, Seal n Devices vibration dampers and SKF bearings.

- Different types of current South African Marketed road rollers. This research area entails the comparison of prices and technical data from different types of machines manufacturers with the documented machine (Appendix 13 -18).

- Some of the Toughest Steel Materials that is used on the market today for heavy engineering jobs and road construction like Toolox, Weldox, Hardox & Armox for vibratory drum strength (Appendix 33).

- Software utilization such as Inventor 2013 Design Accelerator, Stress Analysis, Vibration Analysis and general 2D & 3D drawing techniques.

2. The History of Compaction

The history of compaction is very important concerning this dissertation and must be viewed in context with the current design that is attempted herein. The towed SDVRR was designed during 1940 and needed an extra engine to rotate the eccentric mass assembled to the Main Shaft (MS) which in turn creates drum excitation [6]. The difference between the documented machine and the 1940 machine is that it does not work by utilizing the power of an extra engine, but however, is motioned by means of a tractor PTO.

By the 1950's, the self-propelled roller was introduced with its ability to be driven and excite the compacting drum at the same time with the same engine. This is the modern day single drum vibratory compactor machine that was just further refined in later years [6].

Weinhart [5], Compaction Manuel [6], Wikipedia [7], Intelligent compaction [8]

1. Ancient roads consisted out of old tracks probably, compaction of these roads were carried out by passing humans, animal drawn carts and livestock feet.
   Motion from animal feet and hooves were similar to a modern padded drum that is also known as a sheep's-foot roller (Figure 6).

2. The Romans used a yoked cylindrical stone which was the first road roller known to compact their roads. These roads consisted of removing earth to the width of the road and to a certain depth. The base was compacted and then layers of 230 mm concrete and 150 mm of finer concrete was laid on top of one another from the base surface. Lava stones or stone was embedded into the concrete and then compacted with their yoked stone roller. At the fall of the Roman Empire, the decline of road construction began.

3. During the 1800's horse drawn rollers were used as static rollers where the static weight of the roller was an important factor to achieve compaction.

4. Ancient Chinese used dynamic compaction driven by the principle of variable amplitude as seen in Figure 1. The method was made effective, mainly by alternation of the drum movement with attached ropes.

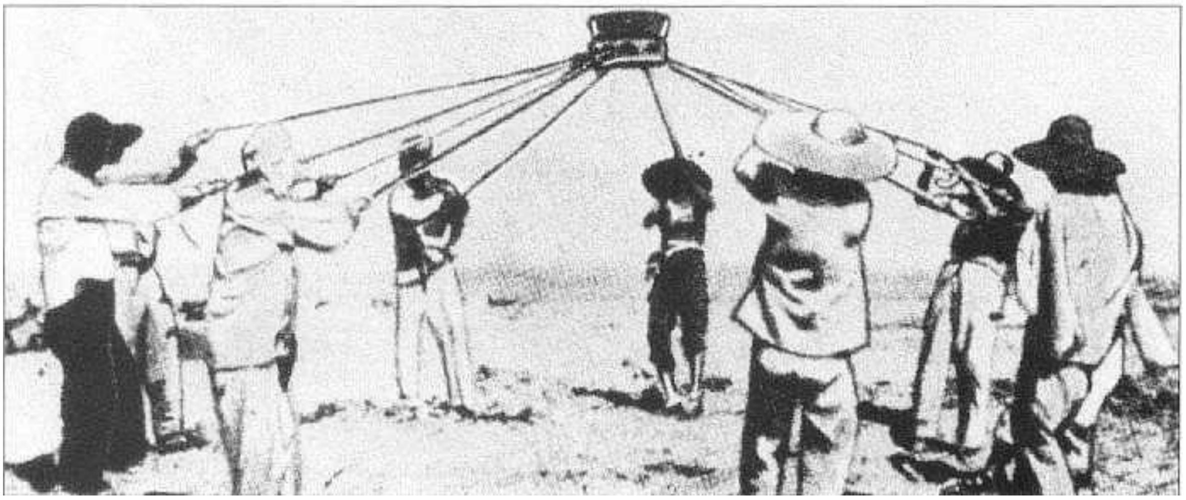


Figure 1: Manual variable amplitude excitation.

5. Around 1815, a Scottish Engineer, John London Macadam introduced a paving by mixing certain ratios of pebbles with clay or rock dust that was compacted. With the right proportional quantities, the mixture almost attained the hardness of concrete. Macadam is still used in road construction theory and practical applications today and is related to a type of road surface seal.

6. The steam roller was invented in England and was tried in New York during 1869. For almost a century the smooth wheeled steam road roller proved very effective in the compaction of roads.

7. Various soil compaction theories and multiple testing methods evolved during 1920. This was where people directed their attention partly to the science of soil, instead of static weight and variable amplitude only.

8. Germany used the first type of vibratory roller during the 1930's to construct their highway system.

9. Moisture content which is one of the most important factors of soil densification was focused on to achieve efficient compaction by R. R. Proctor in 1933. Proctor came up with methods to increase compaction efficiency in the field by focusing on optimum moisture content to achieve the maximum dry density of a soil sample.

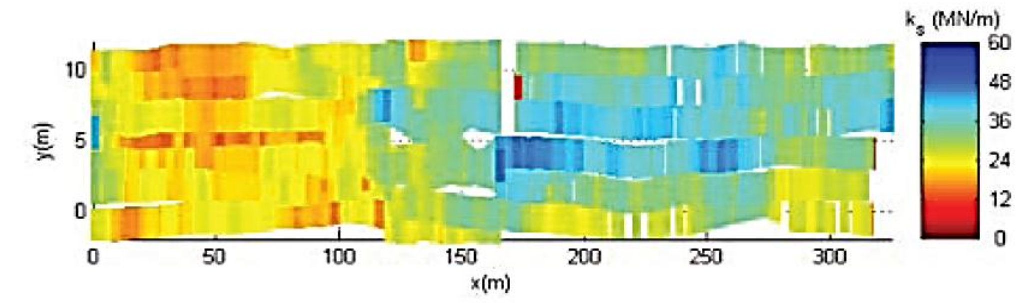
10. Towed type vibratory rollers were manufactured during the 1940’s in the United States of America. These rollers were generally towed by a farm tractor while a separate engine attached to the frame excited the off-centred masses thus causing vibration.

11. Self-propelled vibratory roller designs were introduced during the 1950’s. The popularity of vibratory rollers continued steadily into the 1960’s. Production efficiencies rose especially with cohesive, granular soils and the inclusion of asphalt.

12. During 1969, compaction production capabilities increased even further when the double drum vibratory rollers came on the scene. More road surface could be covered in relatively shorter time spans to achieve desired soil stiffness.

13. Today’s modern compaction rollers measure the underlying material stiffness with the help of sensors that record and feed data to an onboard computer in the cab.
    A vibrating drum traverses the compaction site measuring soil stiffness and collecting GPS coordinates that are together, termed Roller Measurement Values (RMV).

    Figure 2 shows RMV field data from a Bomag vibratory roller, indicating various stiffness values in a width and length of 10 m and 300 m respectively. With this data, the operator has knowledge of the different compaction levels achieved and where the weak spots are, that still need attention. The machine operator is then able to rectify these weak spots by applying more passes until the desired density or soil stiffness is achieved.



RMV data from a test field

Figure 2: Soil stiffness field data.

3. Problem Definition

All criteria and design constraints were discussed in a meeting with the company management interested in manufacturing the compactor, namely Terragrader (Pty) Ltd (M.D. Mr. Luigi Quaroni and Mr. Johan Wessels).

As discussed with them, the design needed to include the following:

- ✓ Compaction must be achieved by towing the machine behind a tractor and so doing move the oscillating drum forward at a minimum speed of 6 km/h and up to a maximum of 12 km/h;
- ✓ The machine must be able to compact the in-situ soil and or gravel into a denser state up to a maximum of 150 mm per layer as with self-propelled SDVRR's (Appendix 13 – 18);
- ✓ The design must ensure the manufacture of the individual components, fabrication of all assemblies, and assembly of the complete machine is simplified for ease of mass production;
- ✓ The price should be lower than that of the competitors in Appendix 13 - 18 and be affordable to rural municipalities;
- ✓ Any rural area tractor driver should be able to operate the machine, unlike any other towed farm equipment;
- ✓ Unskilled labor must be able to operate the machine with as little training as possible.

4. Background information

University of Johannesburg was approached by Demco (PTY) LTD for assistance in the design of various road construction machinery. These machines will then be used for road building, upgrading and the maintenance process in rural South Africa. Roads For Africa (Pty) Ltd (part of DEMCO (Pty) Ltd) focused solely on rural South Africa for vending of their machines.

The company has since change their trading name from Demco (PTY) LTD to Terragrader (PTY) LTD. Apart from the mentioned machine in this dissertation, two other machines have been designed by University of Johannesburg, such as the Terragrader design by FL Martinez [9] and the low cost rural tractor by C. Popa [10]. The mechanical advantaged road roller will be compatible with the low cost tractor designed by C. Popa [10] as well as any other tractor that can deliver the appropriate power to operate the machine.

Letters was drawn up that stipulated a careful and well-constructed plan to empower, train, allow job creation as well as poverty relieve for rural women and their families. A business venture was then pursued which is currently known as Roads for Africa [Appendix 1.1, 1.2 and 1.3].

Key features that are stipulated in the Roads for Africa Letters (PTY) LTD is summarized below; [Appendix 1.1, 1.2 and 1.3]:

- skill shortage alleviation,
- more efficient service delivery,
- the empowerment of rural women by building, maintenance and upgrading of rural roads,
- three year contracts will be supplied to six women which will provide employment for 20 other females under their supervision,
- cycling of teams will be done after each three year cycle to allow for a more efficient spread of wealth and
- general maintenance on buildings, road marking painting, signs etc.

5. Requirements of the solution

The towed mechanical advantaged SDVRR that is being designed in this design dissertation needs to be tractor towed and to be able to compact soil and gravel on rural roads by the use of unskilled labour , which consists mainly of women.

The requirements of the solution are described in point form below:

1. The machine must make use of mechanical advantage to compact soil and gravel on rural dirt roads.
2. Any marketed tractor with the right power specifications must be able to tow the machine.
3. The machine must receive power via a commercial tractors PTO through a drive shaft.
4. The operation of the compactor must be simple enough to be operated by semi and unskilled labour in a relatively short time.
5. Compaction effort must be efficient enough for adequate gravel and soil compaction as required for rural standards.
6. Machine towing speed should not exceed 12 km/h and must be able to operate at lower speeds of up to 6 km/h.
7. The compactor mass must be circa ten tons and must be in the required power range as seen in [Appendix 13 -18].
8. The maximum operation frequency of the machine should be between 31.5 Hz to 40 Hz.
9. The life span of the machine has to be in accordance with conventional construction machinery life specifications.

6 Background Research

6.1 Compaction (Stabilization)

References: Rex Willem Kelfens Dissertation [11], Multiquip Compaction Manuel [12], Caterpillar Compaction Manuel [13].

Compaction is the process through which the volume of the material decreased by reducing air voids between the particles and expel unnecessary water volumes with the aid of mechanical forces. By applying these various mechanical forces, compaction time is shortened as natural compaction of soils can occur overtime as settlement due to the elements, but may span over years.

6.1.1 Goal of Compaction

Referring to Figure 3, the total volume V_t must be decreased to a lesser volume to achieve proper compaction. The total volume of the soil Vs is constant and no soil particles can be lost through compaction effort. Air volume V_a has to be expelled as well as a certain quantity of the water volume V_w. The remaining water volume will then be the moisture content left to assist in achieving adequate compaction by sliding the soil particles over one another more effectively.

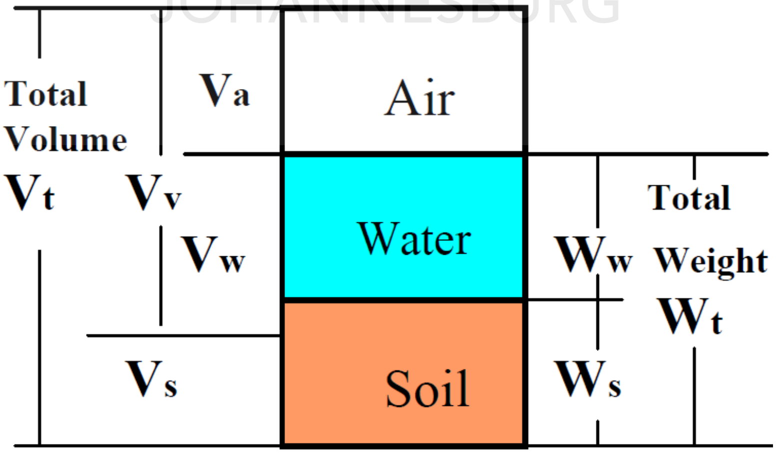


Figure 3: Material volume reduction.

6.1.2 The result of compaction

The actual result of compaction is indicated in Figure 4 where the loose soil particle layout is indicated on the left-hand side (LHS) and on the right-hand side (RHS). The final product is shown in a much denser array packed tightly together. This denser array allows for greater load support that is known as bearing capacity.

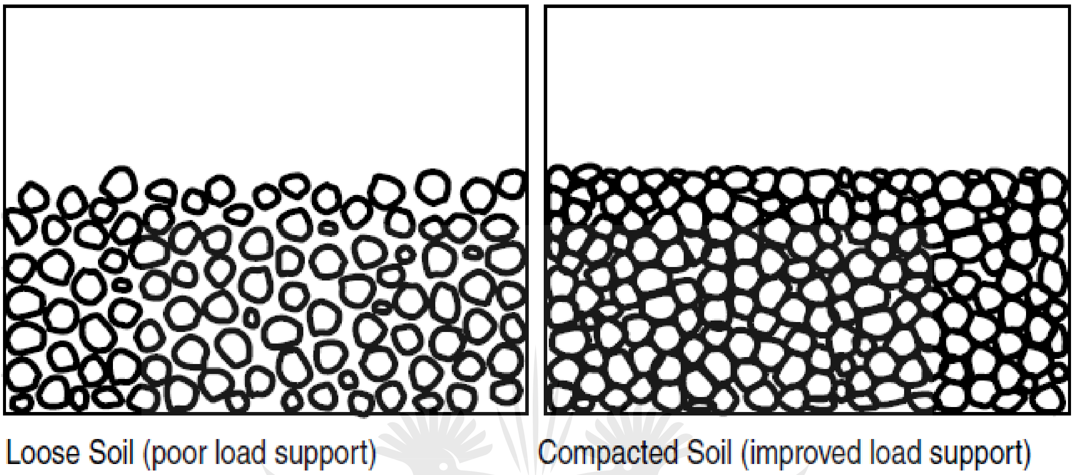


Figure 4: Compaction result.

6.1.3 Benefits of compaction.

To ensure the durability of the road, there are many reasons to compact the soil. References: [11] and [12].

- Increases load bearing characteristics of a soil.
- Prevents soil settlement in the vertical direction with little or no deflection.
- Provides stability between soil particles due to increased shear strength.
- Reduces water seepage, swelling and contraction due to sand particles being packed tightly together.
- Reduces settling of soil after time.
- It is very economical, increases road life and ride quality if done efficiently.
- Frost damage is prevented due to very little water trapped in the voids of the soil.

6.1.4 Forces of Compaction

Static Pressure: In static compaction, the static weight of the compactor is moved over the soil surface causing shear stresses in the soil that promotes sliding of the individual soil particles. The particles will then move into a fewer voids state. See Figure 5.

Figure 5: Static rolling

Kneading: The second compacting force, rearranges particles into a smaller volume and greater density by a kneading process. The process is very effective near the surface of the soil. The longitudinal and transverse kneading action is essential when compacting heavily stratified soils such as clay type soils. It is also the desired process for the compaction of the final wearing surface seal of an asphalt pavement. High pavement sealing is achieved by kneading the final road surface to close all hairline cracks and obtaining a very smooth surface finish. See Figure 6.

Figure 6: Kneading drum kit.

Impact: Impact creates a greater compaction force on the surface than an equivalent static load. This is, because a falling weight has potential energy and speed, which is converted, to energy at the instant of impact. Impact creates a pressure wave, which goes into the soil from the surface. In simple terms, potential energy get converted into kinetic energy just before striking the soil surface and then into work done, thus achieving movement between the soil particles. See Figure 7.

Figure 7: Impact weight compaction

Vibration: This force is created by rotating an eccentric mass on the inside or outside of a geometrical shape, usually round and circular resulting in a double amplitude of compaction. Vibratory compactors produce a rapid succession of pressure waves, which spread in all directions. The vibration creates resonance between the particles of the material being compacted and overcomes the frictional bonds holding them together. When pressure is applied, the particles tend to re-orient themselves in a more dense state. The force used in vibratory compactors is known as the centrifugal force. See Figure 8.

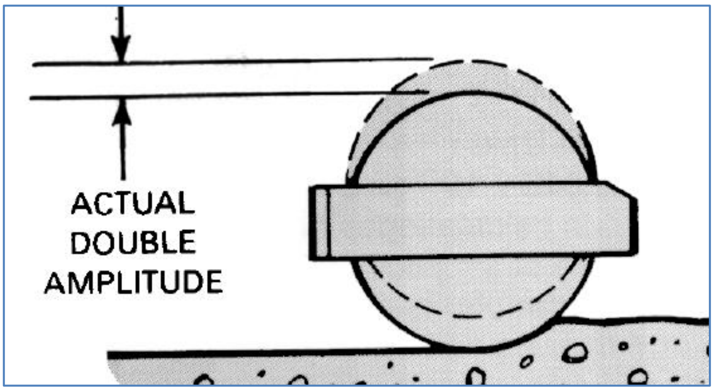


Figure 8: Dual amplitude vibratory drum movement.

6.1.5 Compaction Efficiency

There are over thirty different types of factors influencing compaction, like amplitude, frequency, linear speed, type of force, type of soil, etc. The degree of compaction effort is dependent on the combined characteristics of the compactor (machine) and the soil.

Influence of the compaction material over the compaction efficiency: The influence the material that needs to be compacted, is due to the type, aggregate strength characteristics, layer thickness or lift thickness, gradation, initial density, subsoil base, texture, moisture content, and its supporting capability. The sum effect of these properties is termed mass stiffness and damping. In mechanical vibration modeling it will represent a spring and damper to allow for mathematical analysis. The achievement of maximum compaction is dependent on optimum moisture content, material type and the type of compaction force exerted.

6.1.5.1 Primary focus to achieve optimum compaction

The Optimum Moisture Content (OMC), the type of material and the type of compaction force exerted is primary when it comes to optimum compaction.

OMC: To narrow down all factors of compaction, the main emphasis is placed on the OMC for optimum compaction. The OMC represents the exact amount of moisture needed in material to allow for maximum material densification during compaction. The OMC can be viewed as a type of lubrication to overcome frictional resistance between soil particles. Figure 9 represents the percentage optimum moisture needed to achieve maximum soil densification.

Material type: The material being compacted is dependent on the topography and weather conditions which are very important when it comes to selecting the proper type of roller for the job. The unified soil classification system shows the various soil types by distinguishing between various grain sizes. Rural areas in South Africa have lots of soil types where most can be compacted with a smooth drum type roller.

Materials are found in two types known as granular materials and cohesive materials. These materials are dependent on the topography and weather conditions. The primary mechanism of soil creation is the weathering of rock. All rock types (igneous rock, metamorphic rock and sedimentary rock) may be broken down into small particles to create soil.

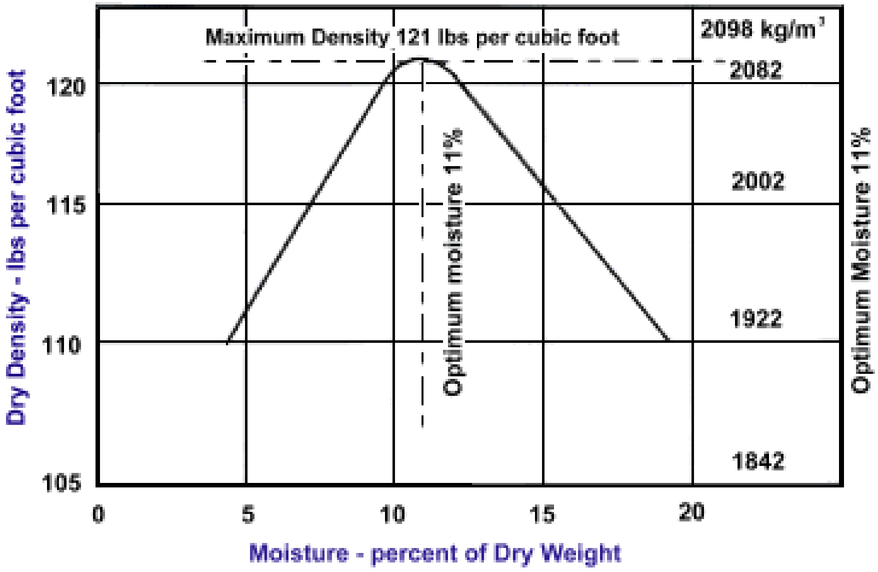


Figure 9: Optimum moisture content curve.

The unified soil classification system shows the various soil types by distinguishing between various grain sizes [14].

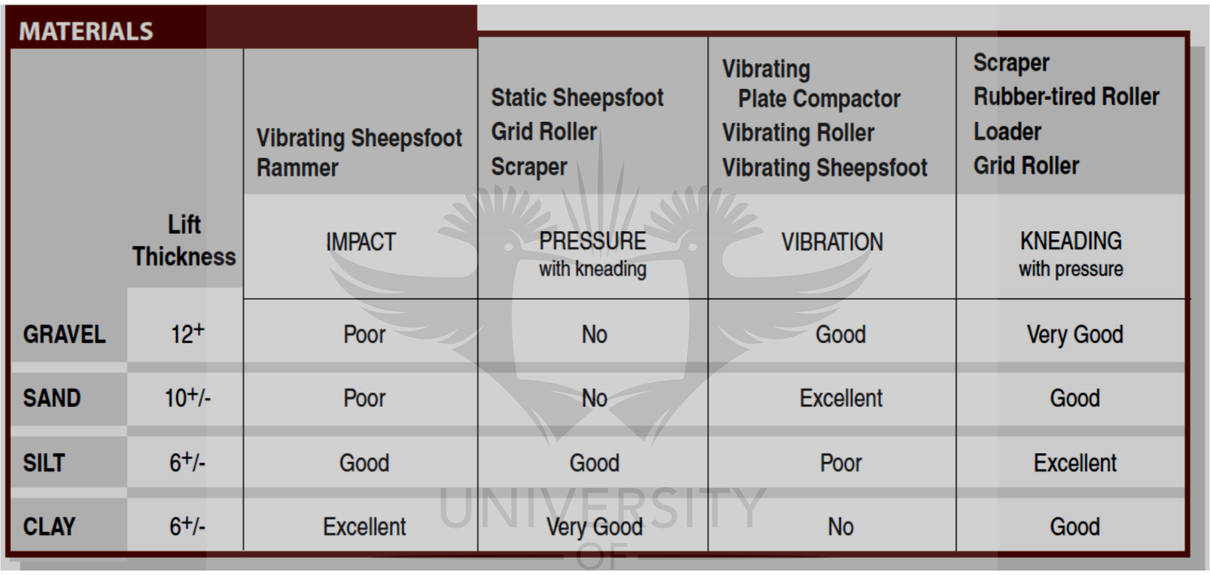
Type of Materials classifications are:

1. *Gravel:* Individual grains vary in size from 2.0 mm to 63 mm in diameter and have a rounded appearance.
2. *Sand:* These are small rock or mineral fragments smaller than 2.0 mm in diameter and semi sharp.
3. *Silt:* Fine grains appearing soft and floury when dry. When moist, silt pressed between the thumb and forefinger will have a crack-like appearance. The grain sizes for silt are betweent 0.002 mm and 0.063 mm.
4. *Clay:* Very fine textured soil with a size classification of 0.002 mm, with a high level of cohesion and forms hard lumps or clods when dried.
5. *Organic:* Organic matter found between soil particles. This matter consists of either partially decomposed vegetation (peats) or finely divided vegetable matter (organics).

Type of force: There are different types of soils as indicated by unified soil classification system. The type of soil indicates what type of compaction force is to be applied to get faster and more efficient compaction results (see Table 1).

The force type is also relevant when choosing a road roller. Table 1 indicates what type of roller is suitable for certain material types. The smooth drum vibrating roller is quite suitable for South African sand and gravel compaction CARNS 2004 [14].
Where gravel is indicated as good and sand is indicated as excellent for smooth roller vibrational compaction.

Table 1: Different compactor types vs material suitability.



MATERIALS		Vibrating Sheepsfoot Rammer	Static Sheepsfoot Grid Roller Scraper	Vibrating Plate Compactor Vibrating Roller Vibrating Sheepsfoot	Scraper Rubber-tired Roller Loader Grid Roller
	Lift Thickness	IMPACT	PRESSURE with kneading	VIBRATION	KNEADING with pressure
GRAVEL	12^{+}	Poor	No	Good	Very Good
SAND	10^{+}/-	Poor	No	Excellent	Good
SILT	6^{+}/-	Good	Good	Poor	Excellent
CLAY	6^{+}/-	Excellent	Very Good	No	Good

6.2 Soil compaction equipment.

Reference: Caterpillar Compaction Manuel [13]

Many factors influence the choice of compaction equipment. The type of equipment selected for a project is sometimes chosen based on the contractor’s previous experience, by the type of soil or by method specifications. Other considerations are how well a machine will conform to the hauling and spreading operation. Climatic and traction conditions are also important. Standardization of equipment sometimes plays a role in the decision-making process. There is no single compactor that will do all things on all jobs. Each type has a definite material and operating range on which it is most economical.

Pneumatic Tyre Compactors: These compactors use pressure and kneading to densify soil and asphalt. The rubber tyres have a specific tyre pattern which compact and seal road surfaces by means of the kneading action caused by the tyre pattern and the pressure caused by the static weight. Tyre pressure and ballast influence the amount of compaction force transmitted to the road surface.

Figure 10: Pneumatic Tyre Compactors.

Sheep's foot Roller: Sheep's foot rollers simulate a herd of sheep's footing motion which was used in the old days to compact soil. The pads on the roller drum penetrate the soil and cause the soil to be compacted from the bottom upward. On exiting the soil, padded drums licks up or fluffs material, this accelerates the drying of silts and clays. Six to ten passes are needed in 203 mm soil lifts to obtain the desired density by the machines pressure and kneading action.

Figure 11: Sheep's foot roller compactor.

Tamping Foot Compactors: Tamping foot compactors usually have four steel padded wheels and are equipped with a dozer blade. Their pads are tapered with an oval or rectangular face. These machines have a very high production output due to the rollers ability to work at very high speeds (24 – 32 km/h). They develop all four types of compaction forces, kneading, pressure, impact and vibration at high speed. Four maximum passes are only needed to get the target density in 203 mm to 305 mm lifts, greatly increasing compaction efficiency.

Figure 12: Tamping Foot Compactor.

Vibratory Compactors: Vibratory compactor has an unbalance mass within their drums that force oscillates the drum at a certain frequency, usually 30Hz to 36.7Hz. The amplitude will then be used to do work on the soil in lifts of 150 mm to 607 mm relative to the centrifugal force. For vibratory compactors, a speed from 2 to 4 mph (3.2 to 6.4 km/h) will provide the best results. This type of roller is particularly suited for gravel and soil compaction that is used in rural road construction.

Figure 13: Vibratory compactor

6.3 Soil compaction preparation and measurement.

References: Multiquip Compaction Manuel [12], Caterpillar Compaction Manuel [13], South African Pavement Engineering Manuel [15].

Considering the compaction efficiency the most important influence factors are: moisture content, the type of material being compacted and the application of the force.

The moisture content is primary between these entities for any material that needs compaction and supplies aid to the applied force to achieve the required density or maximum dry density of the soil.

For the maximum dry density to be achieved, that is linked to the optimum moisture content, see Figure 9. The initial soil density and moisture content of a site from where soil samples were taken , has to be known in order to maintain the proper levels of moisture and density throughout the substrate of a road in preparation for compaction. After compaction, measurements have to be taken again, to give an indication of the level of density achieved.

Various tests have been devised both on field and in the laboratory to measure soil density and moisture content during the road and after road compaction process.

6.3.1 Job site soil density and moisture content testing.

The most common field testing methods are the Nuclear Method, the Sand-Cone Method and the Water Balloon Method, discussed herein.

Nuclear Method: Both moisture content and density is verified with this method at the same time. The method consists of two basic methods to measure density, such as measuring direct transmission and backscatter.

- The direct transmission method ranges of depths from two to twelve inches and yields the best accuracy, least composition error and least surface roughness error.
- The backscatter method has the unit resting on the soil surface and allows for surface measuring, instead of being reliant on an access hole in the compacted soil. The apparatus can only be used in shallow depths of up to a maximum of 76.2 mm and has less accuracy composition, errors are likely.

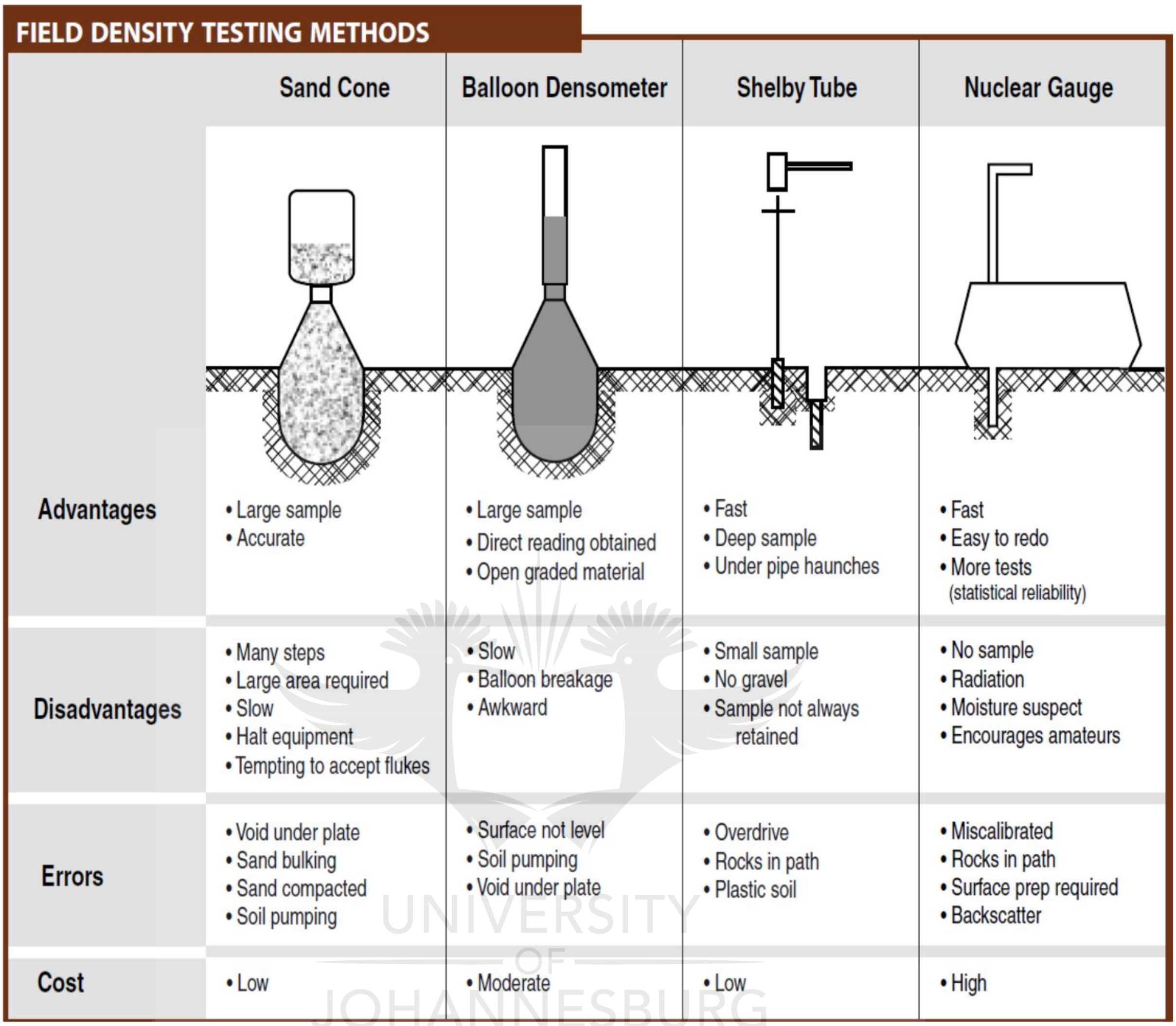
Sand-Cone Method: This method is sometimes used to calibrate nuclear density gauges that are used in the nuclear test method. It is a multi-step procedure that requires more time than the nuclear density method, but with proven accuracy.

- First, a test site is located where there are no disturbances of construction machinery to nullify vibrations that can cause inaccuracies.
- A soil sample is then excavated through the hole in the apparatus plate to a depth of 150 mm. By weighing the wet sample, then drying it and then weighing it again, the soil moisture content can then be determined by the mass difference of both measurements. The volume of the hole on site is measured by filling it with dry, free-flowing sand from a special sand-cone cylinder, where the discharged sand volume will be equal to the volume of the hole.
- By simple calculation of mass = density x sand volume, the density as unknown can be obtained.

Water Balloon Method: The water balloon method is also called the Washington Densometer Test.

- Similar to the Sand-Cone Method, a sample is taken from the job site weighed, dried and then weighed again, by this method soil moisture content is obtained.
- Instead of discharging sand into the hole by means of a sand cone to obtain hole volume, a Washington Densometer is used where a balloon is placed inside the hole with fluid being released into the balloon. The volume of the fluid, plus the balloon volume which is already recorded on the machines calibration initially, will allow for the whole volume to be known.
- The soil density can then be determined by mass = soil density x hole volume.

Figure 14 shows a summary of advantages and disadvantages of the three explained tests and also includes the Shelby Tube test that is not that commonly or widely used.



FIELD DENSITY TESTING METHODS

	Sand Cone	Balloon Densometer	Shelby Tube	Nuclear Gauge
Advantages	• Large sample • Accurate	• Large sample • Direct reading obtained • Open graded material	• Fast • Deep sample • Under pipe haunches	• Fast • Easy to redo • More tests (statistical reliability)
Disadvantages	• Many steps • Large area required • Slow • Halt equipment • Tempting to accept flukes	• Slow • Balloon breakage • Awkward	• Small sample • No gravel • Sample not always retained	• No sample • Radiation • Moisture suspect • Encourages amateurs
Errors	• Void under plate • Sand bulking • Sand compacted • Soil pumping	• Surface not level • Soil pumping • Void under plate	• Overdrive • Rocks in path • Plastic soil	• Miscalibrated • Rocks in path • Surface prep required • Backscatter
Cost	• Low	• Moderate	• Low	• High

Figure 14: Field density and moisture content testing.

6.3.2 Laboratory soil density and moisture content testing.

6.3.2.1 Proctor tests

The Standard Proctor Test was developed in 1933 by Ralph R Proctor. Figure 15 shows a standard Proctor density test equipment where a small soil sample is taken from the jobsite and then a standard 2,495 kg mass is dropped from a height of 304,8 mm at 25 blows per layer. The material is weighed before compaction and then oven dried for 12 hours and then weighed again to determine the moisture content in the soil sample. Standard: ASTM D-698, AASHTO T-99.

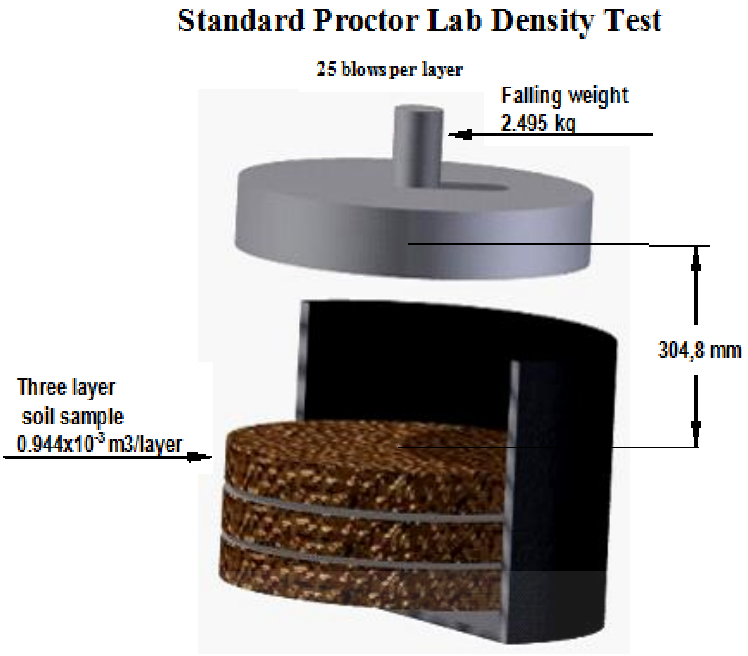


Figure 15: Standard Proctor Test.

Figure 16 shows a modified Proctor density test equipment. This test indicates different parameters, but a similar approach to the Proctor test is used, only this time a hammer is utilized to compact the soil sample. According to the Modified AASHTO procedure (T-180), a 4.536 kg hammer is dropped from a height of 457.2 mm. The soil sample is compacted in five layers with 25 blows per layer. The compaction energy is 4.5 times larger than the Standard Proctor test. High shearing strength is determined in this type of test, as well as the amount of moisture content. Modified: (ASTM D-1557),

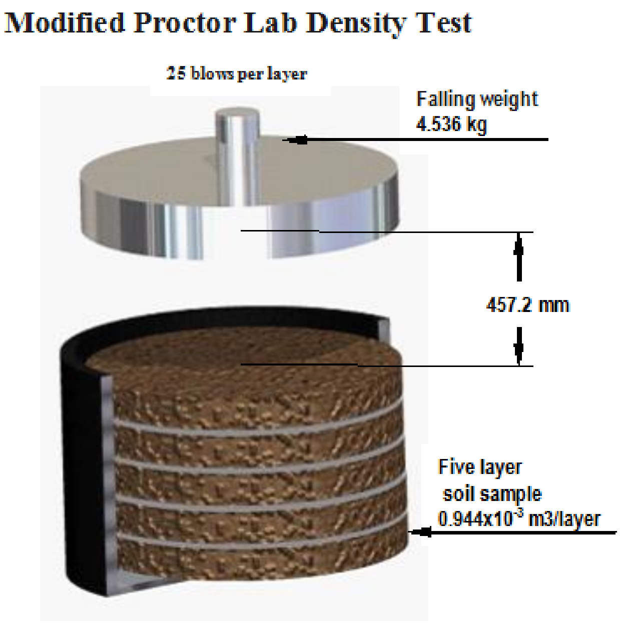


Figure *16:* Modified Proctor Test.

6.3.2.2 Moisture content and void ratio in the Procter tests.

Any of the two Proctor methods may be used to achieve a moisture density curve. The material in the moisture density curve is compacted at different moisture contents to obtain the optimum moisture content, maximum dry density and the least void ratio.

6.3.2.3 California Bearing Ratio CBR

Reference: CG Dyer [16]

The CBR is a comparative measure of the strength of a non-compacted material, i.e. the in-situ soil of an earth road, including the subgrade and granular bases and sub-bases.

- First the OMC is determined,
- The material is then placed in a special mold 152.4 mm in diameter with a height of 127 mm,
- A 4.536 kg rammer is dropped from a height of 457 mm to compact five equal layers at 55 blows per individual layer.
- A plunger of area 1935 mm^2 then penetrates the surface after compaction at a rate of 1.27 mm per minute.
- Standard material stiffness is then obtainable via the applied load and depth penetrated, i.e. 2.54 mm, 13.334 kN; 5.08 mm, 20.016 kN; 7.62 mm, 26.243 kN.

6.4 The properties of soil.

References: Caterpillar Compaction Manuel [13], Soil Mechanics A. Verruijt [14], South African Pavement Manuel [15].

Engineers use a number of terms when defining the characteristics and properties of various soils. Understanding these terms are essential to understanding soil compaction principles and techniques for compaction in soil mechanics.

- *Stiffness:* Metal and concrete is obeying Hooks law up to a certain point, but with soils this is not the case. Soil becomes gradually stiffer when compression is applied, as the particle formation changes, it starts to give a combined resistance, and the stiffness becomes greater, which gives the structure of particles an increased strength.

- *Shear*: Sloping surfaces has to minimise shear as to avoid slope failure. This occurs when soil particles begin to slide over one another due to the slope magnitude. If the shear stresses reach a certain level with respect to the normal stresses, failure of the soil mass may occur.

- *Dilatancy:* "Shear deformations of soils often are accompanied by volume changes.. Loose sand has a tendency to contract to a smaller volume, and densely packed sand can practically deform only when the volume expands somewhat, making the sand looser. This is known as the effect of dilatancy".

- *Creep:* Soil deforms over time and under constant loading which is a result of creep.

- *Capillarity:* Capillarity is the ability of a soil to syphon water upwards and in any direction where sufficient voids or pockets sizes are located in the soil. Granular soil usually has better capillarity than cohesive soils. This is, because of pore spaces that acts like small tubes giving rise to capillarity. Capillarity is an excellent tool when it comes to syphoning water from the subgrade of road surfaces. The wet subgrade material would disintegrate and compromise load bearing capacity if the moisture could not be transported out of the soil by capillarity.

- *Compressibility:* Compressibility of a soil refers to the rate of soil volume reduction during force application. The smaller the soil particles, the less mechanical work are needed for soil compression, where water and air filled voids are expelled from the material. Clay soils usually have higher compressibility than granular soils and have less capillarity.

- *Elasticity:* When a compressive load is applied to a soil, the soil will deform, but will return to its original shape when the load is removed. This is an indication that the soil behaves elastically. Organic matter mixed between soil particles has a very low compressibility which can give rise to elastic deflections during fluctuating loads. This causes fatigue during fluctuating loads and eventual roadway failure will occur.

- *Permeability:* Permeability refers to the rate at which soil absorbs water. Where capillarity expels water from soil, permeability sucks water into the body of soil. “Soil texture, gradation and the degree of compaction influence a soil’s permeability”. The coarser the type of soil, the faster the pores will fill up with water between the soils particles.

- *Plasticity:* Plasticity refers to the degree of compressibility and cohesiveness a soil possesses. The measure of plasticity is expressed as the Plasticity Index (PI). Clayey soils that have high PI values can be compacted easily, because of the high plastic index that encourages cohesion and compressibility in the soil. Moisture content is one of the major components that affect its PI value.

- *Settlement:* Settlement occurs when the voids between soil particles have not been expelled properly due to inadequate compaction. The top surface of the roadway will yield without returning to its original shape, where the amount of settlement is due to the amount of volume expelled from the voids settling out over time.

- *Shear Resistance:* Is the ability of a soil body to resist change when a compaction force is applied to it, this is due to particles constraining sliding movement to a certain degree at their mating surfaces. This is known as shear resistance which is encouraged by friction, cohesion, particles roughness and shape. The higher the shear resistance, the more compaction force is required to achieve the needed density.

- *Shrinkage / Swelling:* Shrinkage and swelling of a soil occurs when water is absorbed and released from the soil. Permeability and capillarity are the two main entities causing the above phenomena. In mechanical materials this is known as fatigue and causes failure when a material is repeatedly stressed and de-stressed after a number of cycles. Failure also occurs in pavements and foundations due to shrinkage and swelling of soil after the required amount of cycles.

- *Moisture Content*: Moisture content affects the compaction advantage by reducing the sliding forces between particles surfaces. For each soil type there is optimum moisture content at maximum dry density. The least compaction energy is required at optimum moisture content. Moisture content is determined by various laboratory tests and field tests.

Atterburg Limits

Soil Limits or Atterburg limits, explain and help with the understanding of how moisture can create better material workability or less workability dependent on the soil type.

- *Liquid Limit (LL):* The limit where internal friction is overcome by liquid separation of soil particle faces. The liquid (moisture) acts as a film that overcomes the cohesion and the friction of the soil particles causing separation. High LL values are associated with soils of high compressibility. Typically, clays have high LL values; sandy soils have low LL values.

- *Plastic Limit (PL):* “This condition exists when a soil changes from a semi-solid to a plastic state. It occurs when the soil contains just enough moisture that a small amount of it can be rolled into a 1/8" (3.2 mm) diameter thread without breaking. The PL of a soil is important. It represents the moisture content at which particles will slide over each other and still possess appreciable cohesion. It is the point at which best compaction occurs with pure clay soils. The strength of the soil decreases rapidly as the moisture content increases beyond the plastic limit”, thus the pore pressure becomes greater and will resist vibration compaction with smooth drum rollers.

- *Plasticity Index (PI):* This is the numerical difference between the LL and the PL.

- *Shrinkage Limit (SL):* When the volume of soil is reduced until no further reduction is possible. This is where further shrinkage cannot be attained and particles are locked tightly together. Maximum soil settlement is then achieved when there is no shrinking left for a soil.

6.5 South African Rural roads

Reference: Carns Consulting Group [2]

“The Department of Transport recognizes that the building of road networks and their maintenance usually constitutes the largest single capital investment made in rural development. However, the absence of a road network usually constrains the delivery of all other services. As such, local or access roads have a considerable impact on social, cultural and economic life of resource poor people. Not only does it provide access to markets, services, employment, business development, transport and communication, it improves personal mobility, crisis management, world view and quality of life.”

6.5.1 Rural road identification and design life.

Reference: Official road construction documents ([17], [18], [19]).

A rural road is defined as one that is not likely to acquire urban characteristics during its design life. TRH 17 [16]

The design of new roads or improvements is based on traffic volume and the design life in conjunction with rural roads is often assumed to be 7 - 20 years and the reliability ranges from 50% to 80%. TRH 4 [17]

“Unpaved roads may be divided into earth tracks, earth roads or gravel roads. Earth roads are classified as those on which no imported gravel is used, but the in-situ material is cleared of vegetation and lightly compacted”. TRH 20 [18]

6.5.2 Rural road hierarchy.

There are three different types of rural roads in South Africa and each has different design standards. The first is Type 7B Local Roads, the second is the Type 7A Local Roads and lastly, the Type 6 District Roads. These three types of roads differ mostly in road width and gravel surface thickness. Carns Consultants Group [2].

“Many unpaved roads, however lightly trafficked at the time of construction, will with the passage of time capture more traffic and increase in use (and importance) as the local population increases. They may eventually be upgraded to higher standard unpaved roads or even relatively lightly trafficked paved roads.” TRH 17 [17]

Type 7B Local Roads

Figure 17 shows the road shape geometry of a Type 7B Local Road and it merely serves as access to and between communities. It is the lowest standard of all three road categories and it is expected that the majority of Local Roads will be constructed to this standard in rural areas. These roads will never qualify to be upgraded to a Type 6 District Road. The gravel thickness or surfacing may be reduced from 150 mm to between 125 mm and 100 mm lift thickness. Traffic is estimated at 30 vehicles per day due to low vehicle ownership in the rural areas.

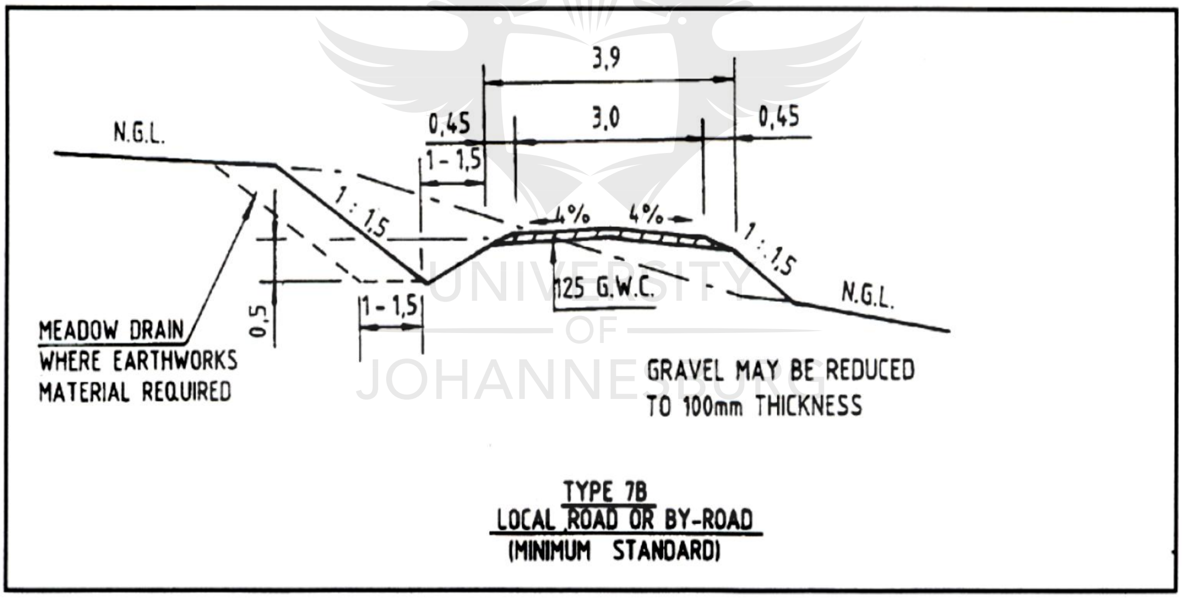


Figure 17: Type 7B Local Road (Minimum standard local road).

Type 7A Local Roads

The Type 7A standard Local Road is the desired standard for a rural road type, but due to cost constraints most roads will be constructed to the Type 7B standard (Figure 17). The Type 7A Local Road is 2 m wider than the Type 7B Local Road and will have a gravel surface of between 150 mm and 100 mm. This type of road can either be upgraded to a Type 6 District Road or downgraded to a narrower longer Type 7B Local Road, this is dependent on the type of traffic and increase or decrease in traffic volume. An average travelling speed of 30 km/h is the acceptable norm for Local Roads. See Figure 18.

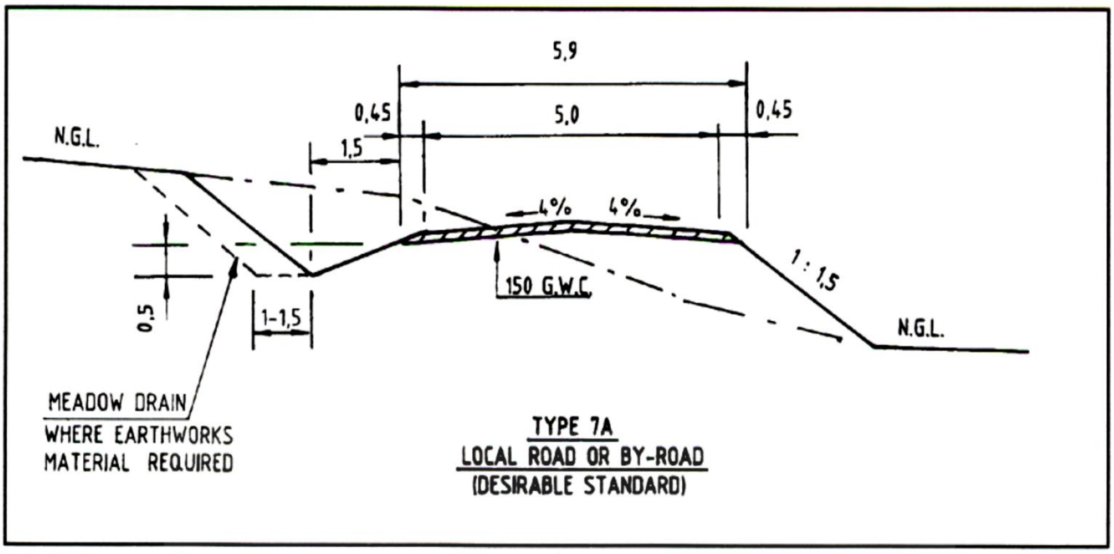


Figure 18: Type 7A local road (Desirable standard local road).

Type 6 District Road

District Roads are designed with a gravel surface layer of 150 mm and is 3 m wider than the Type 7B road (Figure 17) and 1 m wider than the Type 7A road (Figure 18). The design speed is 60 km/h, this is twice the average speed of Local Roads. These roads are used for access between farms and other main arteries. See Figure 19.

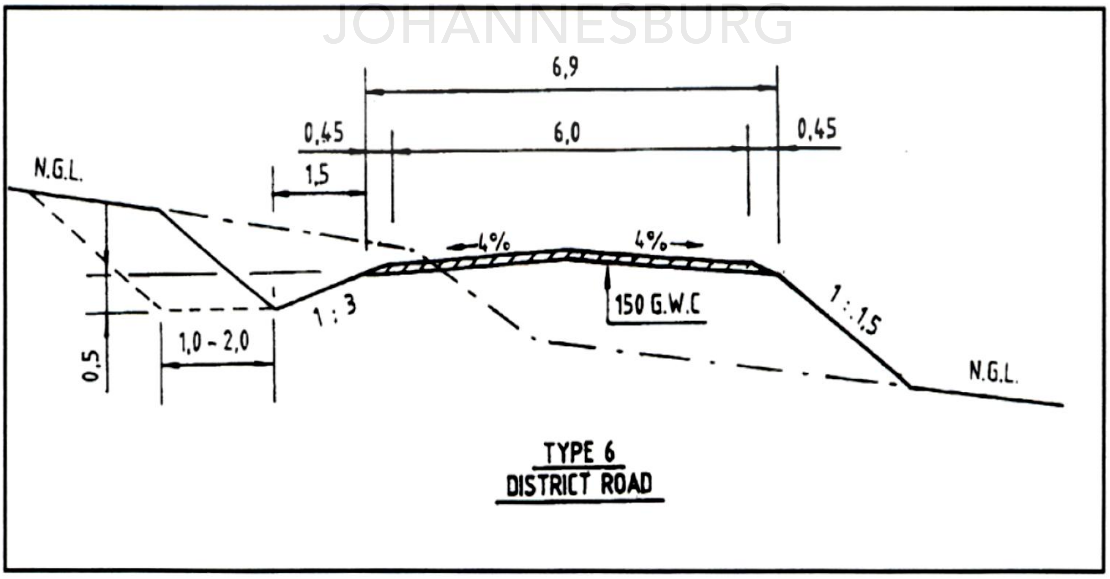


Figure 19: Type 6 District Road

6.5.3 Rural Road Construction and Standards

Road preparation and compaction TRH 17 [17]

- Graders or bulldozers must be used to remove all vegetable matter and organic soil from the road surface.
- The road bed should be ripped and mixed, sprayed with water to reach Optimum Moisture Content (OMC).
- The road must be compacted to a density of at least 90% Mod AASHO (about 95% Proctor).
- Nuclear density techniques must be used to test soil density and OMC or a quick, simple sand replacement test.

Road load bearing requirement TRH4 [18]

Traffic loads in South Africa are defined by soil densification by means of the various available soil testing techniques. In this case the primary load that needs to be considered in the design is the *traffic spectrum* that will be carried by the road. In South Africa, as in many other countries, the standard axle load is 80 kN and the permissible legal load is 88 kN per axle (South Africa, 1996).

Gravel operations TRH 17 [17]

The location, winning and transportation of wearing course gravels are one of the most expensive operations associated with the development of unpaved roads. It is therefore important that the optimum material be located nearby and used to maximum advantage. Most of these available are the G4, G5 and G6 natural gravel, of which G4 is crushed or totally natural (Figure 20).

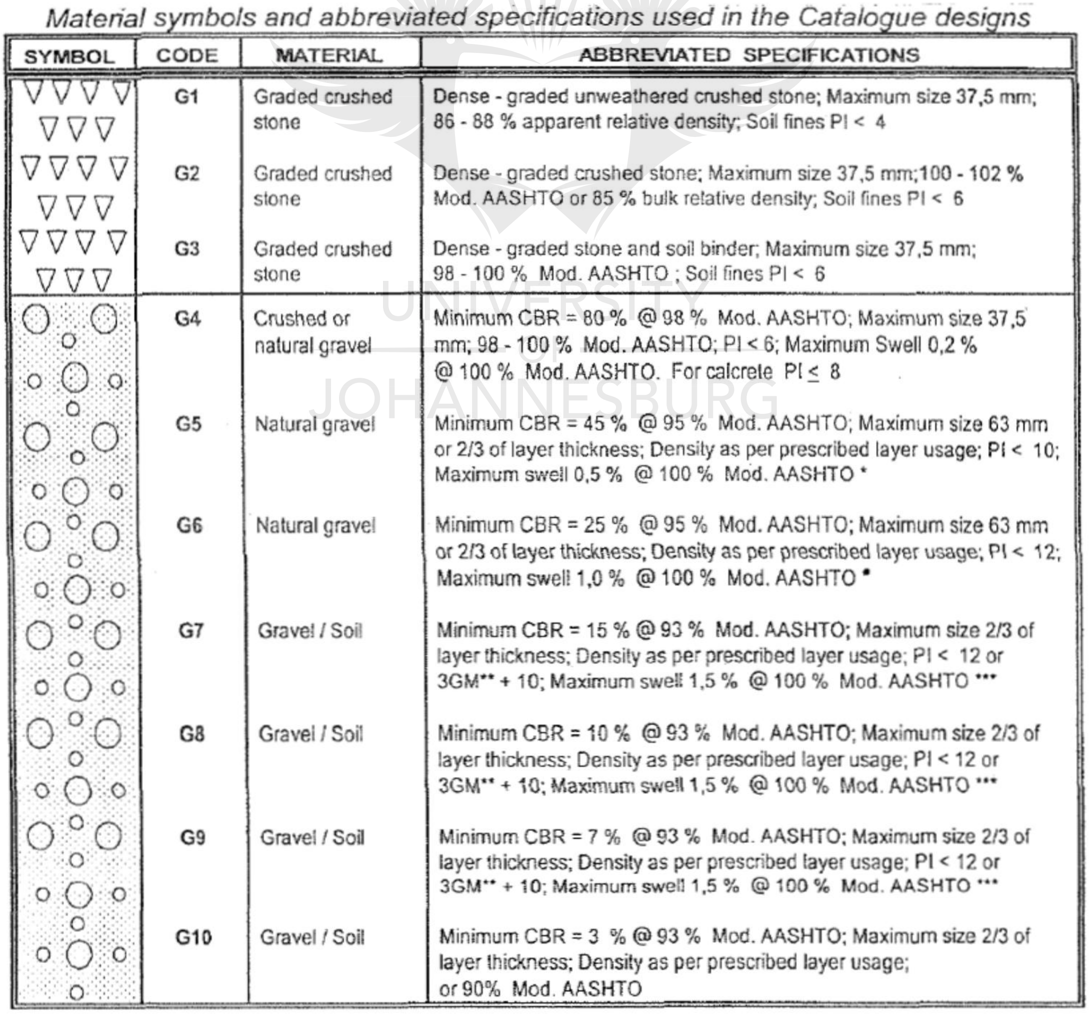
These gravel types are found in borrow pits on the side of the road. It is important to demarcate the extent of the suitable materials in the borrow pit clearly and ensure that material is only obtained from this area. Excavation depth is very important and should be carefully controlled so as not to excavate into less weathered or different material.

Gravel spreading

The thickness must be as consistent as possible over the length of the link to avoid total loss of gravel over portions of the link, only. Material must be dumped on the road at the correct spacing to provide a consistent layer thickness of gravel after spreading and compaction. Inadequate gravel spreading will result in incorrect thickness and will lead to early re-graveling of the road to achieve the required thickness.

Gravel & Sand compaction lift thickness. TRH9 [20]

"The conventionally compacted layer thickness for soil, sand, clay and gravel is 150 mm. However, layers of 200 mm to 300 mm is much thicker in rock and rocky material are more in line with present conditions. The layer thicknesses are dependent on the maximum particle size of the material and the efficiency of the compaction equipment."



Material symbols and abbreviated specifications used in the Catalogue designs

SYMBOL	CODE	MATERIAL	ABBREVIATED SPECIFICATIONS
▽▽▽▽ ▽▽▽	G1	Graded crushed stone	Dense - graded unweathered crushed stone; Maximum size 37,5 mm; 86 - 88 % apparent relative density; Soil fines PI < 4
▽▽▽▽ ▽▽▽	G2	Graded crushed stone	Dense - graded crushed stone; Maximum size 37,5 mm;100 - 102 % Mod. AASHTO or 85 % bulk relative density; Soil fines PI < 6
▽▽▽▽ ▽▽▽	G3	Graded crushed stone	Dense - graded stone and soil binder; Maximum size 37,5 mm; 98 - 100 % Mod. AASHTO ; Soil fines PI < 6
○ ○ ○	G4	Crushed or natural gravel	Minimum CBR = 80 % @ 98 % Mod. AASHTO; Maximum size 37,5 mm; 98 - 100 % Mod. AASHTO; PI < 6; Maximum Swell 0,2 % @ 100 % Mod. AASHTO. For calcrete PI ≤ 8
	G5	Natural gravel	Minimum CBR = 45 % @ 95 % Mod. AASHTO; Maximum size 63 mm or 2/3 of layer thickness; Density as per prescribed layer usage; PI < 10; Maximum swell 0,5 % @ 100 % Mod. AASHTO *
	G6	Natural gravel	Minimum CBR = 25 % @ 95 % Mod. AASHTO; Maximum size 63 mm or 2/3 of layer thickness; Density as per prescribed layer usage; PI < 12; Maximum swell 1,0 % @ 100 % Mod. AASHTO *
	G7	Gravel / Soil	Minimum CBR = 15 % @ 93 % Mod. AASHTO; Maximum size 2/3 of layer thickness; Density as per prescribed layer usage; PI < 12 or 3GM** + 10; Maximum swell 1,5 % @ 100 % Mod. AASHTO ***
	G8	Gravel / Soil	Minimum CBR = 10 % @ 93 % Mod. AASHTO; Maximum size 2/3 of layer thickness; Density as per prescribed layer usage; PI < 12 or 3GM** + 10; Maximum swell 1,5 % @ 100 % Mod. AASHTO ***
	G9	Gravel / Soil	Minimum CBR = 7 % @ 93 % Mod. AASHTO; Maximum size 2/3 of layer thickness; Density as per prescribed layer usage; PI < 12 or 3GM** + 10; Maximum swell 1,5 % @ 100 % Mod. AASHTO ***
	G10	Gravel / Soil	Minimum CBR = 3 % @ 93 % Mod. AASHTO; Maximum size 2/3 of layer thickness; Density as per prescribed layer usage; or 90% Mod. AASHTO

Figure 20: Material Identification chart TRH 4.

6.5.4 General earth road defects that may apply to rural roads.

Reference: TRH20 [19] and (Figures 21 to 29) by FL Martinez [9]).

Potholes are significant in the development of roughness on earth roads that is caused by poor compaction and final road surface finish. A rural earth road that has a poor road geometry finish has been poorly designed during grading and compaction stages. This generally leads to seepage and drainage problems that give rise to pothole formation. Vehicle damage can occur when potholes reach sizes between 250 mm and 1500 mm in diameter with a depth of more than 50 to 75 mm. See Figure 21.

Figure 21: Potholes

Cracks are one of the contributors of potholes in the road surface. Dry seasons are usually the main cause of earth road surface cracking. Scientifically, the material has a very high plastic index (easy deformation) and low shear strength to keep the particles together. Compaction increases the shear strength and lowers the plastic index. See Figure 22.

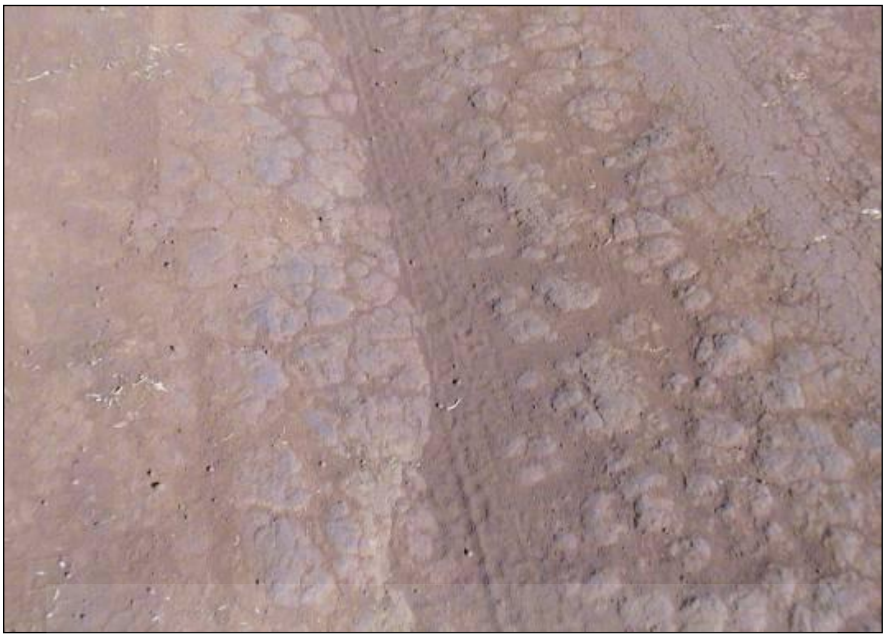


Figure 22: Road bed cracks.

Dustiness involve loose silt particles (2 µm – 75 µm) lying on the earth road surface that gets released by passing vehicles throwing excessive dust into the air forming poor road visibility and health hazards for people living nearby. Dust clouds form with different soil particle sizes and densities and are dependent on the passing vehicles size, speed of travel, tyre thread, wheel diameter, aerodynamic shape and soil properties. Initial compaction quality is very important to keep the dust particles bonded and locked together thus increasing the shear strength and lowering the Plastic Index. See Figure 23.

Figure 23: Loose soil particles can cause dustiness.

Corrugations are horizontal rifts or bumps sticking out of the road surface that causes vehicles to oscillate, compacting the valleys during resonant velocity and compacting the hills during an increased velocity touching only the top surface of the hill. As more vehicles with poor suspension use the road, the valleys will deepen and then result in greater amplitudes of oscillation. The valleys will then stop to go deeper once maximum compaction has been achieved for the specific type of material. See Figure 24.

Figure 24: Corrugation formation

Stoniness refers to stones sticking out of the earth road surface with a particle size equal to or greater than 37,5 mm causing rough rides. Stones prevent inadequate compaction next to and around areas, causing pot holes to form. Disadvantages of stoniness are difficult road maintenance, unnecessary grader maintenance, vehicle maintenance, raveling and corrugations. See Figure 25.

Figure 25: Stoniness

Ruts are parallel depressions in the earth road surface in the wheel tracks which are also parallel to the direction of traffic. They form due to deformation and poor compaction of the road surface. Water retention within the rut depth during down pours cause further surface defects. “The main cause of rutting in southern Africa is the raveling of low-cohesion material under traffic movement. A secondary cause is the deformation of highly cohesive wearing course materials under traffic”. See Figure 26.

Figure 26: Rut formation

Ravelling refers to the formation of loose gravel under traffic and can represent a significant safety problem. In dry seasons more material is loosened from the road surface then during the wet seasons. Variable soil particles sizes allow road surfaces to ravel away. This is due to the lack of cohesion in the fine material and poor particle size. See Figure 27.

Figure 27: Ravelling

Erosion is caused when water flow over the road surface and it takes surface material away over time. Resistance to erosion depends on the shear strength of the road surface; shear strength is increased by compaction forcing the particles tightly together that increases contact forces between particles. (See Figure 28)

.

Figure 28: Erosion

Shape is viewed as the cross-sectional profile of the road surface geometry that stays in shape with proper compaction maintenance and routine grading. The geometrical shape of an earth road prevents inadequate drainage which causes erosion, ruts and potholes. The slope of the road is also a very important factor, if the slope is too small poor drainage and water transportation will occur. When the slope is too large, road surface erosion will occur, because water flowing at high velocity takes away surface material.

Figure 29 is a representation of the official (TRH) road geometry construction documents for South African earth roads. (See Figure 29).

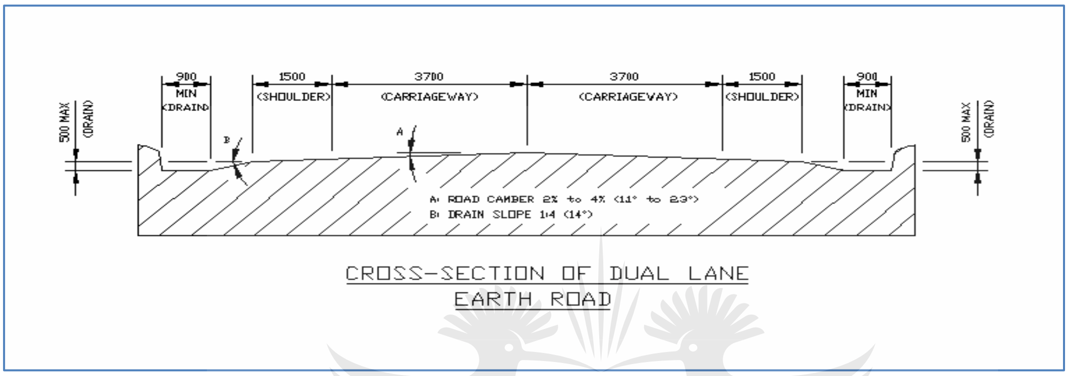


Figure 29: Dual lane earth road geometry (F.L. Martinez [9]).

“Slipperiness is promoted by wet weatherand is caused by excessively fine or plastic material in the wearing course that results from non-standard compaction. Even materials with adequate coarse aggregate may become slippery if the fine silt and clay fraction becomes concentrated near the surface.” See Figure 30.

Figure 30: Slippery road surface.

"Impassability is an objective rarely met by earth roads in wet conditions. The unpaved road is required to provide an all-weather surface which is often a problem where in-situ materials are used to meet these criteria. A wearing course gravel layer is required on the road surface". Poor compaction standards give rise to low material strength that does not comply with all weather conditions. It is generally considered that an adequately high material strength will provide a trafficable surface under all conditions. See Figure 31.

Figure 31: Impassable dirt road.

Gravel loss takes place over time wearing thinner as traffic passes by during various climatic conditions. Replacing gravel at correct intervals is very expensive and requires a lot of maintenance. Gravel is found in borrow pits as close to the site as possible to avoid bringing in foreign gravel that will result in extra transportation costs. See Figure 32.

Figure 32: Road side gravel loss.

7. Machinery overview, specification and cost summary.

7.1 Characteristics of a vibratory road roller.

Any vibratory road roller must be able to comply with the following basic machine specifications:

- Operating Weight (kilograms)
- Drum Diameter (millimetres)
- Drum Width (millimetres)
- Number of Vibrating Drums
- Overall Width (millimetres)
- Operating Height (millimetres)
- Overall Length (millimetres)
- Wheelbase (millimetres)
- Static Weight at Drum(s) and at Tyre(s) (kilograms)
- Centrifugal Force at a Stated Frequency (kilo-Newton,)
- Vibrating Mass (kilograms)
- Eccentric Moment (kilogram meters)
- Frequency (hertz)
- Nominal Amplitude (millimetres)
- Dual Amplitude (millimetres)

.

These are just a few of the needed specifications, but are the most important specifications. Appendix 15 to 20 indicates this type of specifications for current marketed SDVRR's.

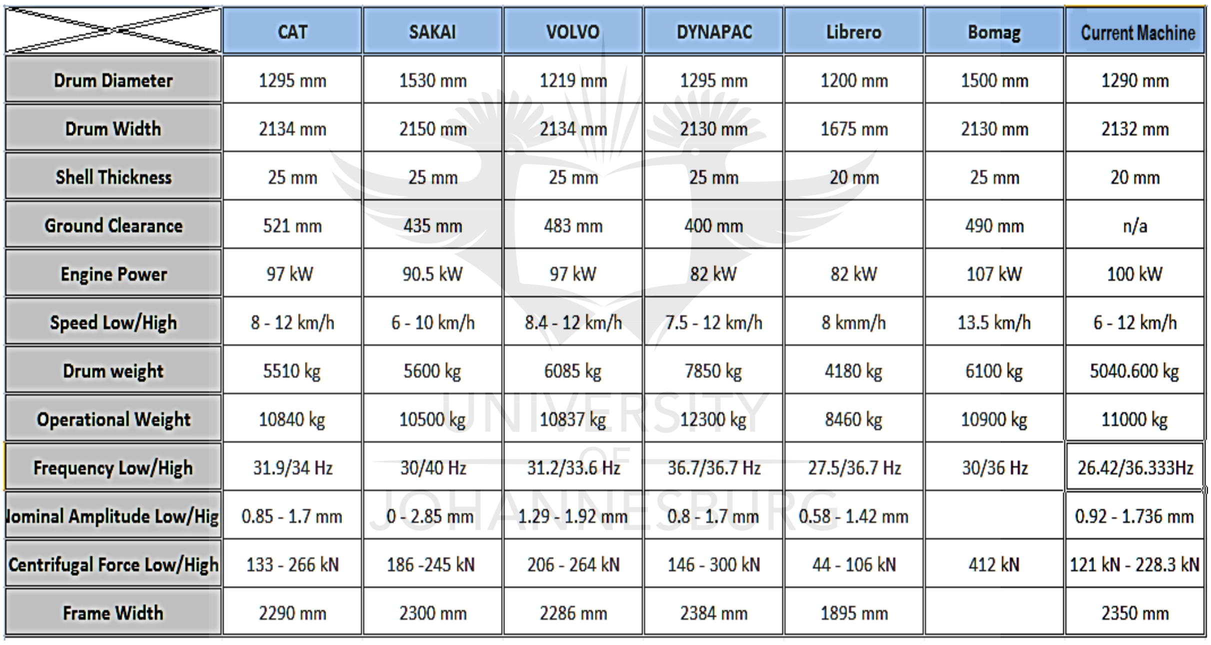
7.2 Current marketed single drum road roller specification comparison.

Table 2 shows a summary of all the road roller manufacturer catalogues in Appendices 15 to 20. These specifications are compared to the actual specifications of the designed machine contained within this document.

The details in Table 2 in the "Current Machine" column are the final design specifications of the machine. It gives a clear indication of the final machine specification outcome of the project and enhances the understanding of further calculations in other chapters.

As per 6.6.1 all relevant specifications are stipulated in Table 2.

Table 2: Overall compactor specification comparison.



	CAT	SAKAI	VOLVO	DYNAPAC	Librero	Bomag	Current Machine
Drum Diameter	1295 mm	1530 mm	1219 mm	1295 mm	1200 mm	1500 mm	1290 mm
Drum Width	2134 mm	2150 mm	2134 mm	2130 mm	1675 mm	2130 mm	2132 mm
Shell Thickness	25 mm	25 mm	25 mm	25 mm	20 mm	25 mm	20 mm
Ground Clearance	521 mm	435 mm	483 mm	400 mm		490 mm	n/a
Engine Power	97 kW	90.5 kW	97 kW	82 kW	82 kW	107 kW	100 kW
Speed Low/High	8 - 12 km/h	6 - 10 km/h	8.4 - 12 km/h	7.5 - 12 km/h	8 kmm/h	13.5 km/h	6 - 12 km/h
Drum weight	5510 kg	5600 kg	6085 kg	7850 kg	4180 kg	6100 kg	5040.600 kg
Operational Weight	10840 kg	10500 kg	10837 kg	12300 kg	8460 kg	10900 kg	11000 kg
Frequency Low/High	31.9/34 Hz	30/40 Hz	31.2/33.6 Hz	36.7/36.7 Hz	27.5/36.7 Hz	30/36 Hz	26.42/36.333Hz
Iominal Amplitude Low/Hig	0.85 - 1.7 mm	0 - 2.85 mm	1.29 - 1.92 mm	0.8 - 1.7 mm	0.58 - 1.42 mm		0.92 - 1.736 mm
Centrifugal Force Low/High	133 - 266 kN	186 -245 kN	206 - 264 kN	146 - 300 kN	44 - 106 kN	412 kN	121 kN - 228.3 kN
Frame Width	2290 mm	2300 mm	2286 mm	2384 mm	1895 mm		2350 mm

The only major difference between the "Current Machine" column and the other machines are the 20 mm thickness of its drum shell. All the others have a drum shell thickness of 25 mm, except the Librero, which is a smaller machine then the other shown in table 2.

The 20 mm drum shell is reinforced by 12 ribs on the inside to increase drum stiffness and decrease costs.

7.3 Brands of road rollers brand specifications.

Appendix 15 to 20 contains six catalogues of 10 ton SDVRR's. These include Caterpillar (CAT), Sakai, Librero, Ingersoll Rand, Volvo, Bomag and Dynapac. These are very efficient machines that have been optimized by years of company experience. This is one of the main reasons why these machines have a very high price tag attached to it, referenced in Appendix 45. Although the price tags of the machines in Appendix 45 are prices of second hand machines from all over the world, it gives some idea of their prices in general.

Table 3 shows a summary of the costs involved with different machines listed in Appendix 45. NB: Only five brands have the price shown in Table 3 and it represents the average price per year. Two to four machines for each brand are evaluated. See Appendix 45 for the costs pertaining to Table 3.

Table 3: Cost summary of the different machines.

Model year	Caterpillar	Bomag	Dynapac	Ingersoll Rand	Volvo
2012	R 935 705 4403 hrs	R 1 119 803 195.5 hrs	R 937 634 430 hrs		
2011	R 946 564 1278 hrs				R 884 056 1270 hrs
2010			R 789 399 841 hrs	R 874 754 1222 hrs	R 904 076 6152 hrs
2009					R 664 381 Unknown
2008				R 830 772 3683 hrs	R 1 018 103 596 hrs

Table 3 indicates that machine price tag is based on the amount of hours it has been used rather than the year of manufacture for the models. The model year has little or no influence, but the hours of operation are primary when it comes to determining the price tag.

<u>Price influenced by hours used</u>

In table 3 Bomag has a price tag of R 1 119 803 for a 2012 machine that has only been used for 195.5 hours, while a 2008 Volvo machine with 596 hours boasts about the same price tag at R 1 018 103. These two models were manufactured four years apart, but yet the price is almost the same, thus machine price is determined relative to its hours of use.

Another example is the 2012 Caterpillar model with its operating hours of 4403 hrs is going for R 935 705 and the 2011 Caterpillar, 1278 hrs model goes for R 946 564. The earlier 2011 model has less hours of operation and is priced higher than the later 2012 model. NB: R 1.2 million to R 1.3 million price tag for a new 2014 model is a good estimate when it comes to Volvo, Caterpillar and Bomag which seems more expensive than Ingersoll Rand and Dynapac machines. Judging from Table 3, Dynapac and Ingersoll Rand machines can be estimated at R 1 million to R 1.1 million for a new 2014 model.

7.4 Documented machine cost

To be able to compete in the overcrowded market of compaction machinery, the documented machine should cost much lesser than the current marketed ones. In the costing section Appendix 51, the final selling price of the designed machine in this dissertation is quantified as R 812 195,1. This final selling price was obtained by different quotes, an estimated labour cost and a mark-up price to generate revenue. Refer to Appendix 51.

By comparing the designed machine selling price with those currently on the market, it is observed that the Dynapac and Ingersoll Rand machines are about R 300 000 more than the designed machine. Where Caterpillar and Bomag are concerned, there is an estimate of a massive R 500 000 difference.

NB: This is a clear indication that the machine is highly cost competitive against the current marketed SDVRR’s, however in Appendix 46, there are some Chinese machinery quotes and the cost of their machines is about R 350 000 including shipping costs. The actual quality of these machines is unknown, but the machines can be overhauled with quality bearings to increase the bearing life of these machines. Other parts such as the hydraulic pump and all its relevant parts need to be monitored for its operating life. With a markup of 60%, the machine can be sold for R 560 000 to rural area municipalities. The high markup is to accommodate the guarantee while the machines life is monitored for product experience purposes as to which parts will fail first.

The other option is to fly to China and get references from road construction companies that have experience of using these types of Chinese machines.

8. Constraints & Criteria

8.1 Constraints

Roller drum width: 2100 mm
Roller drum diameter: 1290 mm
Machine weight: 5000 kg
Tractor weight: 6100 kg
Compacting drum frame width: 2350 mm
Compacting drum frame length: 2545 mm
Pulley and sprocket sizes are constrained as per catalogue recommendations.

8.2 Criteria

During the Design and Development phase the performance criteria for the machine were discussed and agreed on with the future manufacturer Terragrader (PTY) LTD.
These are as follows:

Machine compaction depth or lift: 100 mm - 150 mm
Roller Drum vibrational frequency: 31.5 Hz – 40 Hz
Eccentric weight rotational speed: 2100 r/min – 2200 r/min

9. Description of machine components and their functions.

Figure 33 shows the rough layout of the machine refined in the later chapters. The machine layout in Figure 33 was decided upon after a number of concepts were considered as seen in Appendix 21. The figure sheds light on the different regions where calculations are done as the chapter progresses. The components are explained in detail in the following subchapters.

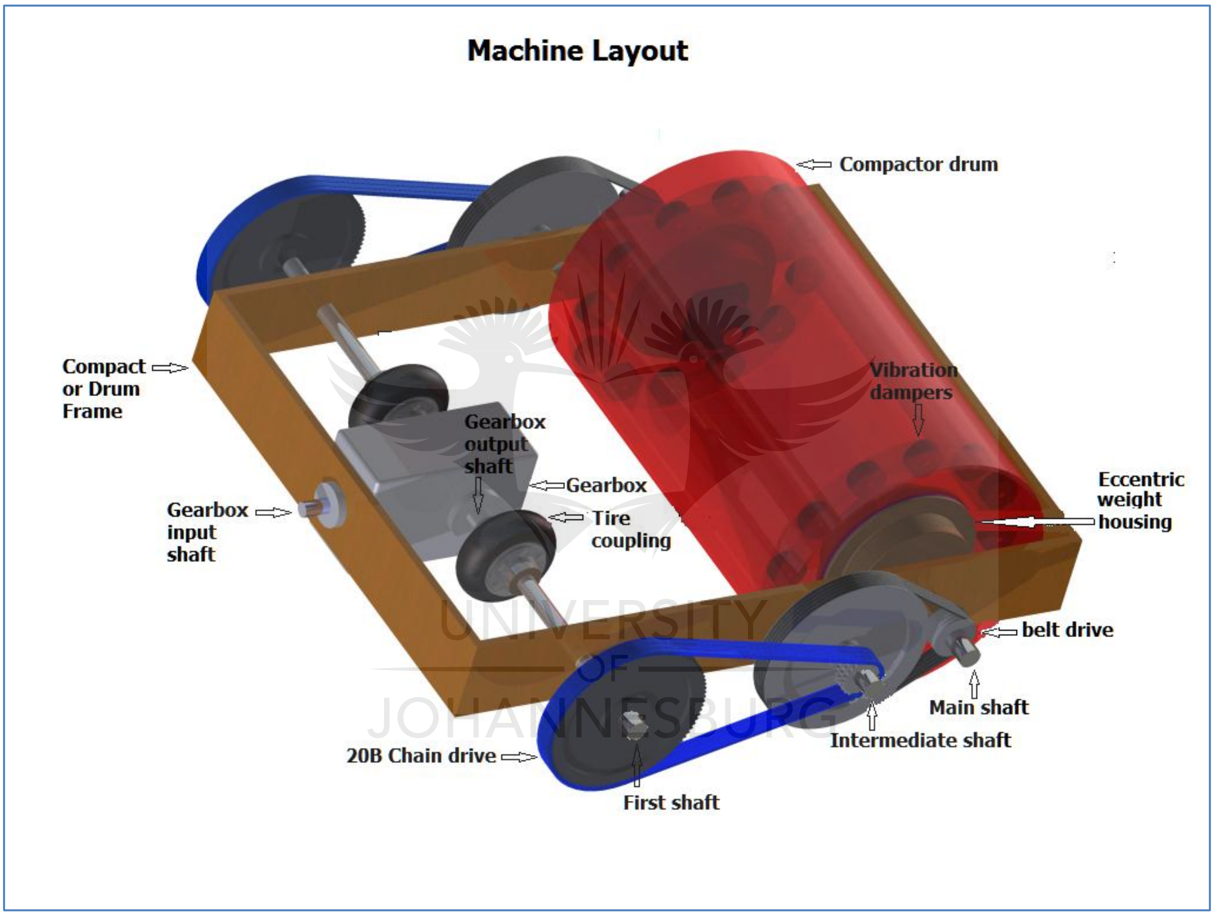


Figure 33: Machine layout

9.1 Drive shaft

The drive shaft is connected between the gearbox input shaft and the tractor PTO (not shown). The drive shaft is running at 1437 r/min transmitting power through its universal joints to the gearbox input shaft.

9.2 Gearbox

At the gearbox, a 10.1:1 ratio is used to step down the input speed to a lesser output speed. The input shaft receives 1437 r/min and outputs a lesser speed of 131.97 r/min.

9.3 Tyre couplings

The location of the tyre couplings are on either side of the gearbox output shafts and promote symmetry in the design as well as the rest of the power transmission components from this point on. Each coupling receives half the power and is running at 131.97 r/min. Primarily the tyre couplings are used to absorb vibration and misalignment between the gearbox output shafts and First Shaft (FS), because of the high damping factor of 0.9 the couplings possess.

9.4 Chain drive

On the input side of the chain drive the larger sprocket is connected to the FS and is running at 131.97 r/min on the output side, while the smaller sprocket is assembled to the Intermediate Shaft (IS) and is moving at a rotational speed of 545 r/min.

9.5 Belt drive

The belt drive has a ratio of 4:1 and steps up the speed further from 545 r/min at the larger pulley to a final output speed of 2180 r/min at the MS where the smaller pulley is located.

9.6 Eccentric weights

Located on the MS the eccentric weights are assembled via a heavy press fit to allow for adequate clamping onto the shaft. The heavy press fit will then create pressure between the eccentric weight hub or sleeve and the MS.

9.6.1 Eccentric weight concepts and designs.

Many other eccentric weight concepts and actual designs are featured in Appendix 24, Appendix 25 and Appendix 26 for information purposes. A step by step Finite Element Design approach was used to optimize most of the conceptually designed eccentric weights mentioned in the above Appendices.

- Appendix 24 features the stress analysis done using the Inventor 2013 software for the actual Caterpillar eccentric weight design for their single smooth drum vibratory compactors. This simulation indicates the proof of understanding on how to achieve an accurate result in the simulation of eccentric weights.

- Appendix 25 features different eccentric weight concepts and their operation. All the shown concepts were abandoned due to the impossibility to reverse the machine drive system as to rotate the drive shaft in the opposite direction at such a high power (120 kW) output. It is also dependent on where the reverser would sit on the drive. The only speed reversers obtainable on the market is from Zeromax and can handle very low power outputs as seen in their catalogues in Appendix 23.

- Appendix 26 shows a constructive stress analysis method of the refined designs of two of the eccentric weights. These eccentric weights can be used in the actual machine, but due to the reverser problems encountered, it was abandoned. It is only included to show that an attempt was made to operate the machine at low and high amplitude at the same rotational speed.

In the current documented design, the different amplitudes are achieved by switching from one PTO speed to a lower speed. The problem with controlling centrifugal force with speed instead of eccentric moment manipulation is that the production rate will be less at low amplitude due to the slower speed. When the eccentric moment is altered, the speed can remain the same and so doing will promote faster production.

9.7 Eccentric weight housing

The eccentric weight housings, houses the eccentric weights and their respective bearings on either side. The bearings are located in the cavity of the housings.

9.8 Vibration absorbsion gaskets

Vibration isolation rubber gaskets are used between the eccentric weight housings and the compactor drum, these isolate the eccentric weight housings that are connected to the MS from any remaining vibration that is caused by the impact load the compactor drum experience during compaction.

9.9 Compactor Drum

The compactor drum is connected to the machine frame via bearings and is used to compact subgrade and roll forward at the same time during compaction. Impact occurs at a point on the circumference along the entire length of the compacting drum.

The stiffer or denser the soil become that is being compacted, the smaller the soil damping coefficient will be and as a result, greater vibration travels back to the machine components due to drum bounce on the harder or stiffer soil subtrate. Vibration will move up towards the MS through the drum rings on the inside of the drum, thus 24 vibration dampers are placed between the drum rings to intercept the vibration before it travels to the MS. The remainder vibration that gets past this point will then be intercepted by the vibration absorbsion gaskets that are isolating the eccentric weights housings from the drum before it gets to the MS. The MS will then experience little or no vibration.

9.9.1 Compactor drum material selection.

In Appendix 39 some of the hardest and hardwearing steels are shown. These include Weldox, Hardox, Armox and Toolox. To select the right type of material for the compacting drum, a closer look has to be taken at its uses.

- Weldox – can be welded to most steels including Hardox and its primary use is structural.
- Hardox – can handle abrasion and impact very well and is widely used in road construction equipment, including vibratory road rollers.
- Armox – is mainly used to resist penetration of the metal.
- Toolox – is used for hard tips in tools to cut and shape different mechanical components.

Hardox is the obvious choice for the drum shell that will scrape on the road surface and experience impact during compaction. The drum shell will therefore be made of Hardox to accommodate for scuffing and impact. See Appendix 41 for full specifications.

Weldox on the other hand will be used for structural support on the inside of the compacting drum and is easily weldable onto Hardox. See Appendix 42 for full specifications.

9.10 Vibration Dampers

Vibration dampers will intercept the vibration that tries to travel up the drum rings to the eccentric weight housing, already mentioned in 8.9.

9.11 Compactor frame

The compactor frame houses the whole assembly and is therefore the main structural support that keeps the machine assembly together. It basically houses all the above mentioned components.

All soil or material that sticks to the drum shell surface is removed via a drum scraper that is assembled on top of the frame. The drum scraper can be adjusted by a set of screws located on top and the front of the plate to move it in the forward or backward direction. The slotted holes on the scraper allows sliding and thus the scraping force can be adjusted to scrape more effectively relative to the cohesion of the material being compacted that sticks to the drum shell. Appendix 11 shows a more complete final assembly of the machine in 3D format and the drum scraper plate is shown in the top view of the frame as it makes contact with the drum shell.

9.12 Drum scraper plate adjustment screws.

The drum scraper plate adjustment screws allow the scraper plate tip to move forward towards the circumference of the drum with each turn. This method allows for the fine adjustment of the contact force between the drum circumference and the scraper plate tip.

The drum scraper plate itself, is designed with a type of spring stiffness to keep continuous contact with the drum surface. Adjustment screws alter the contact force between the drum shell circumference and the plate tip for efficient scraping. This method allows for continuous contact between the two surfaces during vibrational operation. See Appendix 11 for the positioning of the scraper plate.

9.13 Bolts and nuts.

All bolts and nuts to be used in the assembly need to be of grade 8.8 strength to accommodate the high clamping torque needed for vibration purposes.

10. Catalogue design calculations.

All calculations done in this section pertain to the selection of industrial products that are readily available over the counter at the various respective suppliers. The design work from these catalogues in Appendices 3, 6, 7, 8, 9, 12, 13 and 14 is utilised in this section.

Verifications of the selections were made with the various company product engineers involved. In any design work where the designer has no experience, it is always best to call the experts first to avoid any unnecessary problems and design work, but in this case, it was necessary to show competence in using the catalogue design processes before consulting the respective professionals.

Telephonic conversations, emails (Appendix 48) and one on one consultations were used to obtain the best design options available. Trial and error methods with a great amount of engineering sense are demonstrated in this chapter to achieve an optimum working design.

10.1 Consulted companies for design refinement and guidance.

- ZF South Africa (Pty) Ltd (Off Road Department) for the tractor gearbox/PTO combination called the T-7232 tractor transaxle transmission system. Technical Representative: Off-Road vehicles.
- Spicer & Hardy USA (Pty) Ltd. The Spicer & Hardy Off-highway catalogue was only consulted to demonstrate competence in driveshaft selection. Propshaft world (PTY) LTD is the agent that was consulted via telephonic conversation; prop-shaft manufacturing technical department.
- Bonfiglioli South Africa (Pty) Ltd for the selected gearbox; The Company Technical Representative.
- Bearings International (Pty) Ltd is the supplier of all Fenner drives like chains, sprockets, pulleys, couplings etc; OEM Sales Department.
- SKF for all relevant bearing selections.
- MacSteel for shafting requirements.

10.2 Other companies consulted telephonically.

These are companies that yielded almost similar specifications for gearboxes, couplings, Fenner drives and drive shafts. Costs are considered between all these companies, but due to the size of this project, prices are only sourced from companies whose catalogues was used in this design. Further price sourcing for products with similar specifications can be done privately at Terragrader (Pty) Ltd after the finalisation of the design to reduce further costs.

See all alternative companies consulted below:

- David Brown Gear Industries (Pty) Ltd (Gearboxes sourcing);
- Hansen Transmissions South Africa (Pty) Ltd (Gearbox sourcing);
- Renold Crofts (Pty) Ltd (Gearbox sourcing);
- Bearing Man Engineering (BMG) (Pty) Ltd (Gearbox sourcing);
- Rodecon Engineering (Pty) Ltd (Gearbox sourcing);
- Propshaft World (Pty) Ltd (Drive Shaft Spicer Series);
- Bearing Man Engineering (BMG) (Pty) Ltd (Tyre Couplings and Fenner drives);
- FAG bearings alternative to SKF bearings;
- And Timken is also a reliable quality bearing source.

For the allocation of these companies, the monthly magazine, “The Technical Buyers Guide” was consulted. This source had a tremendous amount of suppliers for different Industrial applications.

10.3 PTO output power and speed.

Referring to the T-7232 catalogue from ZF transmissions (Appendix 3) the following data is recorded.

Tractor Engine Power = 149 kW

Gearbox input power is 134 kW after engine losses.

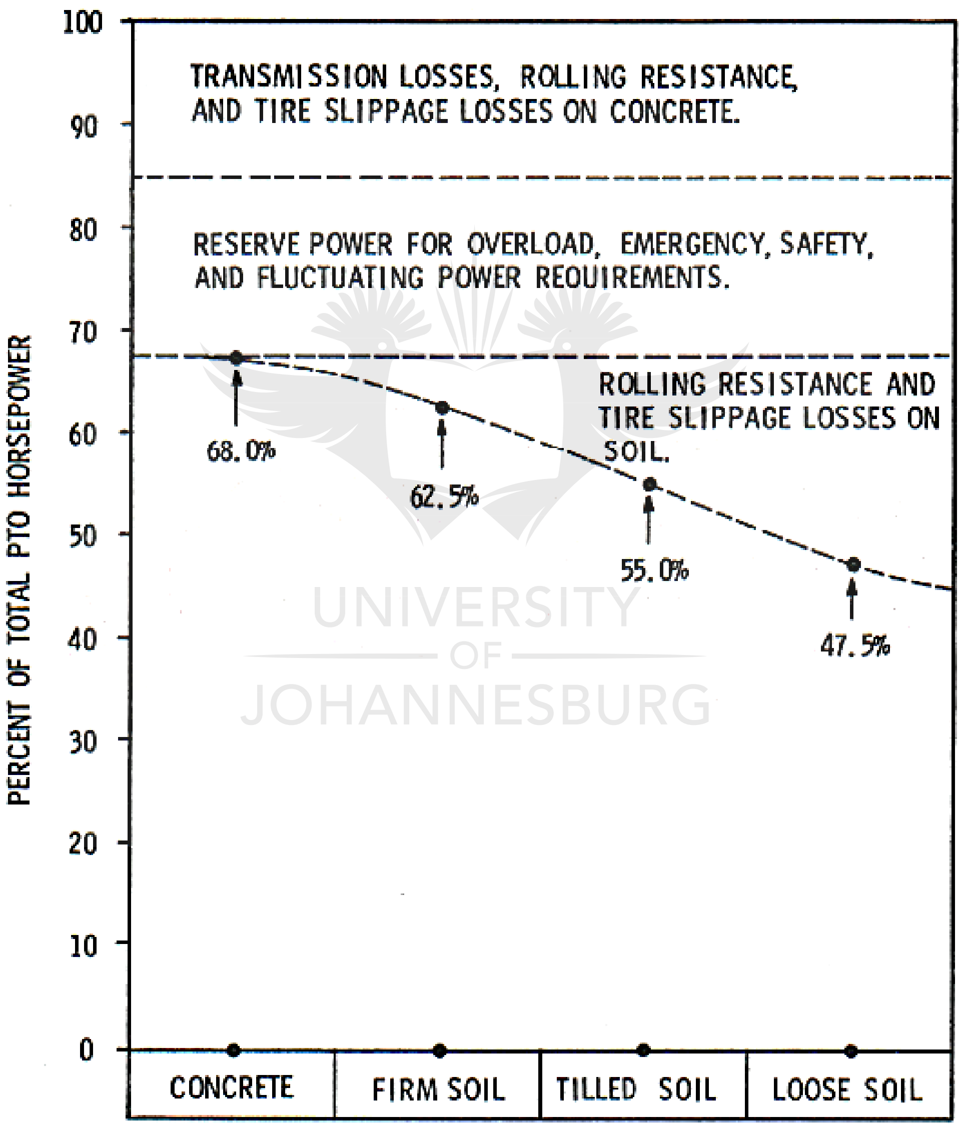
P_{PTO} *(PTO output power) is 120 kW after gearbox and PTO losses.* (Appendix 48: Email from Chris Higino ZF Transmissions)

The chosen Output PTO speed for machine operation is the level 4 speed of 1437 r/min. By selecting the higher speed between the four stipulated PTO output speeds in Appendix 3, the final operating speed at the compactor MS is easier attainable when it comes to the stepping up of the drive, due to the reduction of the gearbox. The operating speed or frequency needed for compaction was one of the requested constraints given by the client (Terragrader (PTY) LTD) and it was then compared with the road roller catalogues in Appendix 15 to 20.

10.3.1 Tractor net drawbar power tests.

Reference: Matching tractor horsepower and Farm implementation size [21].

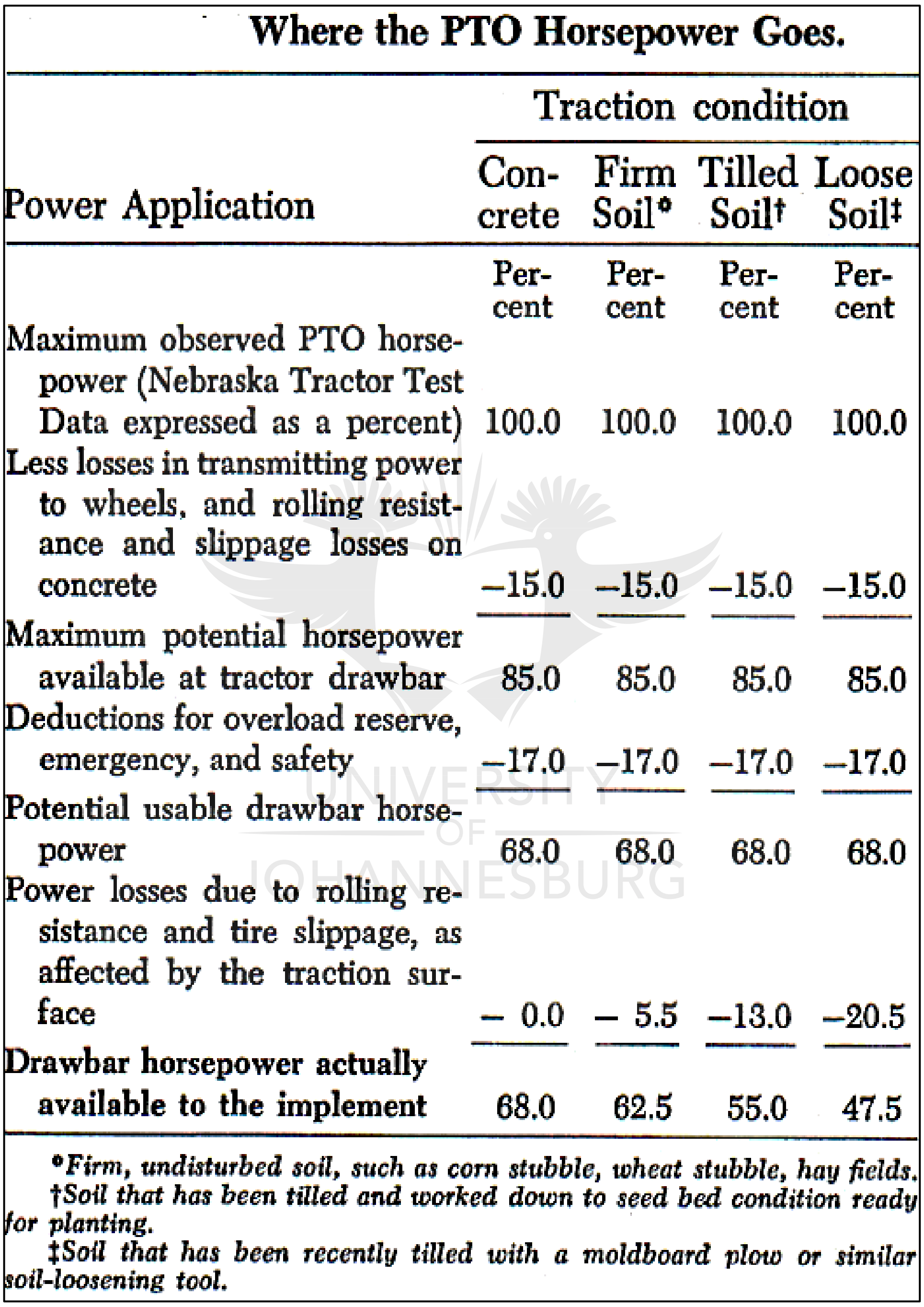
Tractor test results shown in Figures 34 and 35 indicate various losses on different ground conditions for tractors. Since the actual weight of the tractor that is going to tow the machine is not yet known, the following research is a good reliable source to calculate the power lost by the tractor itself.



Where the PTO horsepower goes. Net drawbar horsepower available for use at the tractor drawbar on four surface conditions.

Figure 34: Percentage power loss due to soil state.

In Figures 34 and 35 the tractor drawbar power can be calculated from four different surface conditions. The tractor test that was done on concrete and tilled soil is solely for farming applications, thus it will be ignored. The compactor will only be used on loose soil and firm soil conditions.



Where the PTO Horsepower Goes.

Power Application	Traction condition			
	Concrete	Firm Soil*	Tilled Soil†	Loose Soil‡
	Percent	Percent	Percent	Percent
Maximum observed PTO horsepower (Nebraska Tractor Test Data expressed as a percent)	100.0	100.0	100.0	100.0
Less losses in transmitting power to wheels, and rolling resistance and slippage losses on concrete	–15.0	–15.0	–15.0	–15.0
Maximum potential horsepower available at tractor drawbar	85.0	85.0	85.0	85.0
Deductions for overload reserve, emergency, and safety	–17.0	–17.0	–17.0	–17.0
Potential usable drawbar horsepower	68.0	68.0	68.0	68.0
Power losses due to rolling resistance and tire slippage, as affected by the traction surface	– 0.0	– 5.5	–13.0	–20.5
Drawbar horsepower actually available to the implement	68.0	62.5	55.0	47.5

**Firm, undisturbed soil, such as corn stubble, wheat stubble, hay fields.*

†Soil that has been tilled and worked down to seed bed condition ready for planting.

‡Soil that has been recently tilled with a moldboard plow or similar soil-loosening tool.

Figure 35: Tractor PTO power distribution.

Loose soil conditions can also be further ignored, because the compactor will just be towed with its static weight over this type of terrain without the need to operating the eccentric weights during the tow process. This will ensure a firm type of soil base that flex little under the static weight of the machine. NB: All calculations from this moment on will be done based on firm earth conditions.

For the design of the power transmission equipment such as gearboxes, drives, couplings etc., in Figure 34 and Figure 35 the 17% reserved power will not be deducted for component design purposes. The Net power is needed to select mechanical transmission equipment from manufacturer design catalogues, where input power is of utmost importance to obtain design power, after multiplication of an appropriate service factor (safety factor). This will also ensure that there is no mechanical failure during torque overloading.

10.3.2 Calculation of available PTO power on firm and loose earth.

10.3.2.1 Power lost on loose earth.

The power at the PTO is 120 kW as seen in 9.4. The losses experienced on firm earth amounts to 20.5%. See Figures 34 and 35.

$$P_{PTO\ available} = P_{pto}(P_{\%\ loss.firm.earth} - P_{\%\ loss.concrete})$$

$$= 120(0.85 - 0.205)$$

$$= 77.4\ kW$$

9.3.2.2 Power lost on firm earth by the tractor.

$$\boldsymbol{P_{roll\ resistance\ level\ firm\ earth}} = P_{pto}(P_{\%\ loss.firm.earth} + P_{\%\ loss.concrete})$$

$$= 120(0.055+0.15)$$

$$= 24.6\ kW$$

Thus, the available PTO power available after firm earth losses is as calculated below.

$$P_{PTO\ available} = Ppto - P_{tractor\ drawbar\ loose\ earth}$$

$$= 120 – 24.6$$

$$= 95.4\ kW$$

10.4 Tractor power losses for loose and firm earth.

To verify the soundness of the mentioned research in 9.3, to qualify it to be used in a practical sense, a check has to be done by consulting another reference source. The handbook for heavy construction written by FW Stubbs [22] is an excellent source concerning rolling and inclined resistances to determine the tractor drawbar power lost on firm earth.

10.4.1 Assumption of the tractor mass.

The tractor in Figure 36 has no cladding, which means the mass has to be estimated. At the time of writing this dissertation the tractor that was designed by C Popa [10], had no cladding. See Figure 36 {C. Popa [10]} which is a drawing of the final product without a cabin and cladding. All cladding for the tractors body and other necessary modifications will be added during the fabricators of the tractor, by Terragrader (Pty) Ltd.

Since the tractor PTO output power is 120 kW the mass can be estimated by looking at the tractor catalogues in (Appendix 43), this will allow the determination of the mass power relationship of tractors.

Tractor Power is always rated at the PTO output power according to the Nebraska tractor tests [21], in this case, the PTO power is 120 kW. Looking at the 114 kW tractor, its mass is 5610 kg (Appendix 43).

By power to mass ratio calculation, an estimate can be made. The 5905.263 kg calculated below is the estimated mass of the tractor.

$$\frac{\mathbf{114}}{\mathbf{5635}} = \frac{\mathbf{120}}{\boldsymbol{m}} \therefore \boldsymbol{m} = \mathbf{5931.6}\ \boldsymbol{kg}$$

Assuming the actual tractor mass to be 6000 kg, which will allow a safe margin when calculating the drawbar power losses for the tractor?

10.4.2 Machine operating velocity.

The maximum operating velocity (MOV) of these compactors is about 6 km/h to 12 km/h (Appendix 15 - 20 compactor catalogues), for design purposes 12 km/h will be used as the MOV. Below is the MOV conversion from km/h to m/s.

$$\mathbf{v = \frac{1000 \times 12}{3600} = 3.333\ m/s}$$

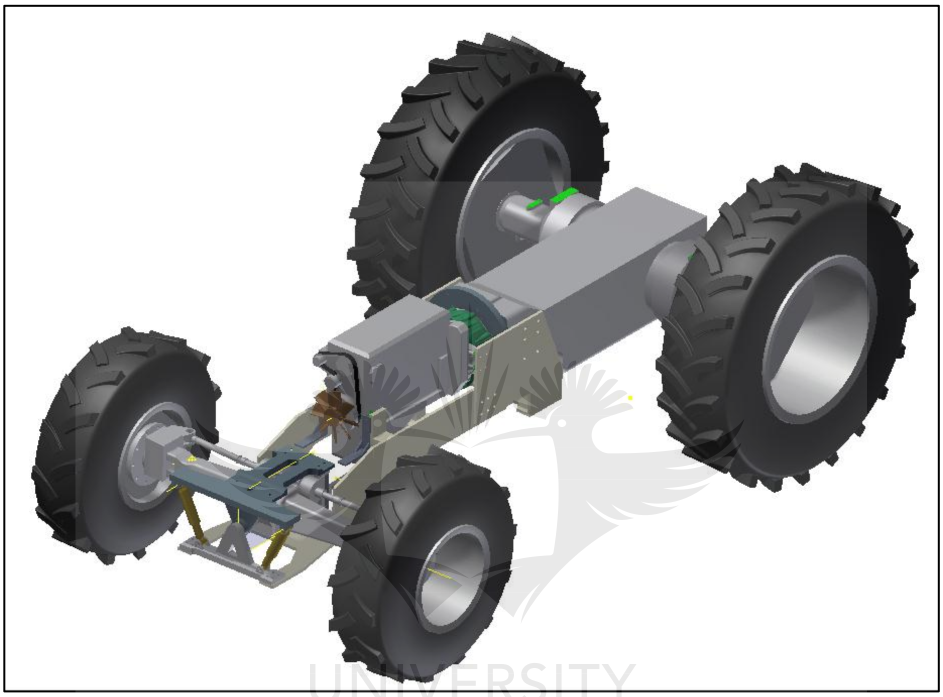


Figure 36: Low cost South African Tractor (C. Popa [10]).

10.4.3 Tractor drawbar power lost due to rolling resistance on level firm earth.

FW Stubbs [22] gives the rolling resistance on level firm earth as 29 kg/ton and 18 kg/ton for miscellaneous factors involved during the rolling process in the Handbook for heavy construction. Miscellaneous factors include tyre stiffness, soil tyre interaction, lateral earth pressure, etc.

Table 4 indicates different earth road conditions and their pulling forces in kg/ton required to overcome rolling resistance. As stipulated in 9.3.1 in the tractor test section, further work will only be done on firm earth conditions. The static weight of the machine will be towed over soft terrain to create a firm smooth roadway flexing slightly under load, thus causing the rolling resistance to be 29 kg/ton, plus the miscellaneous factor of 18 kg/ton.

Table 4: Rolling resitance roadway losses.

Rolling resistance in kg/ton for different level earth road conditions.	
Type of earth road way.	kg/ton
A hard smooth stabilized roadway without penetration under load.	18
A firm smooth rolling roadway flexing slightly under load.	29
A rutted dirt roadway flexing considerably under load.	45
A rutted dirt roadway, no stabilization, somewhat soft under travel.	67
A soft muddy rutted roadway or sand.	89 – 179

The tractor drawbar power needed to overcome its own rolling resistance on level firm earth is quantified below;

$$\boldsymbol{P_{tractor\ rolling\ loss}} \\ = \boldsymbol{rolling\ resistance \times g \times m \times v = (29 + 18) \times 6.1 \times 9.81 \times 3.333} \\ = \boldsymbol{9.374\ kW}$$

Where:

- $P_{tractor\ rolling\ loss}$ = *Tractor drawbar power due to tractor rolling resistance kW.*
- *Roll resistance 29 kg/ton including the miscellaneous factors of 18 kg/ton for firm earth.*
- *m = Tractor mass in ton.*
- *g = gravity, taken to be 9.81* m/s^2.
- *v = maximum tractor velocity allowed for compaction in m/s.*

10.4.4 Tractor drawbar power needed to overcome grade resistance on firm earth.

One of the official road construction design documents TMH 4 [23] indicates a maximum desirable grade of 8% for rural roads where a travel speed of 60 km/h is required.

The Type 6 rural road (District road) stipulated in 6.5.2 is designed for a speed of 60 km/h and thus indicates a maximum desirable grade of 8%.

FW Stubbs [22] indicates furthermore, that the resistance per 1% grade is 9 kg/ton for firm earth road conditions.

Table 5 gives an idea of the grade resistance vs kg/ton required as the grade increases from 1% to 8%.

Table 5: *Grade resistance losses for firm earth.*

Grade resistance in kg/ton for firm earth road conditions.	
% Grade	kg/ton
1%	9
2%	18
3%	27
4%	36
5%	45
6%	54
7%	63
8%	72

The tractor drawbar power needed to overcome its own grade resistance is quantified below. NB: It is important to understand that the grade resistance loss is only due to the elevated grade, no ground rolling resistance is added up the gradient. The final resistance up a gradient (the sum of rolling and grade resistance) will be quantified later in this document.

$$\boldsymbol{P_{tractor\ grade\ loss} = gradient\ resistance \times \max grade \times m \times g \times v}$$
$$= [\mathbf{9 \times 8 \times 6.1 \times 9.81}] \times \mathbf{3.333 = 14.362}\boldsymbol{kW}$$

Where:

- *$P_{tractor\ grade\ loss}$ = Tractor drawbar power due to tractor grade resistance kW.*
- *For each 1% grade, 9kg/ton of resistance is used [22].*
- *Maximum desirable grade for rural roads 60 km/h is 8% [23].*
- *m = Tractor mass in ton.*
- *g = gravity, taken to be 9.81 m/s^2.*
- *v = maximum tractor towing velocity allowed for compaction in m/s.*

10.4.5 Total tractor drawbar power needed to overcome rolling plus grade resistance on firm earth.

$$\boldsymbol{P_{tractor\ db} = P_{tractor\ rolling\ loss} + P_{tractor\ grade\ loss}}$$
$$= \mathbf{9.374 + 14.362 = 23.736}\ \boldsymbol{kW}$$

- $P_{tractor\ db}$ = Tractor draw bar power or pull required to move itself up a gradient (includes grade resistance and rolling resistance on level firm earth) kW.

In 10.3.2.2 the tractor drawbar power lost for the tractor alone without the towing of the machine is 24.6 kW. The 24.6 kW indicates a greater power loss than the 23.736 kW calculated by the method in [22]. Henceforth, the 24.6 kW losses for the tractor will be used in further calculations due to its higher power loss.

10.4.6 Compactor machine mass estimation.

The initial mass of the machine is estimated at about 5000 kg, the tractor already has a mass of 6100 kg (calculated 9.4.1) which will yield a gross mass for both the machine and tractor of 11100 kg. Appendix 15 – 20 the compactors have a gross mass between 10 ton and 11 ton, also bearing in mind that the engine, cab, cladding, back wheels, etc, add to the mass of these compactors and its weight at their drums is also between 5000 kg and 6000 kg.

10.4.7 Tractor drawbar power lost due to machines rolling and grade resistance.

The drawbar power at the tractor tow bar due the machine resistance can be calculated by means of the addition of rolling and grade resistance the machine will experience during operation.

10.4.7.1 Tractor power lost due to machine rolling resistance on level firm earth.

NB: It is important to understand that the grade resistance loss is only due to the elevated grade, no ground rolling resistance is added up the gradient. The final resistance up a gradient (the sum of rolling and grade resistance) is quantified later in this document.

$$\boldsymbol{P_{machine\ rolling\ loss}}$$
$$\boldsymbol{= Resistance\ on\ level\ firm\ earth \times g \times m \times v}$$
$$\boldsymbol{= (29 + 18) \times 5 \times 9.81 \times 3.333 = 7.684\ kW}$$

Where:

- $P_{machine\ rolling\ loss}$ *= Tractor drawbar power due to machine rolling resistance kW.*
- *Roll resistance 29 kg/ton including the miscellaneous factors of 18 kg/ton.*
- *m = Machine mass in ton.*

- *g = gravity, taken to be 9.81 m/s^2.*
- *v = maximum machine tow velocity allowed for compaction in m/s.*

10.4.7.2 Tractor drawbar power needed to overcome machine grade resistance.

$$\boldsymbol{P_{machine\ grade\ loss} = gradient\ resistance \times\ max\ grade \times m \times g \times v}$$
$$\boldsymbol{= [9 \times 8 \times 5 \times 9.81] \times 3.333 = 11.771\ kW}$$

Where:

- *$P_{machine\ grade\ loss}$ = Tractor drawbar power due to machine grade resistance kW.*
- *For each 1% grade, 9kg/ton of resistance is used [22].*
- *Maximum desirable grade for rural roads 60 km/h is 8% [23].*
- *m = Machine mass in ton.*
- *g = gravity, taken to be 9.81 m/s^2.*
- *v = maximum tractor tow velocity allowed for compaction in m/s.*

10.4.7.3 Tractor drawbar power due to machine grade and rolling resistance losses.

$$\boldsymbol{P_{tractor\ db\ machine} = P_{machine\ grade\ loss} + P_{machine\ rolling\ loss}}$$
$$\boldsymbol{= 11.7 + 7.684 = 19.455\ kW}$$

Where:

- $P_{tractor\ db\ machine}$= Tractor draw bar power or pull required to move the machine past its grade and rolling resistance kW.

10.4.7.4 Total tractor drawbar power required to overcome both the addition of machine and tractor losses on firm earth.

$$\boldsymbol{P_{total\ tractor\ db} = P_{tractor\ db} + P_{tractor\ db\ machine}}$$
$$\boldsymbol{= 24.6 + 19.455 = 44.055\ kW}$$

Where:

- *$P_{total\ tractor\ db}$ = Total tractor drawbar power or pull required to move both itself and machine past their rolling and incline resistances kW.*

The total power needed just to move both the tractor and the machine up an 8% gradient (the sum of rolling and grade resistance), without engaging the centrifugal weights through the

PTO or/and accelerate the machine, on an inclined surface, with firm soil conditions, is 44.055 kW.

Grade resistances from 1% to 8% are stipulated in Table 6.

Table 6: Total tractor draw bar power vs grade percentage.

Gradient (%)	Tractor drawbar power to overcome its own grade resistance (rolling + grade) (W)	Tractor drawbar power to overcome machine grade resistance (rolling + grade) (W)	Total Tractor drawbar power to overcome its own + machine grade resistance (rolling + grade) (W)
1	11169.05	9155.35	20324.4
2	12964.10	10626.71	23590.81
3	14759.15	12098.059	26857.21
4	16554.20	13569.41	30123.61
5	18349.25	15040.76	33390.02
6	20144.30	16512.12	36656.42
7	21939.35	17983.47	39922.82
8	24600	19455	44050

10.5 Available PTO power.

The available PTO power is the usable power left for machine operation after the total tractor drawbar power needed to move the machine and tractor up a gradient of 8% has been deducted from the 120 kW at the PTO. This available PTO power that is quantified below will be used to operate the machine power transmission components during towing and for acceleration.

$$\boldsymbol{P_{PTO\,(available)} = P_{PTO\ out} - P_{total\ tractor\ db}}$$

$$\boldsymbol{= 120 - 47.947 \quad = 75.945\ kW}$$

The power available to accelerate the power transmission equipment and to accelerate the machine is 75.945 kW.

10.6 Design of the machine mechanical power transmission system.

Figure 37 shows the ideal mechanical power transmission system concept as a whole. Note that in Figure 37, the drive shaft and the PTO is not shown.

The arrangement in Figure 37 consist of the following; a bevel gear system, two flexible Fenner tyre couplings and two similar friction belt systems connected to the eccentric weight shaft. This drive arrangement would be the ideal drive if there was a 1:1 ratio mitred bevelled gearbox at an input power rating of 100 kW and above, readily available on the market.

Unfortunately such a 1:1 ratio mitred gearbox at 100 kW+ does not exist readily available on the market. Also over the 100 kW, design power must be considered thus inflating the selected gearbox power it can handle.

From all the consultations that were done with the main gearbox suppliers in 9.1 and 9.2 telephonically and by view of their respective catalogue, it was noted that all industrial gearboxes required for this application in the required power range are generally step down gearboxes with the lowest ratios of about 6.3:1 to 7:1. Appendix 8 shows the lowest gear ratios as 6.3:1 for the following three suppliers (Bonfiglioli, Redicon and Hansen). Only three suppliers are used to prove the ratio point, there are other gearbox catalogues that suggest this as well not shown in Appendix 8.

This large step down gearbox ratios generated huge engineering challenges to be overcome, just to get the system to work with readily available over the counter parts that are obtainable locally within South Africa. Stepping up the speed was the main problem and caused step up options to be investigated in-depth in the chain and belt drive Fenner catalogues.

10.6.1 The ideal mechanical drive system concept.

Due to the unavailability of a low ratio gearbox in the required power range (120 kW), the ideal mechanical advantage power drive shown in Figure 37 is not possible. One of the solutions is to design a bevel gearbox with the desired ratio for comfortable step-up ability since the maximum step up ratio for belts are 5:1. Such a gear system was designed with the Autodesk Inventor 2013 Design Accelerator software in Appendix 5 and might not be economical when it comes to the design of a complete new gearbox with seals, tolerances, gaskets, width clearances, bearings, housing, shafting, oil pressure, sensors, etc to achieve the desired goal. Nevertheless, the gear system was designed, because such an option can be investigated in a new design task.

The most feasible option at the moment is to see if a successful power transmission design can be attained by means of readily available over the counter technology.

Figure 37: Ideal mechanical advantaged system.

10.6.2 Machine drive power losses.

As in the design of mechanical components by companies with extensive experience in power transmission, there are various losses between each component and assembly. The summing up of the losses by Womac Engineering sources (Appendix 2) are as follows:

- Efficiency per gear set is normally about 98 %;
- Tyre Coupling efficiency = 99%;
- Belt drive efficiency = 96%;
- Chain drive efficiency = 98%;
- And the Universal joint Efficiency = 98.5% (Estimated see explanation below).

The drive shaft has two driving mechanisms, the front universal joint and the rear universal joint. Assuming each joint act like a pair of mating gears, the efficiency for both joints will be higher than the mating gears. Mainly, because universal joints don’t have to mesh and slide like gears do to transfer power, it’s more like direct power transfer through its pins, this is why 98.5 % is assumed.

10.6.3 Machine maximum power available at the PTO output shaft on a level firm earth road.

To calculate the actual power that can be applied at the machine MS to rotate the eccentric weights, the maximum power available at the PTO must be considered while the machine and tractor is traveling on a straight road and not on a gradient.

When the machine is traveling on a straight road, more power can be applied through the power transmission mechanism when no grade resistance loss occurs at that moment. Thus the total grade resistance for both the tractor and machine must be added to the 75.945 kW available PTO power.

The key of catalogue selection is the inflation of the actual power being transmitted into the design power. At a higher available PTO power, the drive system will have the ability to handle torque overloading more efficiently.

$$\boldsymbol{P_{Max\ available\ \ PTO\ power}}$$
$$= \boldsymbol{P_{pto\ availabe}} + \boldsymbol{P_{loss\ machine\ grade\ resitance}} + \boldsymbol{P_{loss\ tractor\ grade\ resitance}}$$
$$= \mathbf{75.945 + 11.771 + 14.362 = 102.078}\ \boldsymbol{kW}$$

The 102.078 kW power is the actual power that will be used for catalogue component design. Safety or service factors will be employed that will ultimately increase the transmittable power to manufacturer specifications and by this the design power is obtained.

10.7 Drive shaft selection and calculations.

Drive Shaft Technical Literature*: Appendix 6*

The design of the drive shaft was done considering overseas / national available drive shafts and modified to suit the current project conditions. The shaft can be fabricated locally using Hardy and Spicer parts by Prop shaft World (Pty) Ltd and various other drive shaft manufacturers. Any drive shaft that can supply a power of 115 kW to 275 kW (Appendix 48: E-mail from the manufacturer) is acceptable, provided that the ends of the drive shaft will fit onto the PTO output shaft and onto the gearbox input shaft.

NB: All calculations were done according to the strict standards of the Spicer off highway driveshaft standard product catalogue. See Appendix 6.

For drive shaft installation, Appendix 7 should be consulted.

10.7.1 Determination of the equivalent torque for agricultural application.

$\boldsymbol{Te} = \boldsymbol{k}_{\boldsymbol{a}} \times \boldsymbol{k}_{\boldsymbol{l}} \times \boldsymbol{T}_{\boldsymbol{n}}$ *((catalogue) Page 10)*

Where:

- T_e = *Equivalent Torque Nm,*
- K_a =*Angularity factor,*
- K_l = *Life requirement factor,*
- T_n = *Nominal Torque (input torque to the drive shaft) Nm.*

In Appendix 3 the highest PTO speed (1437 r/min) is used as the operating speed for the PTO. This is to ensure that the step-up of the speed after the gearbox has stepped it down, is less difficult to step up to the final input speed (between 2100 – 2200 r/min) of the MS at the vibrating drum.

$$\boldsymbol{T_n = \frac{30 \times P}{\pi \times N} = \frac{30 \times 102078}{\pi \times 1437} = 678.339\, Nm}$$

Where:

- N = *PTO output speed r/min*
- P = *total available PTO power W*

10.7.2 Determination of the drive shaft operation angle.

Figure 38 shows the idea behind the calculation of the drive shaft operating angle, considering the tractor back tyre size relative to the vibrating drum diameter and the drive shaft length.

NB: This applies only to the vertical angle. Figure 38 does not represent the actual machine, but merely the concept for calculation and illustration purposes.

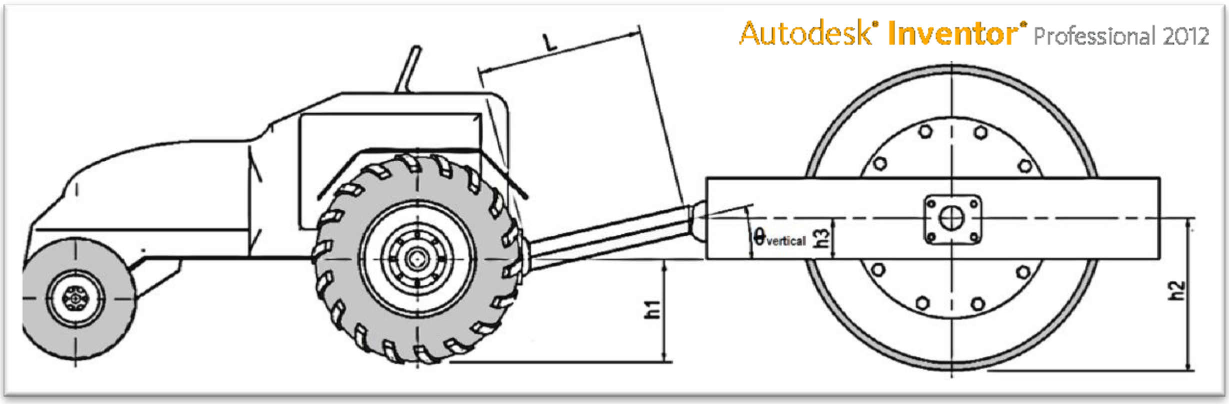


Figure 38: Drive shaft angle.

Referring to Figure 38 above, the following need to be noted;

- *L = Drive shaft length mm.*
- $\theta_{vertical}$ *= Drive shaft operation angle as seen from the side view in degrees (catalogue) Page 13.*
- *h1 = Tractor wheel radius mm (Appendix 3).*
- *h2 = Compacting drum radius plus the amplitude mm.(Appendix 21).*
- *h3 = Vertical height at which the drive shaft must be installed, from the tractor to the machine mm.*

In the ZF T-7000 tractor transaxle data sheet (Appendix 3), the suggested tractor rear tyre size is 42 inches for the T-7232 transmission.

$$\boldsymbol{h_1 = \frac{42 \times 25.4}{2} = 533.4\ mm}$$

The specification table (Appendix 21) is a summarization of all compactor catalogues specifications and has all the relevant data for easy reference.

From the specification table in Appendix 21, the drum diameter for 10 ton compactors, ranges from 1200 mm to 1530 mm. A drum diameter of 1290 mm is assumed for this machine. The drum is going to operate at amplitude during compaction, thus lifting the drum from the soil, causing the vertical angle ($\theta_{vertical}$) operating drive shaft to increase.

Maximum amplitude ranges from 1.7 mm to 2 mm (Appendix 23 Spec. Table) for compacting drums ranging from 1219 mm to 1295 mm. A maximum amplitude of 2 mm was assumed for the machine.

NB: The final machine amplitude may vary and may be less than the 2 mm maximum value dependant on the final vibrating drum mass, eccentric weight mass, eccentric radius, speed of operation, frame mass and all power transmission equipment masses.

$$h_2 = drum\ radius + \max amplitude = \frac{1290}{2} + 2 = 647\ mm$$

$$h_3 = h_2 - h_1 = 647 - 533.4 = 113.6\ mm$$

Since the tractor linkages and drawbar itself is not part of this design project, there is no clear indication as to how long the drive shaft will be, this is why all possible lengths, relative to their vertical drive shaft angles are calculated in the Table 7 below. The shortest length of 1524 mm is calculated first ((catalogue) Page 13).

$$\theta_{vertical} = sin^{-1}\left(\frac{h3}{L}\right) = sin^{-1}\left(\frac{113.6}{1524}\right) = 4.275°$$

For all other available drive shaft lengths, the drive shaft operation angles are as shown in Table 7.

Table 7: Drive shaft length vs operating angle.

Maximum length L (mm)	*Drive shaft operation angle $\theta_{vertical}$ (degrees)*
1524	4.275º
1651	3.945º
1778	3.663º
1905	3.419º
2032	3.205º

10.7.3 Anglularity Factor selection.

Refering to Figure 39, the angularity factor k_a can be read for each calculated drive shaft angle in 9.7.2. (catalogue) Page 3

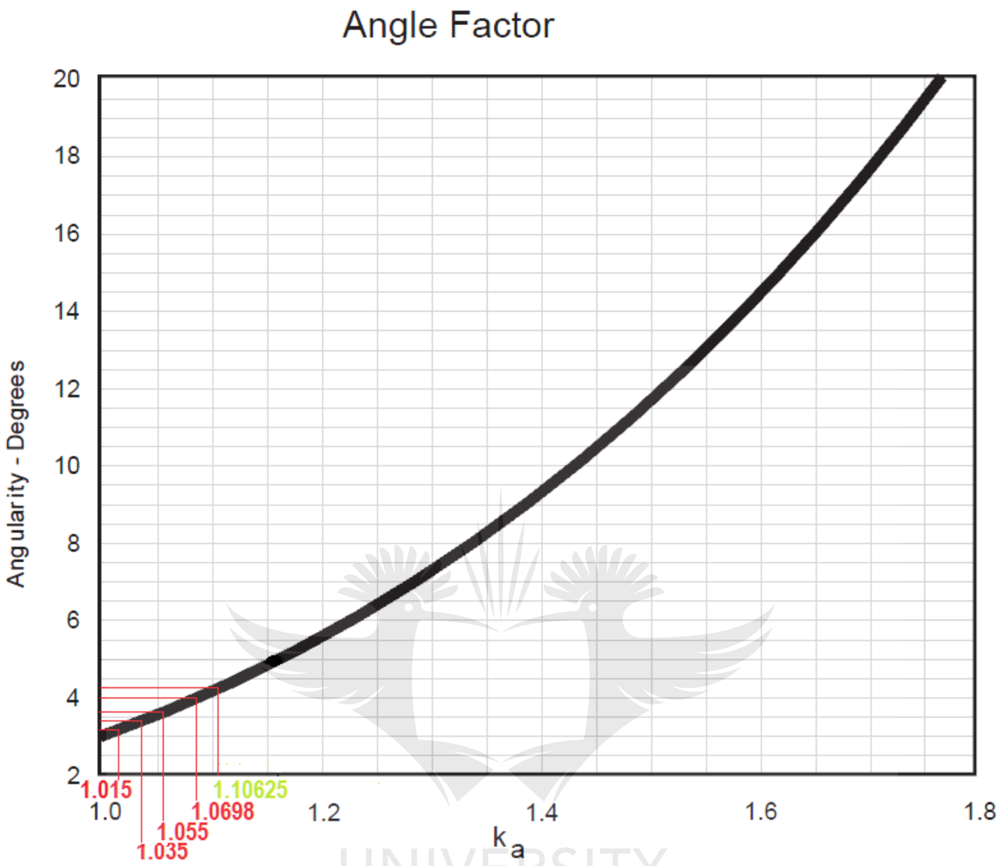


Figure 39: Operation angle vs angularity factor.

The selected angularity factors for all possible drive shaft operating angles are calculated in Table 8 below:

Table 8: Drive shaft operating angle vs angularity factor.

Drive shaft operation angle $\theta_{vertical}$ (degrees)	*Angularity Factor k_a*
4.275º	1.10625
3.945º	1.0698
3.663º	1.055
3.419º	1.035
3.205º	1.015

10.7.4 Life Requirement Factor selection.

Figure 40 show the drive shaft life requirements in hours vs. the Life Factor. The SKF bearing catalogue on (Appendix 33) indicates that the life expectancy for construction machinery should be between 3000 hrs and 8000 hrs. By selecting the maximum of 8000 hrs just for an initial assumption, an appropriate selection can be made for the Life Factor.

It is noted that the 8000 hrs refer to bearing life, however, it is a reliable figure to use for the Life Factor selection.

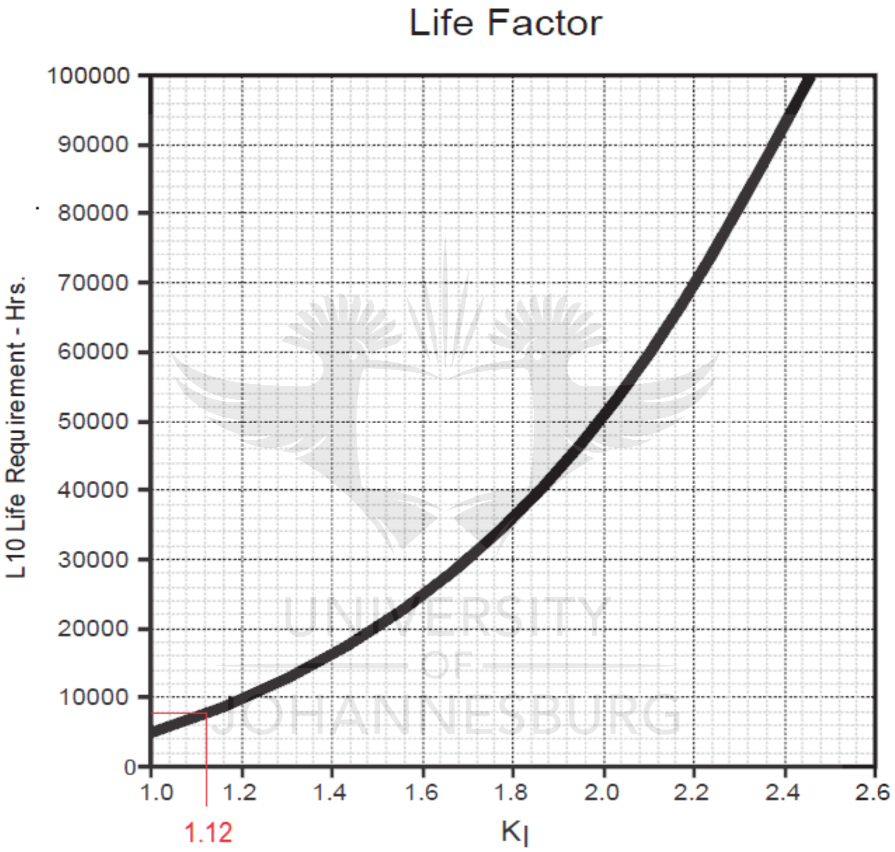


Figure 40: L10 life vs life factor.

From the Figure 40 the K_l Life Factor is 1.12 for 8000 hrs.

The equivalent torque can be calculated. The application is specifically for agricultural machines. (catalogue) Page 9

$$\boldsymbol{T_e(Equivalent\ torque) = k_a \times k_l \times T_n = 1.10625 \times 1.12 \times 678.339}$$
$$\boldsymbol{= 840.462\ Nm}$$

The equivalent torques for all possible Angularity Factors are seen in Table 9.

Table 9: Angularity factor table.

Angularity Factor k_a	*Equivalent Torque T_e Nm.*
1.10625	840.462
1.0698	812.77
1.055	801.525
1.035	786.331
1.015	771.136

10.7.5 Selection of drive shaft series.

The equivalent torque and rotational speed is utilised in selecting the proper drive shaft series. In the (catalogue) Page 5, 6 & 7 three different Charts (Figure 41, Figure 42, and Figure 43) are present to select the drive shaft series that may be appropriate for this design.

10.7.5.1 Series 10 drive shaft selection

The lines in Figure 41 represents the 1437 r/min drive shaft speed vs the equivalent torque rating. The two lines meet in the 10 Series 1550 space that indicates it to be the preliminary selected drive shaft series.

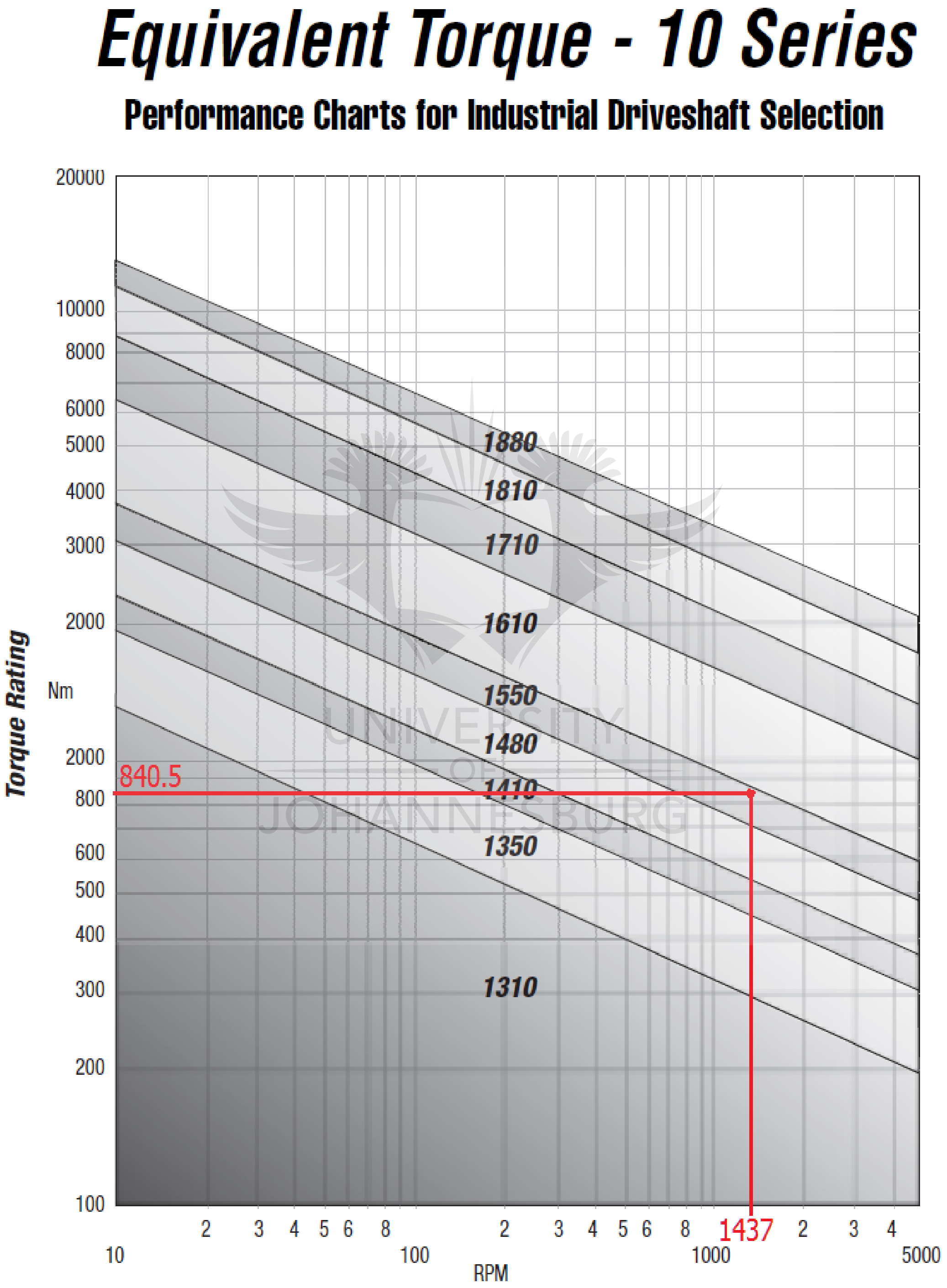


Figure 41: 10 Series 1550 drive shaft series selection.

10.7.5.2 Wing bearing series drive shaft selection.

The same way as with Figure 41 in 10.7.5.1, for the Wing Bearing Series, the 5C model is selected. See Figure 42.

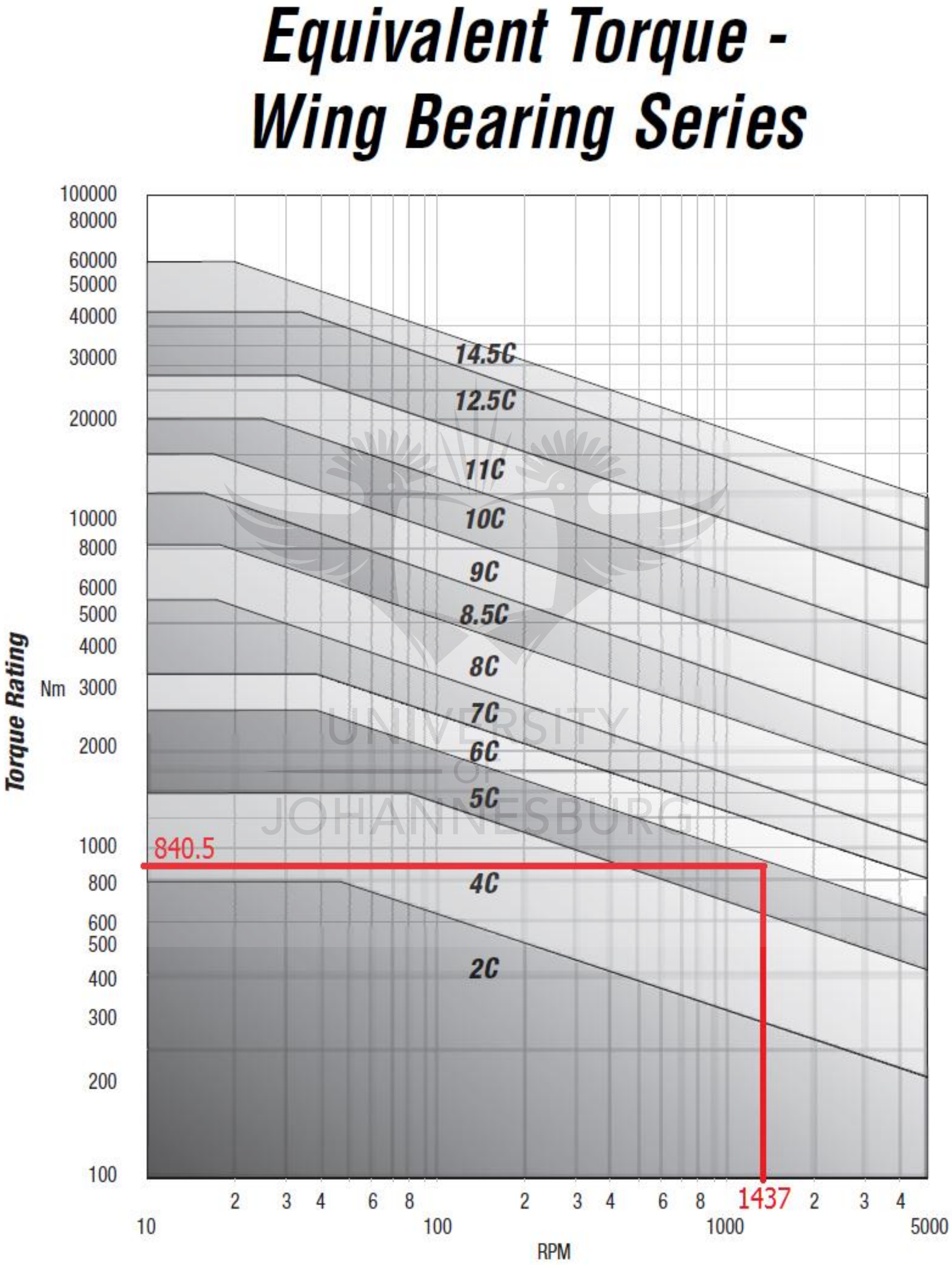


Figure 42: Wing bearing series drive shaft selection.

10.7.5.3 SPL 70 Life Series drive shaft selection.

In Figure 43 the Spicer Life Series (SPL 70) model is selected at similar coordinates (1437 r/min ; 840.5 Nm) as the previous two selections.

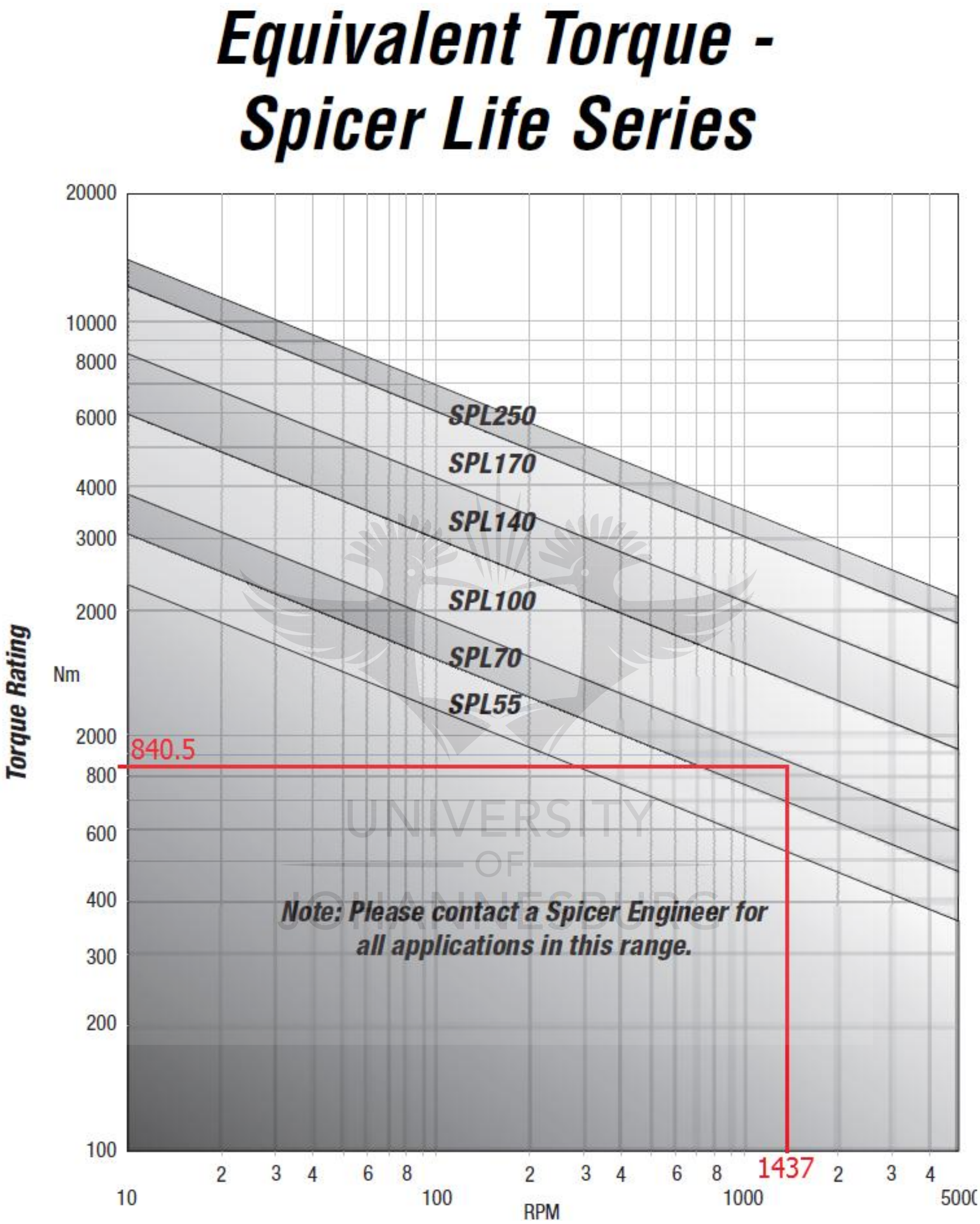


Figure 43: Spicer life series drive shaft equivalent torque.

All of the selected drive shaft models such as the series 10 (1550), the wing bearing series (5C) and the Spicer Life Series (SPL 70), has been selected using the equivalent torque values of 840.462 Nm, rounded off to 840.5 Nm and 1437 r/min rotational speed.

10.7.6 Selection of industrial Torque ratings.

Consult the tables in Appendix 6 for the following details;

- For the series 10 (1550) model the industrial torque rating is given as 5050 Nm.
- The Wing Bearing Series (5C) has an industrial torque rating of 2650 Nm.
- The industrial torque for the Spicer Life Series (SPL 70) is similar to that of the 10 series (1550) at 5050 Nm.

A service factor for drive shaft is selected to accommodate for extreme shock loads and it is given as $k_{sf} = 6$ (vibration conveyers).

NB_1: In order to make a successful selection, one must first check if the industrial torque is greater than the product of the Application Service Factor k_{sf} and the nominal torque T_n calculated as 678.339 Nm in 10.7.4.

NB_2: The industrial torque must be greater than the calculated value of T_n x k_{sf} for design success. The industrial torque comparison for the 10 Series (1550) model is given below;

$$\boldsymbol{T_{ind} > T_n \times k_{sf} \quad \therefore\ 5050 > 678.339 \times 6\ \therefore\ 5050 > 4070}$$

The industrial torque is greater than the product of the service factor and nominal torque and the above calculation applies for all series that will be evaluated.

Evaluation for all other drive shaft series is shown in Table 10.

Table 10: Drive shaft series design evaluation.

Type of Series	*Model Selected*	*Industrial Torque* $T_{ind\ NM}$	*Maximum calculated.* $T_n \times k_{sf}$	*Design Success*
10 Series	1550	5050	4070	Yes
Wing Bearing Series	5C	2650	4070	No
	6C	3400	4070	No
	7C	5700	4070	Yes
Spicer Life series	SPL 70	5050	4070	Yes

Table 10 has indicated that the Wing Bearing Series (5C) type does not meet the design requirements. The calculated $\mathbf{T_n \times K_{sf}}$ value is greater than the rated industrial torque. The next model higher up is the 6C model and it also fails at a rated industrial torque of 3400 Nm.

The final selection is the 7C model at 5700 Nm of industrial torque and it satisfies the design requirements for the Wing Bearing Series.

10.7.7 Drive shaft power verification with the T-7232 transmission PTO output shaft requirements.

Consultants from ZF Services South Africa (Pty) Ltd were consulted regarding all relevant unknown specifications not featured in the T-7000 catalogue (Appendix 6). Please refer to Appendix 6 for the attached email on the suggested power for the drive shaft and other requirements from the manufacturer of the T-7232 off-highway transmission.

The power that the drive shafts can operate at is tabulated in Table 11.

Table 11: Drive shaft power.

Type of Series	*Model Selected*	*Power Kw*
10 Series	1550	125
Wing Bearing Series	5C	145
	7C	220
Spicer Life series	SPL 70	125

As stipulated by the Technical Representative for Off-Road Vehicles of the ZF group, the ISO 500 recommends that the drive shaft should be in the power range of 115 kW to 275 kW. The selected drive shafts are well within specification in Table 11 which ranges from 125 kW to 220 kW.

10.7.8 Approximate drive shaft service life.

Referring again to Appendix 6, the following calculation for drive shaft service life was quantified.

$$\boldsymbol{B_{10} = \frac{1.5 \times 10^6}{N \times \theta} \times \left(\frac{T_d}{T_n}\right)^{\frac{10}{3}}}$$

Where:

- *B_{10} = Service Life – hrs*
- *T_d = Universal Joint Bearing Capacity – Nm*
- *T_n = Driveshaft Torque – Nm*
- *N = Driveshaft Speed - r/min*
- *θ = Universal Joint Angularity – Degrees*

According to the catalogue, the total universal joint angle consists of the vertical operation angle and can be seen in the side view of Figure 38. From the plan view shown in Figure 41, the horizontal operation angle can be seen as 0 ˚ degrees.

Since the angle ($\theta_{horizontal}$) in the plan view is zero, the total universal joint angle will be the vertical angle only.

$$B_{10} = \frac{1.5 \times 10^{6}}{1437 \times 4.275} \times \left(\frac{1900}{678.339}\right)^{\frac{10}{3}} = 7563.44\ hrs$$

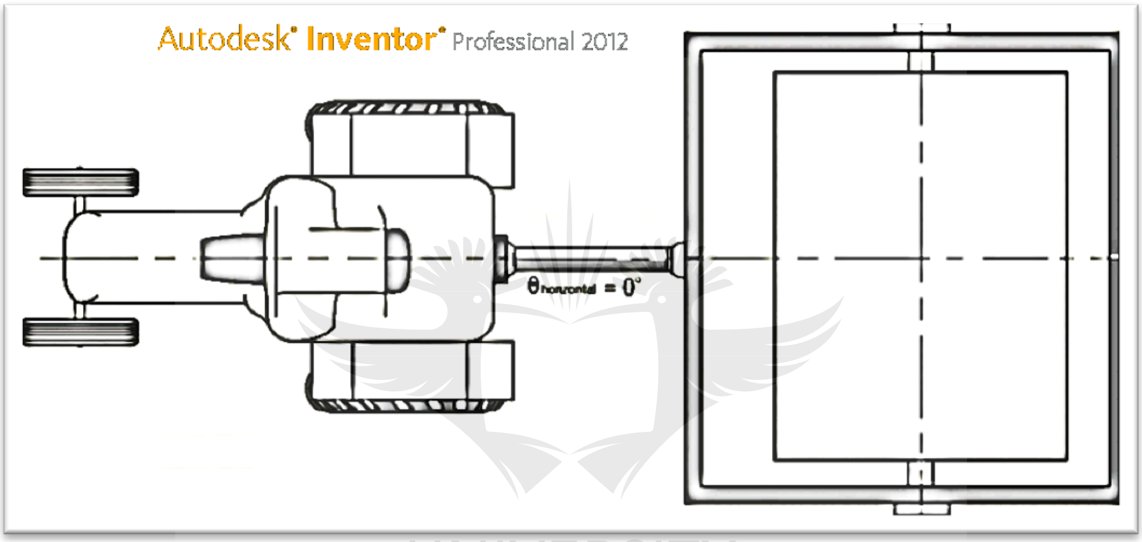


Figure 44: Horizontal drive shaft operating angle.

NB: T_d the bearing capacity in Appendix 6 is similar for the 10 series (1550) and Spicer Life (SPL70) series = 1900 Nm and the Wing Bearing Series 7C, T_d = 3400 Nm

Table 12 shows the B_{10} life in hours, for all driveshaft types with their respective installation angles in degrees.

The estimated life of 8000 hours from the SKF Bearing catalogue (Appendix 33), for the 10 Series and SPL70 drive shaft types is very close to the lowest service life of 7563.44 hrs in Table 12. Since the Wing Bearing Series (7C) has a tremendous amount of life in hours, which is too high for this application, it will be ignored from hence forward. This will amount in unnecessary extra costs, but will give excellent durability.

Table 12: Drive shaft service life vs operating angle.

Series & Model	*B_{10} Service Life hrs*	*Drive shaft operation angle $\theta_{vertical}$ (degrees)*
10 Series 1550 & SPL70	7563.44	4.275º
	8196.12	3.945º
	8827.12	3.663º
	9457.10	3.419º
	10088.53	3.205º
Wing Bearing (7C)	52618.26	4.275º
	57019.78	3.945º
	61409.52	3.663º
	65792.06	3.419º
	70185.04	3.205º

10.7.9 Drive shaft length limitations.

Safe rotational speed

$$\boldsymbol{N_{max}} = \frac{\boldsymbol{68140000\sqrt{D^2 + d^2}}}{\boldsymbol{L^2}} = \frac{\boldsymbol{68140000\sqrt{76^2 + 66^2}}}{\boldsymbol{1524^2}} = \boldsymbol{2953.11, rpm}$$

Where:

- *N_{max} = Safe rotational speed - r/min*
- *L = Drive Shaft Length – mm*
- *D = Tube outer diameter – mm*
- *d = Tube inner diameter – mm (Estimated at a wall thickness of 5 mm)*

The safe rotational speed N_a for all other drive shaft lengths is indicated in Table 13. The table also summarises the main calculated values thus far concerning the selected drive shafts series.

Table 13: *Drive shaft model specification summary.*

10 Series (1550) and Spicer Life Series(SPL70) shafts.					
Shaft operation angle $\theta_{vertical}$ (degrees)	*B10 Service Life hrs*	*Driveshaft Lenth mm*	*Critical shaft Speed N_{max} r/min*	*D - Outer tube diameter mm*	*d - inner tube diameter mm*
4.275º	7563.44	1524	2953.11	76	66
3.945º	8196.12	1651	2516.26	89	-
3.663º	8827.12	1778	2169.63	101	-
3.419º	9457.10	1905	1881.00	114	-
3.205º	10088.53	2032	1661.12	127	-

The catalogue stipulates that the safe maximum operating speed is 5000 r/min for the 10 series (1550), as well as the Spicer Life Series (SPL 70) indicating a design success as the safe speed falls far below the 5000 r/min.

10.7.10 Checking for shaft alignment limitations.

From the (catalogue) Page 13, the resultant output angular acceleration has to be determined to compare with a constant of 300 rad/s^2.

$$\boldsymbol{\alpha = (3.34 \times 10^{-6}) \times N^2(\theta_{vertical} - \theta_{horizontal})}$$

Where:

- *α = Resultant output angular acceleration – rad/sec^2 (calculated)*
- *$\theta_{vertical}$ = Input universal joint angularity in the vertical plane - degrees.*
- *$\theta_{horizontal}$ = Centre universal joint angularity in the plan view - degrees.*
- *N = Drive shaft rotational speed r/min.*

$$\boldsymbol{\alpha = (3.34 \times 10^{-6}) \times 1437^2(4.2750) = 29.7\ rad/s^2}$$

29.7 rad/s^2 < 300 rad/sec^2, suggests that all lower angles in Table 14 are within specification, because all angular acceleration values are lower than the 29.7rad/s^2 value.
For all other acceleration values, their respective drive shaft operating angles are shown.

Table 14: Drive shaft operating angle vs angular acceleration.

Drive shaft operation angle $\theta_{vertical}$ (degrees)	*Angular acceleration rad/s^2*	*Alignment Design Check*
4.275º	29.5	OK
3.945º	27.21	OK
3.663º	25.26	OK
3.419º	23.58	OK
3.205º	22.1	OK

10.7.11 Drive Shaft Inertia calculations.

The inertia of the shaft calculation are mainly done to help calculate the torque required to accelerate the drive shaft. Actual angular acceleration can be quantified later in the design when all power transmission equipment and shafting has been designed. The power left over after all losses in the system has been identified and quantified, can be used to calculate the acceleration of the drive system as a whole. See Appendix 6.

$$I_{driveshaft} = I_{tube} + I_{componant\ masses}$$

Where:

- $I_{driveshaft}$ = *Total mass moment of inertia for the arrangement – kg. cm^2*
- I_{tube} = *Tube moment of inertia in kg.cm^2/100mm of shaft – kg.cm^2*
- $I_{component\ mass}$ = *Moment of inertia for the shaft and components – kg.cm^2*

10.7.11.1 Inertia calculations for the 10 Series (model 1550).

$$I_{tube} = 9.64 \times 15.24 = 147.01\ kg.cm^2$$

$$I_{driveshaft} = 147.01 + 256 = 403.01\ kg.cm^2$$

The drive shaft inertia for all other 10 Series drive shaft lengths (Table 15).

Table 15: 10 Series drive shaft inertia.

10 Series (1550 Model)			
Drive Shaft Length mm	*I_{tube} kg.cm^2*	*$I_{component\ mass}$ kg.cm^2*	*$I_{driveshaft}$ kg.cm^2*
1524	146.91	256	402.91
1651	159.16	256	415.1
1778	171.4	256	427.4
1905	183.64	256	439.64
2032	195.88	256	451.88

10.7.11.2 Inertia calculations for the Spicer Life Series (model SPL70).

$$\boldsymbol{I_{tube} = 12.77 \times 15.24 = \ 194.61\ kg.cm^2}$$

$$\boldsymbol{I_{driveshaft} = 194.61 + 256 = 450.61\ kg.cm^2}$$

Drive shaft inertia for all other Spice Life Series (SPL70) drive shaft lengths (Table 16):

Table 16: Spicer life series drive shaft inertia.

Spice Life Series (SPL70 Model)			
Drive Shaft Length mm	*I_{tube} kg.cm^2*	*$I_{component\ mass}$ kg.cm^2*	*$I_{driveshaft}$ kg.cm^2*
1524	146.91	256	402.91
1651	159.16	256	415.1
1778	171.4	256	427.4
1905	183.64	256	439.64
2032	195.88	256	451.88

10. 7.12 Drive shaft specifications.

The final drive shaft specification is:

- Series = SPL 70 or Series 10
- Power = 125 kW
- Length = 1651 mm
- Operating angle = 3.945^o
- Operating Life = 8196.12 hrs

All other operating angles and drive shaft lengths quantified and tabulated are appropriate to use. However, the minimum power needed for the PTO output shaft is 120 kW as stipulated by the transmission / PTO supplier ZF (Appendix 6, Email section).

The calculated life of 8196.12 hrs is consistent with that of construction equipment of 3000 hrs to 8000 hrs (Appendix 33).

NB: It is important to understand that the length, operating angle and the life in hours are all linked and may change during installation, if a longer or shorter drive shaft is required, but all drive shaft specifications will still stay well within range.

10.8 PRELIMINARY GEARBOX SELECTION.

Gearbox Technical Literature*: Appendix 8*

The entire major gearbox companies within South Africa in 9.2 couldn't supply a 1:1 ratio gearbox neither a step-up gearbox in the required power range of plus minus 100 kW as previously explained.

Only step down gearboxes are available in the required power specification of 99.039 kW and the lowest step-down ratios for these gearboxes are between 6.3:1 and 7:1. In fact, the lowest speed ratio possible for this application in the Bonfiglioli catalogue is 10.9 due to the power rating suggested by the supplier of which it brought the speed right down from 1437 r/min to 131.93 r/min. This caused great difficulty selecting a chain or belt drive for the design due to the huge gap of stepping up the speed from 131.93 r/min to about 2200 r/min.

Design sections for the gearbox selection further on include:

- the gear box input power in kW (Obtained from drive shaft output power after losses),
- the gearbox design power, also known as the minimum rated power P_{n1} - kW,
- the gearbox selection based on the rated power P_{n1} - kW,
- the tabulated justification of the selected gearbox model,
- the tabulation of other suitable models according to their rated power .

NB: All calculations and selections are justified in Appendix 8.

10.8.1 Gearbox input power calculation.

The input power for the gearbox after the driveshaft losses is calculated below.

$$\boldsymbol{P_{in\ gearbox} = P_{Max\ available\ \ PTO\ power} \times \eta_{universal\ joints} = 102.078 \times .985 \times .985}$$
$$\boldsymbol{= 99.039\ kW}$$

Where :

- ✓ $\boldsymbol{P_{in\ gearbox} = P_{r1}}$*, this is the gearbox input shaft power kW (catalogue) Page 1.*
- ✓ *P = Maximum available PTO power output shaft, already calculated in 10.6.3 kW.*
- ✓ *N = Speed from the PTO through drive shaft into the gearbox r/min.*
- ✓ $\boldsymbol{\eta_{universal\ joints}}$ *=Universal joints efficiency.*

10.8.2 Gearbox minimum rated power (P_{n1}) quantification.

(catalogue) Page 13, $\boldsymbol{P_{n1} \geq P_{r1} \times f_s \times f_m = 99.039 \times 2 \times 1.25 = 247.6\ kW}$

The gearbox rated power that needs to be selected must not be less than the above calculated design power of *247.6* kW.

Where:

- ✓ *(catalogue) Page 7 vibration service factor (f_s) = 2 (10 hours and under)*
- ✓ *(catalogue) Page 12 adjusting factor (f_m) = 1.25 (Multi – cylinder internal combustion engine)*
- ✓ *P_{n1} = Selected gearbox rated power kW.*

10.8.3 Gearbox selection based on the calculated rated power P_{n1}.

In the (catalogue) Page 51, the following selections were made based on the calculated design power rating of 247.6 kW:

- ❖ HDO 100 2 gearbox model (Selected at the closest power rating 257 kW up to 247.6 kW);
- ❖ 1750 r/min maximum input speed (1437 r/min is within the selected maximum speed the gearbox can handle and therefore, OK);
- ❖ selected power P_{n1} = 257 kW (the calculated design power of 247.6 kW is less than the selected and is therefore, OK) and
- ❖ the speed reduction ratio of 10.9 (Constraint for the 257 kW rated power model).

10.8.4 Selected HDO 100 2 (257 kW power rating) justification.

The output speed for the gearbox is very low, because of its 10.9 step down ratio. The lowest ratio possible that meet the power requirements was key in this selection and 10.9 was the lowest relative to the needed power of 247.6 kW and most economical. See Table 17.

$$\boldsymbol{N_{gearbox\ out} = \frac{1437}{10.9} = 131.93\ rpm}$$

For all other gearbox models:

Table 17: HDO 100 2 (257 kW) gearbox selection justification.

HDO 100 2 gearbox models.					
Gearbox Model	*P_{n1}(Rated Power) kW*	*Speed ratio*	*Actual gearbox output speed r/min.*	*Maximum suggested output speed. r/min*	*Selected model*
HDO 100 2	433	6.5	221.08	271	no
HDO 100 2	412	7	205.29	249	no
HDO 100 2	333	8	179.63	219	no
HDO 100 2	317	8.7	165.172	201	no
HDO 100 2	269	10	143.7	175	no
HDO 100 2	257	10.9	131.93	161	yes
HDO 100 2	222	12.4	115.89	141	no
HDO 100 2	212	13.5	106.44	130	no

The HDO 100 2 gearbox models in Table 17 are a summarised version of the actual table in Appendix 3 within the catalogue. The HDO 100 2 gearbox models lower down from the 257 kW selected power, has lower power ratings than the required bench mark 247.6 kW.

Higher up the table, the rated power becomes more than required and the ratio becomes less and more appropriate to help step-up the drive speed more effectively, but it is not economical, because of the over designed gearbox power.

The HDO 100 2 model with a rated power of 257 kW is the most economical and efficient choice amongst all tabulated models in Table 16.

10.8.5 Other Suitable HDO gearbox models.

For all other suitable HDO gearbox models higher up from the selected HDO 100 2 base model, Table 18 can be consulted which represents a summarized view of all possible HDO gearboxes close to the bench mark power rating, but much more expensive, heavier and too high step down ratios.

Table 18: Suitable gearbox models above 247.6 kW.

Suitable power and speed specifications of different gearbox models above 247.6 kW.				
Gearbox Model	*P_{n1}(Rated Power) kW*	*Speed ratio*	*Output Speed (N2) r/min*	*Reference*
HDO 100 2	257	10.9	131.93	Page 51
HDO 110 2	263	15.5	92.71	Page 58,59
HDO 120 3	255	21.8	65.92	Page 67
HDO 130 3	262	43.8	32.81	Page 74,75
HDO 140 3	256	55.8	25.75	Page 82

The rated power indicated for each HDO model in Table 18 is the most economical choice in their respective ranges and the closest to the 247.6 kW bench mark. Since the HDO 100 2 model has the higher output speed of 131.93 r/min, the speed step-up process will be easier and thus nullifies the other models with their low output speeds.

10.9 Drive evaluation & choice.

In the following subchapters, several options were calculated and evaluated to try to obtain the best possible solution. Due to one of major constrains for this design i.e. low cost, all attempts were made to use off-the shelf products.

Three possible types of drive systems can be used to step up the speed from the output shaft of the gearbox to the required final speed of 2100 r/min – 2200 r/min at the MS. The first is a chain drive; the second a friction wedge belt drive and lastly a gear transmission.

10.9.1 Gear drive evaluation.

Figure 45 below is a proportional 3D drawing which represents how the gear system will fit from the gearbox shaft to the compacting drum shaft. Due to the large step-up ratio needed, the gear attached to the gearbox shaft will be larger than the compacting drum itself and is therefore not practical.

Three gears in series will also not work, mainly, because the first gear will have the same size as the large gear connected to the gearbox shaft in Figure 45.

Using an epicyclical gearbox on the output gearbox shaft at this high power rating, will be extremely costly. The mitre gearbox already gives an idea of what power rating will be required for the epicyclical gearbox.

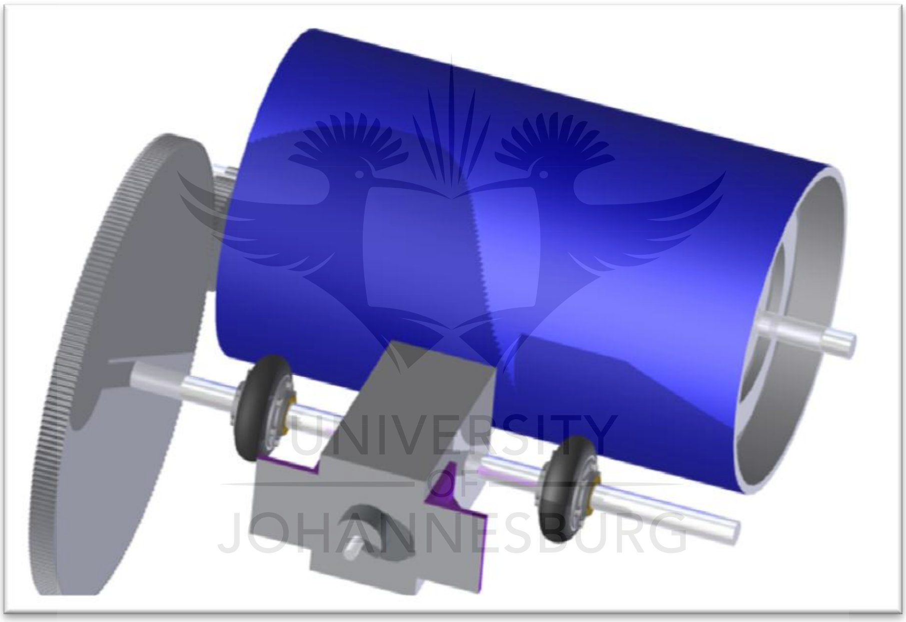


Figure 45: Gear size indication.

10.9.2 Friction wedge belt drive evaluation.

There are a couple of problems with friction wedge belt drives when used in a soil environment. Sand can go between rotating belts and pulleys thus causing belt slip and promote static conditions where the soil will stick to the pulley by means of electro- static forces and in so doing, generate unnecessary heat and further belt slip.

This is where power loss will occur and reduce drive efficiency and ultimately create more downtime due to belt failure. The belt system is not entirely redundant as in the case of the gears and can be looked at, at a later stage, if need be.

10.9.3 Chain and sprocket drive evaluation.

Chains are much more robust and suitable for vibratory applications and also have some problematic areas, but nothing critical that can cause major failure.

Chains can still operate in a soil environment and dirty conditions, even if it is heavily soiled, noise levels will increase, but the design will still last longer than a slipping belt. Less maintenance is required for chains than belts. Covers are also available to shield the chain and sprockets from dirty conditions.

NB: The final evaluation of these drives can only be determined depending on the engineering specifications of the products available on the market. Engineering specification constraints of supplier products must be adhered to, to avoid unnecessary component failure.

10.10 Chain drives investigation.

***Chain Drive Technical Literature**: Appendix 9*

To check if a chain drive will be adequate, the following calculations and investigations are done:

- chain drive design power;
- sprocket speed ratio and final speed;
- and the possible solution.

10.10.1 Chain drive design power calculation.

The HDO 100 2 gearbox model has only two sets of mating gears, in (Appendix 8) and also (catalogue) Page 34 of the Bonfiglioli gearbox catalogue, the efficiency for this model is 96 %. In general it comes down to 2% loss per mating gear set.

The efficiency for flexible tyre couplings is 99% (Womack engineering source). Since the drive system will be symmetrically located on both sides of the gearbox output shafts, the tyre couplings and chain drives will be doubled up. Half of the power will be transferred on one half of the machine and half of the power transferred to the other half of the machine. Refer to the Appendix 21.

$$\boldsymbol{P_{in\ couplings} = \eta_{gearbox} \times P_{in\ gearbox} = 0.96 \times 99.039 = 95.077\ kW}$$

Where:

- $P_{in\ couplings}$ *= Input power to the couplings see figure in 9.7 (kW).*
- $\eta_{gearbox}$ *= Gearbox efficiency, Bonfiglioli gearbox (catalogue) Page 34.*
- $P_{in\ gearbox}$ *= Gearbox input power (kW).*

NB: The 95.077 kW is the sum of total input power for both couplings. For one coupling, the input power has to be halved, $\frac{\mathbf{95.077}}{\mathbf{2}} = \mathbf{47.54\ kW}$ per coupling.

By doing calculations on one side of the power transmission system only, half power calculations can be used for further design work to simplify the design journey further. The input power to the chain drive after the coupling loss, can be calculated using the following equation.

$$\boldsymbol{P_{in\ chain\ drive} = \eta_{coupling} \times P_{in\ couplings} = 0.99 \times 47.54 = 47.065\ kW}$$

Where:

- $P_{in\ chain\ drive}$ *= Input power to the chain drive sprocket (kW).*
- $\eta_{coupling}$ *= Tyre coupling efficiency.*
- $P_{in\ couplings}$ *= Both couplings combined input power (kW).*

10.10.2 Speed ratio and speed calculation.

The slower sprocket is running at 131.93 r/min (N_1) and the maximum target speed at the MS is 2200 r/min (N_2).

$$\boldsymbol{n_{overall\ ratio} = \frac{N_1}{N_2} = \frac{2200}{131.93} = 16.676}$$

The possible speed ratio for the chain drive (Appendix 9) and “Table 4, (catalogue) Page 6” indicates that the maximum possible speed ratio is 9.5:1 and is way less than the required 16.676 ratio.

This proves that a single chain drive is not adequate to meet the step-up requirements for this design.

10.10.3 Possible solution

A possible solution is to create a series connected chain drive to accommodate for the step-up requirement. It is assumed that each chain drive will have a ratio of 4.084 for a first estimate as quantified below. The equal ratio will promote uniformity between the two drives (drive A and B).

$$\boldsymbol{n = \sqrt{n_{overall\,ratio}} = \sqrt{16.676} = 4.0836}$$ *(drive A and drive B has equal ratios)*

$$\boldsymbol{N_A = N_2 \times n = 131.92 \times 4.0836}$$
$$\boldsymbol{= 538.71\,rpm}\ \ (\boldsymbol{Speed\ of\ fastest\ shaft\ for\ drive\ A})$$

$$\boldsymbol{N_B = N_A \times n = 538.71 \times 4.0836}$$
$$\boldsymbol{= 2200\,rpm}\ \ (\boldsymbol{Speed\ of\ fastest\ shaft\ for\ drive\ B})$$

Where :

- $n_{overall\ ratio}$ = *Common ratio between drive A and B.*
- n = *Actual ratio of drive A or B.*
- N_A = *Speed of the fastest shaft for drive A in r/min.*
- N_B = *Speed of the fastest shaft for drive B in r/min.*

10.11 Series chain drive investigation for drives A & B.

To evaluate the step-up requirement for a series connected chain drive, the following calculations and selections need to be considered;

- calculation of the design power for drives A & B,
- the service factor selection,
- the chain pitch selection and
- the conclusion of chain drive limitations.

NB: Although numbers and tables are referred to in the Fenner catalogue for chain drive selection, all steps used in this section can be followed in Appendix 9.

10.11.1 Service Factor selection

From “Table 1, (catalogue) Page 3”, the Service Factor may be selected which is applicable to the drive.

- Prime mover (Tractor) - Internal combustion engine with 4 or more cylinders (Soft Starts).
- Driven machine type (Vibratory roller compactor) - Vibrating (Heavy Duty)
- Hours of operation - 10 and under.
- Service factor (SF) - 1.3

10.11.2 Design power calculations for drive A & B.

$$\boldsymbol{P_{design\ A} = P_{in\ chain\ drive\ A} \times SF = 47.065 \times 1.3 = 61.185\ kW}\ (\textbf{Drive A})$$

$$\boldsymbol{P_{design\ B} = P_{in\ chain\ drive\ A} \times \eta_{chain\ drive\ A} \times SF = 47.065 \times 0.98 \times 1.4}$$
$$\boldsymbol{= 59.96\ kW}\ (\textbf{Drive B})$$

Where:

- P_{design} *A = Design power for drive A (kW).*
- P_{design} *B = Design power for drive B (kW).*
- $\eta_{chain\ drive}$ *A = Efficiency for drive A.*

10.11.3 Chain pitch selection.

The following coordinates will be used to select the applicable chain pitch for the drives in Figure 46.

- Drive A specification: 61.185 kW at 538.71 r/min
- Drive B specification: 59.96 kW at 2200 r/min

From Figure 46 it can be seen that the selection of a suitable chain pitch for drive A was a success (Type 24B or 28B), but for drive B, no chain pitch is available for a shaft running at such a high rotational velocity of 2200 r/min and with a 59.96 kW design power rating.

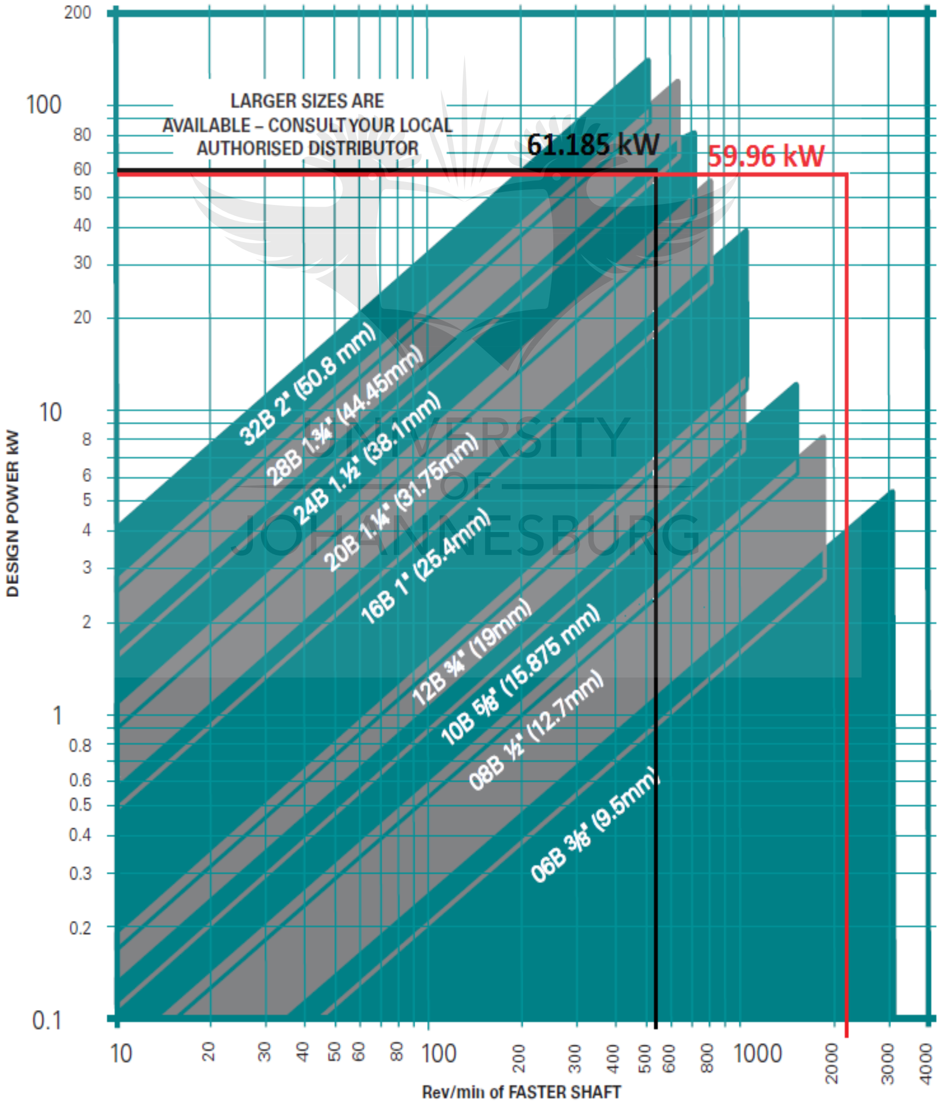


Figure 46: Chain drive pitch selection.

The Figure 46 show that the selection process for drive B was a failure due to speed constraints, thus it is not possible to use a series chain drive combination to solve the problem, see Table 19 below, which summarises all relevant data.

Table 19: Chain drive possibility check.

Drive A and Drive B chain selection data summarization.					
Drive	*Design Power kW*	*(Input) Speed of Slower Shaft r/min*	*(Output) Speed Of Fastest Shaft r/min*	*Speed Ratios*	*Chain type Selected*
A	61.185	131.93	538.71	4.0836	24B or 28B
B	59.96	538.71	2200	4.0836	Not possible

The main failure of the chain drives for this type of design is its speed limitation. Due to the impossibility of using the series connected chain drives, the next step is to evaluate the use of a friction wedge belt drive.

10.12 Fenner friction wedge belt drive selection.

Friction Wedge Belt Drive Technical Literature*: Appendix 12*

The focus for belt drive selection needs to be placed on the capable speed ratios of such a drive to avoid unnecessary design work. If there is no available speed ratios that can satisfy the design criteria, all other work will not be feasible.

The consideration of the speed ratio is the critical area to be taken into consideration, before the manufacturer's process for drive selection stipulated in the catalogue may be attempted.

NB: The justification for all work and reasoning pertaining to this section can be found in Appendix 12.

10.12.1 Speed ratio check.

From the chain drive section, the ratios and speeds are as below. The 16.676 ratio is the required step-up ratio needed to get to the output speed at the MS.

$$n_{overall\,ratio} = \frac{N_1}{N_2} = \frac{2200}{131.93} = 16.676$$

Since the focus is on stepping up the drive, the following centre distance tables in the catalogue were checked first, for the available speed ratios:

- ✓ *the Centre Distance Table (CDT) for SPZ, XPZ & QXPZ Wedge Belt Drives ((catalogue) Page 42) gives a maximum ratio of 5.97;*
- ✓ *the CDT for SPA, XPA & QXPA Wedge Belt Drives ((catalogue) Page 45) yield a maximum speed ratio of 5.94;*
- ✓ *the CDT for SPB, XPB, & QXPB Wedge Belt Drives ((catalogue) Page 49) a maximum ratio of 5.88 and*
- ✓ *the CDT, SPC & QXPC Wedge Belt Drives ((catalogue) Page 51), its maximum ratio is 5.58.*

10.12.2 Conclusion

Referring to Table 20 below, the Fenner catalogue with its low ratios for friction wedge belt drives cannot supply a single drive with a ratio of 16.676. By calculation, for the ratios in 10.12.1, the output speeds for these ratios range from 736.17 r/min to 787.62 r/min and is far below the required 2200 r/min target. Thus, a single belt drive cannot supply the necessary design requirements.

A series drive is the next option to be considered as in the case of the chain drives.

Table 20: Speed ratios available and final speed limitation.

Available possible speed ratios for all belt types.					
Belt Type	*Input speed from gearbox (N_1) r/min.*	*Maximum possible Drive Speed ratio.*	*Output Speed (N_2) r/min*	*Satisfy ratio requirements*	*Reference: Catalogue*
SPZ, XPZ & QXPZ	131.93	5.97	787.62	no	Page 42
SPA, XPA & QXPA	131.93	5.94	783.66	no	Page 45
SPB, XPB & QXPB	131.93	5.88	775.75	no	Page 49
SPC & QXPC	131.93	5.58	736.17	no	Page 51

10.13 Investigation of a series connected friction wedge drive system.

Refer to Appendix 12 for the following design work. To check if the series drive system will work, the proper procedure for belt drive selection has to be followed as stipulated in the Fenner Friction Wedge Belt Catalogue.

The inspection of minimum and maximum pulley sizes has to be done first as stipulated on “(catalogue) Page 38, Table 1”. The minimum pulley size is based on design power requirements and has to be obeyed for a successful design.

10.13.1 Speed ratios.

The same method as in the series chain drive system will be used to further the selection.

$$\boldsymbol{n = \sqrt{n_{overall\,ratio}} = \sqrt{16.676} = 4.084}$$

Both drives A and B will use an equal speed ratio of 4.084 each, to get to the final speed of 2200 r/min.

10.13.2 Service Factor and Speed Increase Factor Selection.

From “Table 3 (catalogue) Page 39”, the Service Factor may be selected which is applicable to the drive. The drive system is a (step-up) speed increasing drive system and an additional speed ratio factor is used in the design power calculation.

From “Table 3” the following items were selected:

- Prime mover - Internal combustion engines with 4 or more cylinders (Soft Starts).
- Driven Machine – Class 4 signifies impact loads and vibration.
- Operation hours *–10 hours and under.*
- Service factor – 1.3
- Speed increase factor – 1.25 (Our speed ratio is 4.084 above the maximum of 3.5)

10.13.3 Design Power for Drive A and B.

Variables for the design power equations for shafts A and B:

- $P_{design\ A}$ *= The safe power needed to make a proper belt selection for drive A (kW).*
- $P_{design\ B}$ *= The safe power needed to make a proper belt selection for drive B (kW).*
- $P_{in\ belt\ drive\ A}$ *= Output power from the coupling to drive A (kW).*
- $\eta_{belt\ drive\ A}$ *= Percentage power loss drive A during power transmission (kW).*
- *SF – Service Factor is generally used as a safety factor in most design catalogues.*
- *SPIF – Speed increase factor, only used if a drive is going to be stepped up in speed.*

The efficiency for the belt drive is 96% (Appendix 2).

$$\boldsymbol{P_{design\ A} = P_{in\ belt\ drive\ A} \times SF \times SPIF = 47.065 \times 1.3 \times 1.25}$$
$$\boldsymbol{= 76.48\ kW\ (\text{Drive A})}$$

$$\boldsymbol{P_{design\ B} = P_{in\ belt\ drive\ A} \times \eta_{belt\ drive\ A} \times SF \times SPIF}$$
$$\boldsymbol{= 47.065 \times 0.96 \times 1.3 \times 1.25 = 73.42\ kW\ (\text{Drive B})}$$

10.13.4 Belt Type Selection.

Belt selection coordinates:

- Drive A – 538.71 r/min ; 76.48 kW (Output speed and design power for drive A)
- Drive B – 2200 r/min ; 73.42 kW (Output speed and design power for drive B)

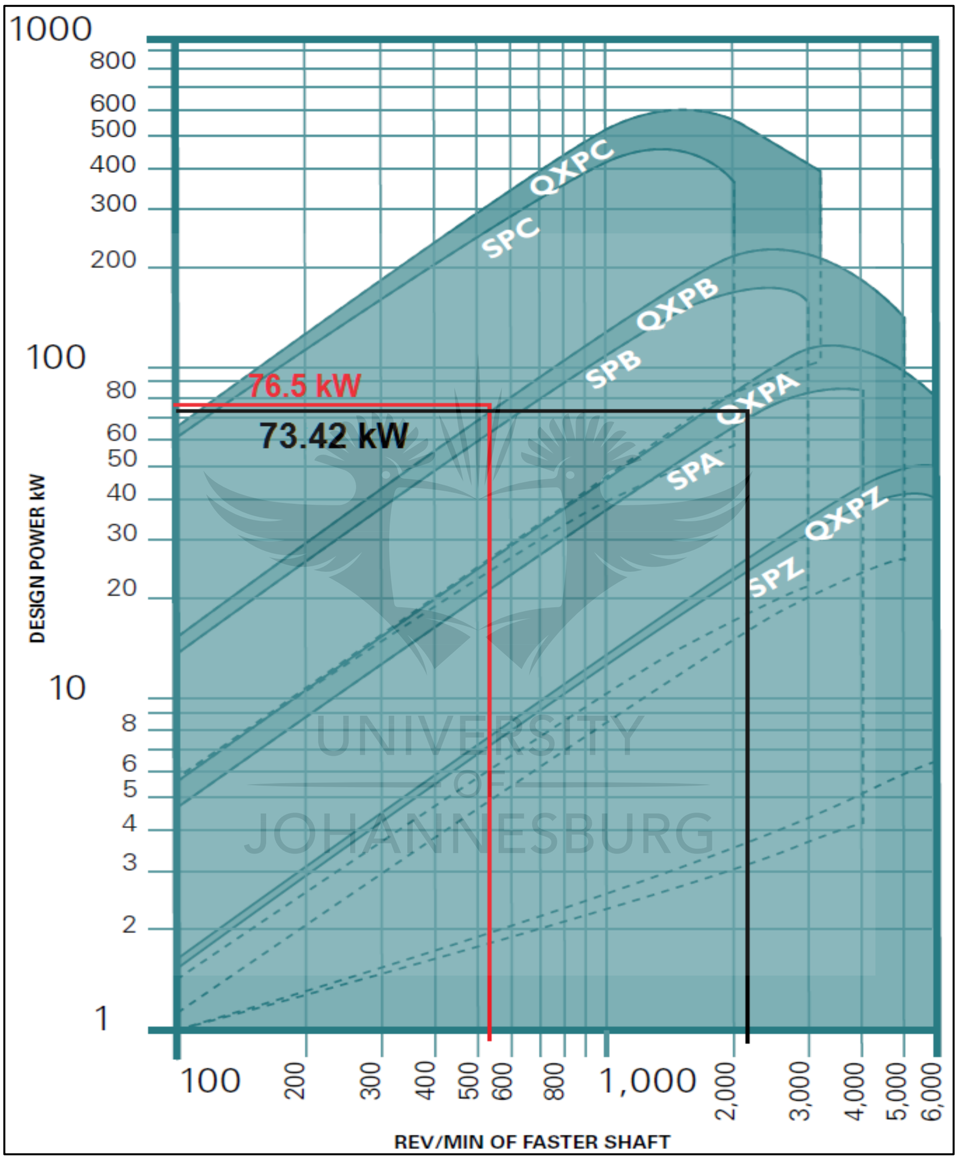


Figure 47: Belt type selection.

Figure 47 shows the belts applicable to the drives A & B.

Drive A – The selected belts are SPC & QXPC.

Drive B - The selected belts are QXPA or SPB & QXPB.

10.13.5 Investigation of the min & max pulley diameters for both drives A & B.

"(catalogue) Page 38, Table 1" the minimum pulley diameter (MIPD) can be selected. Since the MIPD's are not listed in the table, interpolation is used to get the MIPD for the values concerned. The interpolation consists of the speed and MIPD's.

$$\frac{d_{A\,min} - 300}{375 - 300} = \frac{538.71 - 600}{500 - 600} \therefore d_{A\,min} = 345.97 \rightarrow 350\,mm$$

$$\frac{d_{B\,min} - 170}{190 - 170} = \frac{2200 - 2880}{1800 - 2880} \therefore d_{B\,min} = 182.59 \rightarrow 190\,mm$$

In Table 21, drive A the calculated MPD is 350 mm and the suggested design power (SDP) is 90 kW. Multiplying the MPD by the calculated ratio of 4.084, the maximum pulley diameter (MAPD) (1429.4 mm) for drive A is obtained (see Table 21). Similarly for drive B with its 75 kW SDP and 190 mm MPD, the MAPD is 190 x 4.084 = 776 mm.

Table 21: Minimum suggested pulley diameters and maximum.

Calculation of minimum & maximum pulley diameters for Drive A and Drive B.						
Drive	*Calculated Power (kW)*	*Suggested design Power due to min pulley dia.*	*Min Pulley diameter (mm)*	*Ratio*	*Max calculated Pulley Diameter (mm)*	*Reference: Catalogue*
A	76.48	90 kW	350	4.084	1429.4	Page 38,Table 1
B	73.42	75 kW	190	4.084	776	Page 38,Table 1

The 1429.4 mm MAPD is bigger than the 1295 mm diameter for the compacting drum.

The assumed equal ratios of 4.084 for drive A and drive B are not possible due to excessively large pulley diameter for drive A. To possibly solve the problem, the drive ratios need to be revisited to see if the drive ratios can be manipulated in such a way that the maximum pulley sizes falls within an acceptable range.

10.13.6 Re-evaluation of drive ratios.

To reduce the MAPD for drive A, a much smaller ratio needs to be considered for the drive. The maximum possible ratio (MAPR) for drive B must be selected from the catalogue first. This will give the advantage to decrease the ratio for drive A to a minimum possible ratio (MIPR) and then in turn will decrease the MAPD for this drive.

10.13.6.1 Selection of a maximum actual ratio for drive B and the calculation of the MAPD for drive A.

Selecting a MAPD for drive B in accordance with the manufacturer's suggested 190 mm MIPD (Table 1, (catalogue) Page 38).

- On (catalogue) Page 45, for belts SPA & QXPA (drive B), the highest ratio possible (HRP) is 3.15 with an actual minimum pulley diameter (AMIPD) of 200 mm. The ratio is too low for drive B and will increase the drive A ratio to 5.3 to still comply with the final 2200 r/min output speed. This will result in an even larger pulley diameter (1855 mm) than previously calculated. See the calculation prove below.

$$\boldsymbol{n_A = \frac{2200}{131.93 \times 3.15} = 5.3\ (ratio\ too\ large)}$$

$$\boldsymbol{d_{A\,max} = d_{A\,min} \times n_A = 350 \times 5.3 = 1855\ mm} \quad \text{Also;}$$

$$\boldsymbol{d_{B\,max} = d_{B\,min} \times n_B = 200 \times 3.15 = 630\ mm}$$

Where:

- n_A = *Drive A speed ratio.*
- n_B = *Drive B speed ratio.*
- $d_{A\ min}$ = *Minimum pulley diameter for drive A (mm).*
- $d_{B\ min}$ = *Minimum pulley diameter for drive B (mm).*
- $d_{A\ max}$ = *Maximum pulley diameter for drive A (mm).*
- $d_{B\ max}$ = *Maximum pulley diameter for drive B (mm).*

This ratio combination (5.3 drive A; 3.15 drive B) is not possible, due to the compacting drum size. If it was possible, such a large pulley will cause bearing failure, utilise a tremendous amount of power for the start-up torque to overcome its inertia and too heavy for onsite maintenance and transportation. Specialised equipment will then be needed such as a crane, tooling, etc.

10.13.6.2 Recalculation of ratios and MAPD for drive B and drive A.

- On (catalogue) Page 49, for belts SPB & QXPB (drive B), the highest actual ratio possible (ARP) is 5.26 for an AMIPD of 190 mm. The 190 mm AMIPD is the same as the suggested MIPD in "Table 17".

By using the 5.26 ratio obtained for drive B above, the ratio for drive A (3.17) is calculated below including their MAPD's.

$$\boldsymbol{n_A = \frac{2200}{131.93 \times 5.26} = 3.17}$$

$$\boldsymbol{d_{A\,max} = d_{A\,min} \times n_A = 350 \times 3.17 = 1110\,mm}$$

$$\boldsymbol{d_{B\,max} = d_{B\,min} \times n_B = 190 \times 5.26 = 999.4\,mm \approx 1000\,mm}$$

The MAPD for drive A is still too large at 1110 mm and still too close to the compacting drum size of 1290 mm. A lower possible ratio needs to be looked up in the catalogue for drive A.

Since the 3.17 is a calculated value and not an actual value selected from the catalogue, an actual ratio less than 3.17 must be selected before the final speed can be calculated at the drive B output shaft (machine shaft).

10.13.6.3 Actual ratio selection and maximum pulley diameter calculation for the SPC & QXPC (drive A) belts.

- On (catalogue) Page 51, for belts SPC & QXPC (Drive A), the selected ARP is 3.13 and has a MIPD of 400 mm. The 400 mm AMIPD is within specification, because it is a higher value than the suggested MIPD of 350 mm and not less and therefore OK.
- Other APR combinations yield a final speed less than the minimum 2100 r/min MS target speed due to their APR's for this application.

By calculation investigation the actual drive A MAPD is 1250 mm indicated in the calculation below and is still too large.

$\boldsymbol{d_{A\,max} = d_{A\,min} \times n_A = 400 \times 3.13 = 1250\,mm}$ (still too large)

Recalculation of the final output speed at drive B (at machine shaft).

$$\boldsymbol{N_3 = n_B \times n_A \times N = 5.26 \times 3.13 \times 131.93 = 2172.01\, rpm}$$

Where:

- N_3 = Final output speed at MS pulley r/min.

The 2172.01 r/min is within the desired speed range when compared with other compactor speeds in Appendix 21.

10.13.7 Conclusion for the series friction wedge belt drives system.

Table 22: Belt type vs summarised data obtained so far in calculations and selections.

Summary of final selections for Drive A and Drive B.								
Drive & belt types.	*Design Power (kW).*	*Suggested design Power Rating.*	*Suggested Min Pulley diameter (mm).*	*Actual min. Pulley diameter (mm).*	*Actual Ratio*	*Max Pulley Dia. (mm).*	*Success for larger pulley size.*	*Reference: Catalogue*
A-SPC & QXPC	76.48	90 kW	350	400	3.13	1250	no	Page 38, Table 1
B-SPB & QXPB	73.42	75 kW	190	190	5.26	1000	no	Page 38, Table 1

The selection made for the two drives are the most possible efficient selection as proven in 9.13.4 and summarized in the Table 22. However, the MAPD for drive A is still too large at 1250 mm and there is no other solution in the catalogue.

It is therefore not possible to use a series friction wedge belt drive combination due to the awkward actual MIPD and the available APR's. The MAPD for drive A is once again too close to the compacting drum size of 1295 mm diameter and is therefore not feasible.

NB: The last option available is to try a chain drive installed in series with a friction wedge belt drive to see if the engineering gap can be bridged.

10.14 Chain drive and belt drive series combination.

Sections discussed and or calculated are;

- chain drive design power kW,
- belt drive design power kW,
- chain pitch and belt type selection,
- chain & belt drive calculations,
- final drive selection and
- friction wedge belt drive refinement.

10.14.1 Chain drive design power.

Due to the chain drive's incapability of handling the high final output speed of 2200 r/min, it must be installed first and thereafter the belt drive.

- The 47.065 kW was calculated in 10.10.1.
- Service factor (SF) = 1.3 (selected in 10.11.1)

$$\boldsymbol{P_{design\ chain} = P_{in\ chain\ drive} \times SF = 47.065 \times 1.3 = 61.185\ kW}$$

10.14.2 Belt drive design power.

- The belt speed increase factor (SPIF) is 1.25 (4.13 is higher than the 3.5 ratio Appendix 12)
- Chain drive efficiency (Appendix 2)
- Service factor (SF) = 1.3 (Selected in 9.13.3)

$$\boldsymbol{P_{design\ belt} = P_{in\ chain\ drive} \times \eta_{chain\ drive\ A} \times SF \times SPIF}$$
$$\boldsymbol{= 47.065 \times 0.98 \times 1.3 \times 1.25 \quad = 74.95\ kW} \text{ (\textbf{Drive B})}$$

10.14.3 Preliminary chain pitch selection.

Reference: Appendix 9.

For the chain pitch selection, the chain selection coordinates was 61.185 kW; 538.71 r/min in 9.11.3. The chain pitches selected was types 24B or 28B. In the chain (catalogue) Page 5, power rating table, indicates that 24B is already within the needed power range of (88 kW Simplex) at 600 r/min.

The 28B drive has a too high power rating (133.5 kW Simplex) at 600 r/min and will have a much greater centre distance, resulting in a longer compacting drum frame length, causing more bending stress in the frame sections and subsequently increased costs.

To get an even shorter frame length, the 20B chain pitch is also adequate with its 73.5 kW duplex and 108.75 kW triplex at 600 r/min (next higher value than 544.9 r/min). The centre distance (CD) for the 20B chain pitch is 1200 mm, 24B (1250 mm CD) and for the 28B (1500 mm CD).

All calculations will be done from henceforth on the 20B chain pitch, because of its shorter centre distance that will allow the compactor frame to be 150 mm to 300 mm shorter.

10.14.4 Belt type selection

For the belt drive the selection coordinates is 74.95 kW at 2200 r/min. The belts selected are QXPA, SPB & QXPB as indicated in Figure 48.

The minimum pulley diameter interpolated from “Table 1, (catalogue) Page 38” is 182.59 mm, say 190 mm.

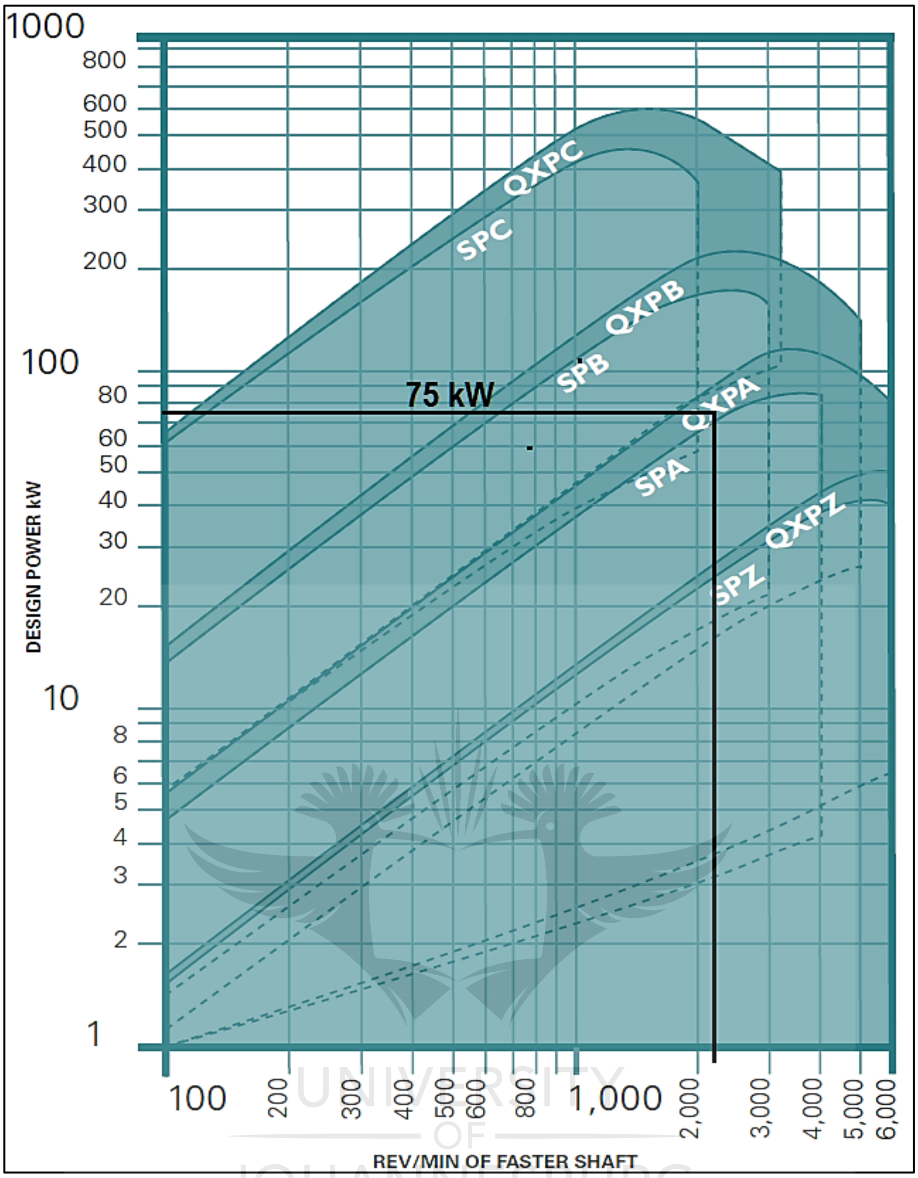


Figure 48: Belt type selection.

10.14.5 Chain & belt drive calculations.

10.14.5.1 Chain drive calculations

See Appendix 9, sprocket factor table and power rating table. Focusing on the 23 teeth sprocket (for impulse application 23 teeth and above is recommended):

- The maximum ratio is 4.13.
- The speed for this ratio is : $\mathbf{131.93 \times 4.13 = 544.9}\ \boldsymbol{rpm}$
- The sprocket factor for 23 teeth is 1.2 and needs to be multiplied with the power in the power rating table to get the final power.

- As mentioned in 9.14.3.1 the 20B chain pitch will be used to achieve a more economical design.
- The selected drive will be 20B triplex (108.75 kW at 600 r/min), although a duplex drive at 88 kW, 600 r/min is adequate, triplex drives are much more reliable when it comes to impulse application (Appendix 9).

$$\boldsymbol{Final\ Power\ = Power\ rating \times Sprocket\ factor\ for\ 23\ teeth = 108.75 \times 1.2}$$
$$\boldsymbol{= 129.6\ kW}$$

For all other speed ratios and power calculations relative to their number of teeth for each sprocket, see Table 23 and see Appendix 9 for the applied method of calculation.

Table 23: 20B pitch chain.

Chain Drive power rating for 20B chain pitch (Triplex)							
Chain Drive Speed Specifications ((catalogue) Page 6, Table 4)				*Power Rating for 24B Pitch Type ((catalogue) Page 5).*			
Chain Pitch	*No of Teeth smaller sprocket.*	*Maximum possible step-up chain speed ratio.*	*Speed of faster sprocket (r/min).*	*Power rating selection speed*	*Sprocket Factor*	*Power Rating selected from 20B power rating table (Triplex)(kW).*	*Power rating multiplied by the sprocket factor (Triplex) (kW).*
20B	19	5	660	700	1	124.88	124.88
20B	20	4.75	626.7	700	1.05	124.88	131.124
20B	21	4.52	596.32	600	1.1	108.75	119.625
20B	22	4.32	567	600	1.15	108.75	125.063
20B	23	4.13	544.9	600	1.2	108.75	129.6

Appendix 9-8 indicates design success for the chain generator design software used in Inventor 2013. This confirms that a 20B chain pitch, 23 teeth small sprocket, 4.13 ratio, triplex drive is adequate for the current application.

10.14.5.2 Belt drive specifications & discussion.

NB: The chain drive input speed is 131.93 r/min and the current output speed is 544.9 r/min and needs to be stepped up to a final output speed of 2200 r/min at the belt drive output pulley.

To make a most efficient selection to obtain the right ratios, the whole series drive has to be taken into consideration. The chain drive and the belt drive need to complement one another to achieve the objective.

By studying the Table 24 below, the 23 teeth sprocket in conjunction with its corresponding pulley and belts, a clearer evaluation can be made to ensure design compliance between the chain drive and the belt drive.

The SPB and QXPB belts satisfy the requirements of the final output speed in conjunction with the 23 teeth sprocket, 2179.5 r/min output pulley speed with its 200 mm MIPD.

The selected belts (QXPA, XPB, SPB and QXPB) in Table 24 are stipulated in 9.14.3.2. The actual ratio was selected from the mentioned belt type ratio tables (Appendix 12).

*Table **24**: Final series drive output speed.*

Belt Drive output speed for QXPA, QXPB & SPB Belts.							
Chain Ratio		*Belt Ratio*		*Pulley*		*Belt drive output speed*	
No of Teeth for smaller Chain drive sprocket.	*Maximum chain drive ratios from the (catalogue) Page 6.*	*Actual ratio selected from belt catalogue (QXPA).*	*Actual ratio selected from belt catalogue (XPB, SPB, QXPB).*	*Min pulley diameter mm (QXPA).*	*Min pulley diameter mm (XPB, SPB, QXPB).*	*Calculated output speed of faster pulley r/min (QXPA)*	*Calculated output speed of faster pulley r/min (XPB, SPB, QXPB).*
23	4.13	3.15	4.0	200	200	1716 (no)	2179.5(yes)

The final output speed for the combination drive is 2179.5 r/min, say 2180 r/min and indicates success for belt types XPB, SPB and QXPB with their 4.0 ratios.

10.15.5 Final drive selection.

10.15.5.1 Centre distance, belt length, combined arc and belt length correction factor selection.

Refering to (catalogue) Page 48, on the XPB,SPB, QXPB CD table, the following selection was made ;

- centre distance = 1176 mm (SPB, QXPB) and 530 mm for XPB (smallest centre distance possible must be used to avoid lenthening of compactor frame), thus 530 mm for the XPB belt type is chosen. Now focusing on XPB only.
- belt length XPB = 2800 mm
- and the combined arch lenth correction factor is 0.85 for XPB.

10.15.5.2 Belt power calculation.

The power rating table for CRE wedge belts in Appendix 12, has minimum pulley diameters up till 150 mm and does not cater for a 200 mm diameter pulley. It is also stipulated above the table "For ratings at other pulley/speed combinations - consult your local Authorised Distributor".

By looking at it in an academic way first and plotting all power and MIPD's on a graph using Excel software, the relationship between the MIPD and power per belt can be observed at the 200 mm MIPD point.

After a graph was plotted in Microsoft Excel 2010 software, it was noted that a linear relationship exists between the power per belt and the MIPD. See Figure 49 for details. The answers are also captured in tabulated format in Table 25 after Figure 49 for a clearer indication.

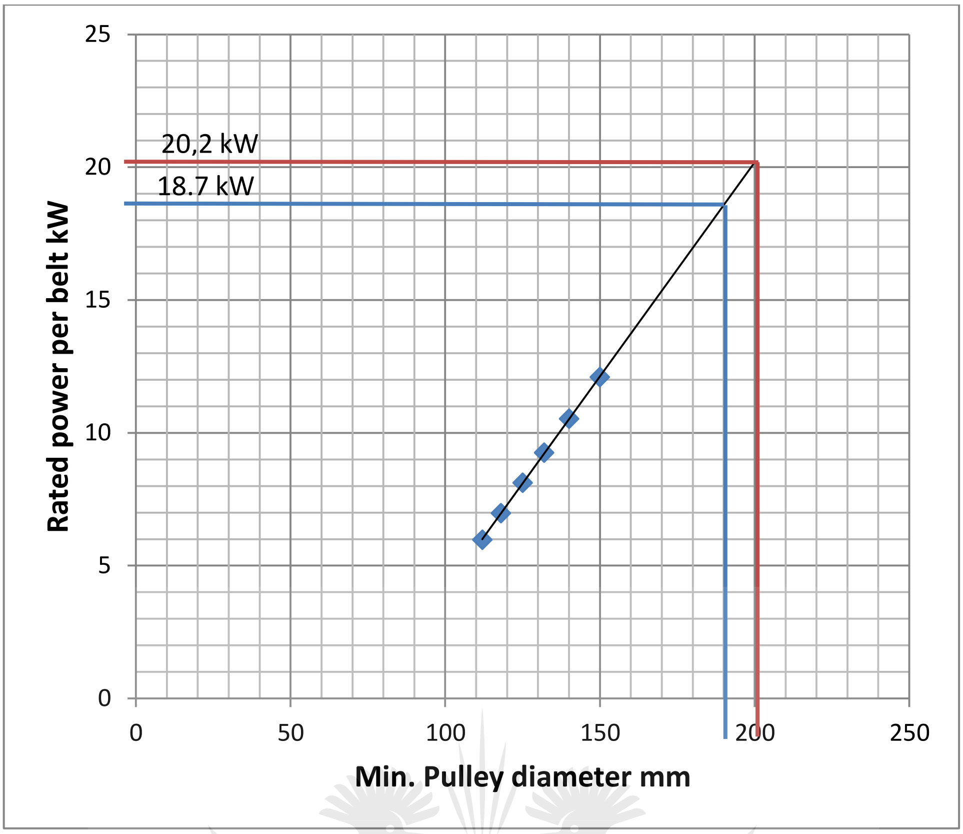


Figure 49: Rated power vs minimum pulley diameter.

The Table 25 shows data typed directly from the table on (catalogue) Page 50, except for the 190 mm and 200 mm diameter pulleys that was read from Figure 49 and typed in for completeness.

Table 25: MIPD vs power per belt.

XPB belt type power rating & minimum pulley diameter data at 2180 r/min	
Power rating per belt (kW).	*Minimum pulley diameter (mm).*
5.96	112
6.96	118
8.11	125
9.24	132
10.52	140
12.09	150
18.7	190
20.2	200

At 2180 r/min the power ratings for the SPB and QXPB belt types, for 190 mm and 200 mm diameter pulleys are as follow:

- (catalogue) Page 54 gives the power per belt rating for the SPB belts as 17.6 kW; 200 mm pulley ;
- for the QXPB belts (catalogue) Page 57 is 23.21 kW; 200 mm pulley
- the additional power per belt is 1.88 kW for the 200 mm pulley, these additional powers per belt ratings also apply for the XPB and QXPB belt types.

10.15.5.2.2 Corrected power per belt for the 200 mm diameter pulley.

$$\boldsymbol{P_{XPB} = (20.2 + 1.78) \times 0.85 = 18.683\ kW/belt}$$

$$\boldsymbol{P_{SPB} = (17.76 + 1.78) \times 0.95 = 18.563\ kW/belt}$$

$$\boldsymbol{P_{QXPB} = (23.21 + 1.78) \times 0.95 = 23.74\ kW/belt}$$

10.15.5.2.3 Number of belts required per belt type.

Calculating the number of belts for the (XPB belt type).

$$\boldsymbol{n_{XPB=}\frac{P_{design}}{P_{XPB}} = \frac{74.95}{17.48} = 4.288 \rightarrow 5\ belts}$$

For all other belt types, see Table 26.

Table 26: Number of belts per belt type.

Number of belts per belt type.				
Minimum pulley diameter (mm).	*Belt type.*	*Corrected Power (kW).*	*Calculated number of belts.*	*Final number of belts.*
200	XPB	18.683	4.012	5
	SPB	18.563	4.038	5
	QXPB	23.75	3.156	4

The selected XPB belt type requires 5 belts to operate within its safe limits.

10.16 Revisiting of the gearbox selection process.

Gearbox Technical Literature*: Appendix 8*

NB: for all relevant referrals in this section, Appendix 8 (i.e. Bonfiglioli Catalogue) can be consulted. Catalogue consultation is optional.

Once the power transmission from the gear box to the vibrating drum was dealt with, the gearbox selection can be completed with its cooling checks and various gearbox dimensions. The preliminary selected model in 10.8, was the HDO 100 2 with a power rating of 257kW and a speed ratio of 10.9. To finalise the selection of this model, the cooling and overhanging load needsto be verified.

10.16.1 Gearbox cooling check.

To make a successful selection of the gearbox cooling system, the P_T (Thermal Power) must be equal or greater than the P_{r1} (Input power of the gearbox) as seen on (catalogue) Page 18 of the Bonfiglioli Catalogue. If it meets the power requirements as stipulated on (catalogue) Page 18, it is not necessary for a cooling system, but if the thermal power P_T is less than the gearbox input power P_{r1}, a cooling system need to be selected.

Referring to (catalogue) Page 51, the thermal power for the HDO 100 2 model is $P_T = 71$ kW and calculated in 9.9.1, the gearbox input power (P_{r1}) is 99.039 kW.

Comparing both the thermal and gearbox input powers, ($P_T < P_{r1}$, 71 kW < 99.039 kW) it is observed that the thermal power is indeed less than the gearbox input power, thus a gearbox cooling system is needed as recommended in the catalogue.

10.16.2 Cooling system selection.

The three cooling systems just above the 99.039 kW gearbox input power, is the P_{TSR} (150 kW), P_{TFAN} (156 kW) and P_{TMCRA5} (159 kW).

The P_{TFAN} (Power Thermal FAN) is selected due to the convenience of installing the fan to the gearbox input shaft to cause forced ventilation, (catalogue) Page 32.

The engineering drawing for the fan cooling device can be found on (catalogue) Page 32 and (catalogue) Page 33 for the HDO 100 2 model. Although the P_{TSR} value of 150 kW is closest to the 99.039 kW, there is no sufficient drawing information in the catalogue to show how the arrangement will fit onto the gearbox, however the PTFAN with its 156 kW thermal capacity has all relevant data and drawings in Appendix 8.

The 156 kW thermal power is greater than the gearbox input power of 99.039 kW and therefore indicates design success.

10.16.3 Over hanging load check for chain drive attachment.

The overhanging load is the final process in confirming a successful selection for the HDO 100 2 model gearbox. See (catalogue) Page 14 for the selection criteria.

Since a Fenner Flex Coupling is going to be attached between the gearbox output shaft and the chain drive FS, the overhanging load can be neglected. This is due to the fact that the gearbox output shaft is not directly assembled to the chain drive as per the manufacturers understanding of the overhanging load requirement, thus the overhanging load is nullified.

10.16.4 Gearbox selection decision.

The HDO 100 2, 257 kW gearbox with its 156 kW thermal capacity is the most efficient gearbox for the compactor.

10.16.5 Gearbox Specification.

HDO 100 2 10.9 LP D 1 GJ 257 B7

- HDO – Gearbox Type
- 100 – Gear frame size
- 2 – Number of gear reductions
- 10.9 – Reduction gear ratio
- LP – Output shaft configuration
- D – Shaft arrangement
- 1 – Execution
- GJ – Input configuration
- 257 – Gearbox power
- B7 – Mounting position

11 General Engineering Calculations

In this calculation section, forces are quantified by means of free body diagrams and mathematical models. Autodesk Inventor 2013 is utilised for shaft, mass and Inertia calculations, in so doing, strength can be determined on the different types of machine components.

For a lot of the complex calculations, simulations has been used in order to determine mass, stresses, force allocation, shaft sizes, pulley and sprocket sizes.

NB: It is important to understand that design refinement for optimal strength at a reasonable cost is a very laborious task and cannot be done entirely in this dissertation document. What this section does demonstrate is engineering sense and judgement on how to go about to get a successful design. The cost component relative to design strength will be refined after the first prototype.

11.1 Machine amplitude of vibration calculations.

The machines vertical acceleration can be calculated by taking into consideration the centrifugal force, the weight of the machine and the inertia force generated during vibrational acceleration. Dependant on the direction of the centrifugal force, either up or down, the maximum and minimum acceleration can be quantified.

11.1.1 Machine mass distribution.

The forces acting on the machine during operation are shown in Figure 50 (Free body diagram), where the inertia force F_a and gravity force W_t act downwards while the centrifugal force F_c act in the upward direction. The minimum vertical acceleration can then be calculated.

The total machine mass m_t was estimated initially in 10.4.6 as 5000 kg. Both centrifugal weights mass m_e connected to both MSs were calculated as 51.418 kg altogether and are kept constant at this stage unless stated otherwise.

The frame mass *mf is* calculated by adding the drum mass m_d and eccentric weight mass, and then subtracted from the main estimated mass below.

$$\boldsymbol{m_t = m_d + m_f + m_e \therefore 5000\ kg = 2500 + m_f + 51.418 \therefore m_f = 2448.58\ kg}$$

Where:

- *m_t = Total estimated machine mass kg;*
- *m_d = Estimated mass of the vibratory drum and components kg;*
- *m_f = Estimated drum frame and components mass kg;*
- *m_e = Eccentric weight mass kg.*

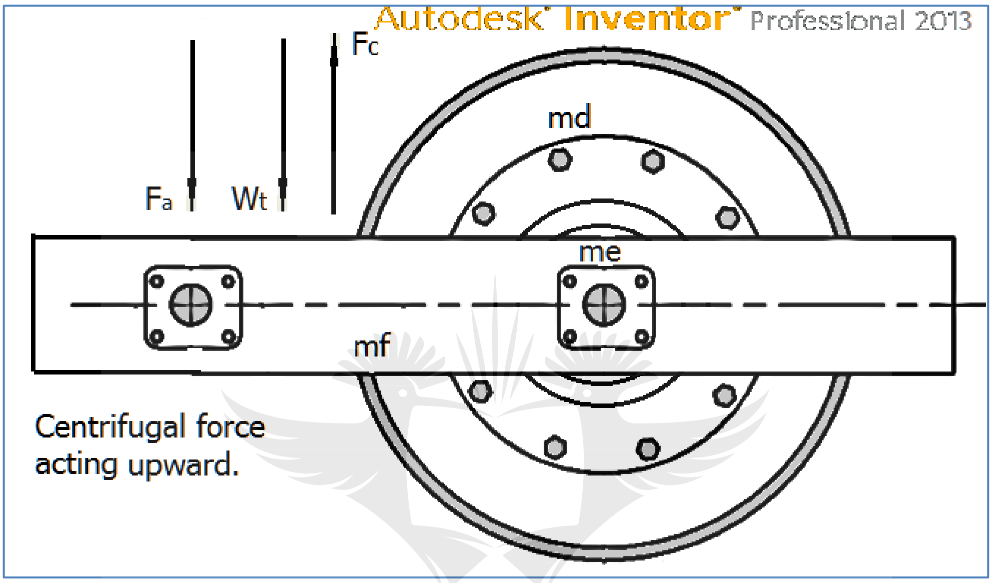


Figure 50: Minimum vertical acceleration free-body diagram.

11.1.2 Total Machine weight

$$\boldsymbol{W_t = m_t \times g = 5000 \times 9.81 = 49050\ N}$$

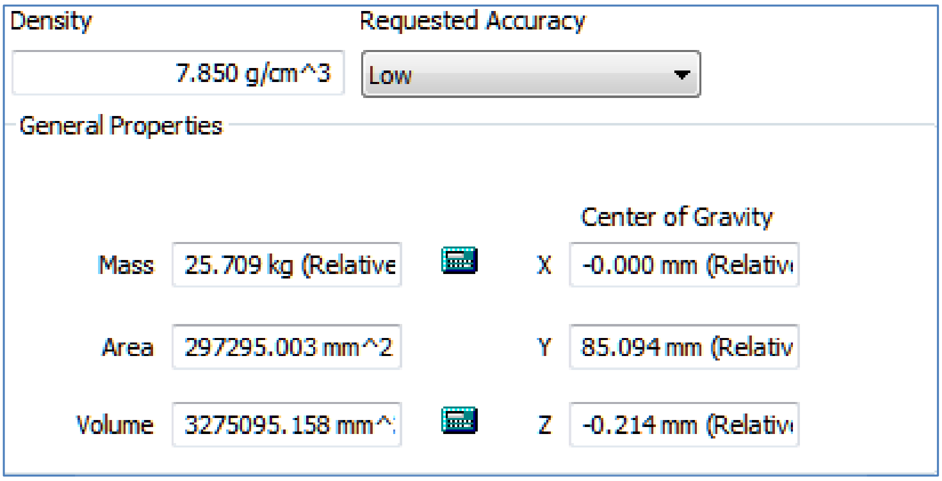
Where:

- *W_t = Total machine weight kg;*
- *g = gravitation acceleration m/s^2*

11.1.3 The total centrifugal force that excites the machine.

Given in Table 27 are the eccentric moment (85.094 mm – Inventor 2013 moment units) and the eccentric weight mass (25.709 kg for one weight only).

Table 27: Centrifugal weight mass and eccentric moment.



Density: 7.850 g/cm^3

Requested Accuracy: Low

General Properties

Mass: 25.709 kg (Relative

Area: 297295.003 mm^2

Volume: 3275095.158 mm^:

Center of Gravity

X: -0.000 mm (Relativ

Y: 85.094 mm (Relativ

Z: -0.214 mm (Relativ

The centrifugal force generated by both eccentric weights.

$$\boldsymbol{F_c = 2 \times m_e \times r \times \omega^2 = 2 \times 25.709 \times 0.085094 \times \left(\frac{2 \times \pi \times 2180}{60}\right)^2 = 228.3\ kN}$$

Where:

- *F_c = Centrifugal force kN.*
- *r = Eccentric distance m.*

11.1.4 Amplitude of vibration.

The formula for the nominal vibration amplitude is given in the Vibratory Roller Handbook [6] as;

$$\boldsymbol{A_{nominal} = \frac{m_{e\,total} \times r_e}{M} = \frac{2 \times m_e\ \times r_e}{M} = \frac{2 \times 25.709 \times 0.085094}{5000} = 0.8751\ mm}$$

Where:

- *$A_{nominal}$ = Nominal amplitude of vibration mm.*
- *$m_{e\ total}$ = Total mass for both eccentric weights kg.*
- *r_e = Eccentric moment mm.*
- *m = Estimated machine mass kg.*

For the double amplitude required as seen published in all catalogues in their specification section for SDVRR compactors in Appendix 15 to Appendix 20;

$$\boldsymbol{A_{double} = 2 \times A_{nominal} = 2 \times 0.8751 = 1.75\ mm}$$

At a later stage after the design has been finalised and the machines final mass has been quantified by Inventor 2013, both the nominal and double amplitude can be recalculated for the actual specification.

11.2 Minimum vertical machine acceleration.

Minimum vertical acceleration occurs when the centrifugal force acts upward see Figure 50.

$$\boldsymbol{F_c = W_t + F_a \therefore 5000 \times a_{min} = 228300 - 49050 \therefore a_{min} = 35.855\ m/s^2}$$

Where:

- *F_a = Acceleration force to oscillate the machine N;*
- *a_{min} = Minimum acceleration m/s^2.*

Maximum vertical acceleration occurs when the centrifugal force acts downward see Figure 51.

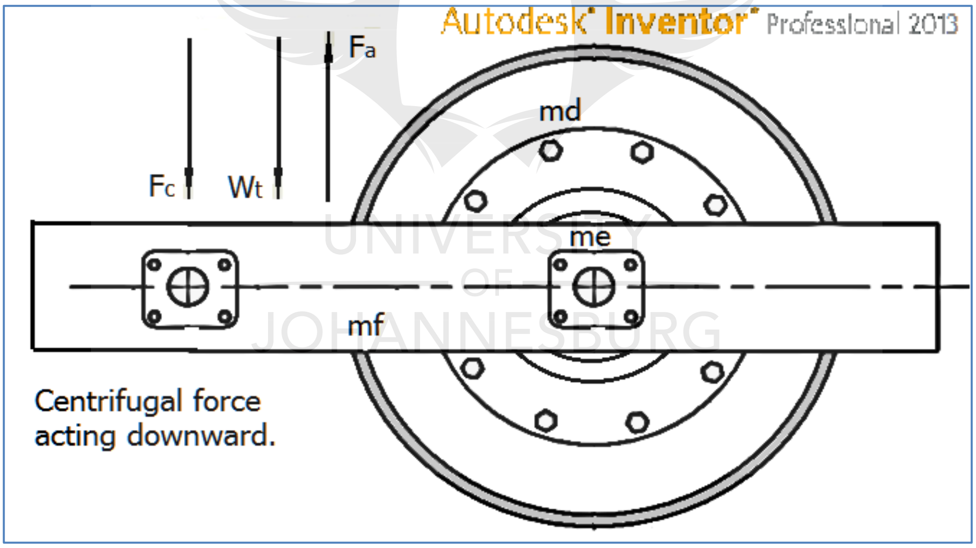


Figure 51: Maximum vertical acceleration free-body diagram.

$$\boldsymbol{F_a = F_c + W_t \therefore 5000 \times a_{max} = 228300 + 49050 \therefore a_{max} = 55.475\ m/s^2}$$

Where:

- *F_a = Acceleration force to oscillate the machine N;*
- *a_{max}= Maximum vertical acceleration m/s^2.*
- *F_c = Centrifugal force that excites the machine N.*
- *W_t = Machine weight N.*

11.3 MS force vector calculations.

Both minimum and maximum accelerations are evaluated in further calculations to determine which of the two accelerations will yield the greatest reactions.
A decision will then be taken as to which of the two yields the greatest reaction and then calculations will be done with the maximum reaction only further onward.

The calculations done in this section are as follow:

- the difference in frame weight, inertia that will yield the frame resultant force;
- the small pulley weight calculation;
- the horizontal reaction on the MS due to belt tensions;
- the centrifugal force on each point minus gravity, or plus gravity;
- the MS support bearing reactions with the centrifugal force downward
- and the MS support bearing reactions with the centrifugal force upward.

NB: All calculations are done on one half of the machine only, due to the symmetry of the design.

11.3.1 Frame resultant force due to weight and inertia.

The compacting drum frame and its components have been estimated in 10.2.1 to be 2449.69 kg, ≈ 2500 kg. Since half of the machine is focused on, half of the frame plus its component mass would be $\frac{\mathbf{2500}}{\mathbf{2}} = \boldsymbol{1250\ kg}$.

Most of the heavy components are assembled at the front of the frame, like the gearbox, couplings, larger sprocket, chain and drive shaft which is supported by the tractor connector links and drawbar (Not part of this project).

It is reasonable to assume that the MS will carry about two thirds of the 1250 kg mass, $\boldsymbol{\left(\frac{2}{3}\right) x\ 1250 = \ 833.33\ kg}$ and the tractor linkages at frame front the other 416.67 kg. The estimate is sound due to the fact that the gearbox alone at the front of the frame weighs 660 kg (Appendix 8).

Dependant on the direction of the centrifugal force, either maximum or minimum frame inertia forces can be calculated. The acceleration of the frame will be exactly the same as that of the machine, since the frame moves in unison with the machine as a whole.

Minimum and maximum inertia forces due to frame inertia.

$$F_{frame\ max} = m_f \times a_{max} = 833.33 \times 55.475 = 46229\ N\ \ (upward)$$

$$F_{frame\ min} = m_f \times a_{min} = 833.33 \times 35.855 = 29879\ N\ (downward)$$

Where:

- $F_{frame\ min}$ = *Minimum frame inertia force N*
- $F_{frame\ max}$ = *Maximum frame inertia force N*

The force due to frame weight.

$$W_{frame} = m_f \times g = 833.33 \times 9.81 = 8175\ N\ (downward)$$

The resultant forces due to the frame weight and the minimum and maximum inertia forces.

$$F_{f\ res\ 1} = 46229 - 8175 = 38054\ N$$

$$F_{f\ res\ 2} = 29879 + 8175 = 38054\ N$$

11.3.2 Determination of the MS pulley weight.

The Pulley mass attached to the MS = approximately 17 kg as given by Inventor 2013. See the mass value shown in Table 28.

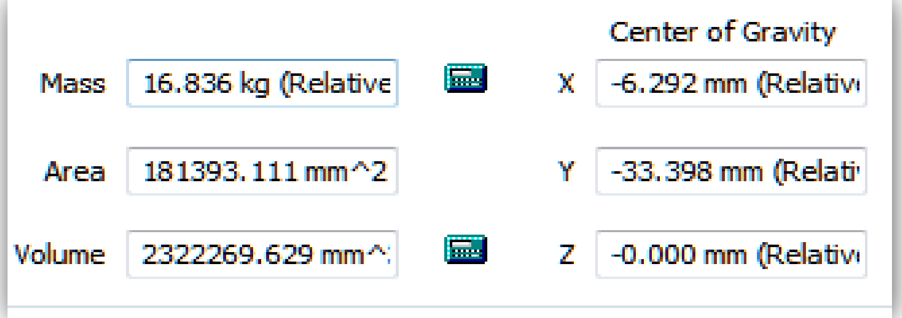
The pulley weight is calculated as:

$$W_{pulley} = m_{pulley} \times g = 17 \times 9.81 = 166.77\ N$$

Where:

- W_{pulley} = pulley weight attached to the MS N.

Table 28: Small Pulley mass as calculated by Inventor 2013.



			Center of Gravity
Mass	16.836 kg (Relative	X	-6.292 mm (Relativ
Area	181393.111 mm^2	Y	-33.398 mm (Relati
Volume	2322269.629 mm^	Z	-0.000 mm (Relativ

11.3.3 Determination of MS reaction due to pulley belt tensions.

$$\boldsymbol{T_{nominal}} = (\boldsymbol{T_1} - \mathbf{T_2}) \times \frac{\mathbf{d}}{\mathbf{2}} \therefore\ \mathbf{197.963} = (\boldsymbol{T_1} - \mathbf{T_2}) \times \frac{\mathbf{0.207}}{\mathbf{2}} \ldots\ldots\ldots.\mathbf{1}$$

The tensions for the belt drive could not be located in the Fenner catalogue; this is why another method had to be used to determine the belt tensions for the drive.

Engineeringtoolbox.com [24] gives the coefficient of friction between rubber and steel as $\boldsymbol{\mu = 0.5}$ (**rubber to steel**).

$$\frac{\boldsymbol{T_1}}{\boldsymbol{T_2}} = \mathbf{e}^{\boldsymbol{\mu\theta}} = \mathbf{e}^{\mathbf{0.5}\times\frac{\boldsymbol{\pi}}{\mathbf{180}}\times\mathbf{111°}} = \mathbf{2.634} \ldots\ldots\ldots\mathbf{2}$$

By solving both equations simultaneously, we get;

$\boldsymbol{T_2} = \mathbf{1170.263\ N}$ **and** $\boldsymbol{T_1} = \mathbf{3082.474\ N}$

Force / reaction R can now be calculated as shown in Figure 52. Force R is the shaft reaction due to the belt pull. The same reaction R will be experienced by the IS later in the document due to the reaction on the other larger pulley.

$$\boldsymbol{R^2} = \boldsymbol{R_1}^2 + \boldsymbol{R_2}^2 - \mathbf{2} \times \boldsymbol{R_1} \times \boldsymbol{R_2} \times \mathbf{COS\theta}$$

$$= \sqrt{\mathbf{3082.474^2} + \mathbf{1170.263^2} - \mathbf{2} \times \mathbf{3082.474} \times \mathbf{1170.263} \times \mathbf{COS111°}}$$

$$= \mathbf{3668.33\ N}$$

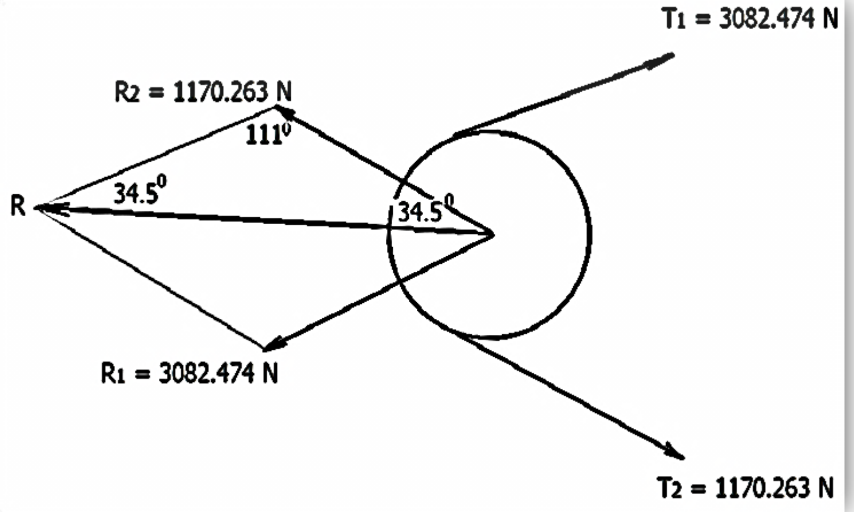


Figure 52: Small Pulley resultant force.

11.3.4 Minimum & maximum eccentric weight resultant.

The total force generated by both eccentric weights in 10.2.3 is 228326 N. The force for one centrifugal weight is $\frac{\mathbf{228326}}{\mathbf{2}} = \mathbf{114163\ N}$, each centrifugal weight consists of two lobes, the force for one of the lobes is $\frac{\mathbf{114163}}{\mathbf{2}} = \mathbf{57081.5\ N}$.

When gravity is added or subtracted, the maximum and minimum resultant force for the lobe can be calculated. The total mass for the eccentric weights were calculated as 50.314 kg, for one weight $\frac{\mathbf{50.314}}{\mathbf{2}} = \mathbf{25.157\ kg}$ and for one lobe $\frac{\mathbf{25.157}}{\mathbf{2}} = \mathbf{12.579\ kg}$.

$$\begin{aligned} \boldsymbol{F_{res\,\max c}} &= \mathbf{57081 + 9.81 \times 12.579} \\ &= \mathbf{57204.4\ N}\ (\boldsymbol{when\ the\ centrifugal\ force\ acts\ downwards}) \end{aligned}$$

$$\begin{aligned} \boldsymbol{F_{res\,\min c}} &= \mathbf{57081 - 9.81 \times 12.579} \\ &= \mathbf{56957.6}\ (\boldsymbol{when\ the\ centrifugal\ force\ acts\ upwards}) \end{aligned}$$

NB: The actual forces used in further calculations for the calculated centrifugal components are a little more than the above. This was due to adding and subtracting the whole eccentric weight mass to the centrifugal component. The actual force is about ±0.21 % (120.6 N) larger than the one calculated.

11.3.5 MS reactions evaluation.

Figure 53 shows the arrangement of the MS while Figure 54 shows the force loading on the MS. Forces moments are used to determine the reactions at the bearings “A” & “D”. The centrifugal force that excites the machine is shown in Figures 54 and 55 acting either in the upward or downward direction to investigate where the maximum shear force and bending moment is located.

One of the two answers will then be eliminated and focus will be placed on the most critical outcome of the two.

11.3.5.1 3D MS presentation.

Figure 53 is a 3D view concept of the MS with all its relevant components. This will allow for a clear understanding as to where calculations are done on the different shaft regions.

The following must be noted in Figure 53:

- at bearings “A” & “D” the reactions for the whole arrangement is located,
- points “B” & “C” is where the eccentric weight resultant force is located,
- the small pulley for the belt drive at point “E” is creating a horizontal reaction R, due to belt pull and a downward pulley weight component,
- the bearing at point “F” is assembled onto the frame to create drum roll when the frame is towed, the calculated resultant force is 38055 N at this point.

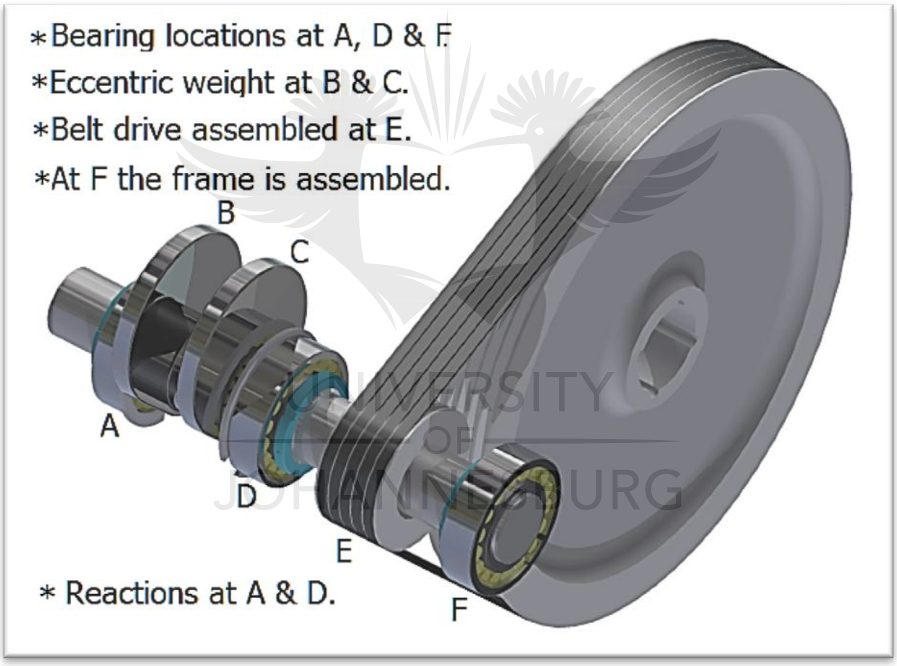


Figure 53: MS Arrangement

11.3.5.2 Taking moments when the centrifugal force is acting upwards.

Figure 54 and Figure 55 contain both the calculated forces (calculated in 10.1.1 to 10.3.4). The two figures represent two different positions for the eccentric weights, Figure 54 has the eccentric weight in the upward position and Figure 55 has the eccentric weight in the downward position.

The aim is to determine which eccentric weight position will yield the maximum reactions on the shaft. Once the maximum reactions have been established, all calculations will then be focused on the maximum reaction force arrangement.

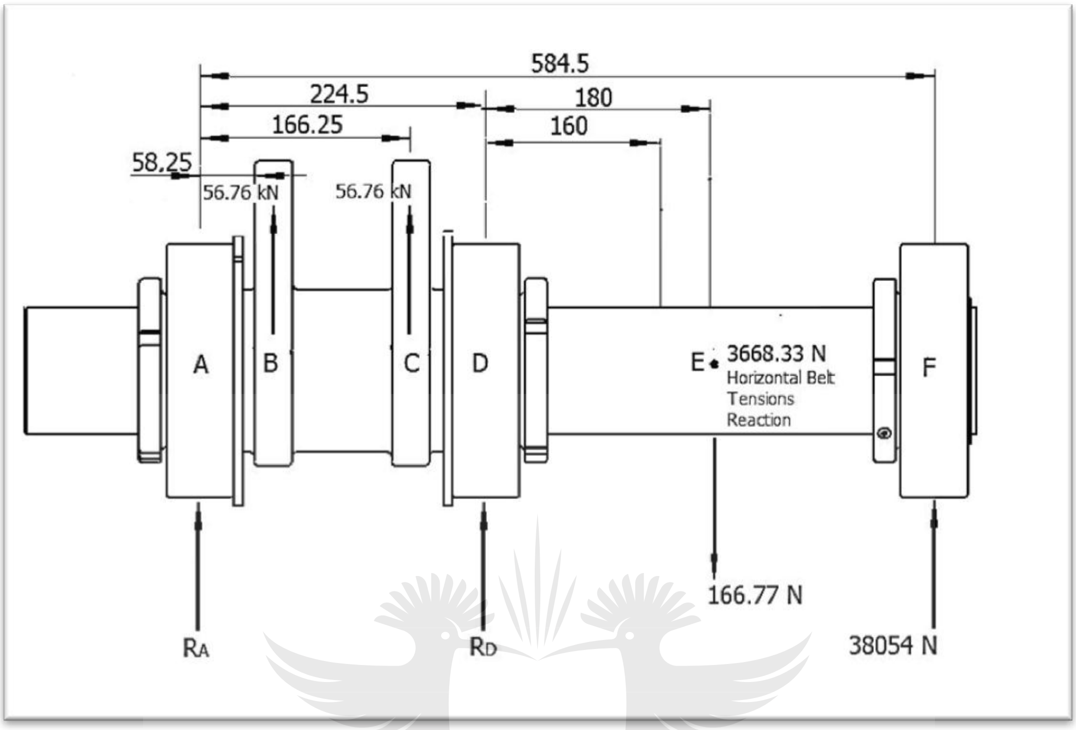


Figure 54: MS loading minimum vertical acceleration diagram.

Figure 54 is in the 2D line digram format, but simmilar to the arrangement in Figure 53. Here the centrifugal force is acting in the upward direction. Moments are taken to calculate the reactions at bearings "A" and "D".

Taking moments at R_A:

$$\boldsymbol{R_D \times 224.5 + (58.25) \times 56760 + 166.25 \times 56760}$$
$$\boldsymbol{= +(405.5) \times 166.77 + 38054 \times 584.5}$$
$$\therefore \boldsymbol{R_D = 42617,23\ N\ (upward)}$$

Taking moments at R_D:

$$\boldsymbol{180 \times 166.77 + (584.5 - 224.5) \times 38054 + (224.5 - 166.25) \times 56760}$$
$$\boldsymbol{+ (224.5 - 58.25) \times 56760 + R_a \times 224.5 = 0}$$
$$\therefore \boldsymbol{R_A = 117915.72\ N\ (downward)}$$

11.3.5.3 Taking moments with the centrifugal force acting downwards.

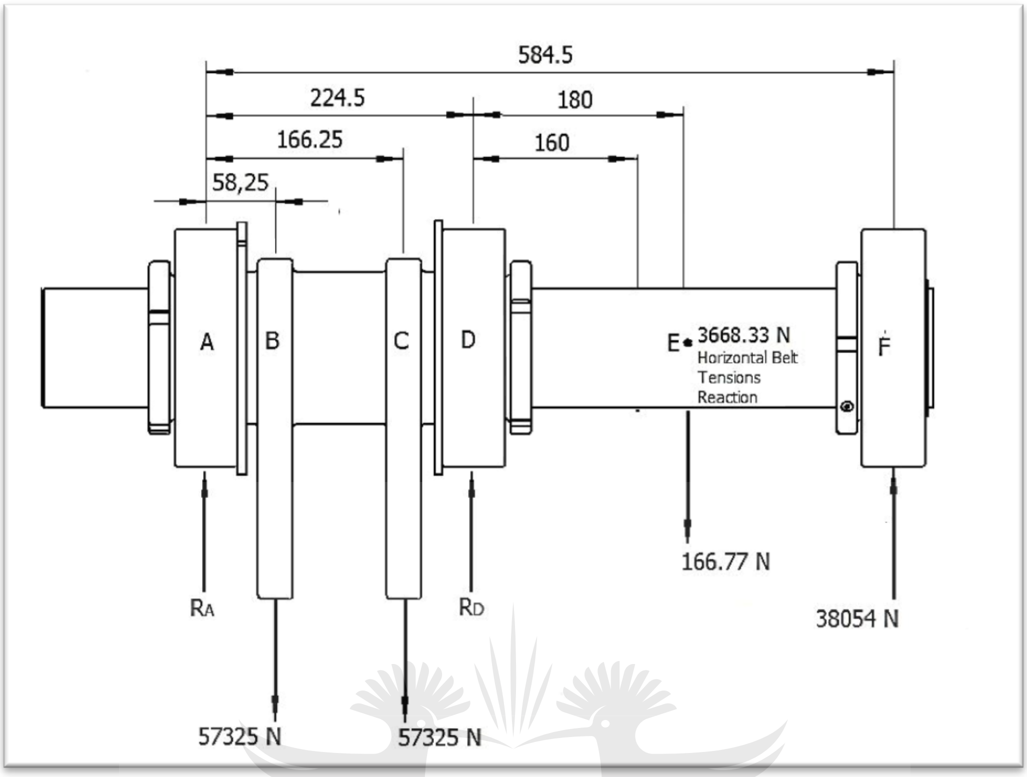


Figure 55: MS loading maximum vertical acceleration.

In Figure 55, the centrifugal force is now in the downward direction and is also a representation of the 3D configuration in Figure 53.

Taking moments at R_A:

$$\boldsymbol{R_D} \times \mathbf{224.5} = (\mathbf{38054}) \times (\mathbf{584.5}) + (\mathbf{166.25}) \times \mathbf{57325} + (\mathbf{58.25}) \times \mathbf{57325}$$
$$+ (\mathbf{224.5} + \mathbf{180}) \times \mathbf{166.77}$$
$$\therefore \boldsymbol{R_D} = \mathbf{156701.48}\ \boldsymbol{N}\ (\boldsymbol{upward, maximum\ shear\ force})$$

Taking moments at R_D:

$$(\mathbf{584.5} - \mathbf{224.5}) \times \mathbf{38055} + \mathbf{180} \times \mathbf{166.77} + \boldsymbol{R_A} \times \mathbf{224.5}$$
$$= +(\mathbf{224.5} - \mathbf{58.25}) \times \mathbf{57325} + (\mathbf{224.5} - \mathbf{166.25}) \times \mathbf{57325}$$
$$\therefore \boldsymbol{R_A} = \mathbf{3832.32}\ (\boldsymbol{downward})$$

To get the total reaction at “A” the horizontal shaft reaction due to belt tensions at “E” need to be taken into consideration. Moments will only be taken at R_D. Since the reaction at “D” is very large, the answer will not change much if a horizontal component is added of this small magnitude.

$$\boldsymbol{R_{A\,total}} = \sqrt{\boldsymbol{R_{A\,horizontal}}^2 + \boldsymbol{R_{A\,vertical}}^2} = \sqrt{\left(\frac{\mathbf{180 \times 3668.33}}{\mathbf{224.5}}\right)^2 + \mathbf{3832.32^2}}$$

$$= \mathbf{4830.87}\ \boldsymbol{N}\ (\mathbf{52.5}°)$$

The total reaction ($R_{A\ total}$) is 4830.87 N acting at 52.5º on bearing “A”. The high reaction at bearing “D” is an indication of the reaction for the MS and the above procedure will yield little change, hence the focus of the MS design is now distinctly placed on the downward force vectors of the eccentric weight that is capable of yielding the maximum shear force of 156701.48 N.

At bearing “D” where the greatest shear force has been observed, is the ideal point to calculate the maximum bending moment.

At point “E” the small pulley is attached with its 166.77 N force, the bending moment will be ignored, this is due to the small magnitude of its force when compared to the reaction at bearing “D”. The force at point “D” is the only point that needs to be considered to calculate the maximum bending moment due to its large force.

$$\boldsymbol{M_{max}} = \mathbf{38054} \times (\mathbf{0.5845} - \mathbf{0.2245}) = \mathbf{13699.44}\ \boldsymbol{Nm}$$

11.4 MS size design.

Calculations done in this section are based on:

- JE Shigley [25] shaft design method to determine the MS size which includes the approximate diameter, material selection, shear, factor of safety and ultimately the shaft final minimum diameter calculation;
- the Inventor 2013 shaft design software is utilised to determine shear force, bending moment and diameter size;
- the comparison of manual shear force, bending moment and JE Shigley [25] diameter calculations to that of the Inventor 2013 design software;
- the Inventor 2013 generated shaft size compared to the calculated JE Shigley [25] shaft size and
- the Inventor 2013 shaft diameter vs the JE Shigley [25] shaft diameter step-up percentage.

11.4.1 JE Shigley fatigue shaft design method.

The method is mainly based on fatigue design. It is a very refined method of shaft design where all needed design factors are taken into account to get a reliable final product.

The design method is very detailed and includes every possible way that might cause a shaft to fail during its lifetime. It includes material type, shear at the endurance limit, yield in shear Brinell hardness, torsion, bending, material finish, reliability, dimension factors, stress concentration due to steps, holes and grooves, manufacture, transportation, corrosion, type of loading, hub pressure factor, temperature expansion and use and abuse, etc.

11.4.1.1 MS torque determination.

The input power from the IS to the belt drive is 47.065 kW (calculated in 9.13.3) and the belt drive efficiency is 96%.

So the torque supplied to the MS is;

$$\boldsymbol{T = \frac{30 \times 0.96 \times P_{in\,belt\,drive}}{\pi \times N} = \frac{30 \times .96 \times 47065}{\pi \times 2179.5} = 197.963\ N.m}$$

11.4.1.2 Approximate diameter calculation.

By using the maximum bending moment of 13699.44 Nm already calculated in 10.3.5.3, the approximate shaft diameter can be determined.

$$\boldsymbol{d_{approx} = 6 \times (M + T)^{\frac{1}{3}} = 6 \times (13699.3 + 197.963)^{\frac{1}{3}} = 144.25\ mm \rightarrow 145\ mm}$$

With the approximated diameter known, the final diameter at "D" can be calculated after the factor of safety is determined.

11.4.1.3 Shaft material selection & shear calculations.

The design shaft material used is 080M40 heat treatment Q (Appendix 42).
The $S_{yt} = 385\ MPa$ and $S_{ut} = 625\ MPa\ to\ 775\ MPa$ with a Brinell hardness of 179 to 229. The 080M40 (EN8) steel is also suitable for shafts and keys Appendix 42.

$$\boldsymbol{\tau_Y = 0.577 \times S_{yt} = 0.577 \times 385 = 222.145\ MPa}$$

An average is assumed for the S_{ut}, because of the two different values given in Appendix 40. On purchasing of the steel, verifications have to be made so that the steel S_{ut} must not be less than 700 MPa.

$$S_{ut\ average} = \frac{625 + 775}{2} = 700\ Mpa$$

$$\tau_{EL} = 0.577 \times 0.5 \times S_{ut} = 0.577 \times 0.5 \times 700 = 201.95\ MPa$$

Where:

- τ_y = Yield in shear MPa.
- τ_{EL} = Shear at the endurance limit MPa.
- $S_{ut\ average}$ = Average ultimate tensile strength MPa.

11.4.1.4 Factor of safety determination.

The following factors below are as given by JE Shigley [25] to determine the safety factor:

$$fos = k_1 \times k_2 \times k_3 \times k_4 \times k_5$$

- Surface smoothness factor k_1 =1.333 (machined at a UTS of 600 MPa);
- Reliability factor k_2 = 1.152 (95% reliability);
- Dimension factor k_3 = 1.367 (145 mm approx. diameter);
- Stress concentration factor k_4 = 1 (No steps, grooves or holes will be machined into the shaft to avoid the increase of the shaft diameter due to stress raisers. Instead taper locks will be used for the pulley at "E" and spherical roller bearings with withdrawal sleeves will be used to clamp the bearing arrangement to the shaft.

  If shaft size increases, bearing sizes will also need to be increased to compensate for the larger shaft diameter. Since six withdrawal sleeve spherical roller bearings will be installed onto the MS's shaft, costs will increase greatly with bigger bearings.

The miscellaneous factor k_5 is broken up into various factors by J.E. Shigley i.e., press fit factor k_p, corrosion factor k_c, type of loading factor k_l and use, abuse and transportation factor k_a. There is also a temperature rise factor k_t.

Assuming of the factors for k_5:

- k_p = 1.2 (heavy press fit for eccentric weight sleeve);
- k_c = 1.1 [Corrosion can be expected due to soil moisture content and road wetting, the machine might be stored outside in the rain (not advised). A light lubrication can be applied to the shaft during storage or operation to avoid corrosion.
- k_l = 5 (For impact loading and shock, it might be decreased to a lesser value due to all the vibration or shock dampers installed in the machine).
- k_a = 1 (No abuse is foreseen during machine use. Transportation must be done by careful instructions).
- k_t = 1 (It is not advised to compact hot tar with the machine, the tar will have to cool before compaction, but since the design is specifically for rural roads, the focus is mainly placed on the construction, maintenance and upgrading of soil and gravel roads.)

$$\boldsymbol{k_5 = k_p \times k_c \times k_l \times k_a \times k_t = 1.2 \times 1.1 \times 5 \times 1 \times 1 = 6.6}$$

$$\boldsymbol{fos = k_1 \times\ k_2 \times k_3 \times k_4 \times k_5 = 1.333 \times 1.152 \times 1.367 \times 1 \times 6.6 = 13.855}$$

11.4.1.5 Minimum diameter calculation at shaft region "D".

$$\boldsymbol{d_{min} = \left[\frac{16 \times fos}{\pi} \times \sqrt{\left[\left(\frac{T}{\tau_Y}\right)^2 + \left(\frac{M}{\tau_{EL}}\right)^2\right]}\right]^{\frac{1}{3}} d_{min}}$$

$$\boldsymbol{= \left[\frac{16 \times 13.855}{\pi} \times \sqrt{\left[\left(\frac{197.963}{222.145 \times 10^6}\right)^2 + \left(\frac{13699.3}{201.95 \times 10^6}\right)^2\right]}\right]^{\frac{1}{3}}}$$

$$\boldsymbol{= 168.53\ mm \rightarrow 170\ mm}$$

For further understanding of how the diameter can be affected, if the loading factor is altered as mentioned, Table 29 indicates all other possible answers if the diameter needs to be stepped down to a smaller diameter. However, for further design purposes the 170 mm diameter will be used to cater accurately for shock loading.

Table 29: Loading factor vs minimum shaft diameter.

k_l (loading factor)	k_5 (miscellaneous factor)	fos (factor of safety)	d_{min} (Minimum diameter required calculated)	d_{min} (Minimum diameter required converted)
5	6.6	13.855	168.53 mm	170 mm
4.5	5.94	12.469	162.71 mm	165 mm
4	5.28	11.084	156.45 mm	160 mm
3.5	4.62	9.698	149.64 mm	150 mm
3	3.96	8.313	142.15 mm	145 mm

11.4.2 Inventor 2013 MS design.

Shaft design using Inventor 2013 has yielded the results shown henceforth. Not all results are included in the following calculations and simulations. To see all the relevant results Appendix 27 need to be consulted.

All forces in Figure 56 have been exactly placed as shown in Figure 55.

Three diagrams are shown:

- Figure 56 (Free body Diagram)
- Figure 57 (Shear Force Diagram SFD),
- Figure 58 (Bending Moment Diagram BMD) and
- Figure 59 (Ideal Diameter Diagram IDD).

The data is only enough to make an accurate comparison between the manual JE Shigley [25] method and the simulated Inventor 2013 shaft design method.

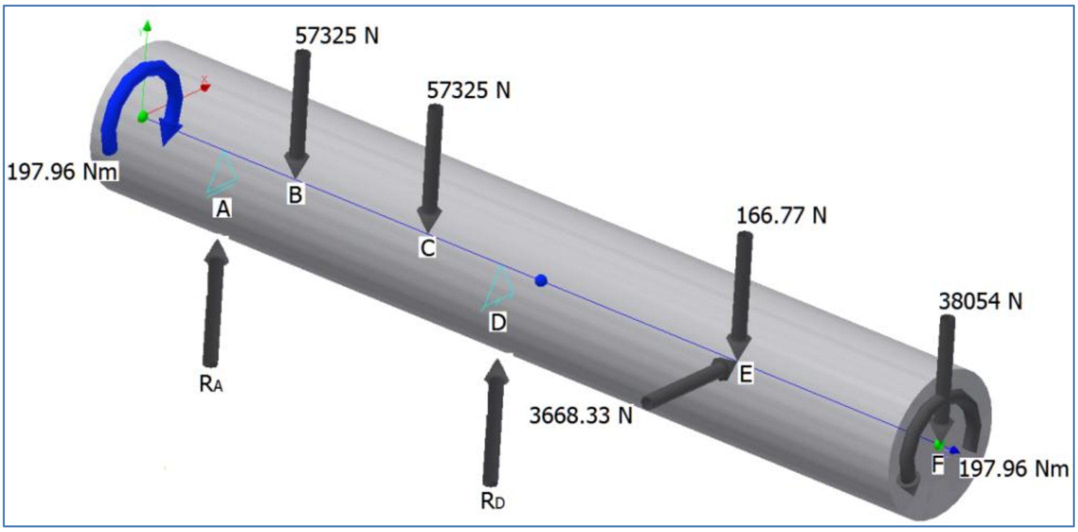


Figure 56: MS Loading 1

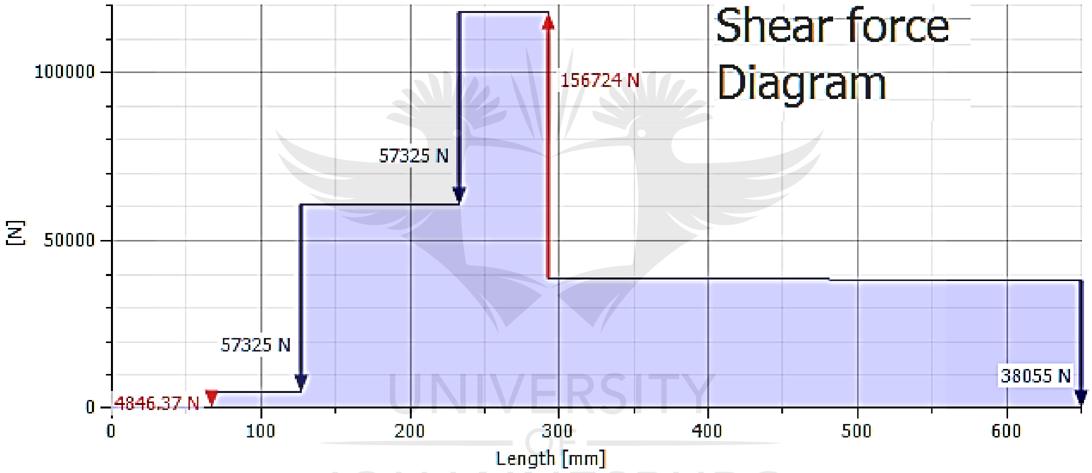


Figure 57: Shearforce Diagram 1, MS

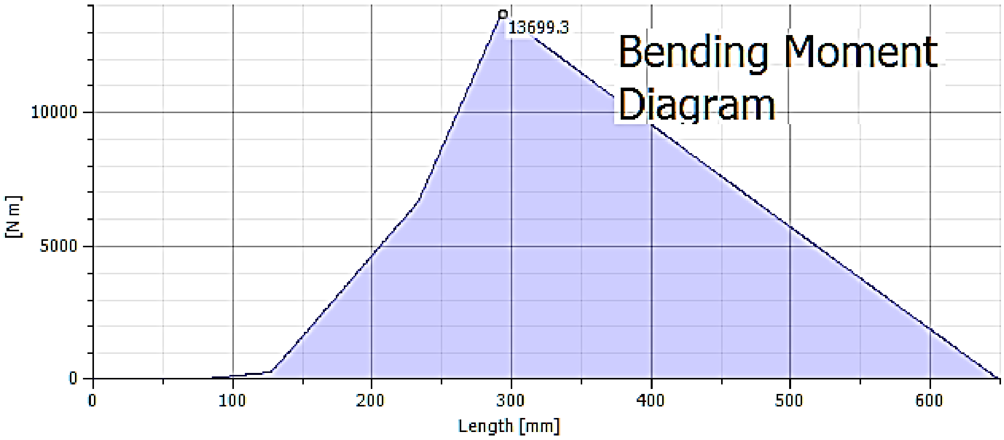


Figure 58: Bending Moment Diagram 1, MS

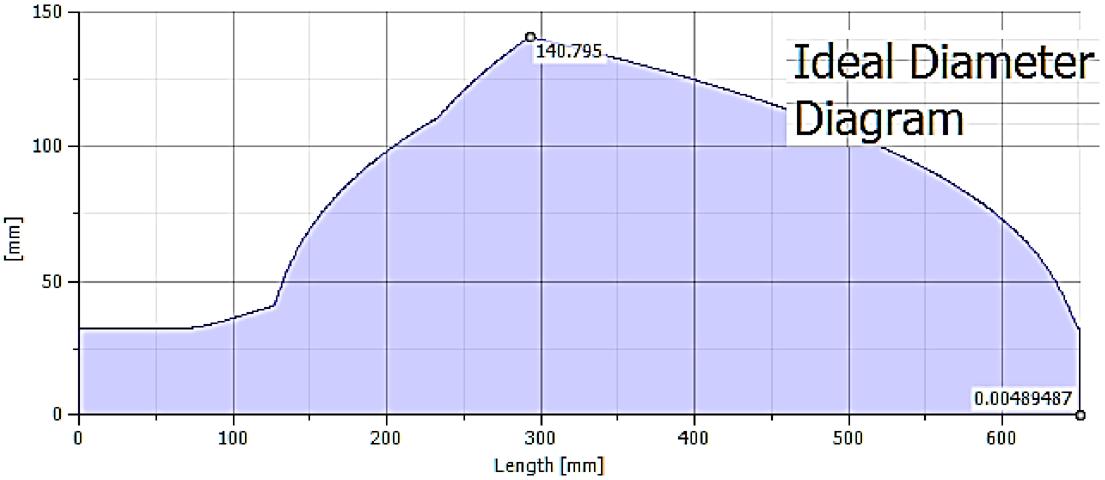


Figure 59: Ideal Diameter Diagram 1, MS

11.4.3 Comparison of Inventor 2013 shaft design and manual shaft calculations.

Shear Force comparison at shaft regions "D" and "A".

The manual calculations yielded the following: $R_A = 4830.87$ N

$R_D = 156701.48$ N

Inventor 2013 shaft design calculations (Figure 57): $R_A = 4846.37$ N

$R_D = 156724$ N

The shear force at region "D" for both design methods is very close, indicating correlation between the two methods of calculating shear force.

Bending Moment comparison at shaft region "D".

The manual calculations yielded the following: $M_D = M_{max} = 13699.44$ Nm

Inventor 2013 shaft design calculations (Figure 58): $M_D = M_{max} = 13699.3$ Nm

The bending moments complement one another with a similar result at shaft region "D".

Shaft Diameter comparison at region "D".

The manual calculations yielded the following: $\boldsymbol{d_{min} = 168.53\,mm \rightarrow 170\,mm}$

Inventor 2013 shaft design calculations (Figure 59): $\boldsymbol{d_{ideal} = 140.795\,mm \rightarrow 145\,mm}$

Inventor 2013 suggests a shaft diameter at region “D” of 140.795 mm → 145 mm (Rounded off to the nearest upper 5 mm due to the fact that most manufactured shaft sizes is found in increments of 5 mm, unless the shaft diameter is very small.)

NB: To get an idea by what percentage the two diameters differ and to avoid the long manual laborious method of calculating the minimum shaft diameter for future shaft calculations, the Inventor 2013 Ideal Diameter (ID) can be used and then stepped up by the difference in percentage, this will yield an accurate approximation of the shaft size for the JE Shigley [25] method.

$$\boldsymbol{Shaft\ step\ up}\ \% = \left(\mathbf{1} - \frac{\mathbf{140.795}}{\mathbf{168.53}}\right) \times \mathbf{100} = \mathbf{16.457}\ \%$$

The calculated percentage can now be applied, by multiplying the above percentage as a ratio of 1.16457 by the Inventor 2013 ID for further shaft design purposes.

NB: The calculated 170 mm shaft diameter is too high and needs to be stepped down to a more efficient size that will alleviate the need for larger bearings and other related components.

11.4.4 MS diameter refinement and size recalculation.

Due to the large moment the compactor frame is causing on the MS at bearing region “F” (Figure 55), the reaction at “A” (4846.37 N) and “D” (156701.48 N) Figure 54, has a great force difference due to the large bending moment. The only way to alleviate the bending moment the frame is creating, is to move the belt drive which is on the inside of the compactor frame to the outside of the compactor frame.

The aim is to move the frame as close as possible to Bearing D and thus decrease the large shear force (SF) reaction at Bearing D and ultimately the bending moment. This will ensure a smaller shaft diameter in the end.

11.4.4.1 Decreasing of the compactor frame bending moment.

Figure 59 shows the 3D drawing for the rearrangement of the MS components. A decreased distance between the compactor frame Bearing at “E” and Bearing “D” is created by carefully calculating assembly distances. The belt drive at “F” is now located on the outside of the frame, see Figure 60.

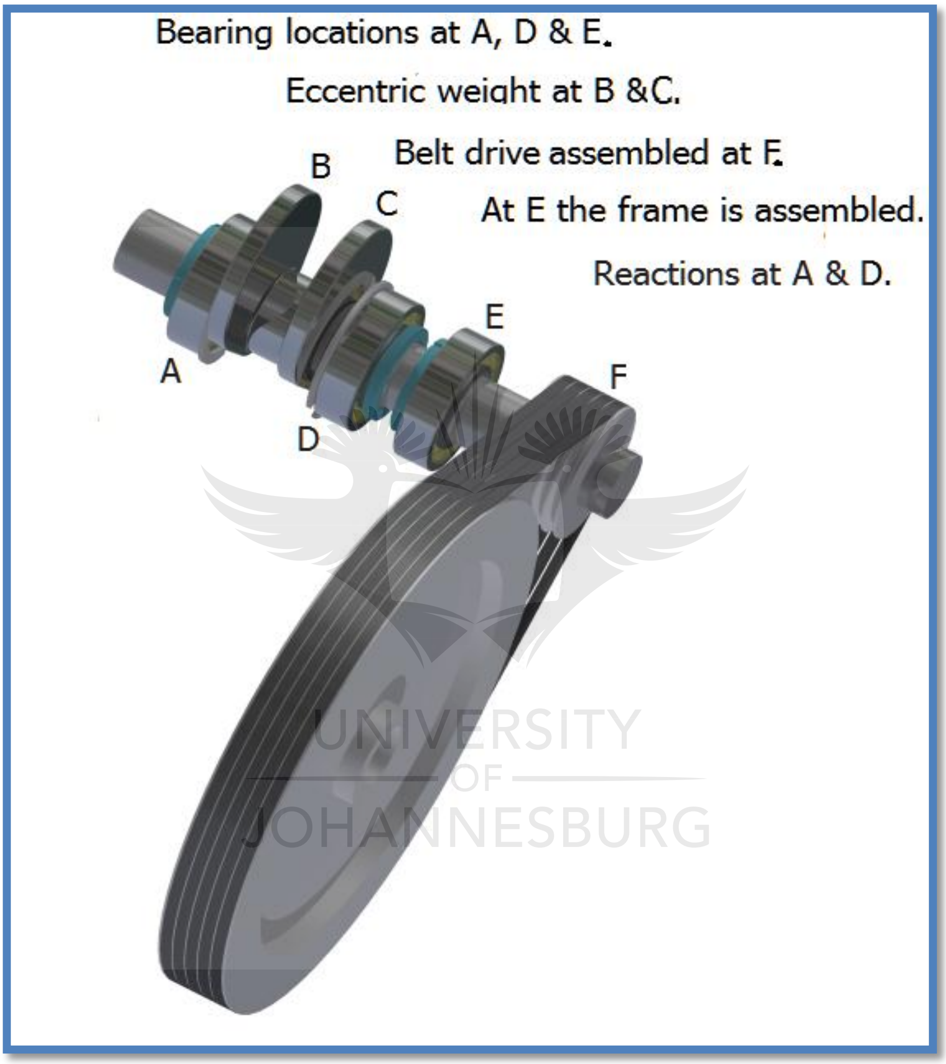


Figure 60: MS Rearrangement

11.4.4.2 Inventor 2013 design results for the second Ideal Diameter attempt.

The shaft diameter can now be re-evaluated by checking the ID in Inventor 2013 first. The ID (93.3479 mm) obtained from Figure 64 below, can be stepped up by the calculated ratio (1.16457) in 11.4.3. For the full simulation results for the MS second attempt, consult Appendix 28.

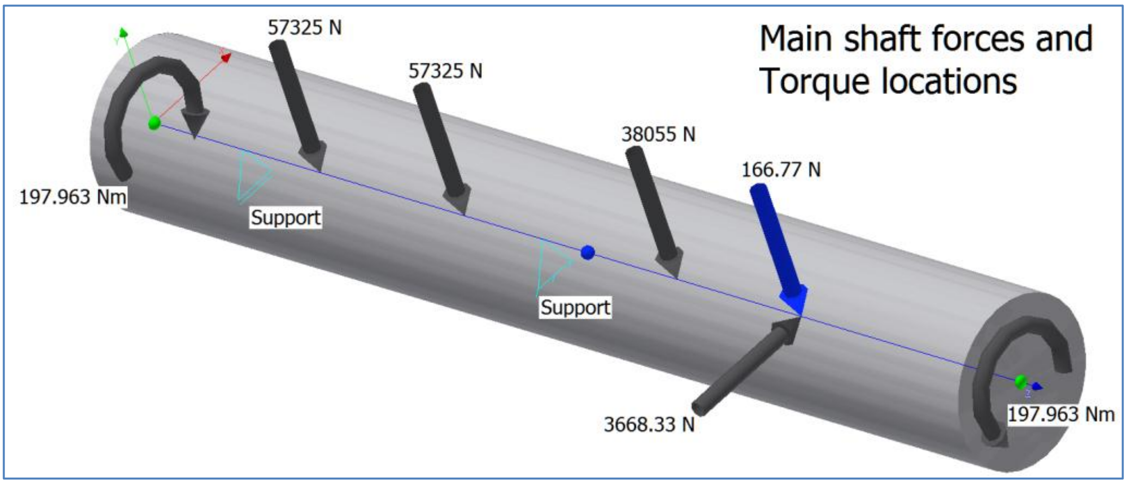


Figure 61: MS Loading 2

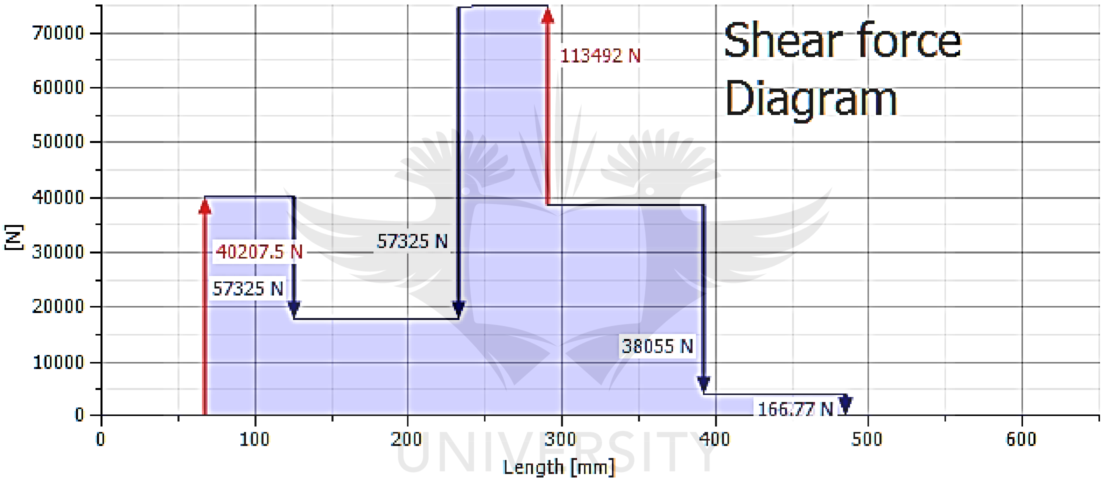


Figure 62: Shear Force Diagram 2, MS

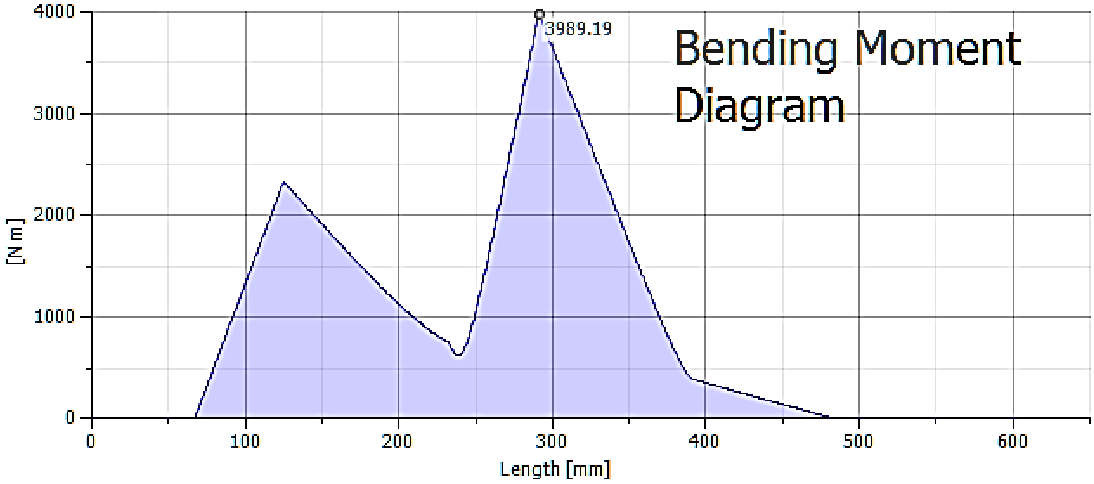


Figure 63: Bending Moment Diagram 2, MS

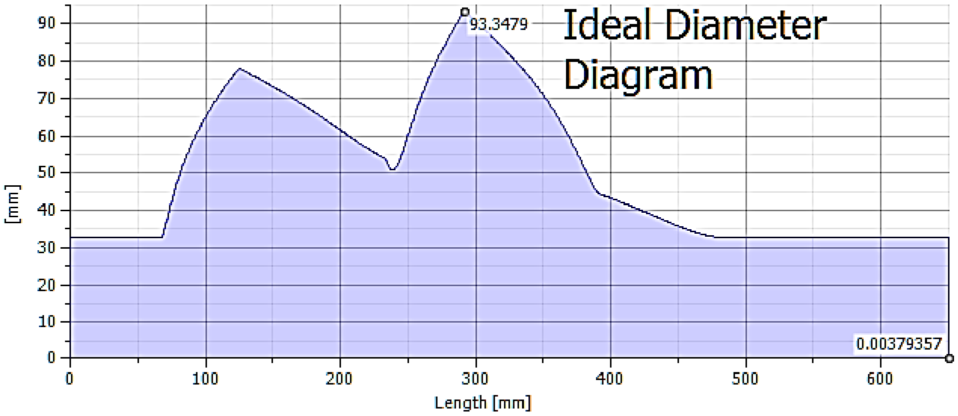


Figure 64: Ideal Diameter Diagram 2, MS

- In the SF Diagram (Figure 62) there is an improvement of the reaction at "$\boldsymbol{D}$" of $\mathbf{156701.48 - 113492 = 43209.5}\ \boldsymbol{N}$ less, much of the load is transferred to "A" thus receiving $\mathbf{40207.5 – 4846.37 = 34361.113}\ \boldsymbol{N}$ more load.

- The BM Diagram (Figure 63) also has an excellent improvement from 13699.3 Nm to 3989.19 Nm. By bringing the frame closer to bearing "D", a better result was achievable.

- The ID Diagram (Figure 64) reveals a much smaller shaft diameter at Bearing "D" where the maximum BM and maximum SF are located. The diameter has reduced from 140.795 mm to 93.348 mm with a difference of 47.447 mm.

11.4.4.3 Final MS diameter conversion and calculation.

As proven in 10.4.3, the JE Shigley [25] diameter is 1.16457 much larger than the ID yielded by the Inventor 2013 software.

$$\boldsymbol{d_{min} = d_{ideal} \times \%upgrade = 93.3479 \times 1.16457 = 108.71mm \rightarrow 110\ mm}$$

The 110 mm shaft diameter is the final MS diameter without the inclusion of a keyway.

<u>Investigation of a keyway machined into the MS</u>

The final shaft size is then 110 mm, however to compensate for a keyway, stress concentration needs to be taken into account.

NB: In Figure 64 the 93.3479 mm ID, before step-up, is only applicable to shaft region "D" about 290 mm from the LHS. It is important to note that the pulley is attached 480 mm from the shaft LHS and the ID given by Inventor 2013 at this point is about 32.5 mm, say 33 mm.

This 33 mm ID is the diameter that needs to be used in the calculation of shaft stress concentration and not the rounded of value of 110 mm. This technique is used to avoid unnecessary inflation of the shaft size, rounding of the shaft size will be done after keyway stress concentration has been calculated.

Stepping up of the 33 mm ID to the JE Shigley [25] shaft size yields;

$$\boldsymbol{dmin = 33 \times 1.16457 = 38.43\, mm}$$

The formula below is a simplified formula that is used to increase the minimum designed shaft area by 15% to compensate for the stress disturbance at the considered keyway, the formula has been simplified to yield final keyway diameter instead of area.

$$\boldsymbol{d_{new} = d_{min} \times \sqrt{1.15} = 38.43 \times \sqrt{1.15} = 41.21\, mm \rightarrow 45\, mm}$$

The calculation indicates that the shaft region where the keyway will be machined into, only requires a final diameter of 45 mm, thus 110 mm is adequate to be used as a solid smooth shaft throughout and no further shaft increase is needed.

Investigation of the maximum bush size for the small MS pulley.

In Appendix 13 the pulley tables indicate that the maximum diameter for bush number 4030 is 75 mm and therefore will not be able to accommodate the required 110 mm diameter shaft size.

This is an indication that the shaft size needs to be stepped down from 110 mm to 75 mm at the pulley region. Stress concentration for the step can be ignored for the following reasons;

1. the shaft will be stepped down with an appropriate radius to allow for a smoother stress flow from the 110 mm diameter to the 75 mm diameter and thus alleviate stress at the stepped region and

2. the required diameter as quantified in the strength calculation previously is only 45 mm after keyway stress concentration has been taken into account and 75 mm is more than enough to justify the shaft strength at this region, thus the stress concentration created at this region, can be ignored based on discussions 1 and 2.

NB: Hence the final MS size will be 110 mm stepped down via a 5 mm radius to 75 mm to accommodate the pulley bush.

11.5 IS design.

For the IS diameter, the Inventor 2013 simulation will be first and then the step-up percentage method will be applied to yield the final result as previously discussed. No manual calculations will be attempted as in the MS section.

In section 11.4.3 it was proven that the Inventor software is very accurate in comparison to manual calculations, when it comes to the SF and BM calculations.

Located on the IS (Figure 65) is the larger size pulley of 807 mm diameter and the smaller size sprocket of 233.17 mm pitch diameter (refer to Appendix 9–6 for pulley pitch diameters).

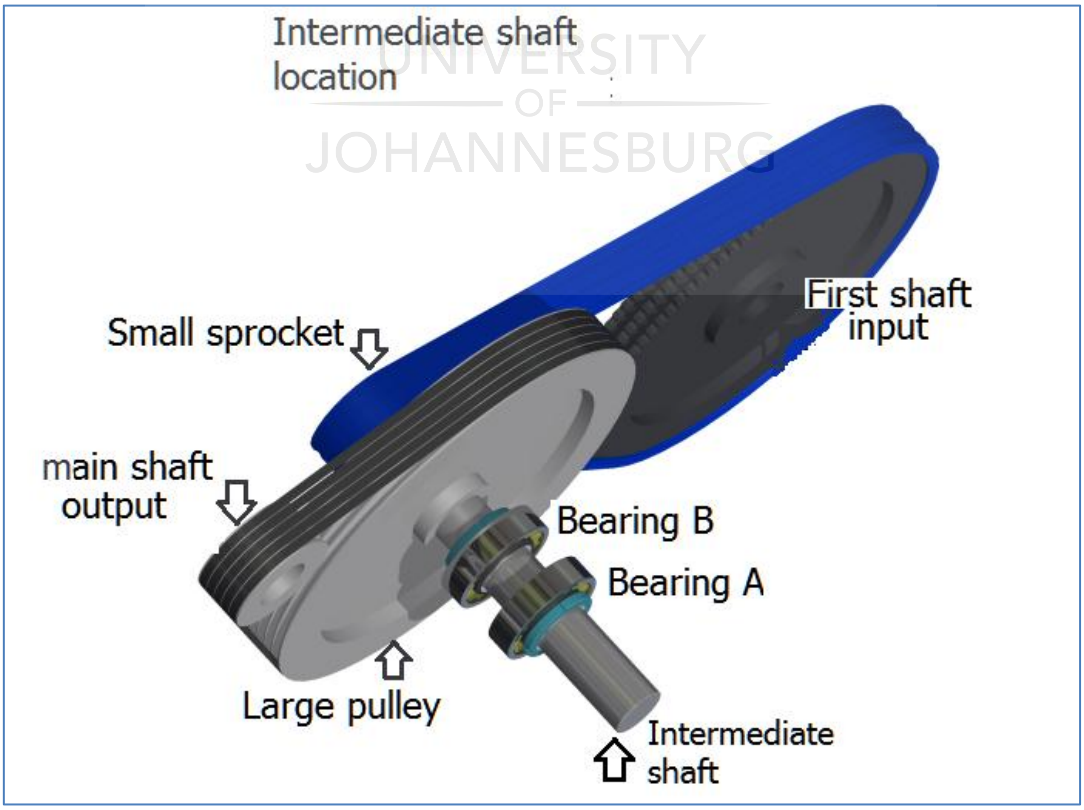


Figure 65: IS location.

Both the larger pulley and small sprocket, including their portions of belt and chain wrapped around them will add to the radial loads at that point on the IS. The weight of the pulley and sprocket, plus the inertia forces caused by the vibration amplitude, inclusive of the horizontal force vectors caused by belt and chain tensions will yield the final reactions at the bearings A and B.

11.5.1 IS component masses

11.5.1.1 Big pulley mass calculation

In Appendix 11, the table for the larger pulley indicates a diameter of 807 mm and a width of 101 mm which is a pulley type 4 with five grooves. The profile of the type 4 pulley shows that a lot of mass has been removed to make it lighter. By assuming a solid disk and then subtract 35 % of the pulley mass for the grooves, shaft diameter hole and side material removal, a proper estimate can be obtained.

$$\boldsymbol{Big\ pulley\ mass = V_{big\ pulley} \times \rho_{steel} \times 0.65 = \frac{0.807^2 \times \pi}{4} \times 0.101 \times 7850 \times 0.65}$$

$$\boldsymbol{= 263.6\ kg\ say\ 270\ kg}$$

Where:

- $V_{big\ pulley}$ = Volume of big pulley m^3.
- ρ_{steel} = Density of steel kg/m^3.

11.5.1.2 Small sprocket mass calculation

In Appendix 9-6, the following dimensions can be read from the 20B plate wheel sprocket table.

The width (B3) = 91mm, no. of teeth = 23, de = 233.17 mm.

The sprocket profile also indicates mass removal to make it lighter. Assuming a solid disk as well, as in the big pulley mass calculation and that 30% of the mass is removed due to its smaller size.

$$\boldsymbol{Small\ sprocket\ mass = V_{small\ sprocket} \times \rho_{steel} \times 0.7}$$

$$\boldsymbol{= \frac{0.23317^2 \times \pi}{4} \times 0.091 \times 7850 \times 0.65 = 21.35\ kg\ say\ 22\ kg}$$

Where:

- $V_{small\ sprocket}$ = small sprocket volume m^3.
- ρ_{steel} = density of steel kg/m^3.

11.5.2 IS loading

10.5.2.1 Big pulley loading

The calculated maximum acceleration in 11.2.5 is 55.475 m/s^2. The total vertical radial force on the shaft is as calculated below for the large pulley:

$$\boldsymbol{F_{big\ pulley} = m \times a_{max} + m \times g = 270(55.475 + 9.81) = 17627\ N\ (vertical)}$$

Where:

- $F_{big\ pulley}$ = Total force on the big pulley due to inertia and weight N.
- m = mass of the big pulley kg.
- a_{max} = maximum vertical acceleration as quantified in 11.2.5 m/s^2.
- g = gravitational pull m/s^2.

The horizontal radial force due to belt tension was quantified in 11.3.3 as 3668.33 N on the small pulley and the reaction on the larger pulley will be exactly the same at 3668.33 N due to the reactive force.

10.5.2.2 Small sprocket loading

The total vertical radial force caused by the small sprockets mass is;

$$\boldsymbol{F_{small\ sprocket} = m \times a_{max} + m \times g = 22(55.475 + 9.81) = 1436.27\ N\ (vertical)}$$

Where:

- $F_{small\ sprocket}$ = Total force due to inertia and weight N.
- m = mass of the sprocket kg.
- a_{max} = maximum vertical acceleration as quantified in 11.2.5 m/s^2.
- g = gravitational pull m/s^2.

The horizontal force due to chain pull that is acting on the small sprocket is quantified below. The pitch diameter is d_{pitch} = 233.17 mm, Appendix 2 chain drive efficiency = 98%, the input power to the chain drive is 47.065 kW (calculated in 10.10.1) at the larger sprocket and the output speed of the chain drive is 545 *rpm* (calculated in 10.14.4.1).

The sprocket tangential velocity is,

$$\boldsymbol{v = \frac{\pi \times d_{pitch} \times N}{60} = \frac{\pi \times 0.22317 \times 545}{60} = 6.654\ m/s}$$

The input power to the smaller sprocket is,

$$\boldsymbol{P_{in\ small\ sprocket} = P_{in\ chain\ drive} \times drive\ efficiency = 47.065 \times 0.98 = 46.124\ kW}$$

Where:

- $P_{in\ small\ sprocket}$ = input power to the small sprocket kW.
- $P_{in\ chain\ drive}$ = input power to the larger sprocket kW.
- Drive efficiency = chain drive power transfer efficiency.

The total horizontal radial force experienced by the chain, is the centrifugal force due to the chain mass and the chain tension.

The difference in chain tensions is:

$$\boldsymbol{P_{in\ small\ sprocket} = F \times v \therefore F = \frac{46124}{6.654} = 6932.001\ N}$$

The chain angular velocity is,

$$\boldsymbol{\omega = \frac{2 \times \pi \times 545}{60} = 57.072\ rad/s}$$

Where:

- F = the force caused by chain tension N
- v = the tangential velocity of the sprocket m/s
- ω = rotational velocity in rad/s

The chain length is 4.4 kg/m in Appendix 9-8 and has a minimum tensile strength of 190 kN for a 31.75 mm pitch.

The centrifugal force caused by a portion of the chain length wrapped at 146.18 degrees (Figure 66) need to be taken into consideration around the small sprocket as well.

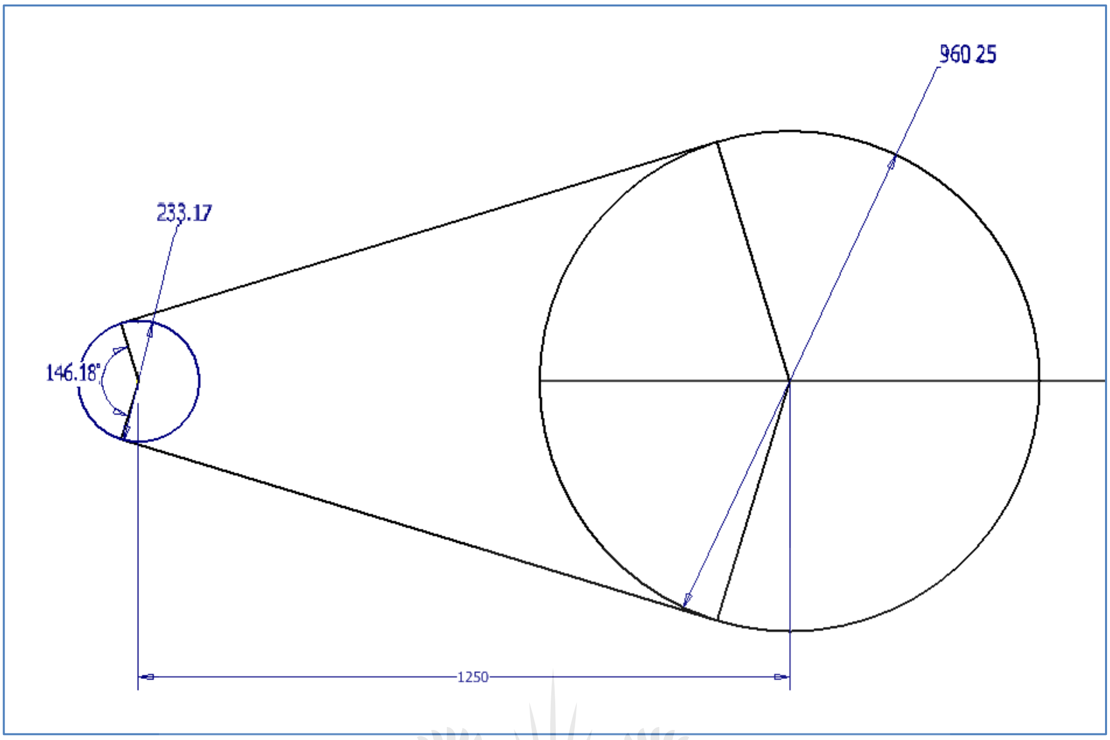


Figure 66: Small pulley contact angle.

- *The mass of the chain portion subjected to the centrifugal force is,*

$$\boldsymbol{m_{chain}} = \frac{\boldsymbol{Contact\ arc}}{\mathbf{57.3}} \times \frac{\boldsymbol{d_{pitch}}}{\mathbf{2}} \times \boldsymbol{m'} = \frac{\mathbf{146.18}}{\mathbf{57.3}} \times \frac{\mathbf{0.23317}}{\mathbf{2}} \times \mathbf{4.4} = \mathbf{1.3087}\ \boldsymbol{kg}$$

- *The centrifugal force is,*

$$\boldsymbol{F_{centrifugal}} = \frac{\boldsymbol{m_{chain}} \times \boldsymbol{d_{pitch}} \times \boldsymbol{\omega}^2}{\mathbf{2}} = \frac{\mathbf{1.3087 \times 0.23317 \times 57.072^2}}{\mathbf{2}}$$

$$= \mathbf{497}\ \boldsymbol{N}\ (\boldsymbol{horizontal})$$

- *The maximum chain tension experienced by the chain is,*

$$\boldsymbol{F_{total}} = \boldsymbol{F} + \boldsymbol{F_{centrifugal}} = \mathbf{6932.001 + 497 = 7429}\ \boldsymbol{N}\ (\boldsymbol{horizontal})$$

- *The torque on the smaller sprocket is,*

$$\boldsymbol{Torque\ small\ sprocket} = \frac{\mathbf{30} \times \boldsymbol{P}}{\boldsymbol{\pi} \times \boldsymbol{N}} = \frac{\mathbf{30 \times 46124}}{\boldsymbol{\pi} \times \mathbf{545}} = \mathbf{808.17}\ \boldsymbol{Nm}$$

Where :

- m_{chain} = chain portion mass effected by the centrifugal force kg.
- d_{pitch} = Small sprocket pitch diameter m.
- *Contact arc* = angle of chain wrap in degrees.
- $\boldsymbol{\omega}$ = angular velocity rad/s

11.5.2.3 IS loading Inventor 2013 design verification.

The final answer to the chain drive loading was the addition of the centrifugal force caused by the chain and the difference in chain tension that yielded the maximum chain tension in turn. In Appendix 9-8, an Inventor 2013 chain drive design is also shown, this was to verify the manually calculated results.

The exact same parameters were typed into the software, such as power, ratio, pulley sizes, speed, 20B chain pitch selection, triplex option, required life, driver, driven, etc and were indicated by the software as a design success.

It was observed that the following results were visible (See Table 30):

Table 30: Maximum chain tension manual calculation vs Inventor 2013 calculation.

	Manual Calculation	Inventor 2013
Centrifugal Force	497 N	487 N
Chain Tension	6932.001 N	6932.005 N
Maximum (Chain Tension)	7429.001 N	7419.005 N
Reference	11.5.2.2	Appendix 9-8

Table 30 confirms that there is correlation between the manual calculation and the Inventor 2013 design method. Maximum chain tension for the manual calculation is found to be 7429.001 N and the Inventor 2013 design method's maximum chain tension is 7419.005 N, only 10 N difference. The confirmation indicates design success.

11.5.2.4 IS design

Figure 67 indicates how the IS experiences loading and Figure 65 can be consulted for further clarity on the position of the small sprocket and larger pulley.

The 17627 N is caused by the larger pulley inertia and weight, while the 3608.33 N is caused by the larger pulley belt tensions.

NB: For all other relevant IS simulation results, refer to Appendix 29.

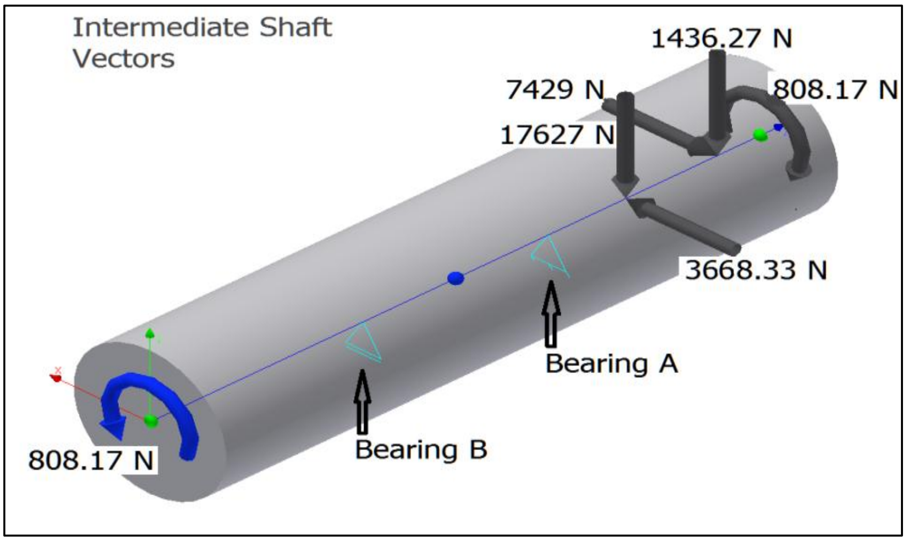


Figure 67: IS loading

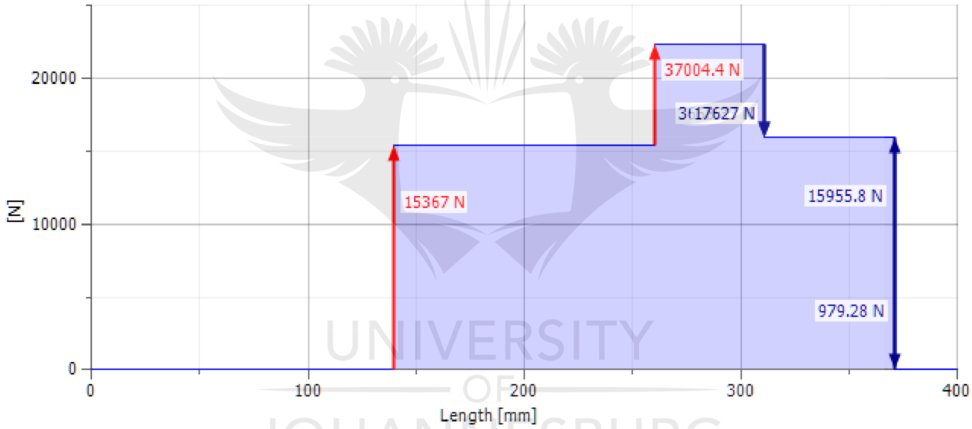


Figure 68: Shear Force Diagram, IS

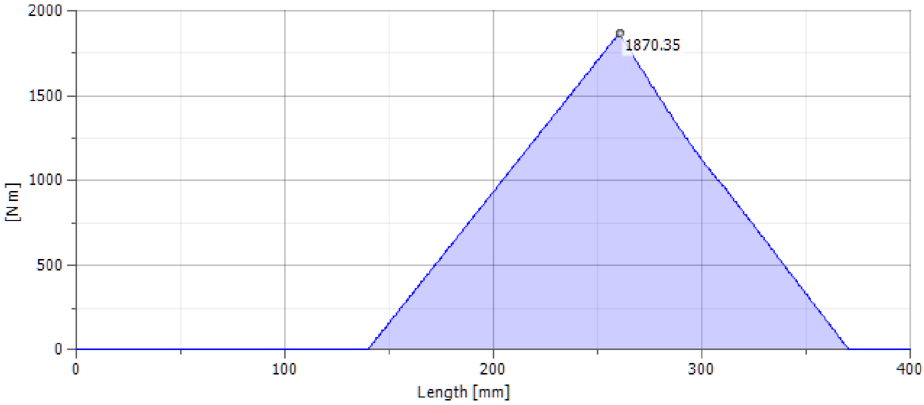


Figure 69: Bending Moment Diagram, IS

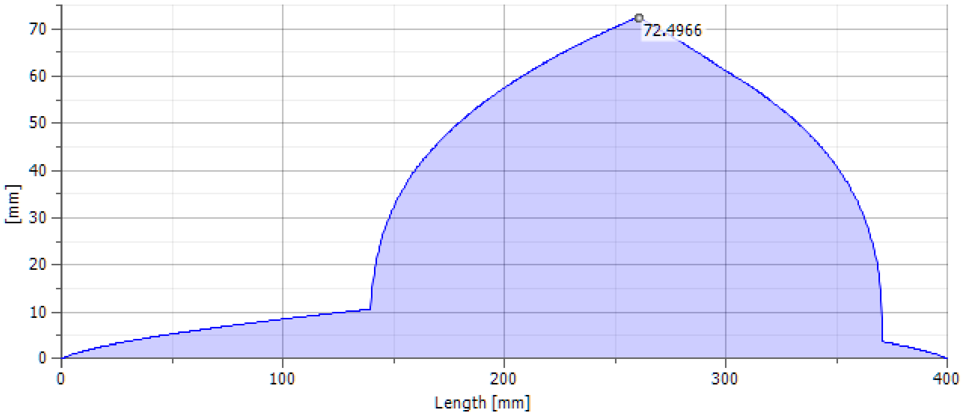


Figure 70: Ideal Diameter Diagram, IS

By studying the IS Inventor 2013 design data, the following is noted;

- Figure 68 – IS SF diagram, the reactions at Bearing A & Bearing B is 15367 N and 37004.4 N respectively. The maximum shear force 37004.4 N is acting at a distance of 260 mm on the shaft from the LHS.
- Figure 69 – BM diagram, the maximum bending moment is 1870.35 Nm at 260 mm on the shaft from the LHS.
- Figure 70 - The ID is now found to be 72.5 mm, also at a distance of about 260 mm on the shaft from the LHS.

11.5.2.4 IS final size quantification

As discussed in 10.4.4.3, IS sizes can be stepped up by 16.457 % to accommodate for the more detailed JE Shigley [25] shaft design. The 20B taper lock sprocket requires a key to be installed and as a result will cause stress concentration on the IS keyway region.
The ID diameter in Figure 70 is 72.5 mm and stepping it up will yield:

$$\boldsymbol{d_{min} = d_{ideal} \times \%upgrade = 72.5 \times 1.16457 = 84.43\ mm \rightarrow 85\ mm}$$

<u>Key stress concentration investigation.</u>

As discussed in 11.4.4.3, the shaft regions where the large pulley and small sprocket is located, needs to be investigated for stress concentration and not the region where the max diameter is needed. This is to avoid unnecessary shaft increase.

<u>Large pulley</u>

Figure 70 gives the shaft diameter where the large pulley is located to be about 58 mm.

$$d_{new} = d \times upgrade\% \times \sqrt{1.15} = 58 \times 1.16457 \times \sqrt{1.15} = 72.43\ mm\ \rightarrow 75\ mm$$

Small sprocket

Figure 70 gives the shaft diameter where the small sprocket is located as about 34 mm.

$$d_{new} = d \times upgrade\% \times \sqrt{1.15} = 34 \times 1.16457 \times \sqrt{1.15} = 42.46\ mm\ \rightarrow 45\ mm$$

The determined 85 mm IS diameter above is adequate to be used throughout the shaft, as the shaft regions where keyway stress concentrations occur, requires smaller shaft sizes.

Investigation of maximum bush diameter and shaft hole diameter.

Large pulley bush diameter

In Appendix 13 the pulley selection tables indicate that the maximum diameter for bush number 4030 is 115 mm. The IS size was calculated as 85 mm which is less than the 315 mm diameter and therefore it is acceptable.

Pulley specification:

Part Number = 031B0415
Bush Number = 4020 – 80 mm diameter
Number of grooves = 5
Pulley type 4
Outside diameter = 807 mm
Pitch diameter = 800 mm
Centre distance = 530 mm

Small sprocket bore

The small 23 teeth, 233 mm diameter plate wheel sprocket in Appendix 9-6 comes in a stock bore diameter of 25 mm basic size, but it can be drilled to suit the 85 mm final shaft diameter.

NB: The 85 mm shaft diameter is the absolute final design diameter throughout the IS.

Sprocket specification:

20B Pitch (Plate-wheel sprocket)
Number of teeth = 23
Outer diameter = 248.3 mm
Pitch diameter = 233.17 mm
Triplex sprocket
Centre distance = 1200 mm

11.6 FS design

Located on the FS (Figure 71) is the larger size sprocket of 960.25 mm pitch diameter and the smaller size sprocket of 233.17 mm pitch diameter.

The weight of the larger sprocket and its inertia will cause a vertical force due to the vibration. The maximum chain pull (7429.001 N) for the larger sprocket is due to the reactive maximum chain pull (7429.001 N) on the smaller sprocket and will act horizontally.

On the other end of the shaft is the F200 type F Fenner flex coupling (Appendix 14), the coupling will be assumed to transfer half its mass to the FS on the left-hand side.

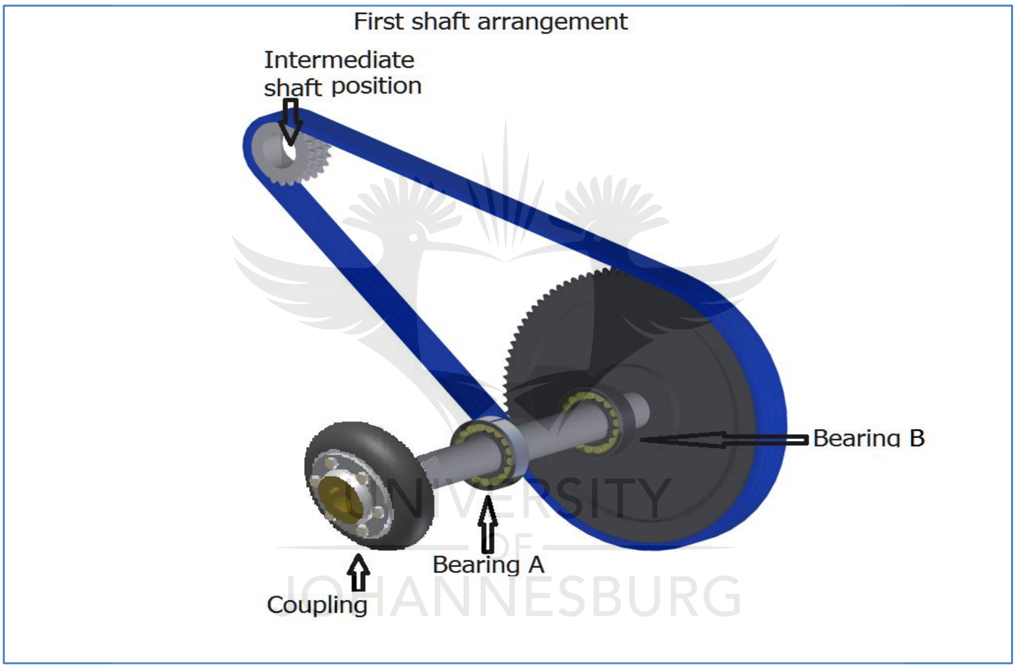


Figure 71: FS drive arrangement.

11.6.1 FS component mass calculation

11.6.1.1 Larger sprocket mass

In Appendix 9-6 the following dimensions can be read from the 20B plate wheel sprocket table (width B3 = 91 mm, teeth = 95, de = 960.25 mm).

By assuming a solid disk and then subtracting the empty space on the sprocket, the mass is estimated. The larger sprocket plus its chain mass is estimated at 70 % of the total solid disk weight. The empty space will then occupy 30 % of the mass.

The large sprocket mass is then,

$$\boldsymbol{Large\ sprocket\ mass = V_{sprocket} \times \rho_{steel} \times 0.70}$$

$$\boldsymbol{= \frac{0.96025^2 \times \pi}{4} \times 0.091 \times 7850 \times 0.70 = 362.13\ kg\ say\ 365\ kg}$$

Where:

- $V_{sprocket}$ = the sprocket volume in m^3.
- ρ = density of steel in kg/m^3.

11.6.1.2 Shaft Loading

There are two components creating forces on the shaft, the first is half of the Fenner Flex F200-F coupling where the one side is on the gearbox shaft side and the other on the FS. The coupling has a mass of 56.6 kg (Appendix 14). On the RHS of the shaft (Figure 71) the big sprocket with part of its chain resting 360° - 146.18° = 213.82° around its circumference.

The vertical radial force on the sprocket is,

$$\boldsymbol{F_{large\ sprocket} = m \times a + m \times g = 365(55.475 + 9.81) = 23828.66\ N\ (vertical)}$$

The vertical force for the coupling is (Appendix 14),

$$\boldsymbol{F_{coupling} = \frac{m}{2} \times (a + g) = \frac{53.6}{2}(55.475 + 9.81) = 1749.64N\ (vertical)}$$

The horizontal reaction is calculated in 11.5.2.2 on the smaller sprocket as 7429.001 N and will have the same force magnitude on its larger counterpart. The following data was already calculated in other sections;

$\boldsymbol{P_{in\ large\ sprocket} = 47.065\ kW}$ (Calculated in 10.10.1)
Sprocket speed (N) = 131.93 r/min (Calculated in 10.8.4)

The larger sprocket torque is,

$$\boldsymbol{Torque\ large\ sprocket\ = \frac{30 \times P}{\pi \times N} = \frac{30 \times 47065}{\pi \times 131.93} = 3406.64\ Nm}$$

11.6.2 FS Inventor 2013 design.

Figure 72 indicates the coupling weight and inertia force (1749.64 N) on the LHS and on the RHS the vertical radial force (23828.66 N) caused by the larger sprocket inertia and weight, the 7429 N force is the horizontal reaction due to chain pull. The torque is shown on both sides of the shaft as 3406.64 Nm. Bearing A and bearing B is resistive towards all indicated vectors. Consult Appendix 30 for the full simulation results for the FS.

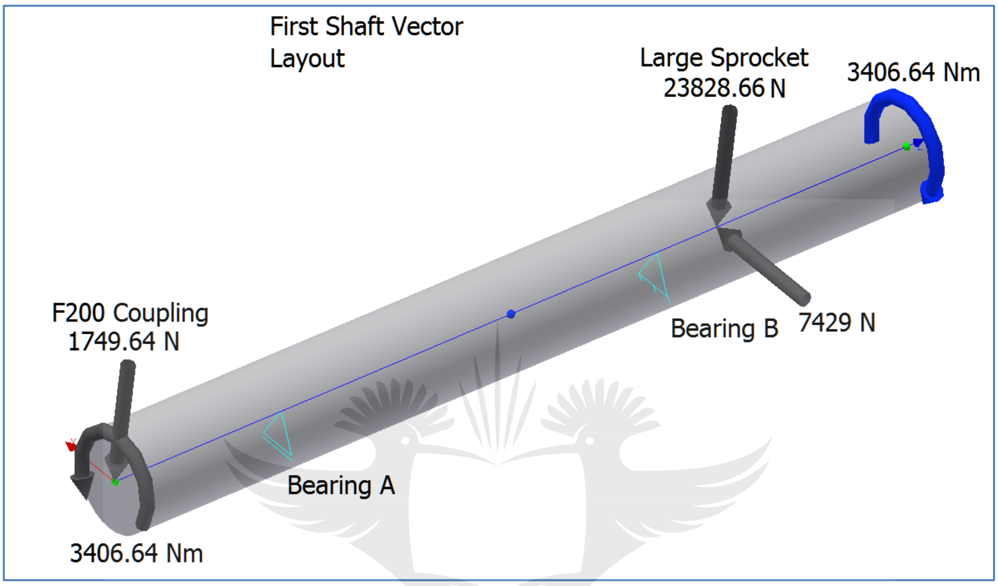


Figure 72: FS Loading

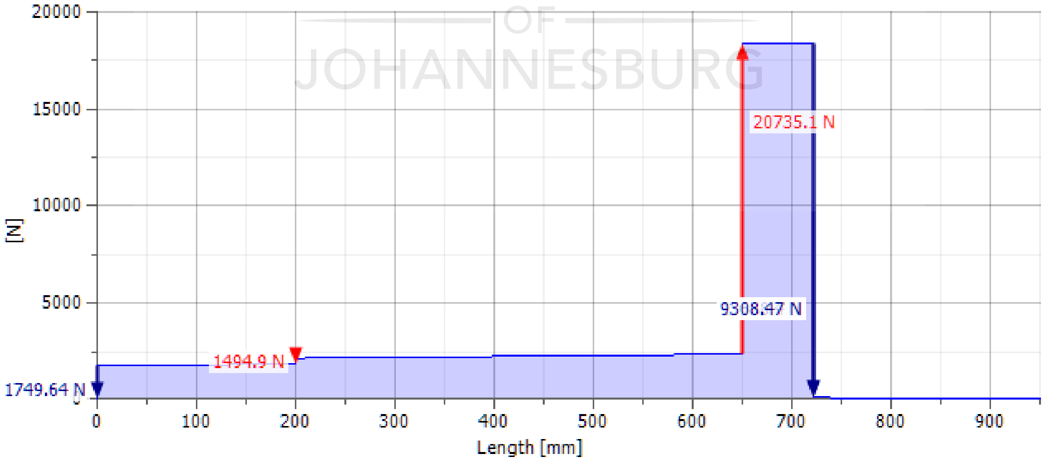


Figure 73: Shear Force Diagram, FS

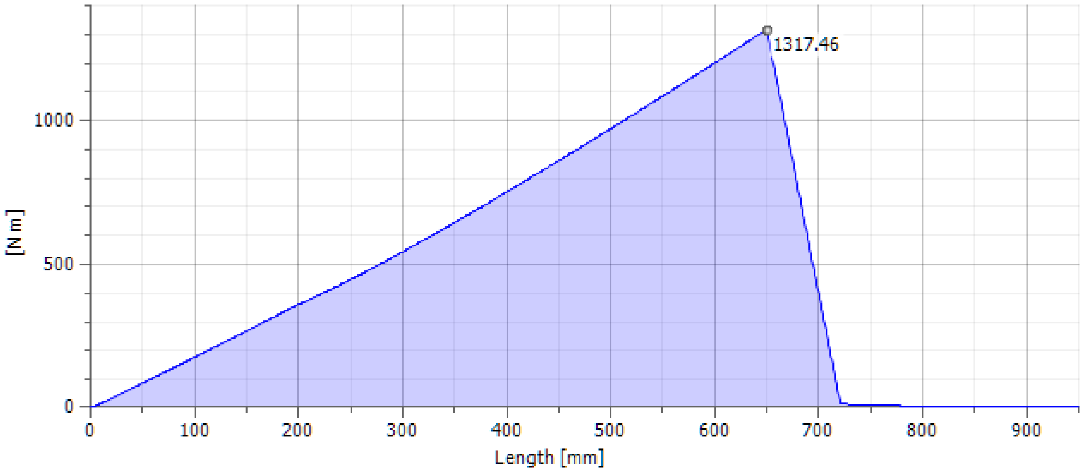


Figure 74: Bending Moment Diagram, FS

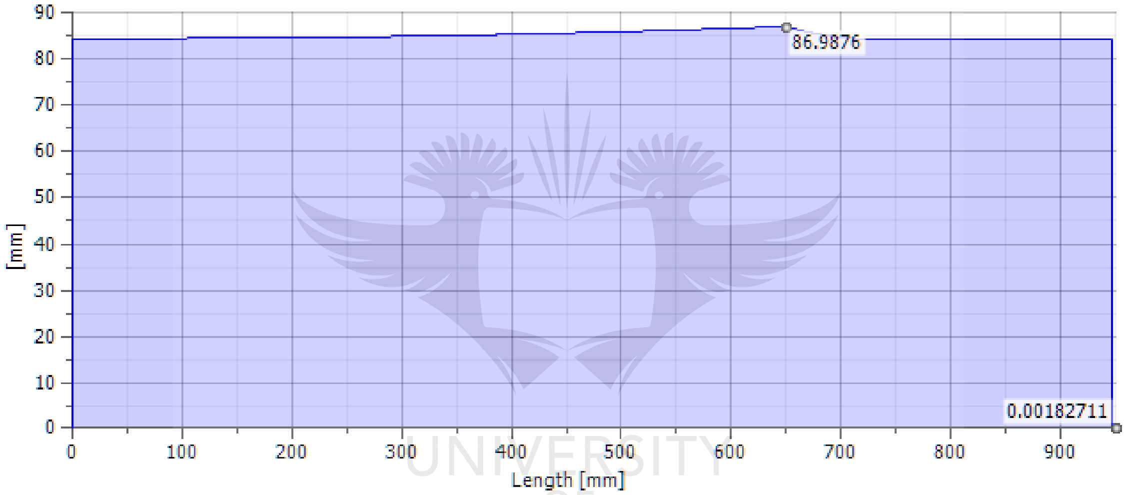


Figure 75: Ideal Diameter Diagram, FS

By studying the Inventor 2013 FS design data, the following are noted;

- Figure 73: FS, SFD, the reactions of Bearing A & Bearing B is 1494.9 N and 20735.1 N respectively. The maximum shear force 20735.1 N is acting at a distance of 650 mm on the shaft from LHS.

- Figure 74: BM diagram, the maximum BM is 1317.46 Nm at 650 mm on the shaft from the LHS.

- Figure 75: The FS diameter is now found to be 87 mm also located in a region situated at 650 mm on the FS.

11.6.3 Calculation of the final diameter for the FS.

After stepping up the shaft with the 16.457 % the following FS diameter was attained.

$$d_{min} = \ d_{ideal} \times \%upgrade = 87 \times 1.16.457 = 101.32\ mm \rightarrow 105\ mm$$

To compensate for key groove stress concentration, 15 % will be added to increase the shaft cross section. The simplified formula for the 15 % shaft area increase is;

Key groove compensation:

$$d_{new} = d_{min} \times \sqrt{1.15} = 101.32 \times \sqrt{1.15} = 108.651\ mm\ \rightarrow 110\ mm$$

Investigation of the larger sprocket bore size.

Appendix 9-6 the stock bore size for a 95 teeth, 960.25 mm diameter 20B Plate wheel sprocket is 40 mm and it can be machined to the needed 110 mm shaft diameter for a perfect fit.

11.7 Tyre coupling selection.

Technical Literature: Appendix 14

NB: For all calculations and selection in this section, refer to Appendix 14. The coupling catalogue is an optional reference guide.

To couple the FS to the gearbox output shaft, a rigid coupling would not work properly, because of vibration being present. The best solution is to use a flexible coupling to damp some of the vibration between these two shafts and accommodate for a small amount of parallel misalignment.

The Fenner Flex Tyre Couplings with its damping factor of 0.9 and excellent parallel misalignment and end-float capabilities are the ideal coupling to dampen vibrations between the mentioned shafts.

11.7.1 Coupling design power.

The power input into the coupling from the gearbox is 47.54 kW (calculated in 9.10.1).

$$\boldsymbol{P_{design} = P_{in} \times SF}$$

Where:

- *P_{design} = Design power of the coupling.*
- *P_{in} = The input power from the gearbox to the coupling.*
- *SF = The factor of safety the manufacturer uses due to their product experience.*

11.7.1.1 Service Factor (SF) selection.

From the Service Factor "Table, (catalogue) Page 194", the Service Factor may be selected that is applicable to the drive.

- *Prime mover - Internal combustion engine.*
- *Driven machine type – Vibrating (Heavy Duty).*
- *Hours of operation– 10 and under.*
- *Service factor (SF) – 2.8*

11.7.1.2 Design power, nominal and maximum torque calculations.

$$\boldsymbol{P_{design} = 47.54 \times 2.8 = 133.112\ kW}$$

$$\boldsymbol{T_{nominal} = \frac{30 \times P_{in}}{\pi \times N} = \frac{30 \times 47540}{\pi \times 131.93} = 3406.064\ Nm}$$

$$\boldsymbol{T_{max} = T_{nominal} \times SF = 3441.018 \times 2.8 = 9634.852\ Nm}$$

11.7.2 Power rating and preliminary coupling selection.

On (catalogue) Page 195, Power Rating Table, the following is noted for the coupling being selected:

- the coupling input speed is 131.93 r/min and the next speed up on the table is 200 r/min;
- the coupling power higher than the 133kW is 195 kW and
- the preliminary selected coupling type is a F200.

11.7.3 Physical characteristics check.

See Appendix 14 for Table 31 evaluation.

Table 31: F200 coupling final selection.

Characteristic	*F200 specs.*	*Calculated*	*Approval*
Maximum speed rev/min	1300	131.93	yes
Nominal Torque Nm	9325	3406	yes
Maximum Torque Nm	23508	9635	yes
Max, parallel misalignment mm	5,3	-	yes
Maximum end float mm	6,6	-	yes
Mass kg	8.8	-	yes

Table 31 is a comparison of the calculated values vs the manufacturer coupling specification values for the selection. The checks indicate that using the F200 coupling in this design is adequate.

11.7.4 Bore sizes check.

On (catalogue) Page 196 the couplings dimensions table, the maximum bore sizes, bush numbers and coupling types can be selected.

Gearbox shaft diameter in Appendix 8 is given as 120 mm and the FS diameter calculated in 11.6.3 is 110 mm.

The F200 – Type F coupling with its maximum bore of 125 mm will accommodate both the gearbox shaft diameter (120 mm) and the FS (110 mm) will be able to fit on either side of the coupling .

11.7.5 Coupling specification.

F200-F - BUSH NO: 4535 diameter 120 mm (gearbox side)

4535 diameter 110 mm (FS designed side)

11.8 Key Calculations for the MS, IS and FS.

The key calculations done in this section will be used as safety devices to avoid torque overloading. Alternatively a torque limiter can be installed on one of the chain sprockets for future alterations to the machine. A torque limiter slips as soon as the maximum allowable torque is exceeded or overloaded and will regain its power transfer ability after the torque has reached an acceptable level again.

To accommodate for torque overloading, the key will be designed weaker than the shaft, to shear and eliminate the torque supply to protect all assembled parts on the shaft as well as the shaft itself. In other words, it will act similar to a circuit breaker in an electrical circuit when the current overloads the circuit, the resulting magnetic field will open the circuit by cutting off the current flow.

11.8.1 MS key calculations.

Shaft size d = 75 mm (calculated in 11.4.4.3)
Torque T = 197.693 Nm (calculated in 10.4.1.1)
Material EN8 - 080M40 heat treatment Q (Selected for all shafts Appendix 42)
Yield Stress S_{yt} = 385 MPa; Tensile Stress S_{ut} = 700 MPa (calculated in 11.4.1.3)

Assuming a weaker key then the shaft.

Since the key will be used as a protection device, it has to be designed weaker than the shaft in order to protect all machine power transmission equipment.

To achieve a weaker key design, it will be assumed that the safety factor of the key will be 10 % less than that of the MS in 11.4.1.1. The MS factor of safety was calculated using the JE Shigley [25] fatigue design method in 11.4.1.4 and is 13.855.

$$\boldsymbol{fos_{key} = 0.9 \times FOS_B = 0.9 \times 13.855 = 12.47}$$

Key dimensions for a 75 mm (see 11.4.4.3) diameter shaft according to BS 4235: Part 1: 1972

b x h = 20 x 12 mm (See Appendix 32 for keyway tables)

Where:

- b = Key width (mm).
- h = key high or thickness (mm).

Allowable shear and crushing yield strengths:

$$\boldsymbol{S_c = \frac{S_{ut}}{fos} = \frac{700}{12.47} = 56.135\ MPa}$$

$$\boldsymbol{S_s = \frac{0.577 \times S_{yt}}{fos} = \frac{0.577 \times 385}{12.47} = 17.814\ MPa}$$

Where:

- S_c = allowable crushing strength MPa.
- S_s = allowable shear strength MPa.
- fos = factor of safety.

<u>Operational key length.</u>

For crushing

$$\boldsymbol{T = \frac{S_c \times L \times h \times d}{4} \therefore L = \frac{197.693 \times 4}{0.012 \times 0.075 \times 56.135 \times 10^6} = 15.65\ mm \rightarrow 20\ mm}$$

For shearing

$$\boldsymbol{T\ = \frac{S_s \times L \times b \times d}{2} \therefore L = \frac{197.693 \times 2}{0.020 \times 0.075 \times 17.814 \times 10^6} = 14.8\ mm \rightarrow 15mm}$$

NB: The key lengths are too short when using EN8 - 080M40 heat treatment Q material, the width of the pulley that will be used to be driven by the key is 101 mm in Appendix 13. To yield a longer more engaging key length, a less stronger material has to be used and the factor of safety has to be left at 13.855, because the material used is already weaker than that of the shaft.

EN9 has a too high S_{ut} = 600 MPa and S_{yt} = 310 MPa value and will therefore not be suitable to get a shorter key length (Appendix 42). Suggesting EN8 - 080M40 normalised material of S_{yt} = 245 MPa and S_{ut} = 510 MPa (Appendix 42) .

Revised MS key length calculation.

All answers given below have been recalculated with the exact same formulae and methods as the above, except for the weaker key material mentioned above and the 13.855 factor of safety was used due to the already weaker key material then the shaft.

S_c = 36.81 MPa
S_s = 10.216 MPa

For crushing L = 23.87 mm = 25 mm
For shearing L = 25.801 mm = 30 mm (the key length only improved by 10 mm and is still too short).

Due to the small amount of torque (197.693 Nm), the MS has to transmit, will make it difficult to design a key length on par with the pulley width of 101 mm (Appendix 13).

Just for further clarification, the calculation below indicates that a material with a tensile strength of 120.53 MPa will allow for a key length of 101 mm to be used. There is no steel currently on the market with such a low tensile strength.

Thus for academic purposes the key specification will contain the length of 30 mm. To make a final decision, key lengths for the FS and IM will have to be calculated first, in the meantime, the key specification below will suffice.

$$\boldsymbol{T = \frac{S_{ut} \times L \times h \times d}{fos \times 4} \therefore 197.693 = \frac{S_{ut} \times 0.101 \times 0.012 \times 0.075}{13.855 \times 4}}$$

$$\boldsymbol{\therefore\ S_{ut} = 120.53\ MPa}$$

Key Specification: b x h x L = 32 x 18 x 30 mm

NB: If the 30 mm length is not long enough for engagement purposes, a material equivalent to the 120.53 MPa tensile strength can be used to suffice. The MS and MS key calculations can also be reworked as to accommodate the pulley width with a standard suggested key size from the manufacturer.

11.8.2 IS key calculations.

Shaft size d = 85 mm (calculated in 11.4.4.3)

Torque T = 808.17 Nm (Calculated in 10.5.2.2)

Material EN8 - 080M40 normalised (Selected for al shafts Appendix 42)

Yield Stress S_{yt} = 280 MPa; Tensile Stress S_{ut} = 510 MPa (Calculated in 11.4.1.3)

$\boldsymbol{fos_{key} = 13.8557}$ (Material is already weaker than the shaft)

Key dimensions for an 85 mm diameter shaft according to BS 4235: Part 1: 1972

b x h = 22 x 14 mm (Appendix 32)

Where:

- b = Key width (mm)
- h = key high or thickness (mm)

Shear and crushing yield strengths:

$$S_c = \frac{S_{ut}}{fos} = \frac{510}{13.8557} = 36.808\, MPa$$

$$S_s = \frac{0.577 \times S_{yt}}{fos} = \frac{0.577 \times 245}{13.8557} = 10.21\, MPa$$

Key length required for operation:

For crushing

$$T = \frac{S_c \times L \times h \times d}{4} \therefore L = \frac{808.17 \times 4}{0.014 \times 0.085 \times 36.808 \times 10^6} = 73.803\, mm \to 75\, mm$$

For shearing

$$T = \frac{S_s \times L \times b \times d}{2} \therefore L = \frac{808.17 \times 2}{0.022 \times 0.085 \times 10.216 \times 10^6} = 84.13\, mm \to 85\, mm$$

Key Specification: b x h x L = 22 x 14 x 85 mm

The key specification is for both the small sprocket and the large pulley on the IS.

Final decision.

The pulley has a width of 101 mm in Appendix 13 and the sprocket has a width of 91 mm in Appendix 9-6, thus the 85 mm key length will suffice.

11.8.3 FS key calculations.

11.8.3.1 Large sprocket

Shaft size d = 110 mm (Calculated in 11.4.4.3)

Torque T = 3406.64 Nm (Calculated in 10.5.2.2)

Material EN8 - 080M40 Q8 (Appendix 42)

Yield Stress S_{yt} = 380 MPa; Tensile Stress S_{ut} = 700 MPa (Calculated in 11.4.1.3)

$$\boldsymbol{fos_{key} = 0.9 \times 13.8557 = 12.47}$$

Key dimensions for a 110 mm diameter shaft according to BS 4235: Part 1: 1972

b x h = 28 x 16 mm (Appendix 32)

Where:

- b = Key width (mm)
- h = key high or thickness (mm)

Shear and crushing allowable stresses.

$$\boldsymbol{S_c = \frac{S_{ut}}{fos} = \frac{700}{12.47} = 56.135\ MPa}$$

$$\boldsymbol{S_s = \frac{0.577 \times S_{yt}}{fos} = \frac{0.577 \times 385}{12.47} = 17.814\ MPa}$$

Required key length for operation.

For crushing

$$\boldsymbol{T=\frac{S_c\times L\times h\times d}{4}\therefore L=\frac{3406.64\times 4}{0.016\times 0.110\times 56.135\times 10^6}=137.9\,mm\rightarrow 140\,mm}$$

For shearing

$$\boldsymbol{T\ =\frac{S_s\times L\times b\times d}{2}\therefore L=\frac{3406.64\times 2}{0.028\times 0.110\times 17.814\times 10^6}=124.18\,mm\rightarrow 130\,mm}$$

The calculated key lengths are too large and needs to be reduced to accommodate the 91 mm triplex sprocket (Appendix 9-6) width.

Revised key length factor of safety.

Assuming a key length of 91 mm (Sprocket width Appendix 9-6).

$$\boldsymbol{T=\frac{S_{ut}\times L\times h\times d}{fos\times 4}\therefore 3406.64=\frac{700\times 0.091\times 0.016\times 0.11}{4\times fos}\therefore fos=8.275}$$

$\boldsymbol{Percentage\ key\ weaker\ then\ shaft\ =\ 1-\frac{8.275}{13.8557}=40.28}$ % (The key is 40.8% weaker, but is still able to transmit the torque comfortably).

Key Specification: b x h x L = 28 x 16 x 91 mm

10.8.3.2 Coupling key size calculation between the gearbox shaft and FS.

Shaft sizes $d_{gearbox}$ = 120 mm (Appendix 8-4); FS_{diameter} = 110 mm

Torque T = 3406.64 Nm (Calculated in 10.5.2.2)

Key dimensions for a 110 mm diameter shaft according to BS 4235: Part 1: 1972
b x h = 28 x 16 mm (Appendix 32)

Where:

- b = Key width (mm)
- h = key high or thickness (mm)

In Appendix 14 for a F200 type F coupling, the length of the keyway that fits into the bush is 89 mm. The coupling itself is supplied with a standard key included. Judging by this, the key specification will then be: 28 x 16 x 89 mm.

There is no need to make the key weaker than the shaft, because it has already been compensated for at the large sprocket attached to the FS. See Figure 71 for positioning.

On the gearbox side, there is a standard key supplied for the 120 mm gearbox output shafts and has a length of 195 mm each. See gearbox engineering drawing in Appendix 8-4. The couplings torque (9325 Nm) ability to handle torque, is much higher than that of the FS (3406.64 Nm), see Table 28, thus only the key at the larger sprocket will satisfy the design for torque overloading.

11.9 Bearing calculations for the MS, IS and FS.

References: [28] and [29]

The vibratory function of the machine needs bearings that can handle radial loads, axial loads, impact, vibration, inertia, gradient resistance and misalignment. These bearings are called spherical roller bearings and are specifically designed for vibratory applications Appendix 34.

11.9.1 Machine acceleration.

Machine and tractor acceleration on level firm earth.

The usable PTO power on level firm earth is 102,078 kW (calculated 10.6.2). The PTO power left after all drive losses for machine acceleration is:

$$\begin{aligned} P_{PTO\ available\ less\ losses} &= P_{PTO\ available} \times Drive\ losses \\ &= \frac{102078 \times 0.985 \times 0.985 \times 0.96}{2} \times 0.99 \times 0.98 \times 0.96 \times 2 \\ &= 88.55\ kW \end{aligned}$$

The acceleration is:

$$P_{PTO\ available\ less\ losses} = m_{machine+tractor} \times\ a_{machine+tractor} \times velocity$$

$$\therefore a_{machine+tractor} = \frac{88550}{11000 \times 3.3333} = 2.415\ m/s^2$$

11.9.1.1 Inertia force reaction addition to the FS, IS and MS.

Pulleys, sprocket and compacting drum inertia need to be considered to add to the belt and chain tensions for the larger pulley and larger sprocket. The smaller sprocket and smaller pulley can be neglected due to its inertia vectors and tensions will subsequently decrease its final horizontal vectors.

1. The force on the big sprocket at the FS, after the addition of the inertia force is:

$$\boldsymbol{F_{big\ sprocket} = F_{\max chain\ tension} + m_{big\ sprocket} \times a_{machine+tractor} =}$$

$$\boldsymbol{= 7429 + 365 \times 2.31 = 8272.15\ N(horizontal)}$$

2. The force on the big pulley at the IS after the addition of inertia force is:

$$\boldsymbol{F_{big\ pulley} = F_{belt\ tensions} + m_{big\ pulley} \times a_{machine+tractor} =}$$

$$\boldsymbol{= 3668.33 + 270 \times 2.31 = 4292.03\ N\ (horizontal).}$$

3. The reaction on the MS frame bearings due to the inertia force when towed is:

$$\boldsymbol{F_{Frame\ bearings} = R_{loose\ earth} \times m_{drum} \times g + m_{drum} \times a_{machine+tractor} - F_{belt\ tensions}}$$

$$\boldsymbol{\therefore 2500 \times 2.31 = 5775\ N\ (horizontal)}$$

All new horizontal forces can be added to Figures 76, 77 and 78 and then be quantified hereafter.

<u>Vector identification.</u>

Figure 76 is a free body diagram representing the MS, the two 57325 N forces represents the two eccentric weight lobes, the 38055 N (Bearing F) is where the drum frame is attached and the vertical 166.77 N and horizontal 3668.33 N forces represents the small pulley. Refer to Figure 60 for the 3D drawing of the arrangement.

NB: As quantified above in point 3, the horizontal component (5775 N horizontal) must be added to the 38055 N vertical force at the frame.

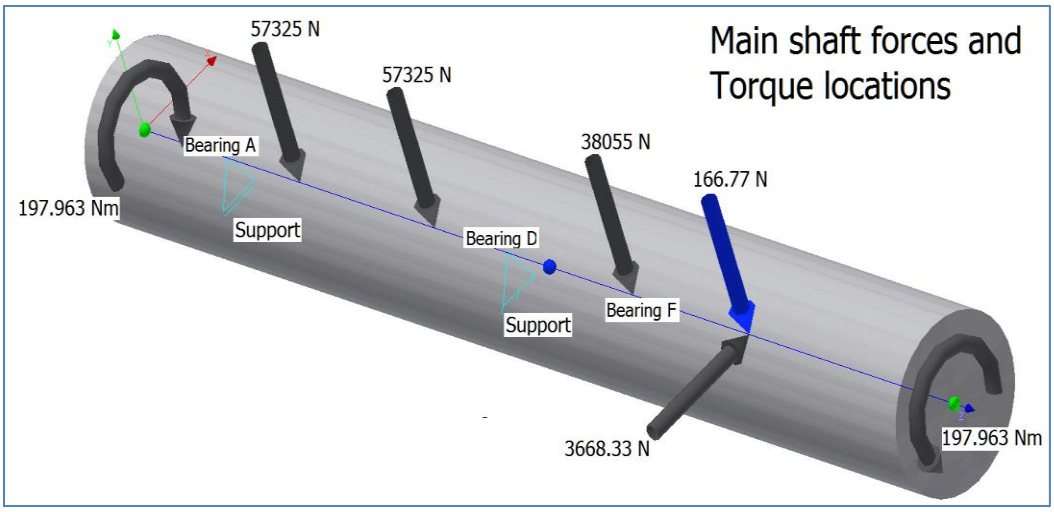


Figure 76: MS free body diagram.

<u>Bearing reactions</u>

Appendix 35 indicates that belt pull has to be taken into consideration when a belt drive is installed on a rolling element bearings supported shaft. The force for the belt pull needs to be multiplied by the belt pull factor stipulated by SKF and the factor for this type of application is indicated as 2.5. This factor is the maximum due to the vibration, impact and short center distance of the belt drive (Appendix 35).

Thus the horizontal force for the small pulley will change to:

$$\boldsymbol{F_{belt\ pull} = 3668.33 \times 2.5 = 9170.825\ N}$$

By considering the position of the bearings from the 9170.825 N belt pull force, reasonable assumptions can be made. See Figure 60 for 3D positioning in 10.4.4.2 and Figure 62, SF diagram for the positioning of the bearings.

Assuming that Bearing F which is situated closest to the force will absorb 4800 N, Bearing D is next to Bearing F with a force of 3500 N and Bearing A will carry the rest left over force, which is:

7910.825 – (4800 + 3500) = 870.825 N of the total belt pull (horizontal force).

For Bearing F, the total horizontal force will be 3500 N + 5775 N = 9275 N.

NB: Due to the distance, the 5775 N force is located from Bearing A and Bearing D, also due to the large vertical reactions the final design of these two bearings will result in very little change.

Referring to the MS SF diagram shown in Figure 63 (in 10.4.4.2) the following vertical forces were considered:

- the reaction at Bearing A = 40207.5 N (non-locating);
- the reaction at Bearing D = 113492 N (locating);
- and the reaction at Bearing F = 38055 N (non-locating).

After using addition of forces via the Pythagoras formula, $\boldsymbol{Reaction} = \sqrt{\boldsymbol{F_v}^2 - \boldsymbol{F_H}^2}$, the final reactions can be quantified at the bearings;

- the reaction at Bearing A = 40216.93 N (almost no change),
- the reaction at Bearing D = 113545.96 N (almost no change),
- and the reaction at Bearing F = 39169 N (minor change).

NB: Judging by the final reaction for all three bearings, the shaft sizes that was already calculated will not change, also the method of shaft diameter calculation was done with a very high factor of safety (13.857).

Shaft expansion

The MS has three bearings, the shaft will be allowed to experience axial expansion in both directions, where bearing D will locate the shaft and bearings A and F will allow for expansion. Spherical roller bearings can absorb high axial loads, but due to the nature of these bearings, their clearances will nullify the effect of thermal expansion.

Appendix 34 indicates an installation tightening angle of about 90 degrees plus and will be used for the locating bearing, this tightening angle is to allow the adaptor sleeve to grip fully onto the shaft.

To make bearings A and F non-locating, the tightening angle for installation can be reduced or the recommended SKF VA406- bearings can be used as stipulated in Appendix 34-2.

Bearing Life Calculations

For the sake of uniformity, symmetry and maintenance purposes bearing A and bearing D will be of the same type. The most highly loaded bearing D, can be calculated first and then bearing A will have the same bearing characteristics as bearing D. This method will apply for the IS and FS bearings as well, later on.

$F_a = 0$ $F_r = 113.546\ kN$

Bearing Number: 22324 CCK/W33 with adaptor sleeve H2324

$C_o = 1120\ kN$ $C = 965\ kN$ $e = 0.35$

$\frac{F_a}{F_r} = \frac{0}{113.546} = 0$ $\frac{F_a}{Fr} \leq e \therefore 0 < 0.35$

The ratio of the axial to the radial load is less than the "e" value, the following formula has to be used (Appendix 34);

$$P = F_r + Y_1 \times F_a = 113.546 + 1.9 \times 0 = 113.546\ kN$$

$$L_{10h} = \frac{10^6}{60 \times n} \times \left(\frac{C}{P}\right)^p = \frac{10^6}{60 \times 2180} \times \left(\frac{965}{113.546}\right)^{3.333} = 9570\ hours$$

For the next bearing number (23224 CCK/W33), with a lower *C* value: *C* = 695 kN, the L_{10h} life =3205.09 hours (calculated using the same method as above, also see Table 32). Appendix 33 indicates the life requirements for construction machine bearings and it is within the range of 3000 to 8000 hours; the 3205.09 hrs is too close to the minimum bearing life requirement for construction machines, thus the 9570 hours will be selected.

For all other bearings attached to the MS, Table 32 indicates all values for life in hours;

Table 32: MS bearings life speculation.

Bearing	Radial load F_r (kN)	Axial load (kN)	*Ratio* F_a/F_r	*e*	Y_1	*C*	(*P*)Equivalent dynamic bearing load (kN)	L_{10h} life in hours
D	113.546	0	0	0.35	1.9	965	113.546	9570
A	40.217	0	0	0.35	1.9	965	40.217	304315
F	39169	0	0	0.35	1.9	965	39169	333077

Bearing A has 31.8 times more life in hours than Bearing D, but it is important to have symmetry on a vibratory shaft or any other shaft to avoid unnecessary loads that can shorten bearing life due to size variation.

NB: Bearing F has a very high life (333077 hrs) in hours and can be reduced for economic reasons.

Refinement of Bearing F life in hours.

Table 33 represents the bearing life in hours for Bearing F. The first two bearings are calculated based on the $\boldsymbol{P = F_r + Y_1 \times F_a}$ formula, because the axial to radial force ratio is zero and smaller than the *e* value (Appendix 35-4).

NB: $F_a = 0$

The final selected most economical bearing in Table 33 is bearing 23024 CCK/W33 in row 5. This bearing has the closest amount of hours 48305 hrs to the suggested 3000 to 8000 hrs for construction equipment in Appendix 33.

*Table **33**: Bearing F life calculation refinement.*

Row no	Bearing number for Bearing F	*C kN*	*Co kN*	*e*	Y_1 or Y_2	(*P*)Equivalent dynamic bearing load (kN)	L_{10h} life in hours
1	22324 CCK/W33	965	1120	0.35	Y1 = 1.9	39169	332311
2	23224 CCK/W33	695	930	0.35	Y1 = 1.9	39169	111289
3	22224 EK	630	765	0.26	Y2 = 3.9	39169	80226
4	23124 CCK/W33	510	695	0.28	Y2 = 3.6	39169	39667
5	23024 CCK/W33	355	510	0.22	Y2 = 4.6	39169	11858

Bearing alignment check.

In Appendix 34-3 there is an alignment check for sealed spherical roller bearings. A maximum misalignment of 0.5º is suggested for sealed roller bearings. If the 0.5º is exceeded, the seal will be affected and will start to loose grease, thus ultimately compromise the life specification of the bearing.

To check if sealed bearings can be used, Figure 77 shows a maximum MS deflection of 0.01969 degrees and suggests that a sealed bearing will be appropriate.

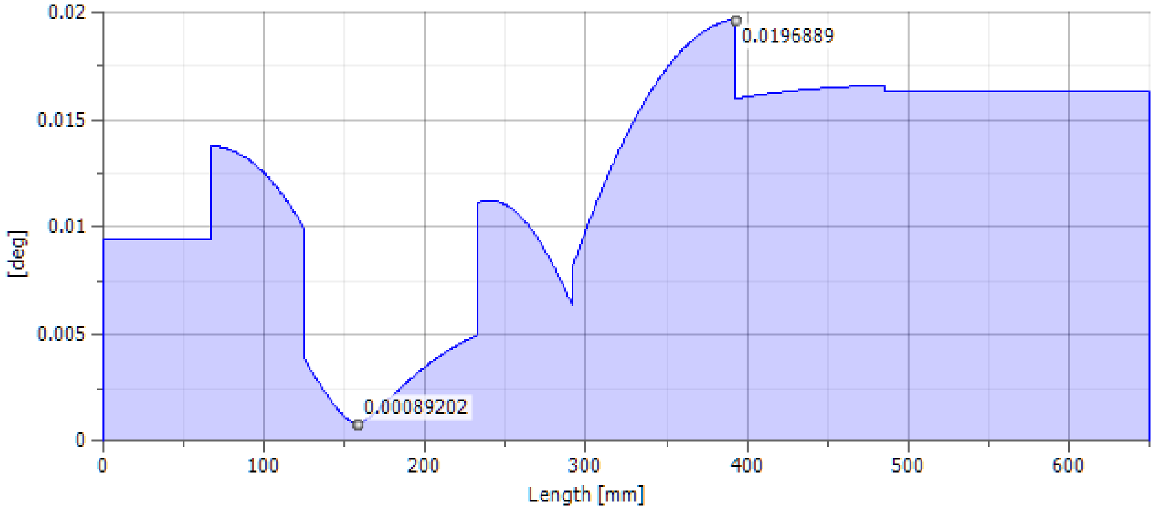


Figure 77: MS maximum deflection angle.

<u>Bearing grease selection</u>

Appendix 38-1 indicates the operating temperature in traffic light format, where green indicates operation with predictable life, yellow for short periods of use and red indicates "do not use".

The predictable temperature inside the drum is from 40ºC to 70ºC due to the drum being closed and has the ability to store heat. However, the drum is not insulated as with solar applications where insulated closed objects can reach temperatures from 100º to 180º.

The basic operating hours for a sealed bearing is dependent on the load magnitude, speed ratio and operating temperature as seen in Appendix 38-2.

There are two types of greases that can be used for this application, the LGEP 2 VT143 which is standard for a sealed spherical roller bearing (Appendix 38-2) and the LGHB 2 GEM9. The minimum life for this grease is about 18000 hrs. And it is well suitable for construction machinery, due to their low life requirement of 3000 to 8000 hrs.

Looking at the traffic light diagram, both greases prove competent when it comes to temperature, but a load magnitude check for the LGEP 2 VT143 grease can be made via this formula:

$$\boldsymbol{P \leq 0,1 \times C}$$

The formula shows that 10 % of the basic dynamic bearing load "C" must be equal or greater than the equivalent dynamic bearing load "P" then the LGEP 2 VT143 grease is appropriate, if not, the LGHB 2 GEM9 grease can be used or any other appropriate grease suggested by the supplier.

For bearings A ,D;

- C = 965 N
- P = 40.217 N (Bearing A), 113546 N (Bearing D)

Bearing A: 40.217 N < 96.5 N (LGEP 2 VT143 grease is ok)

Bearing D: 113.546 > 96.5 (LGEP 2 VT143 grease is not ok, but LGHB 2 GEM9 will be used instead).

For Bearing F

- C = 35.5 N; P = 39.169

Bearing F: 39.169 N > 35.5 N (LGEP 2 VT143 grease is not ok, but LGHB 2 GEM9 will be used instead).

<u>MS Bearing Specifications.</u> (Appendix 37)

- Bearings A and D: 2 off 22324 CCK/W33 with adaptor sleeve H2324 [sealed bearing with grease specification LGHB 2 GEM9 (Bearing A) and LGHB 2 GEM9 (bearing D) (Appendix 34)].

- Bearing F: 1 off 23024 CCK/W33 with adaptor sleeve H2324 [sealed bearing with grease specification LGHB 2 GEM9 (Appendix 34)].

The order can be doubled due to the symmetry of the design. Works within the green light temperature range 40º+ (Appendix 38).

11.9.2 IS bearing calculation

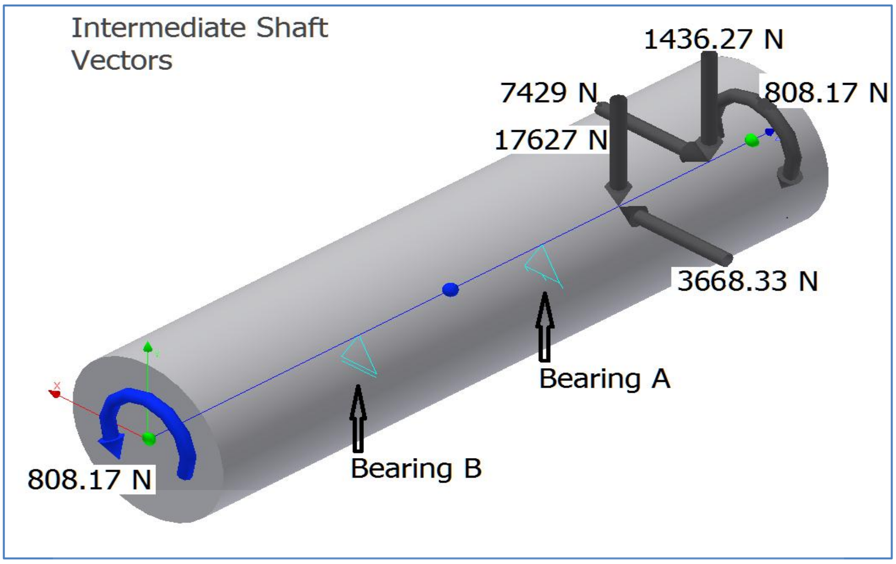


Figure 78: IS free body diagram.

<u>IS vector identification.</u>

Bearing A and bearing B are indicated as supports in the free body diagram in Figure 78. To the right of bearing A, the large pulley is located (forces 17627 N vertical and 3668.33 N horizontal). Also to the right of the larger pulley is the small sprocket (forces 1436.27 N vertical and 7429 N horizontal) of the chain drive that connects to the FS via its larger sprocket.

<u>Bearing reactions</u>

Since Bearing A has the belt and chain drives on its RHS, it is best to make bearing B the locating bearing in the arrangement, due to the fact that expansion can cause slight misalignment of the belts and chains. The clearances and the high standard of spherical roller bearings manufacturing , can handle temperature change.

In 11.8.1 the 3668.33 N force was multiplied by a belt pull factor of 2.5 (Appendix 35-1) to accommodate for the drive centre distance, vibration, etc, and the quantified force was **9170.825 N** for the purpose of bearing selection. The larger pulley attached to the IS has the same reaction (9170.825 N) as experienced by the other belt drive pulley.

By taking moments in Figure 68, it is found that the two bearing reactions point in opposite directions;

R_A = 13756.24 N (horizontal) and R_B = - 4585.413 N (horizontal)

Vector balance check: $F_{belt\ pull} = R_A + R_B$ = - 4585.413 + 13756.24 = 9170.827 N (OK)

Referring to the IS SF diagram Figure 68 the vertical reaction can be read;

- the vertical reaction at bearing A = 15367 N (locating)
- the vertical reaction at bearing B = 37004.4 N (non-locating)

By applying the Pythagoras formula to add the horizontal and vertical vectors, the final reactions or radial forces can be obtained at the bearings.

$$\boldsymbol{R = \sqrt{F_v{}^2 - F_H{}^2}};$$

- the radial force at bearing A = 16036.54 N
- and the radial force at bearing B = 39478.6 N.

Bearing Life Calculation.

Bearing A:

$F_a = 0$	$F_r = 16.037\ kN$	
Bearing Number:	22219 E with adaptor sleeve H319	
$C_o = 450\ kN$	$C = 380\ kN$	$e = 0.24$
$\boldsymbol{\frac{F_a}{F_r} = \frac{0}{16.037} = 0}$	$\boldsymbol{\frac{F_a}{Fr} \leq e \therefore 0 < 0.24}$	

If the ratio of the axial to the radial load is less than the e value, the following formula has to be used (Appendix 34);

$$\boldsymbol{P = F_r + Y_1 \times F_a = 16.037 + 1.9 \times 0 = 16.037\ kN}$$

$$\boldsymbol{L_{10h} = \frac{10^6}{60 \times n} \times \left(\frac{C}{P}\right)^p = \frac{10^6}{60 \times 545} \times \left(\frac{380}{16.037}\right)^{3.333} = 1168444\ hours}$$

Bearing B:

$F_a = 0$ $\quad F_r = 39.479\ kN$

Bearing Number: 22219 E with adaptor sleeve H319

$C_o = 450\ kN$ $\quad C = 380\ kN$ $\quad e = 0.24$

$\frac{F_a}{F_r} = \frac{0}{39.479} = 0$ $\quad \frac{F_a}{Fr} \leq e \therefore 0 < 0.24$

The ratio of the axial to the radial load is less than the e value, the following formula has to be used (Appendix 34);

$$P = F_r + Y_1 \times F_a = 39.479 + 1.9 \times 0 = 39.479\ kN$$

$$L_{10h} = \frac{10^6}{60 \times n} \times \left(\frac{C}{P}\right)^p = \frac{10^6}{60 \times 545} \times \left(\frac{380}{39.479}\right)^{3.333} = 58011\ hours$$

Both bearing A and bearing B are similar bearings and both are above the required range of 3000 to 8000 hrs (Appendix 33) for construction equipment (Appendix 33). However, it is the most economical choice for the 85 mm diameter shaft using this bearing type.

<u>Bearing alignment check.</u>

In Appendix 34-3 there is an alignment check for sealed spherical roller bearings. A maximum misalignment of 0.5º is suggested for sealed roller bearings. If the 0.5º is exceeded, the seal will be affected and will start to loose substance, thus ultimately the life specification of the bearing.

To see if sealed bearings can be used, Figure 79 shows a maximum MS deflection of 0.020316 degrees and suggests that a sealed bearing is adequate.

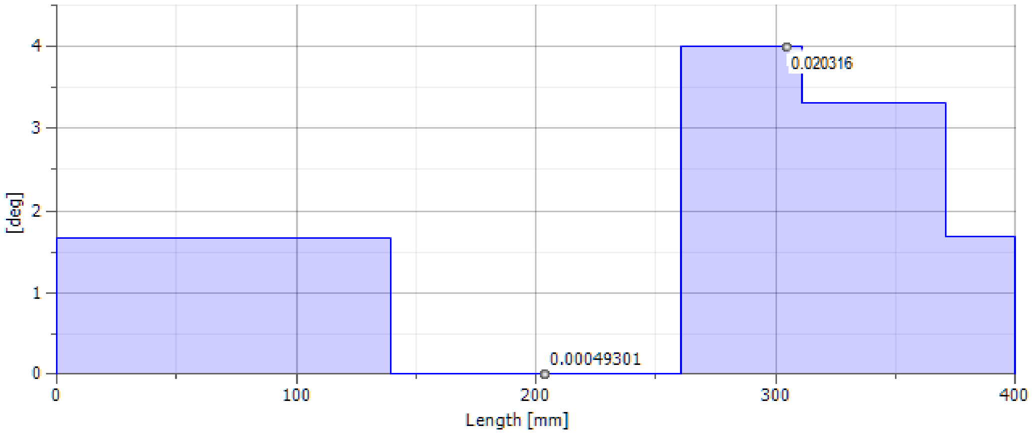


Figure 79: IS maximum deflection angle.

Bearing Specification (See Appendix 37)

Bearing Number:	22219 E with adaptor sleeve H319 (LGEP 2 VT143)
	22219 E with adaptor sleeve H319 (Sealed bearing with appropriate grease lubrication)

Works within the green light temperature range 40°+ (Appendix 38)

11.9.3 FS bearings calculations.

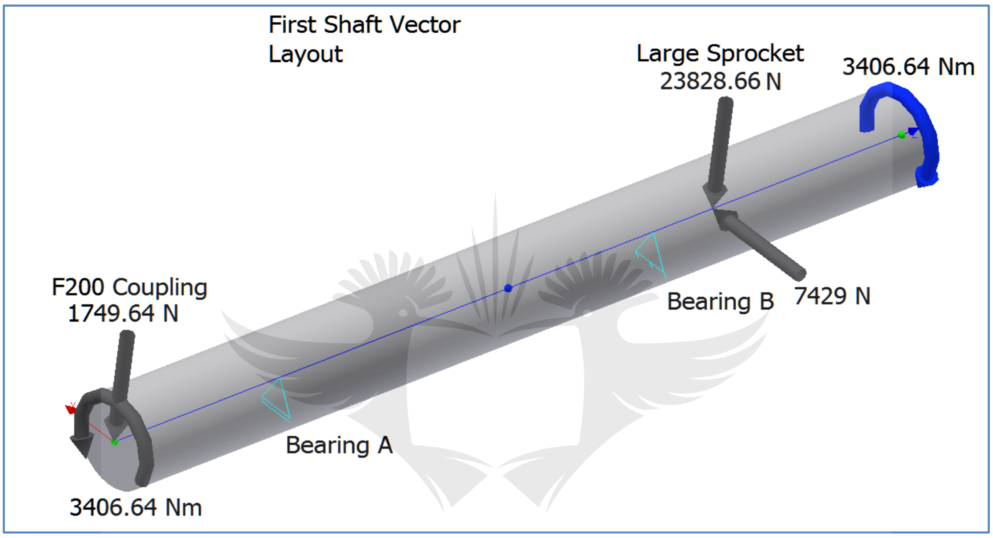


Figure 80: FS free body diagram.

FS vector identification.

Bearing A and Bearing B are indicated as supports in the free body diagram Figure 80. To the left of Bearing A, the F200 – F coupling is located (force 1749.64 N) and to the right of Bearing B, the large sprocket is at this region with a vertical force of 23828.66 N and horizontal force [7429 + 843.15 (inertia force calculated in 10.9.1.1 at the actual bearing regions, this force won't make a difference due to its location and magnitude) = 8272.15 N].

Bearing reactions

Since bearing A will be the non-locating bearing due to the flexible coupling that can handle 6.6 mm of end-float on its LHS, bearing B is the locating one to avoid chain misalignment due to shaft expansion on the LHS.

Referring to the FS, SFD on Figure 73 the vertical reactions can be read;

- the vertical reaction at Bearing A = 1494.9 N (locating)
- the vertical reaction at Bearing B = 20735.1 N (non-locating)

$$\boldsymbol{L_{10h} = \frac{10^6}{60\times n} \times \left(\frac{C}{P}\right)^p = \frac{10^6}{60\times 131.9} \times \left(\frac{355}{20.7351}\right)^{3.333} = 987920\ hours}$$

<u>Bearing Life Calculation.</u>

IS = 110 mm diameter

Bearing A:

F_a =0 *F_r = 1494.9N*

Bearing Number: 23024 CCK/W33 with adaptor sleeve H 3024

C_o = 510 kN *C = 355 kN* *e =0.22*

$\boldsymbol{\frac{F_a}{F_r} = \frac{0}{1.4949} = 0}$ $\boldsymbol{\frac{F_a}{Fr} \leq e \therefore 0 < 0.24}$

The ratio of the axial to the radial load is less than the "e" value, the following formula has to be used (Appendix 34);

$$\boldsymbol{P = F_r + Y_1 \times F_a = 1494.9 + 1.9 \times 0 = 1.4949\ kN}$$

$$\boldsymbol{L_{10h} = \frac{10^6}{60 \times n} \times \left(\frac{C}{P}\right)^p = \frac{10^6}{60 \times 131.9} \times \left(\frac{355}{1.4949}\right)^{3.333} = 1045\ \times 10^6\ \ hours}$$

The shaft can be stepped down at this region to accommodate a smaller diameter bearing with a lower C (Static load rating) value. It will not be attempted further in this dissertation, but at a later stage if need be, with permission from Terragrader (PTY) LTD modifications can be made.

NB: For now, this bearing satisfies the design, but it is not the most economical choice.

Bearing B:

F_a =0 *F_r* = 20.7351 *kN*

Bearing Number: 23024 CCK/W33 with adaptor sleeve H 3024

C_o = 510 kN *C = 355 kN* *e =0.24*

$\boldsymbol{\frac{F_a}{F_r} = \frac{0}{20.7351} = 0}$ $\boldsymbol{\frac{F_a}{Fr} \leq e \therefore 0 < 0.24}$

The ratio of the axial to the radial load is less than the e value, the following formula has to be used (Appendix 34);

$$\boldsymbol{P = F_r + Y_1 \times F_a = 20.7351 + 1.9 \times 0 = 20.7351\ kN}$$

Bearing A and bearing B will be identical and are above the required range of 3000 to 8000 hrs for construction equipment (Appendix 33), however, it is the most economical choice for the 110 mm diameter shaft using this bearing type.

Bearing alignment check.

In Appendix 34-3 there is an alignment check for sealed spherical roller bearings. A maximum misalignment of 0.5º is suggested for sealed roller bearings. If the 0.5º is exceeded, the seal will be affected and will start to loose substance, thus ultimately the life specification of the bearing.

To see if sealed bearings can be used, Figure 81 shows a maximum FS deflection of 0.01642º and therefore suggests that sealed bearings will be appropriate for the application.

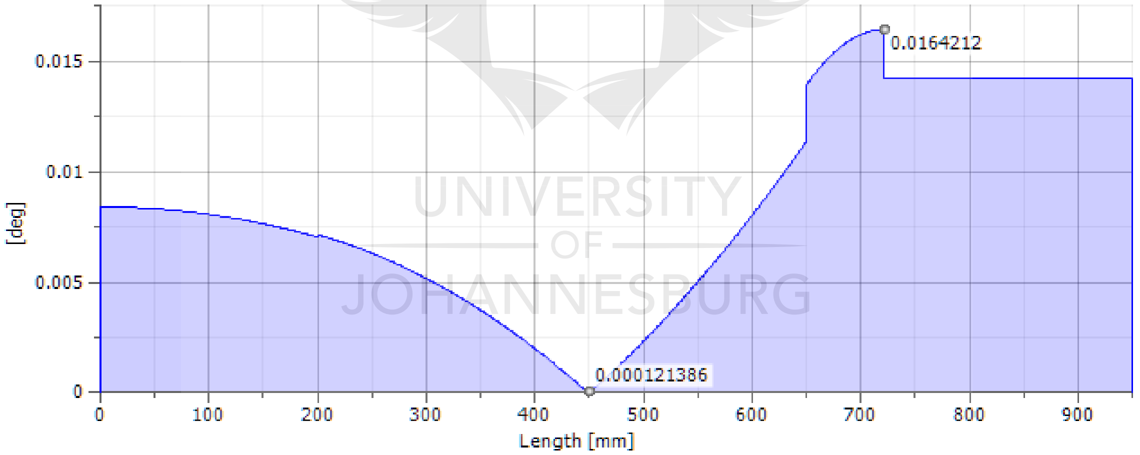


Figure 81: FS maximum deflection angle.

Bearing Grease selection

Works within the green light temperature range 40ºC + (Appendix 38)

Bearing Specification (Appendix 37)

Bearing Number:	23024 CCK/W33 with adaptor sleeve H 3024
	23024 CCK/W33 with adaptor sleeve H 3024

NB: Both bearings will be sealed bearings with standard grease supply on the inside.

11.10 MS eccentric weight heavy press fit.

Table 34: Hole and shaft tolerances.

Description	Basic size mm	Tolerance grade & class	Tolerances mm	Final size mm	Min. and Max diameter descriptions
Shaft	110	S6	+0.101	110.101	Max Shaft
			+0.079	110.079	Min Shaft
Hole	110	H7	+0.035	110.035	Max Hole
			+0	110	Min Hole

Table 34 shows the preferred tolerances and the precision class for the calculated shaft diameter.

Calculation of the pressure between the MS and eccentric weight sleeve will give an indication of the torque that can be transmitted through such an arrangement and is determined below. Also the importance of the amount of force that is required to assemble the eccentric weight sleeve to the MS is determined in 11.1.2.

11.10.1 Maximum radial interference

$$\boldsymbol{\delta max} = \frac{\boldsymbol{Max\ Shaft - Min\ Hole}}{\mathbf{2}} = \frac{\mathbf{110.101 - 110}}{\mathbf{2}} = \mathbf{0.0505}\ \boldsymbol{mm}$$

11.10.2 Eccentric weight sleeve pressure.

$$\boldsymbol{P = \frac{E \times \delta_{max}}{r_b} \times \frac{({r_a}^2 - {r_b}^2) \times ({r_b}^2 - {r_c}^2)}{2 \times {r_b}^2 \times ({r_a}^2 - {r_c}^2)}}$$

Where:

- *E = Youngs Modulus (GPa)*
- *P = Pressure created at the contact surface of both parts due to press fit (MPa)*
- *δ_{max} = Maximum radial interference between shaft and eccentric weight hole (m)*
- *r_a = Outer radius of eccentric weight sleeve (m)*
- *r_b = Contact radius of both hole and shaft (m)*
- *r_c = Inner radius of shaft which is 0 m, because of its solid profile (m).*

The pressure between the shaft and eccentric weight hub hole diameter is:

$$\boldsymbol{P = \frac{206 \times 10^{9} \times 0.0505 \times 10^{-3}}{0.055} \times \frac{(0.065^{2} - 0.055^{2}) \times (0.055^{2} - 0^{2})}{2 \times 0.055^{2} \times (0.065^{2} - 0^{2})}}$$

$$\boldsymbol{= 26.861\ MPa}$$

11.10.3 The required assembly force.

The force to assemble the centrifugal weight onto one of the MSs is:

$$\boldsymbol{F = \mu \times \pi \times 2 \times r_{b} \times L \times P}$$

Where:

- *μ = Coefficient of friction between two steel surfaces [24].*
- *L = Length of the eccentric weight hole (m).*
- *F = Heavy press fit assembly force (kN).*

$$\boldsymbol{F = 0.8 \times \pi \times 2 \times 0.055 \times 0.154 \times 26.861 \times 10^{6} = 1143.603\ kN}$$

11.10.4 The heavy press fit transmittable torque.

$$\boldsymbol{T_{transmittable} = F \times r_{b} = 1444.62 \times 10^{3} \times 0.055 = 62.898\ kN.m}$$

The machine torque at the MS is 197.963 N.m and will never be able to exceed the 628.98 Nm of torque. This indicates the press fit is acceptable.

11.11 Machine start up conditions.

11.11.1 Machine power transmission inertia calculation.

The moments of inertia for all rotating designed components are calculated and for all selected components, are as given by their respective catalogues.

$\boldsymbol{I_{drive\ shaft} = 0.045188\ kg\ m2}$ (Calculated in 11.7.11.1, Table 15)

$\boldsymbol{I_{gearbox} = 0.1182\ kg\ m2}$ (Appendix 8 – 5)

$\boldsymbol{I_{couplings} = 2 \times 1.281 = 2.562\ kg\ m2}$ (Appendix 14)

$$I_{First\,Shafts} = 2 \times \frac{m \times d^2}{8} = 2 \times \left(\frac{\pi \times d^2}{4} \times L \times \rho\right) \times \frac{d^2}{8}$$

$$= 2 \times \left(\frac{\pi \times 0.11^2}{4} \times 0.95 \times 7850\right) \times \frac{0.11^2}{8} = 0.2144\ kg\ m^2$$

$$I_{large\,sprockets} = 2 \times \frac{m \times d^2}{8} = 2 \times \frac{365 \times 0.96025^2}{8} = 84.14\ kg\ m^2$$

$$I_{Intermediate\,Shafts} = 2 \times \frac{m \times d^2}{8} = 2 \times \left(\frac{\pi \times d^2}{4} \times L \times \rho\right) \times \frac{d^2}{8}$$

$$= 2 \times \left(\frac{\pi \times 0.085^2}{4} \times 0.45 \times 7850\right) \times \frac{0.085^2}{8} = 0.0362\ kg\ m^2$$

$$I_{small\,sprockets} = 2 \times \frac{m \times d^2}{8} = 2 \times \frac{23 \times 0.233^2}{8} = 0.312\ kg\ m^2$$

$$I_{large\,pulleys} = 2 \times \frac{m \times d^2}{8} = 2 \times \frac{270 \times 0.807^2}{8} = 43.959\ kg\ m^2$$

$$I_{Main\,Shafts} = 2 \times \frac{m \times d^2}{8} = 2 \times \left(\frac{\pi \times d^2}{4} \times L \times \rho\right) \times \frac{d^2}{8}$$

$$= 2 \times \left(\frac{\pi \times 0.11^2}{4} \times 0.55 \times 7850\right) \times \frac{0.11^2}{8} = 1.241\ kg\ m^2$$

$$I_{small\,pulleys} = 2 \times \frac{m \times d^2}{8} = 2 \times \frac{17 \times 0.203^2}{8} = 0.1751\ kg\ m^2$$

$$I_{eccentric\,weights} = 2 \times (m \times {r_e}^2) = 2 \times (25.71 \times 0.0851^2) = 0.3724\ kg\ m^2$$

$$\begin{aligned}\boldsymbol{I_{total}} &= \boldsymbol{I_{drive\ shaft}} + \boldsymbol{I_{\ gearbox}} + \boldsymbol{I_{First\ Shafts}} + \boldsymbol{I_{large\ sprockets}} + \boldsymbol{I_{Intermediate\ Shafts}} \\ &\quad + \boldsymbol{I_{small\ sprockets}} + \boldsymbol{I_{large\ pulleys}} + \boldsymbol{I_{Main\ Shafts}} + \boldsymbol{I_{small\ pulleys}} \\ &\quad + \boldsymbol{I_{eccentric\ weights}} \\ &= \mathbf{0.045188 + 0.1182 + \ 2.562 + 0.2144 + 84.14 + 0.0362 + 0.312} \\ &\quad \mathbf{+ 43.959 + 1.241 + 0.1751 + 0.3724 = 133.1755}\ \boldsymbol{kg\ m^2}\end{aligned}$$

Where:

m = mass of respective component kg.

I = moment of inertia of respective component kg m^2.

d = diameter of respective component m.

r_e = eccentric moment of the eccentric weights m.

I_{total} = total moment of inertia of all rotating machine components kg m^2.

11.11.2 Power transmission acceleration calculation.

The input torque to the drive shaft is:

$$\boldsymbol{T = \frac{30 \times 120000}{\pi \times 1437} = 797.436\ Nm}$$

The usable torque that can be put through the system after drive losses is:

$$\begin{aligned}\boldsymbol{T_{usable}} &= \frac{\mathbf{797.436 \times 0.985 \times 0.985 \times 0.96}}{\mathbf{2}} \times \mathbf{0.99 \times 0.98 \times 0.96 \times 2} \\ &= \mathbf{691.786}\ \boldsymbol{Nm}\end{aligned}$$

The maximum angular acceleration this torque can accommodate when the machine is standing still and the centrifugal weights are accelerated is:

$$\boldsymbol{T_{usable} = I_{total} \times \alpha_{max} \therefore \alpha_{max} = \frac{691.786}{133.1755} = 5.194\ rad/s^2}$$

11.11.3 Eccentric weight startup time when the machine is stationary.

The final speed at the MS was calculated as 2180 r/min (228.29 rad/s) in 9.14.4.2

$$\boldsymbol{\omega_{final} = \omega_{initial} + \alpha_{max} \times t \ \ \therefore 228.29 = 0 + 5.194 \times t \ \ \therefore t = 43.95\ seconds}$$

The machine will take 43.95 seconds to get up to full rotational speed at the MS.

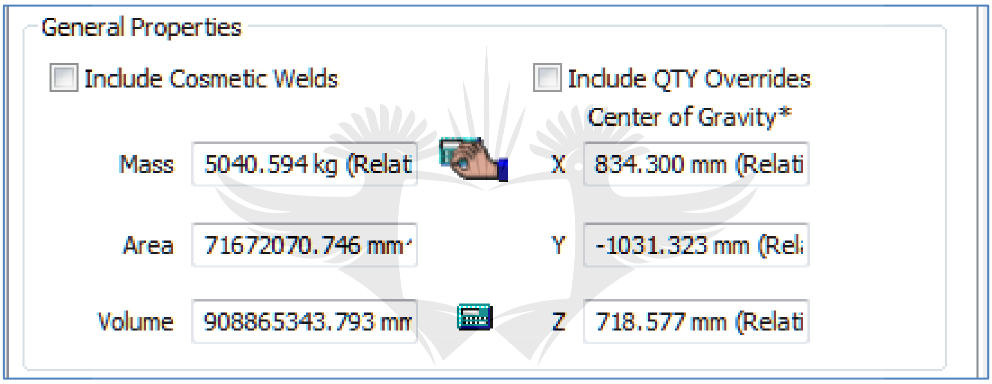
NB: The large amount of time the machine requires starting up uses all the machine's energy and therefore towing and startup must not be done at the same time, but separately. The eccentric weights must first get up to full speed, then towing can be initiated.

11.11.4 Recalculation of the machine high amplitude

Previously in 9.4.6, the machine mass was estimated at 5000 kg, an actual mass check has to be done through the Inventor 2013 software to know the amplitude the machine is going to operate at.

Machine mass

Table 35 Machine's final mass.



General Properties

☐ Include Cosmetic Welds

☐ Include QTY Overrides

Center of Gravity*

Mass 5040.594 kg (Relat

Area 71672070.746 mm'

Volume 908865343.793 mm

X 834.300 mm (Relati

Y -1031.323 mm (Rel

Z 718.577 mm (Relati

The machine's final mass as given by Inventor 2013, is 5040.6 kg in Table 35. The estimated mass of 5000 kg is very close to the actual and the difference is very small.

Machine operation amplitude

$$A_{nominal} = \frac{m_{e\,total} \times r_e}{M} = \frac{2 \times m_e \times r_e}{M} = \frac{2 \times 25.709 \times 0.085094}{5040.6} = 0.868\ mm$$

$$A_{double} = 2 \times A_{nominal} = 2 \times 0.8751 = 1.736\ mm$$

The actual operational amplitude of the machine is 1.736 mm. In Appendix 10 this amplitude can be compared in the universal table with other brands.

12. Reference List

[1] NATIONAL DEPARTMENT OF TRANSPORT, *A DRAFT RURAL TRANSPORT STRATEGY FOR SOUTH AFRICA*, November 2003.

[2] *LOCAL ROADS FOR RURAL DEVELOPMENT*, CARNS Consultants Group, Anderson Vogt & Partners CORD, Jeffares & Green Inc Jonsson & Associates Stewart Scott Inc Theron Burke & Isaac Uhlmann Witthaus & Prins, KWAZULU-NATAL MARCH 1997.

[3] Advancing the Public Interest: *A case of surfacing gravel roads*, information Series 2, Sabita, 2000.

[4] *State of Rural Transport & Development Research in South Africa*: unravelling current and future CSIR, Convention Centre 7-8 December 2009.

[5] Weinert , H.H. (1980).*The natural road construction materials of Southern Africa* Pretoria, H & R Academica.

[6] *Vibratory Roller Handbook with standardized terms and definitions,* SECOND EDITION Developed by the Compaction and Paving Machinery Technical Committee (CPMTC) Bituminous and Aggregate Equipment Bureau (BAEB) of the Construction Industry Manufacturers Association Copyright@ 1994.

[7] www.wikipedia.com/roadroller/

[8] Spatial analysis of modern soil compaction roller measurement values Daniel K. Heersinka*, Reinhard Furrera, *University of Zurich, Winterthurerstrasse 190 8049 Zurich, Switzerland,* 1st Spatial Statistics Conference 2011.

[9] Masters in Technology Dissertation, *The design and development of a working prototype machine to mechanically refurbish/create earth roads in South Africa,* Francisco Lardin Martinez,University of Johannesburg, 2010.

[10] Masters in Technology Dissertation, *Terra Tractor, Design Optimization of a Low Cost, Low Maintenance Tractor,* Christina Elena Popa, University of Johannesburg February 2013.

[11] Master of Science Dissertation, *Vibratory Hammer Compaction of Bitumen Stabilized Materials*, Rex Willem Constantyn Kelfens, University of Stellenbosch, December 2008.

[12] *Soil Compaction a basic hand book*, MULTIQUIP INC. © COPYRIGHT 2004, MULTIQUIP INC, Rev. 3 (08-04).

[13] Caterpillar Compaction Manuel

[14] *SOIL MECHANICS*, A. Verruijt , Delft University of Technology, 2001, 2011.

[15] *South African Pavement Engineering Manual*, November 2011, Revision 1.0 © 2009 South African National Roads Agency Ltd. All rights reserved. First edition published 2011, Printed in the Republic of South Africa.
SET: ISBN 978-1-920158-61-3, CHAPTER: ISBN 978-1-920158-XXXICAN NATIONAL ROADS AGENCY LTD.

[16] Dyer, C. G. (1982). *Road construction technology in South Africa*. South Africa, Juta. ISBN 0 7021 1338 7.

[17] TRH 17, *Geometric design of rural roads*, 1988, ISBN 0 7988 33122, Pretoria, South Africa, 1988.

[18] TRH 4, *Draft TRH 4: 1996, Structural Design of flexible pavements for interurban and rural roads*, ISBN 1-86844-21 8-7, DRAFT TRH4, pp 1-101, Pretoria, South Africa, 1996.

[19] TRH 20, *THE STRUCTURAL DESIGN, CONSTRUCTION AND MAINTENANCE OF UNPAVED ROADS*, 1990, ISBN 0 908381 87 5, Pretoria, South Africa, 1990.

[20] TRH 9, *CONSTRUCTION OF ROAD EMBANKMENTS*, ISBN 0 7988 2272 4, Pretoria, South Africa, 1982.

[21] *Matching Tractor Horsepower and Farm implement size*, Cooperative extension service, Michigan State University, Extension Bulletin E-1152 SF 11 November 1977, by Robert G White, Proffessor Emeritus , Agricultural Engineering.

[22] Stubbs, F.W. (1959).*Handbook of heavy construction*. New York, McGraw-Hill.

[23] TMH 4, *Geometrical Design Standards for Rural Two-Lane Two-Way roads*, Pretoria, South Africa September 1978 by the National Institute of Transport and Road Research of the Counsel for Scientific and Industrial Research. ISBN 0 7988 1396 2.

[24] Engineeringtoolbox.com

[26] Shigley, J. E. (2003). *Mechanical Engineering Design, Sixth Metric Edition*. U.S.A, McGraw Hill.

[25] Wenger, Karl F., (1984) *Forestry Handbook*. Canada John Wiley & Sons, Inc. ISBN: 0 471 06227 8.

[27] Drotsky, J.G. (2005). *Strength of Materials for Technicians*. Sandton, Heinemann.

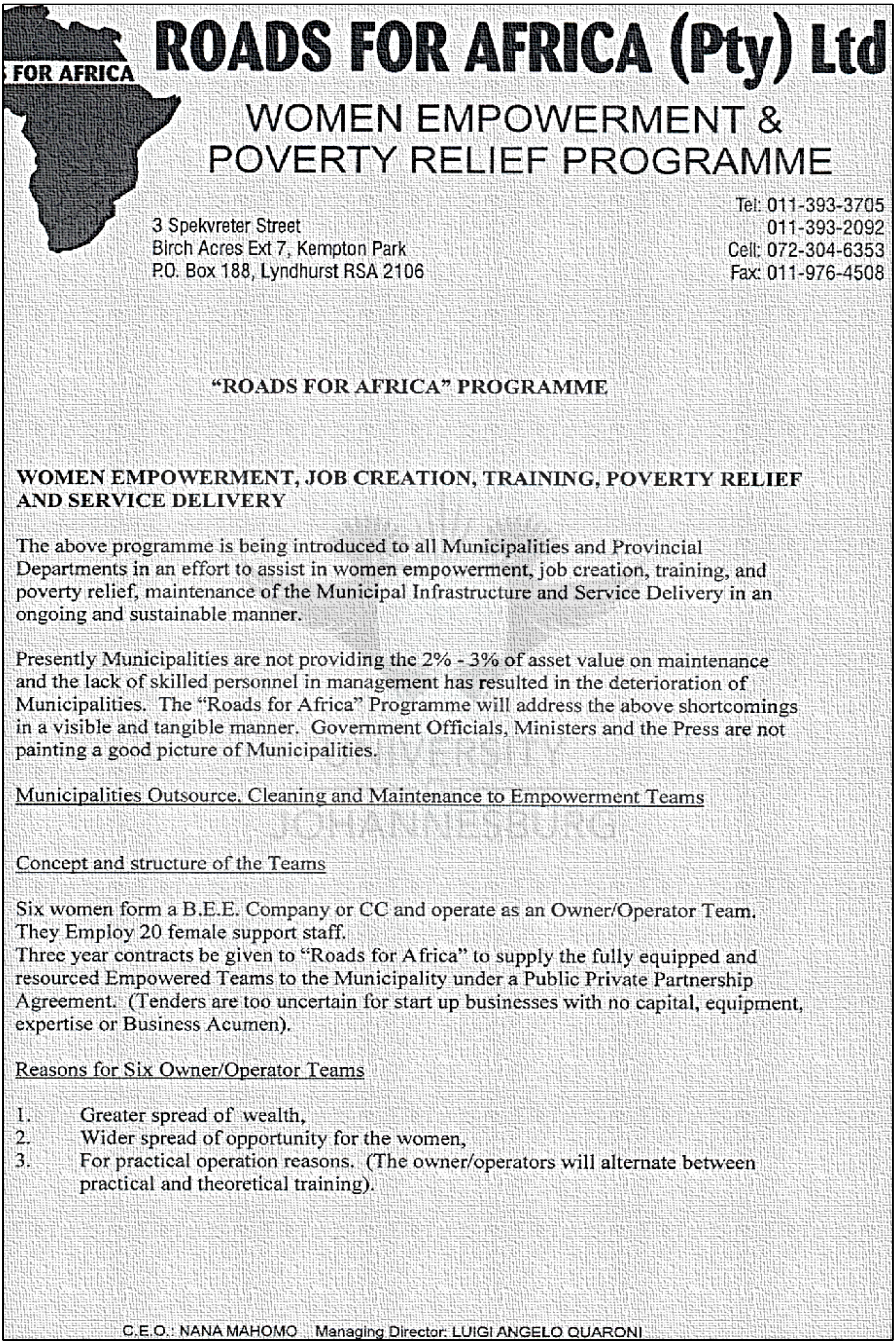
[28] SKF Group. (2008). *SKF General Catalogue*. Catalogue 6000 EN, Sweden.

[29] SKF Group. (October 2012). *SKF General Catalogue*. Catalogue PUB BU/P1 10000 EN.

[30] Autodesk, *Inventor 2013 Professional*, Autodesk is a trademark of the Autodesk Group.

13. Appendices

Appendix 1: Letters from Mr Luigi Quaroni (Managing Director).



ROADS FOR AFRICA (Pty) Ltd

WOMEN EMPOWERMENT &
POVERTY RELIEF PROGRAMME

3 Spekvreter Street
Birch Acres Ext 7, Kempton Park
P.O. Box 188, Lyndhurst RSA 2106

Tel: 011-393-3705
011-393-2092
Cell: 072-304-6353
Fax: 011-976-4508

"ROADS FOR AFRICA" PROGRAMME

WOMEN EMPOWERMENT, JOB CREATION, TRAINING, POVERTY RELIEF AND SERVICE DELIVERY

The above programme is being introduced to all Municipalities and Provincial Departments in an effort to assist in women empowerment, job creation, training, and poverty relief, maintenance of the Municipal Infrastructure and Service Delivery in an ongoing and sustainable manner.

Presently Municipalities are not providing the 2% - 3% of asset value on maintenance and the lack of skilled personnel in management has resulted in the deterioration of Municipalities. The "Roads for Africa" Programme will address the above shortcomings in a visible and tangible manner. Government Officials, Ministers and the Press are not painting a good picture of Municipalities.

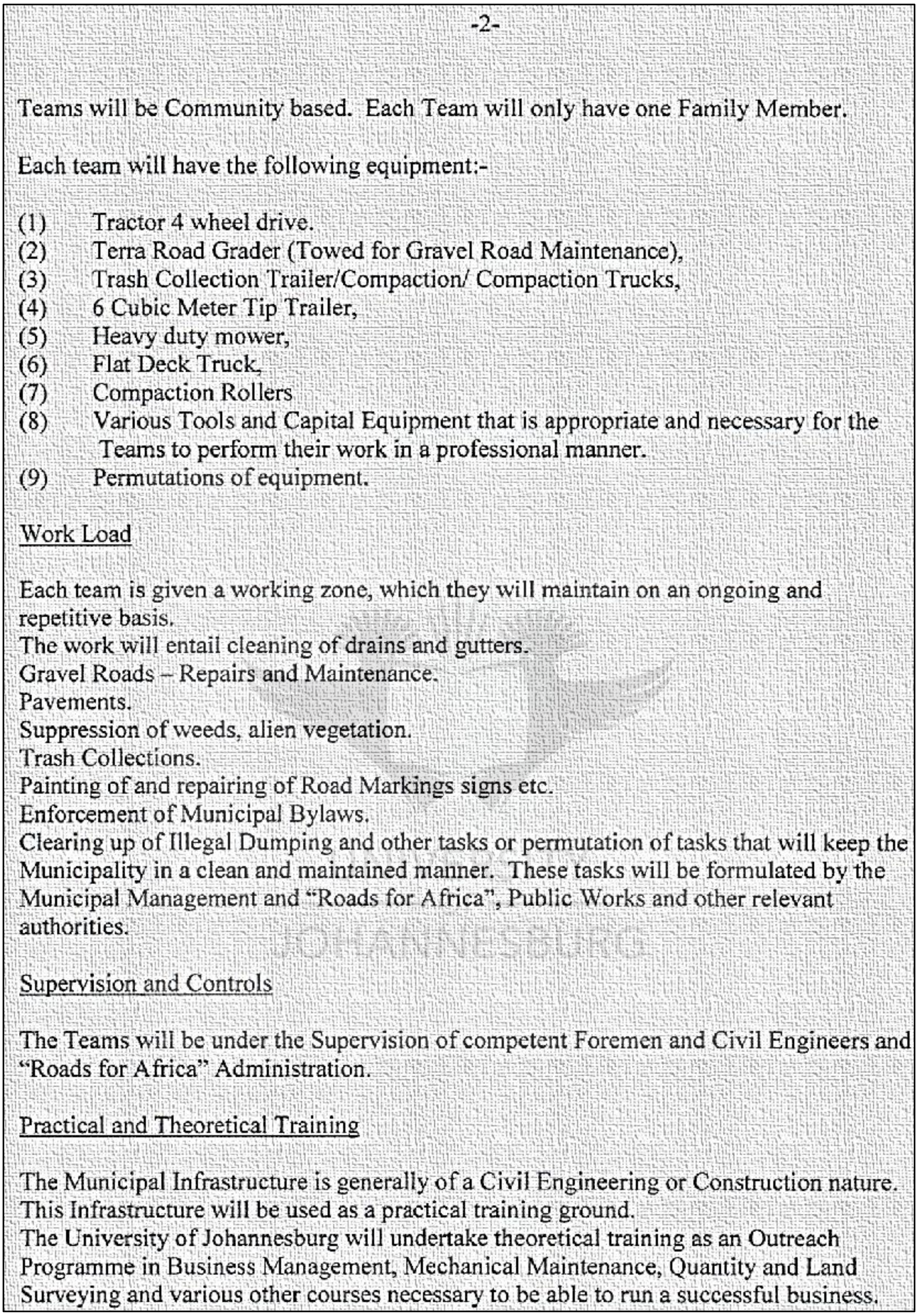
Municipalities Outsource, Cleaning and Maintenance to Empowerment Teams

Concept and structure of the Teams

Six women form a B.E.E. Company or CC and operate as an Owner/Operator Team. They Employ 20 female support staff.
Three year contracts be given to "Roads for Africa" to supply the fully equipped and resourced Empowered Teams to the Municipality under a Public Private Partnership Agreement. (Tenders are too uncertain for start up businesses with no capital, equipment, expertise or Business Acumen).

Reasons for Six Owner/Operator Teams

1. Greater spread of wealth,
2. Wider spread of opportunity for the women,
3. For practical operation reasons. (The owner/operators will alternate between practical and theoretical training).



Teams will be Community based. Each Team will only have one Family Member.

Each team will have the following equipment:-

(1) Tractor 4 wheel drive.
(2) Terra Road Grader (Towed for Gravel Road Maintenance),
(3) Trash Collection Trailer/Compaction/ Compaction Trucks,
(4) 6 Cubic Meter Tip Trailer,
(5) Heavy duty mower,
(6) Flat Deck Truck,
(7) Compaction Rollers
(8) Various Tools and Capital Equipment that is appropriate and necessary for the Teams to perform their work in a professional manner.
(9) Permutations of equipment.

Work Load

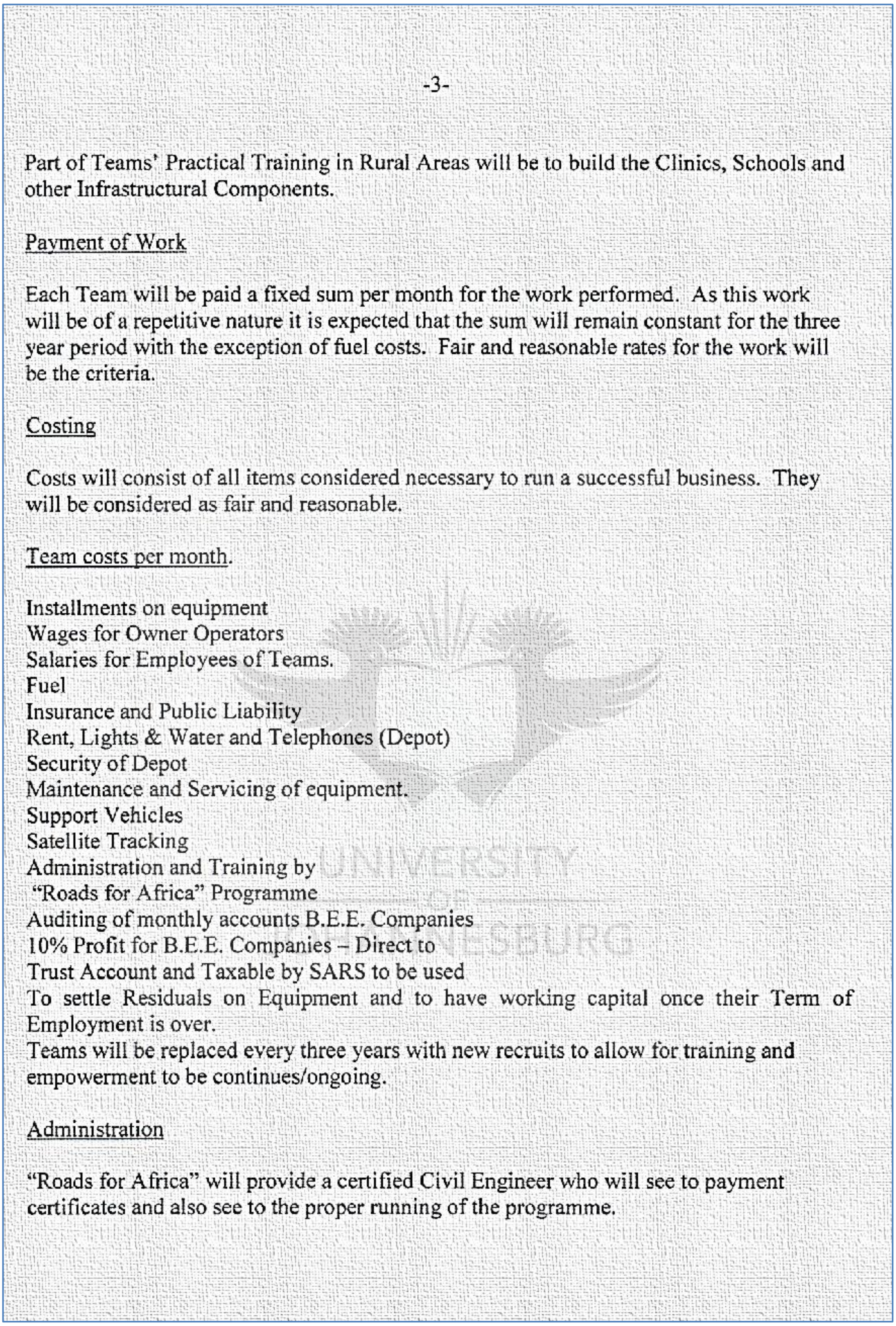
Each team is given a working zone, which they will maintain on an ongoing and repetitive basis.
The work will entail cleaning of drains and gutters.
Gravel Roads – Repairs and Maintenance.
Pavements.
Suppression of weeds, alien vegetation.
Trash Collections.
Painting of and repairing of Road Markings signs etc.
Enforcement of Municipal Bylaws.
Clearing up of Illegal Dumping and other tasks or permutation of tasks that will keep the Municipality in a clean and maintained manner. These tasks will be formulated by the Municipal Management and "Roads for Africa", Public Works and other relevant authorities.

Supervision and Controls

The Teams will be under the Supervision of competent Foremen and Civil Engineers and "Roads for Africa" Administration.

Practical and Theoretical Training

The Municipal Infrastructure is generally of a Civil Engineering or Construction nature.
This Infrastructure will be used as a practical training ground.
The University of Johannesburg will undertake theoretical training as an Outreach Programme in Business Management, Mechanical Maintenance, Quantity and Land Surveying and various other courses necessary to be able to run a successful business.



Part of Teams' Practical Training in Rural Areas will be to build the Clinics, Schools and other Infrastructural Components.

Payment of Work

Each Team will be paid a fixed sum per month for the work performed. As this work will be of a repetitive nature it is expected that the sum will remain constant for the three year period with the exception of fuel costs. Fair and reasonable rates for the work will be the criteria.

Costing

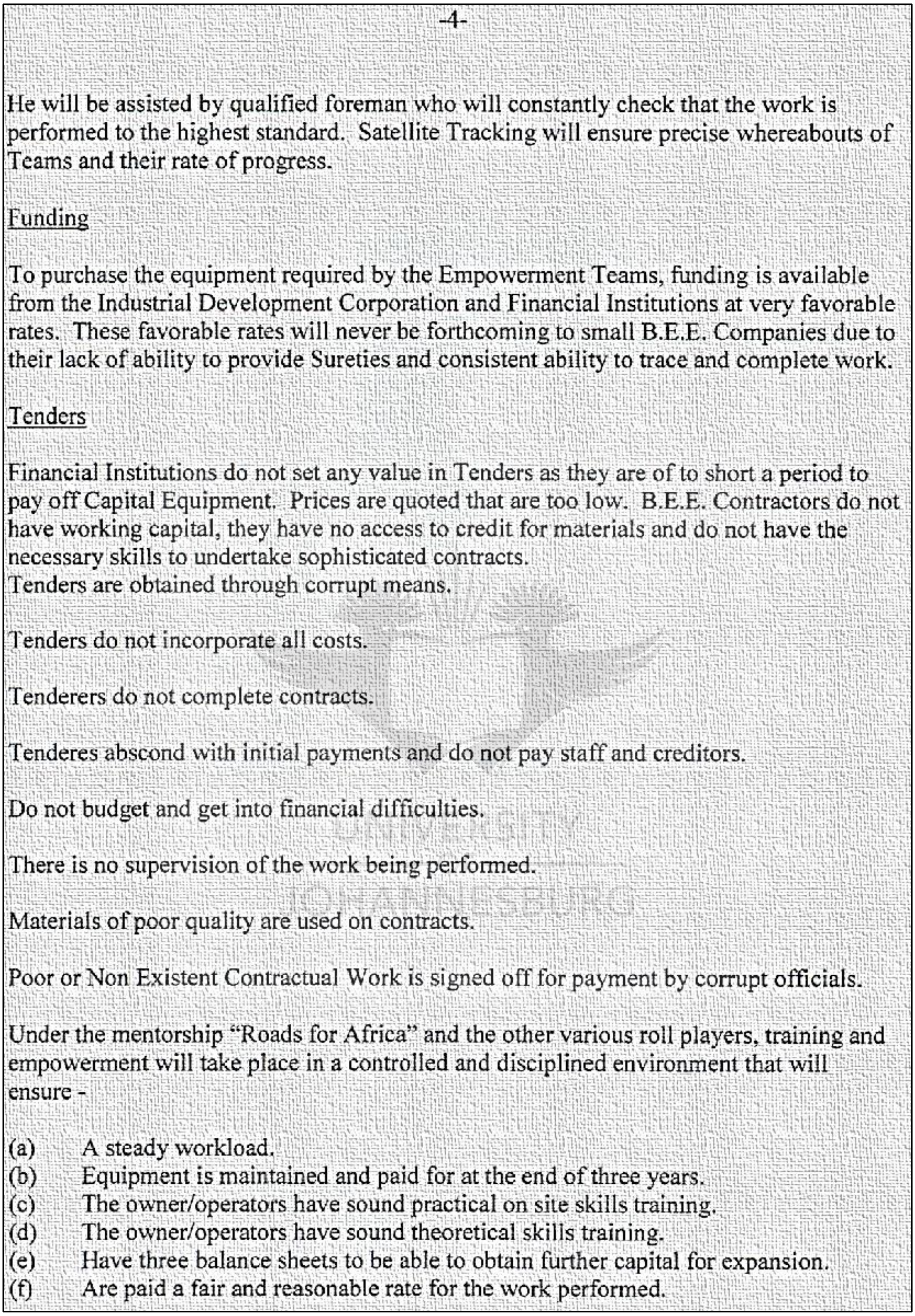
Costs will consist of all items considered necessary to run a successful business. They will be considered as fair and reasonable.

Team costs per month.

Installments on equipment
Wages for Owner Operators
Salaries for Employees of Teams.
Fuel
Insurance and Public Liability
Rent, Lights & Water and Telephones (Depot)
Security of Depot
Maintenance and Servicing of equipment.
Support Vehicles
Satellite Tracking
Administration and Training by
"Roads for Africa" Programme
Auditing of monthly accounts B.E.E. Companies
10% Profit for B.E.E. Companies – Direct to
Trust Account and Taxable by SARS to be used
To settle Residuals on Equipment and to have working capital once their Term of Employment is over.
Teams will be replaced every three years with new recruits to allow for training and empowerment to be continues/ongoing.

Administration

"Roads for Africa" will provide a certified Civil Engineer who will see to payment certificates and also see to the proper running of the programme.



He will be assisted by qualified foreman who will constantly check that the work is performed to the highest standard. Satellite Tracking will ensure precise whereabouts of Teams and their rate of progress.

Funding

To purchase the equipment required by the Empowerment Teams, funding is available from the Industrial Development Corporation and Financial Institutions at very favorable rates. These favorable rates will never be forthcoming to small B.E.E. Companies due to their lack of ability to provide Sureties and consistent ability to trace and complete work.

Tenders

Financial Institutions do not set any value in Tenders as they are of to short a period to pay off Capital Equipment. Prices are quoted that are too low. B.E.E. Contractors do not have working capital, they have no access to credit for materials and do not have the necessary skills to undertake sophisticated contracts.
Tenders are obtained through corrupt means.

Tenders do not incorporate all costs.

Tenderers do not complete contracts.

Tenderes abscond with initial payments and do not pay staff and creditors.

Do not budget and get into financial difficulties.

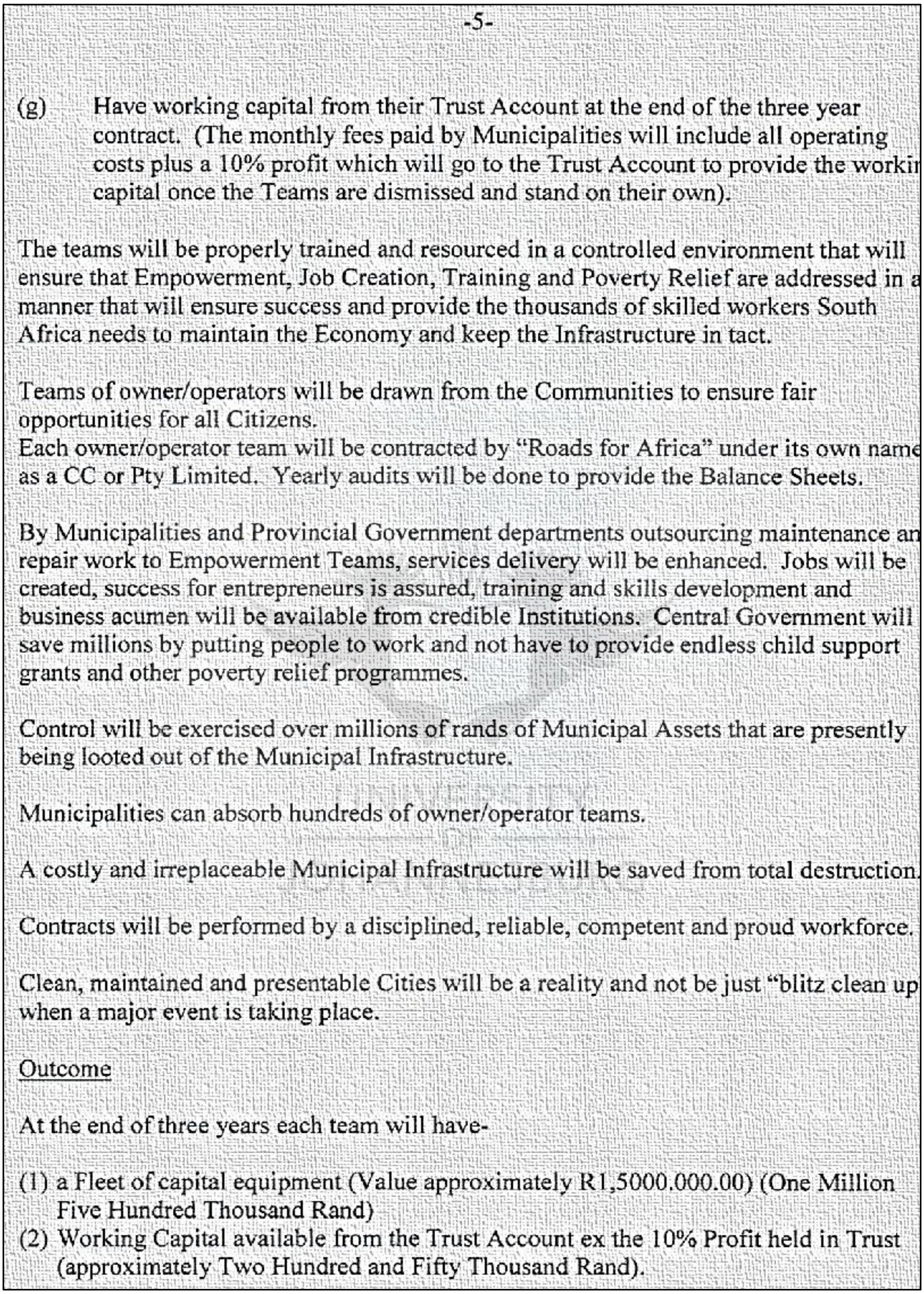
There is no supervision of the work being performed.

Materials of poor quality are used on contracts.

Poor or Non Existent Contractual Work is signed off for payment by corrupt officials.

Under the mentorship "Roads for Africa" and the other various roll players, training and empowerment will take place in a controlled and disciplined environment that will ensure -

(a) A steady workload.
(b) Equipment is maintained and paid for at the end of three years.
(c) The owner/operators have sound practical on site skills training.
(d) The owner/operators have sound theoretical skills training.
(e) Have three balance sheets to be able to obtain further capital for expansion.
(f) Are paid a fair and reasonable rate for the work performed.



(g) Have working capital from their Trust Account at the end of the three year contract. (The monthly fees paid by Municipalities will include all operating costs plus a 10% profit which will go to the Trust Account to provide the workin capital once the Teams are dismissed and stand on their own).

The teams will be properly trained and resourced in a controlled environment that will ensure that Empowerment, Job Creation, Training and Poverty Relief are addressed in a manner that will ensure success and provide the thousands of skilled workers South Africa needs to maintain the Economy and keep the Infrastructure in tact.

Teams of owner/operators will be drawn from the Communities to ensure fair opportunities for all Citizens.
Each owner/operator team will be contracted by "Roads for Africa" under its own name as a CC or Pty Limited. Yearly audits will be done to provide the Balance Sheets.

By Municipalities and Provincial Government departments outsourcing maintenance an repair work to Empowerment Teams, services delivery will be enhanced. Jobs will be created, success for entrepreneurs is assured, training and skills development and business acumen will be available from credible Institutions. Central Government will save millions by putting people to work and not have to provide endless child support grants and other poverty relief programmes.

Control will be exercised over millions of rands of Municipal Assets that are presently being looted out of the Municipal Infrastructure.

Municipalities can absorb hundreds of owner/operator teams.

A costly and irreplaceable Municipal Infrastructure will be saved from total destruction.

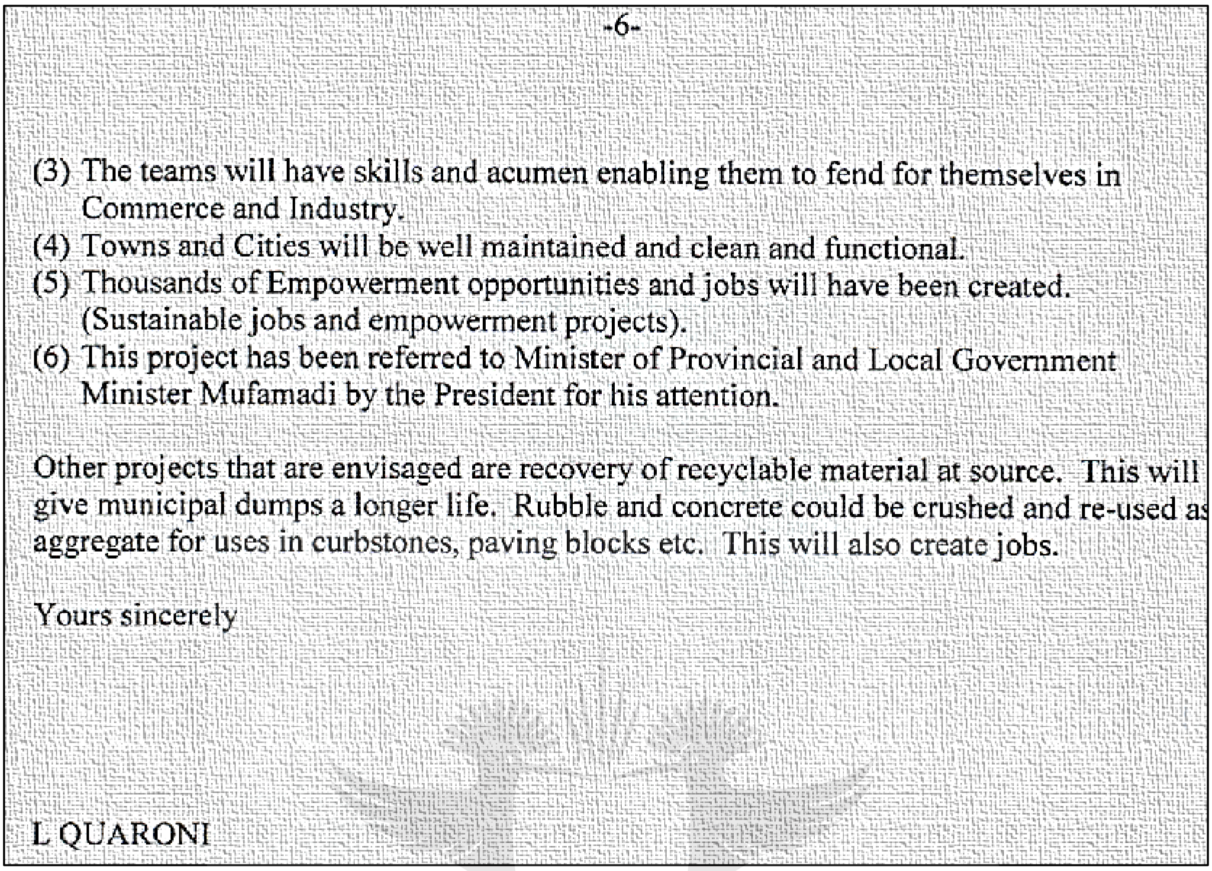
Contracts will be performed by a disciplined, reliable, competent and proud workforce.

Clean, maintained and presentable Cities will be a reality and not be just "blitz clean up when a major event is taking place.

Outcome

At the end of three years each team will have-

(1) a Fleet of capital equipment (Value approximately R1,5000.000.00) (One Million Five Hundred Thousand Rand)
(2) Working Capital available from the Trust Account ex the 10% Profit held in Trust (approximately Two Hundred and Fifty Thousand Rand).

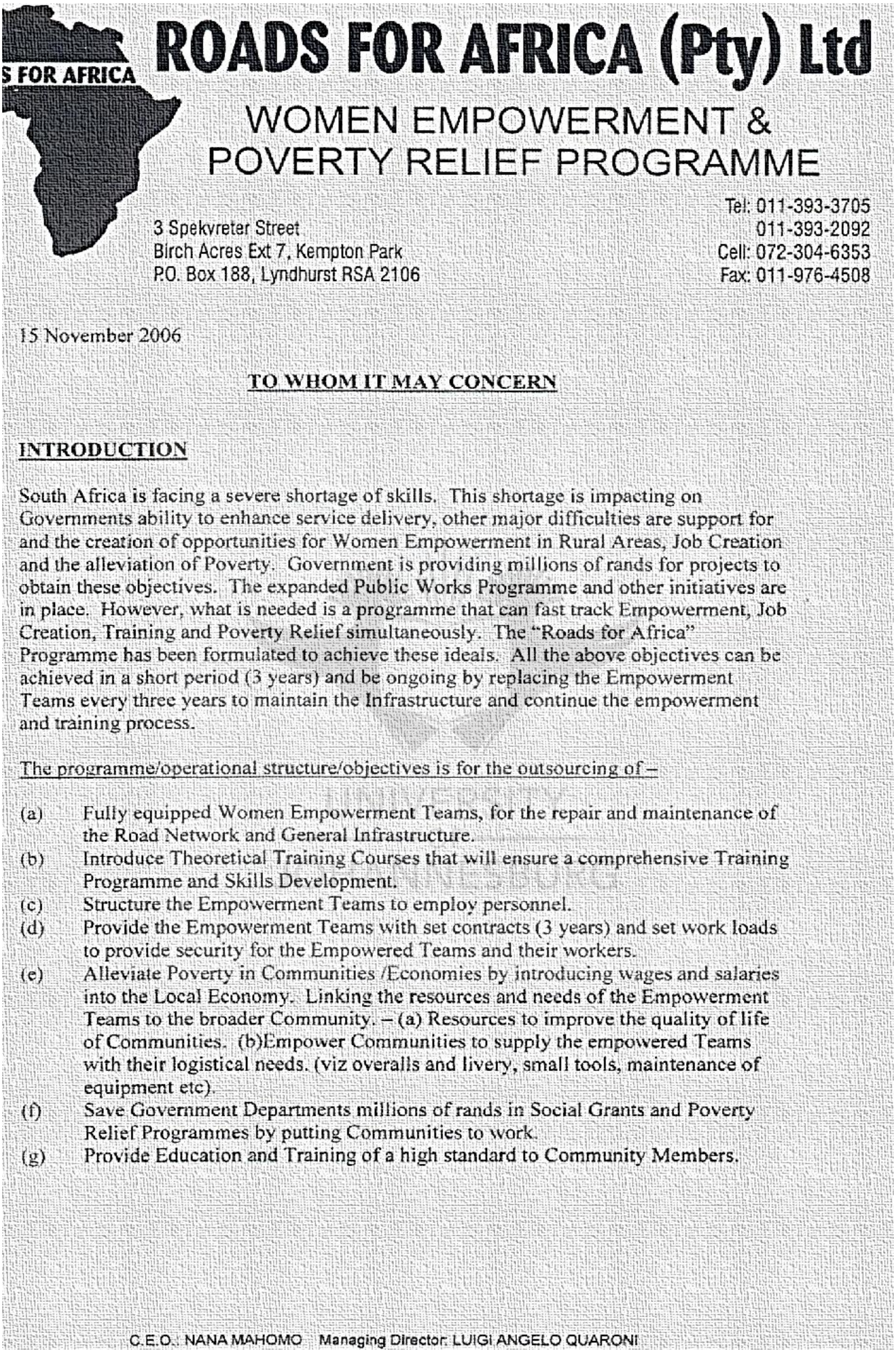


(3) The teams will have skills and acumen enabling them to fend for themselves in Commerce and Industry.
(4) Towns and Cities will be well maintained and clean and functional.
(5) Thousands of Empowerment opportunities and jobs will have been created. (Sustainable jobs and empowerment projects).
(6) This project has been referred to Minister of Provincial and Local Government Minister Mufamadi by the President for his attention.

Other projects that are envisaged are recovery of recyclable material at source. This will give municipal dumps a longer life. Rubble and concrete could be crushed and re-used as aggregate for uses in curbstones, paving blocks etc. This will also create jobs.

Yours sincerely

L QUARONI



ROADS FOR AFRICA (Pty) Ltd

WOMEN EMPOWERMENT &
POVERTY RELIEF PROGRAMME

3 Spekvreter Street
Birch Acres Ext 7, Kempton Park
P.O. Box 188, Lyndhurst RSA 2106

Tel: 011-393-3705
011-393-2092
Cell: 072-304-6353
Fax: 011-976-4508

15 November 2006

TO WHOM IT MAY CONCERN

INTRODUCTION

South Africa is facing a severe shortage of skills. This shortage is impacting on Governments ability to enhance service delivery, other major difficulties are support for and the creation of opportunities for Women Empowerment in Rural Areas, Job Creation and the alleviation of Poverty. Government is providing millions of rands for projects to obtain these objectives. The expanded Public Works Programme and other initiatives are in place. However, what is needed is a programme that can fast track Empowerment, Job Creation, Training and Poverty Relief simultaneously. The "Roads for Africa" Programme has been formulated to achieve these ideals. All the above objectives can be achieved in a short period (3 years) and be ongoing by replacing the Empowerment Teams every three years to maintain the Infrastructure and continue the empowerment and training process.

The programme/operational structure/objectives is for the outsourcing of –

(a) Fully equipped Women Empowerment Teams, for the repair and maintenance of the Road Network and General Infrastructure.
(b) Introduce Theoretical Training Courses that will ensure a comprehensive Training Programme and Skills Development.
(c) Structure the Empowerment Teams to employ personnel.
(d) Provide the Empowerment Teams with set contracts (3 years) and set work loads to provide security for the Empowered Teams and their workers.
(e) Alleviate Poverty in Communities /Economies by introducing wages and salaries into the Local Economy. Linking the resources and needs of the Empowerment Teams to the broader Community. – (a) Resources to improve the quality of life of Communities. (b)Empower Communities to supply the empowered Teams with their logistical needs. (viz overalls and livery, small tools, maintenance of equipment etc).
(f) Save Government Departments millions of rands in Social Grants and Poverty Relief Programmes by putting Communities to work.
(g) Provide Education and Training of a high standard to Community Members.

(h) Provide the necessary qualified on-site supervision and administration that will ensure the protection of all role players in the Programme. (Foremen, Engineers and Administration).
(i) Provide credible Diplomas and Certificates of Competency.
(j) Provide each Team of Owner Operators with yearly balance sheets.
(k) To use the work load as the Practical Training Medium.
(l) To use a Public Private Partnership to deliver the service.

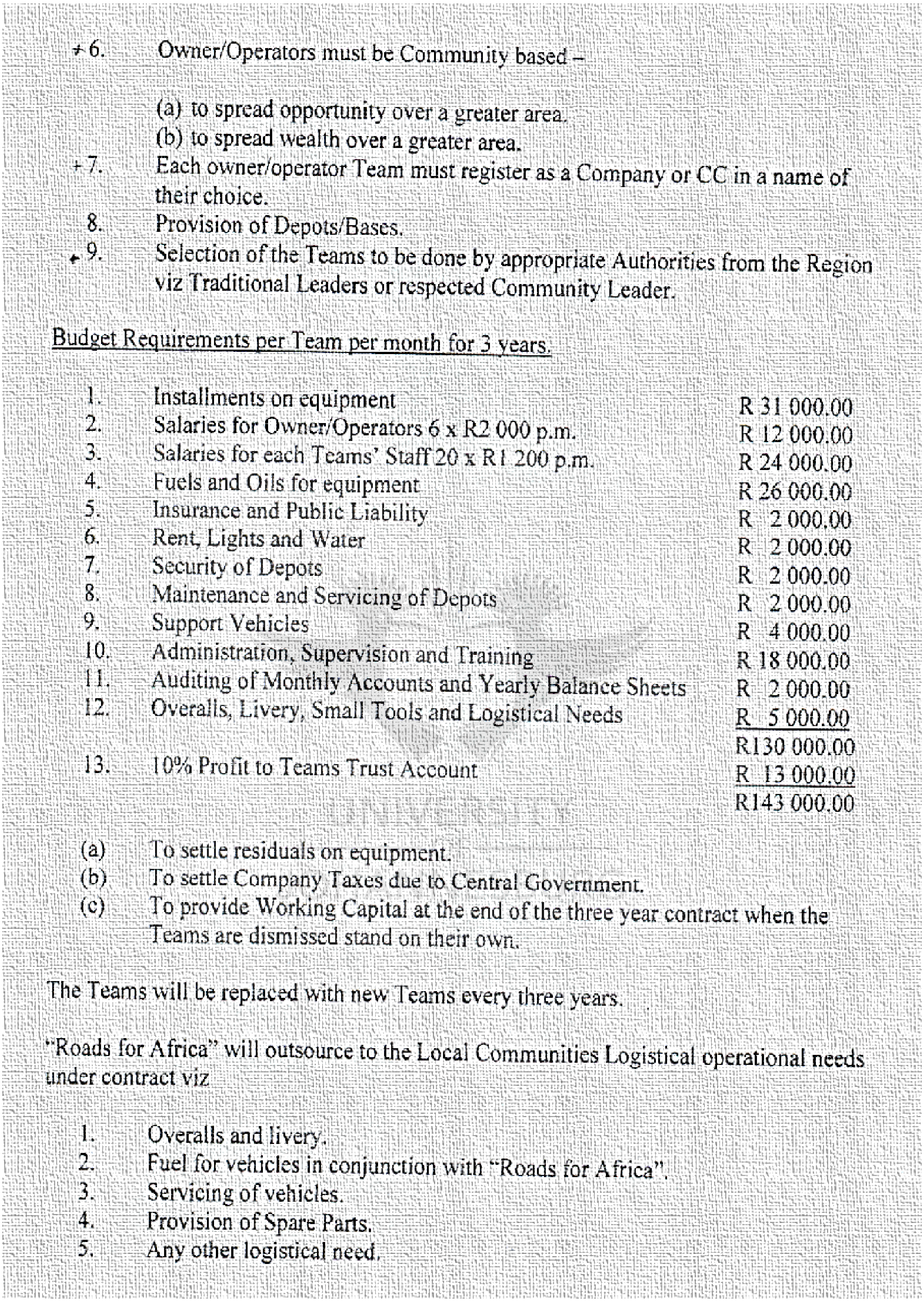
"Roads for Africa" will provide under a Public Private Partnership arrangement the following.

1. Supply the appropriate equipment to maintain Urban and Rural Roads and auxiliary Infrastructure work.
2. Trace and provide funding to acquire the necessary equipment required by the Empowerment Teams.
3. Trace funding at the lowest interest rate for the equipment.
4. Formulate a work load in conjunction with the relevant authorities that will cover the operational costs of each Empowerment Team. (A fair and reasonable work load).
5. Provide the Administrative Infrastructure for on-site Supervision and Training under the control of a competent Foreman and a District engineer (responsible person) and suitable depots.
6. Provide the necessary Theoretical Training under the control and guidance of a reputable Institution.
7. Provide monthly Trading Accounts and Yearly Balance Sheets to the Empowerment Teams (three balance sheets).

We request the support of the Premier of Limpopo and the Traditional Leaders to consider the Introduction of the Empowerment Teams to maintain the General Infrastructure.

<u>The support we request is as follows:-</u>

1. Sufficient Funding to pay for each Teams' monthly work load (approximately R140 000.00 (One hundred and forty thousand rand) for three years.
* 2. Formation of the Teams viz six owner operators per Team. (One owner operator per family).
* 3. Recruitment of each Team Labour force (20 personnel) is the responsibility of the owner operators. (One member per family).
* 4. Each owner/operator to have minimum of standard six.
* 5. Each owner/operator must have a valid learner's license.



+6. Owner/Operators must be Community based –

(a) to spread opportunity over a greater area.
(b) to spread wealth over a greater area.

+7. Each owner/operator Team must register as a Company or CC in a name of their choice.

8. Provision of Depots/Bases.

+9. Selection of the Teams to be done by appropriate Authorities from the Region viz Traditional Leaders or respected Community Leader.

<u>Budget Requirements per Team per month for 3 years.</u>

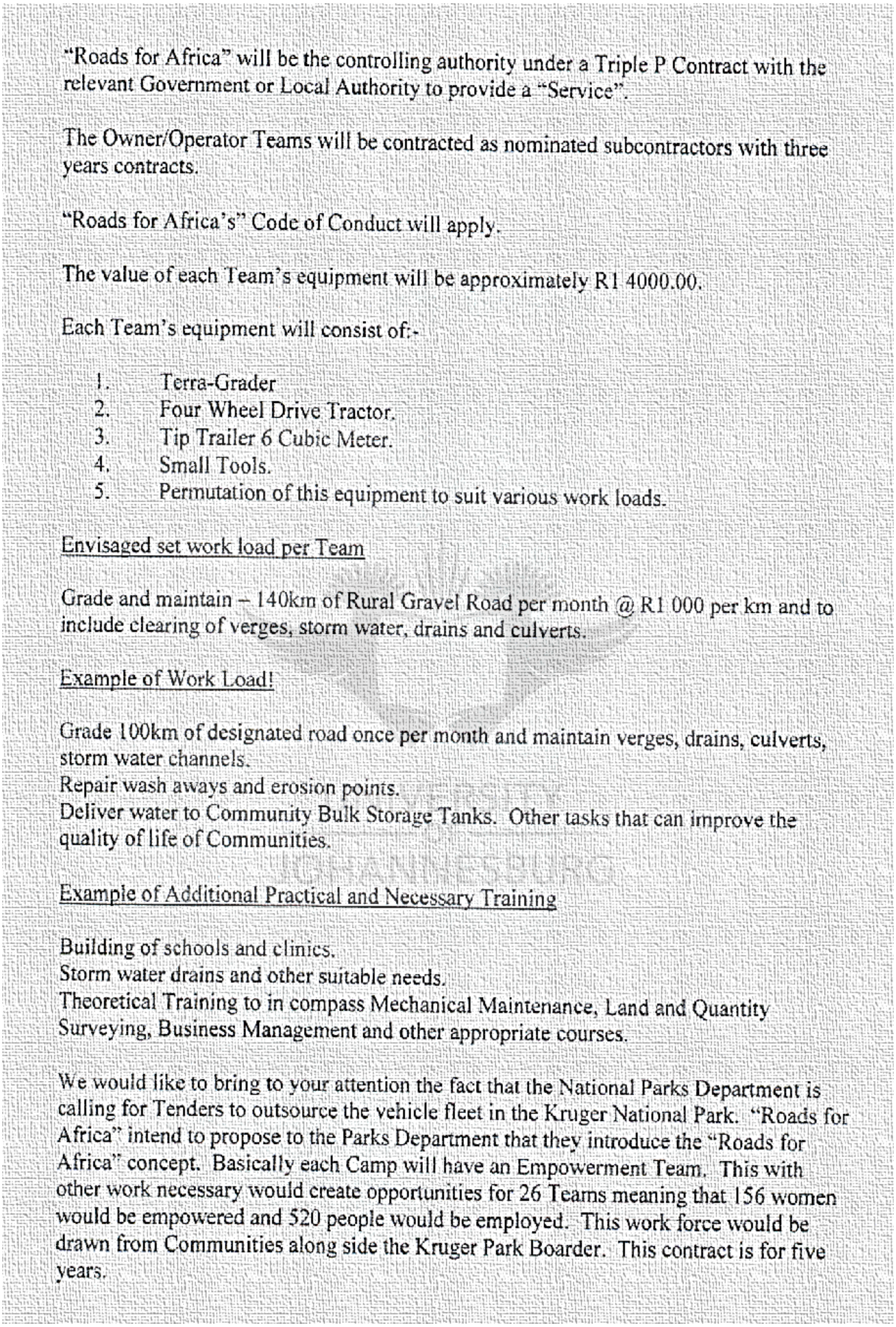
1.	Installments on equipment	R 31 000.00
2.	Salaries for Owner/Operators 6 x R2 000 p.m.	R 12 000.00
3.	Salaries for each Teams' Staff 20 x R1 200 p.m.	R 24 000.00
4.	Fuels and Oils for equipment	R 26 000.00
5.	Insurance and Public Liability	R 2 000.00
6.	Rent, Lights and Water	R 2 000.00
7.	Security of Depots	R 2 000.00
8.	Maintenance and Servicing of Depots	R 2 000.00
9.	Support Vehicles	R 4 000.00
10.	Administration, Supervision and Training	R 18 000.00
11.	Auditing of Monthly Accounts and Yearly Balance Sheets	R 2 000.00
12.	Overalls, Livery, Small Tools and Logistical Needs	R 5 000.00
		R130 000.00
13.	10% Profit to Teams Trust Account	R 13 000.00
		R143 000.00

(a) To settle residuals on equipment.
(b) To settle Company Taxes due to Central Government.
(c) To provide Working Capital at the end of the three year contract when the Teams are dismissed stand on their own.

The Teams will be replaced with new Teams every three years.

"Roads for Africa" will outsource to the Local Communities Logistical operational needs under contract viz

1. Overalls and livery.
2. Fuel for vehicles in conjunction with "Roads for Africa".
3. Servicing of vehicles.
4. Provision of Spare Parts.
5. Any other logistical need.



"Roads for Africa" will be the controlling authority under a Triple P Contract with the relevant Government or Local Authority to provide a "Service".

The Owner/Operator Teams will be contracted as nominated subcontractors with three years contracts.

"Roads for Africa's" Code of Conduct will apply.

The value of each Team's equipment will be approximately R1 4000.00.

Each Team's equipment will consist of:-

1. Terra-Grader
2. Four Wheel Drive Tractor.
3. Tip Trailer 6 Cubic Meter.
4. Small Tools.
5. Permutation of this equipment to suit various work loads.

Envisaged set work load per Team

Grade and maintain – 140km of Rural Gravel Road per month @ R1 000 per km and to include clearing of verges, storm water, drains and culverts.

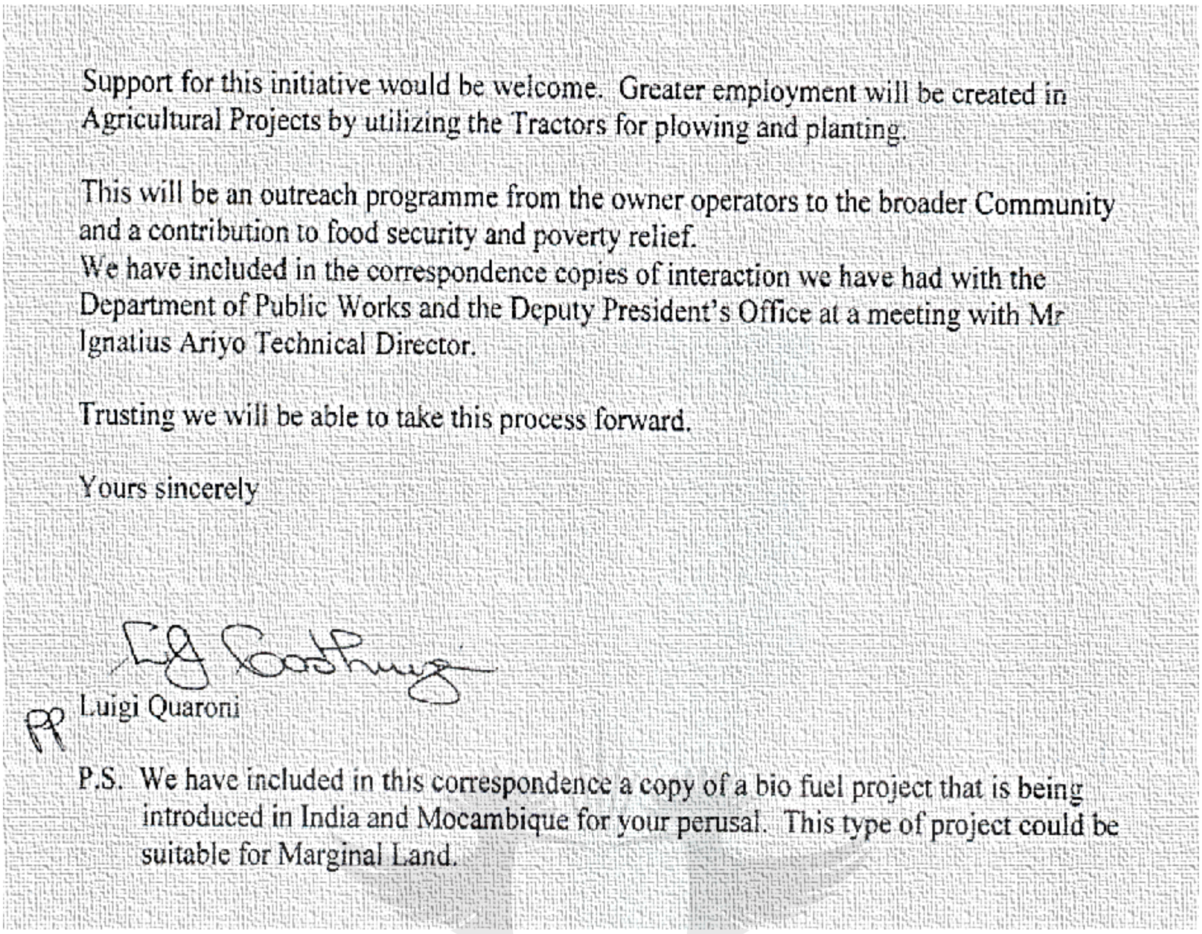
Example of Work Load!

Grade 100km of designated road once per month and maintain verges, drains, culverts, storm water channels.
Repair wash aways and erosion points.
Deliver water to Community Bulk Storage Tanks. Other tasks that can improve the quality of life of Communities.

Example of Additional Practical and Necessary Training

Building of schools and clinics.
Storm water drains and other suitable needs.
Theoretical Training to in compass Mechanical Maintenance, Land and Quantity Surveying, Business Management and other appropriate courses.

We would like to bring to your attention the fact that the National Parks Department is calling for Tenders to outsource the vehicle fleet in the Kruger National Park. "Roads for Africa" intend to propose to the Parks Department that they introduce the "Roads for Africa" concept. Basically each Camp will have an Empowerment Team. This with other work necessary would create opportunities for 26 Teams meaning that 156 women would be empowered and 520 people would be employed. This work force would be drawn from Communities along side the Kruger Park Boarder. This contract is for five years.



Support for this initiative would be welcome. Greater employment will be created in Agricultural Projects by utilizing the Tractors for plowing and planting.

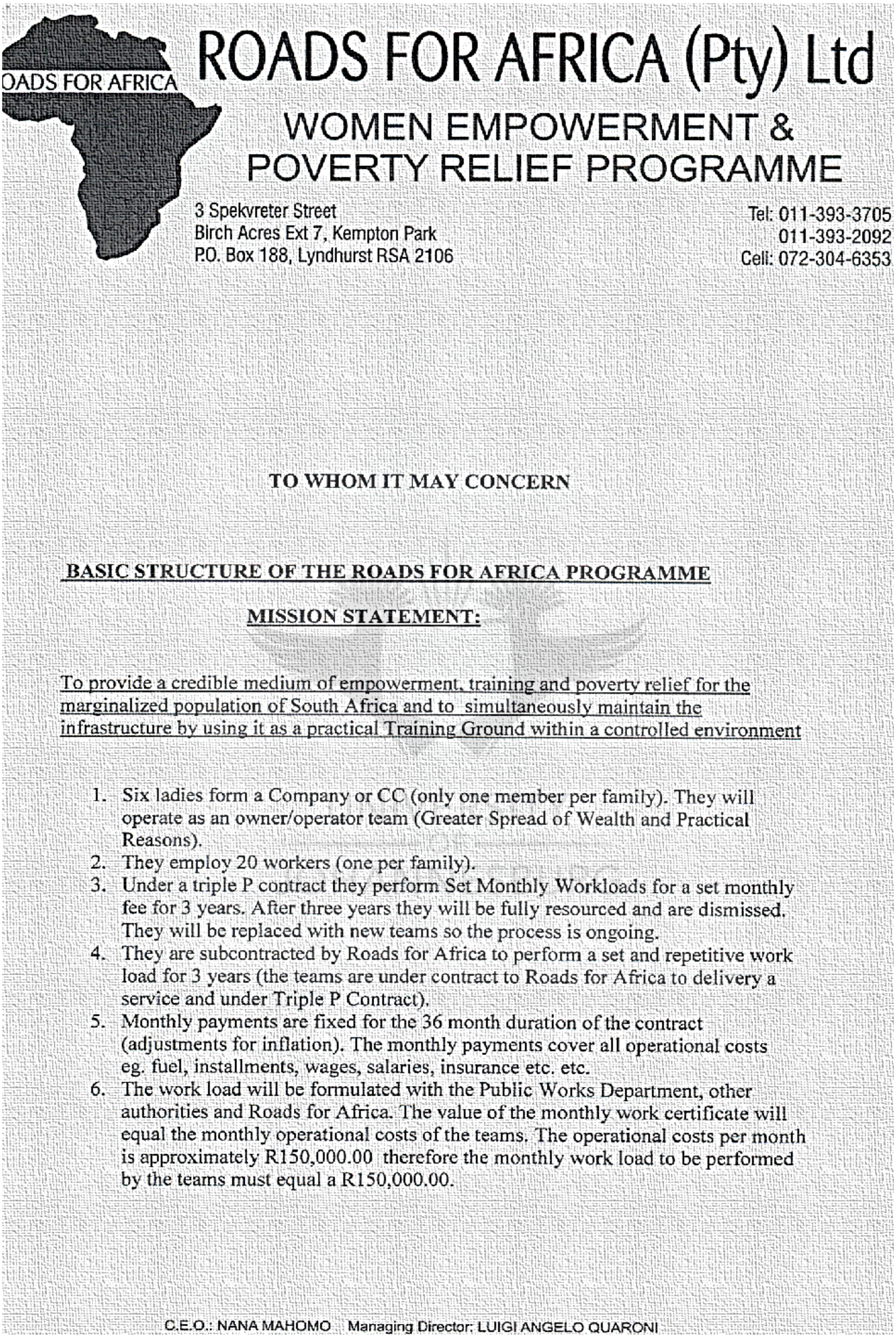
This will be an outreach programme from the owner operators to the broader Community and a contribution to food security and poverty relief.
We have included in the correspondence copies of interaction we have had with the Department of Public Works and the Deputy President's Office at a meeting with Mr Ignatius Ariyo Technical Director.

Trusting we will be able to take this process forward.

Yours sincerely

PP Luigi Quaroni

P.S. We have included in this correspondence a copy of a bio fuel project that is being introduced in India and Mocambique for your perusal. This type of project could be suitable for Marginal Land.



ROADS FOR AFRICA (Pty) Ltd
WOMEN EMPOWERMENT &
POVERTY RELIEF PROGRAMME

3 Spekvreter Street
Birch Acres Ext 7, Kempton Park
P.O. Box 188, Lyndhurst RSA 2106

Tel: 011-393-3705
011-393-2092
Cell: 072-304-6353

TO WHOM IT MAY CONCERN

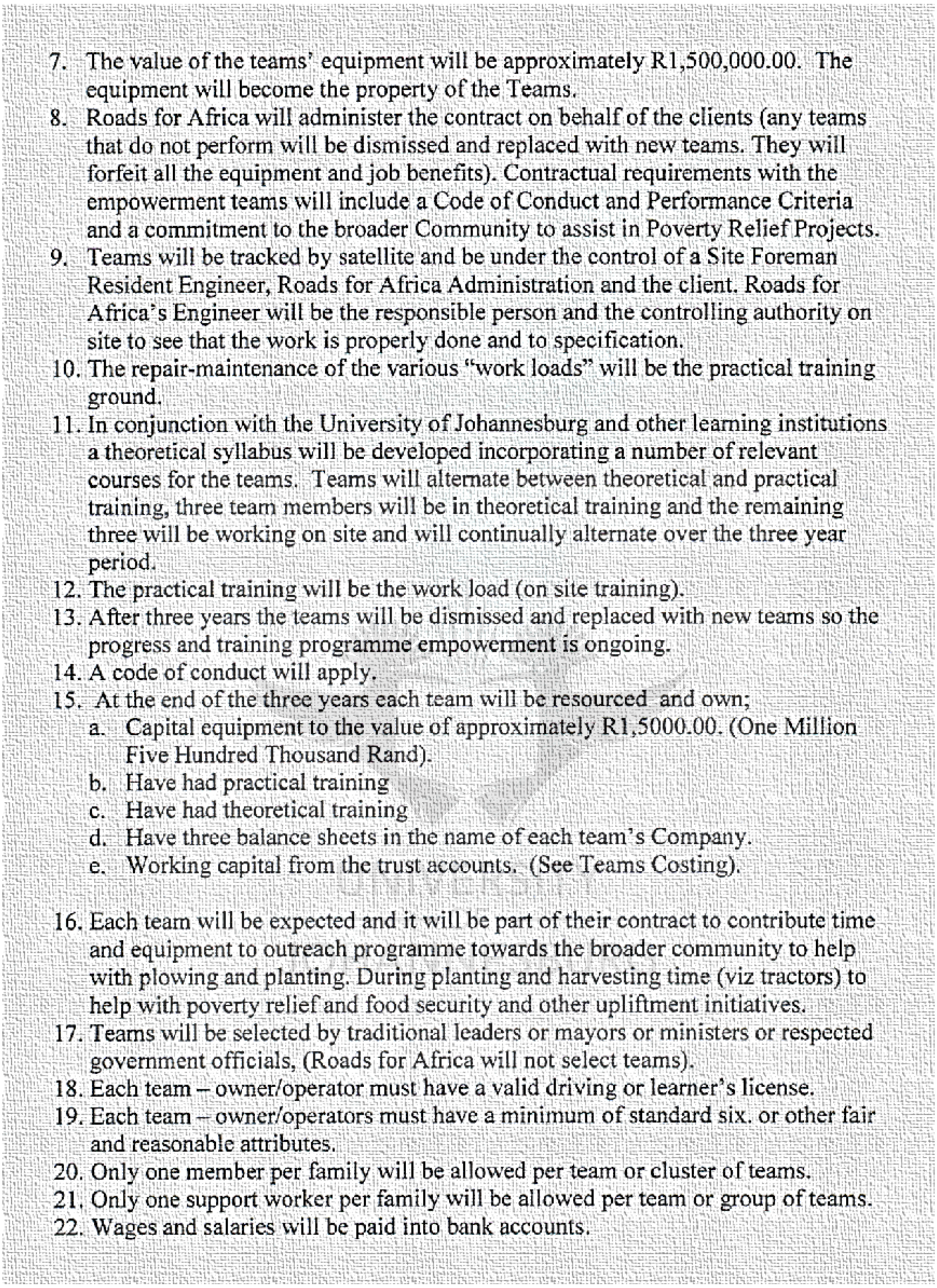
BASIC STRUCTURE OF THE ROADS FOR AFRICA PROGRAMME

MISSION STATEMENT:

To provide a credible medium of empowerment, training and poverty relief for the marginalized population of South Africa and to simultaneously maintain the infrastructure by using it as a practical Training Ground within a controlled environment

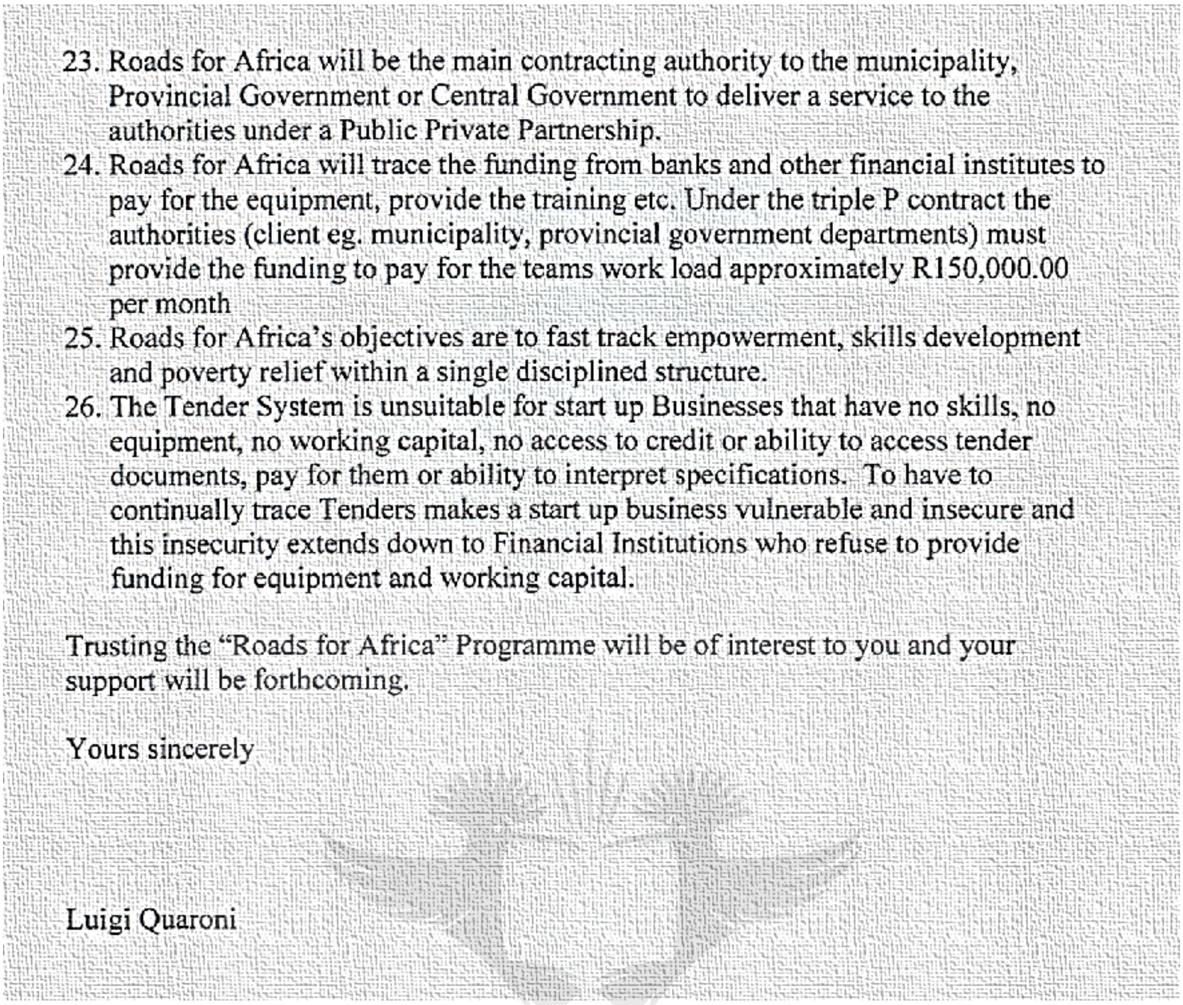
1. Six ladies form a Company or CC (only one member per family). They will operate as an owner/operator team (Greater Spread of Wealth and Practical Reasons).
2. They employ 20 workers (one per family).
3. Under a triple P contract they perform Set Monthly Workloads for a set monthly fee for 3 years. After three years they will be fully resourced and are dismissed. They will be replaced with new teams so the process is ongoing.
4. They are subcontracted by Roads for Africa to perform a set and repetitive work load for 3 years (the teams are under contract to Roads for Africa to delivery a service and under Triple P Contract).
5. Monthly payments are fixed for the 36 month duration of the contract (adjustments for inflation). The monthly payments cover all operational costs eg. fuel, installments, wages, salaries, insurance etc. etc.
6. The work load will be formulated with the Public Works Department, other authorities and Roads for Africa. The value of the monthly work certificate will equal the monthly operational costs of the teams. The operational costs per month is approximately R150,000.00 therefore the monthly work load to be performed by the teams must equal a R150,000.00.

C.E.O.: NANA MAHOMO Managing Director: LUIGI ANGELO QUARONI



7. The value of the teams' equipment will be approximately R1,500,000.00. The equipment will become the property of the Teams.
8. Roads for Africa will administer the contract on behalf of the clients (any teams that do not perform will be dismissed and replaced with new teams. They will forfeit all the equipment and job benefits). Contractual requirements with the empowerment teams will include a Code of Conduct and Performance Criteria and a commitment to the broader Community to assist in Poverty Relief Projects.
9. Teams will be tracked by satellite and be under the control of a Site Foreman Resident Engineer, Roads for Africa Administration and the client. Roads for Africa's Engineer will be the responsible person and the controlling authority on site to see that the work is properly done and to specification.
10. The repair-maintenance of the various "work loads" will be the practical training ground.
11. In conjunction with the University of Johannesburg and other learning institutions a theoretical syllabus will be developed incorporating a number of relevant courses for the teams. Teams will alternate between theoretical and practical training, three team members will be in theoretical training and the remaining three will be working on site and will continually alternate over the three year period.
12. The practical training will be the work load (on site training).
13. After three years the teams will be dismissed and replaced with new teams so the progress and training programme empowerment is ongoing.
14. A code of conduct will apply.
15. At the end of the three years each team will be resourced and own;
    a. Capital equipment to the value of approximately R1,5000.00. (One Million Five Hundred Thousand Rand).
    b. Have had practical training
    c. Have had theoretical training
    d. Have three balance sheets in the name of each team's Company.
    e. Working capital from the trust accounts. (See Teams Costing).

16. Each team will be expected and it will be part of their contract to contribute time and equipment to outreach programme towards the broader community to help with plowing and planting. During planting and harvesting time (viz tractors) to help with poverty relief and food security and other upliftment initiatives.
17. Teams will be selected by traditional leaders or mayors or ministers or respected government officials, (Roads for Africa will not select teams).
18. Each team – owner/operator must have a valid driving or learner's license.
19. Each team – owner/operators must have a minimum of standard six. or other fair and reasonable attributes.
20. Only one member per family will be allowed per team or cluster of teams.
21. Only one support worker per family will be allowed per team or group of teams.
22. Wages and salaries will be paid into bank accounts.



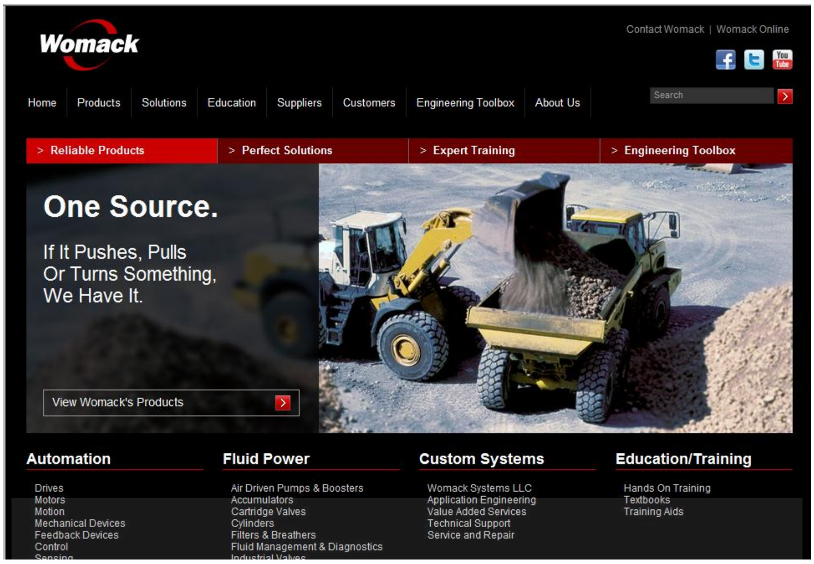
23. Roads for Africa will be the main contracting authority to the municipality, Provincial Government or Central Government to deliver a service to the authorities under a Public Private Partnership.
24. Roads for Africa will trace the funding from banks and other financial institutes to pay for the equipment, provide the training etc. Under the triple P contract the authorities (client eg. municipality, provincial government departments) must provide the funding to pay for the teams work load approximately R150,000.00 per month
25. Roads for Africa's objectives are to fast track empowerment, skills development and poverty relief within a single disciplined structure.
26. The Tender System is unsuitable for start up Businesses that have no skills, no equipment, no working capital, no access to credit or ability to access tender documents, pay for them or ability to interpret specifications. To have to continually trace Tenders makes a start up business vulnerable and insecure and this insecurity extends down to Financial Institutions who refuse to provide funding for equipment and working capital.

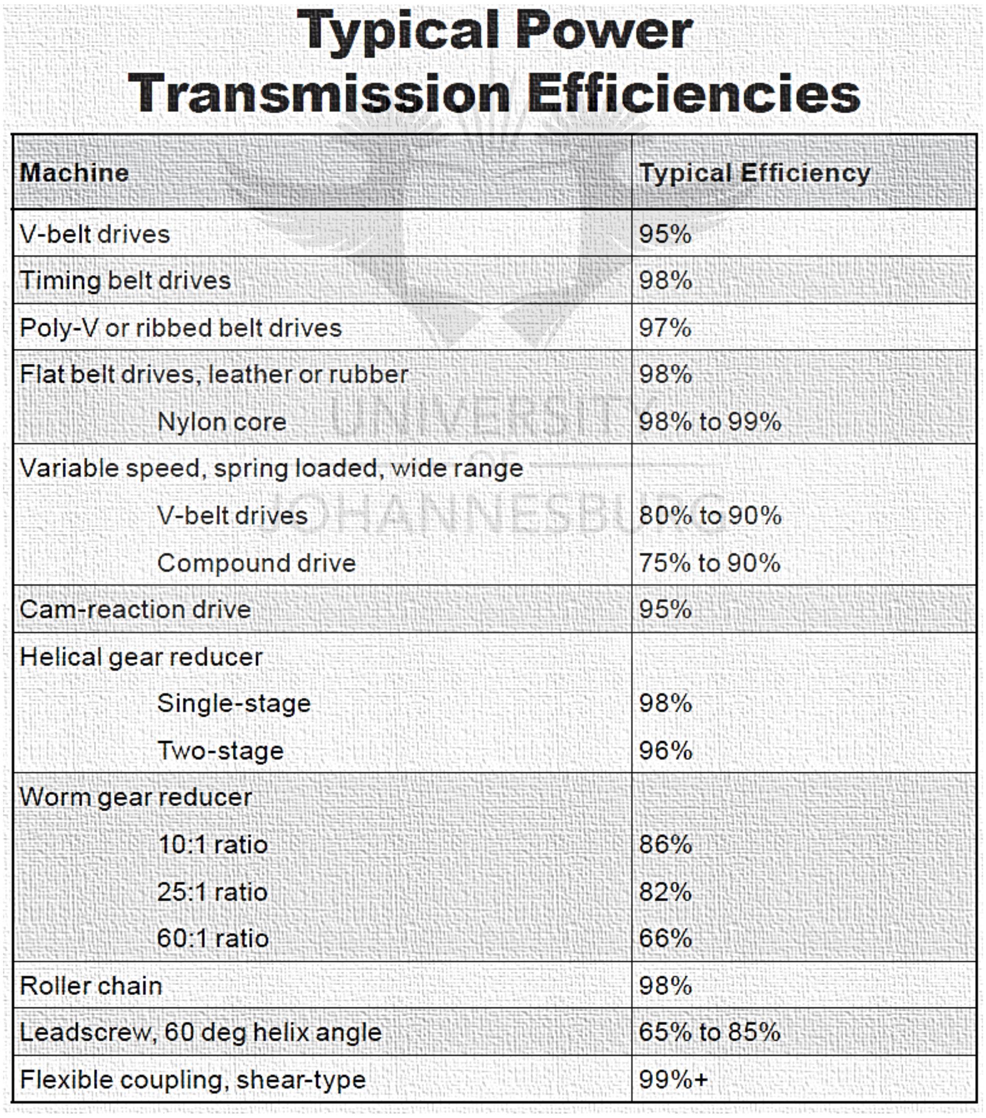
Trusting the "Roads for Africa" Programme will be of interest to you and your support will be forthcoming.

Yours sincerely

Luigi Quaroni

Appendix 2: Womac (PTY) LTD typical power transmission efficiencies.

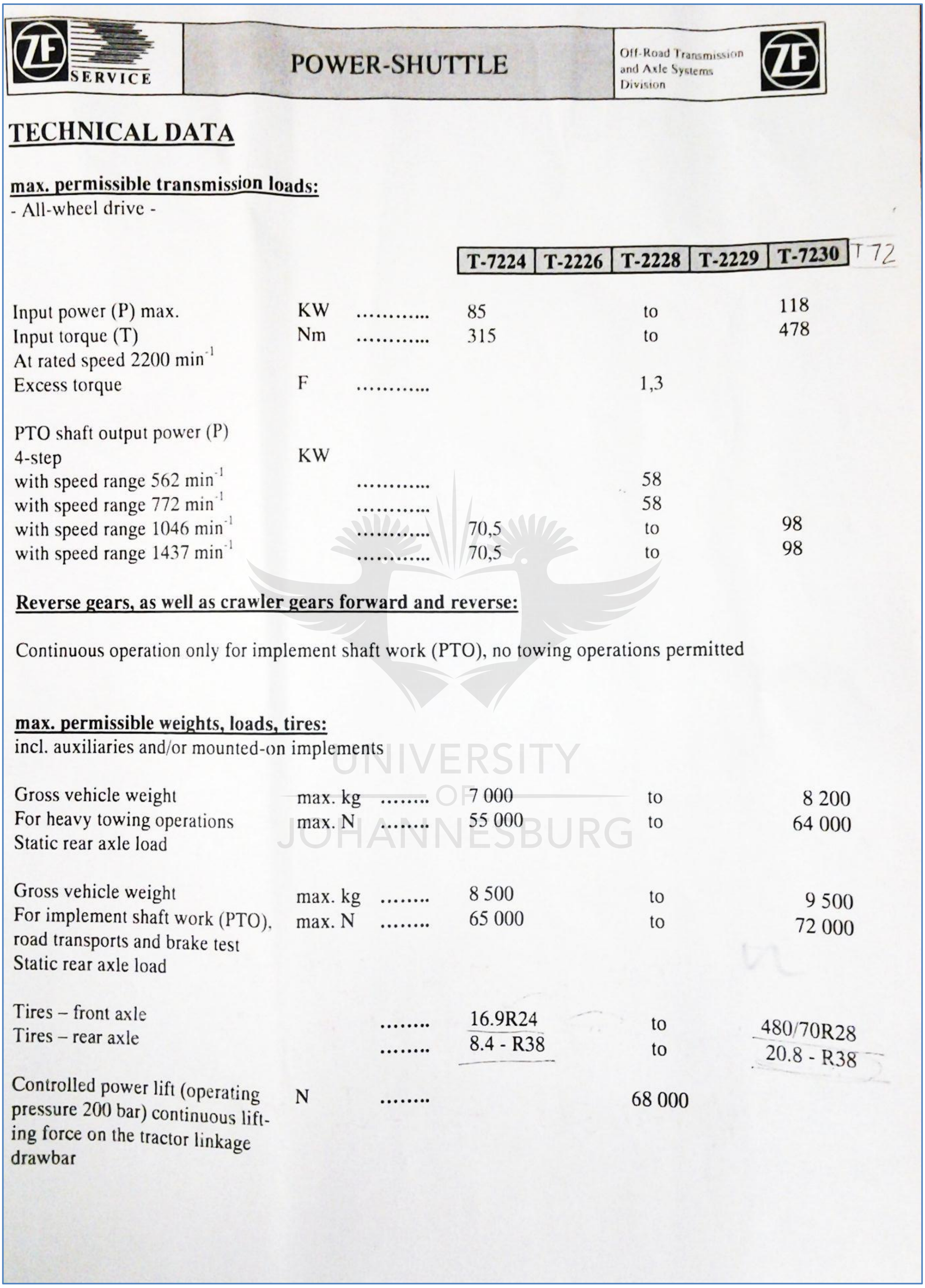




Typical Power Transmission Efficiencies

Machine	Typical Efficiency
V-belt drives	95%
Timing belt drives	98%
Poly-V or ribbed belt drives	97%
Flat belt drives, leather or rubber	98%
Nylon core	98% to 99%
Variable speed, spring loaded, wide range	
V-belt drives	80% to 90%
Compound drive	75% to 90%
Cam-reaction drive	95%
Helical gear reducer	
Single-stage	98%
Two-stage	96%
Worm gear reducer	
10:1 ratio	86%
25:1 ratio	82%
60:1 ratio	66%
Roller chain	98%
Leadscrew, 60 deg helix angle	65% to 85%
Flexible coupling, shear-type	99%+

Appendix 3: ZF Power Take OFF (PTO) specification data.



ZF SERVICE

POWER-SHUTTLE

Off-Road Transmission
and Axle Systems
Division

TECHNICAL DATA

max. permissible transmission loads:
- All-wheel drive -

			T-7224	T-2226	T-2228	T-2229	T-7230	T72
Input power (P) max.	KW	………..	85		to		118	
Input torque (T)	Nm	………..	315		to		478	
At rated speed 2200 min^{-1}								
Excess torque	F	………..			1,3			
PTO shaft output power (P)								
4-step	KW							
with speed range 562 min^{-1}		………..			58			
with speed range 772 min^{-1}		………..			58			
with speed range 1046 min^{-1}		………..	70,5		to		98	
with speed range 1437 min^{-1}		………..	70,5		to		98	

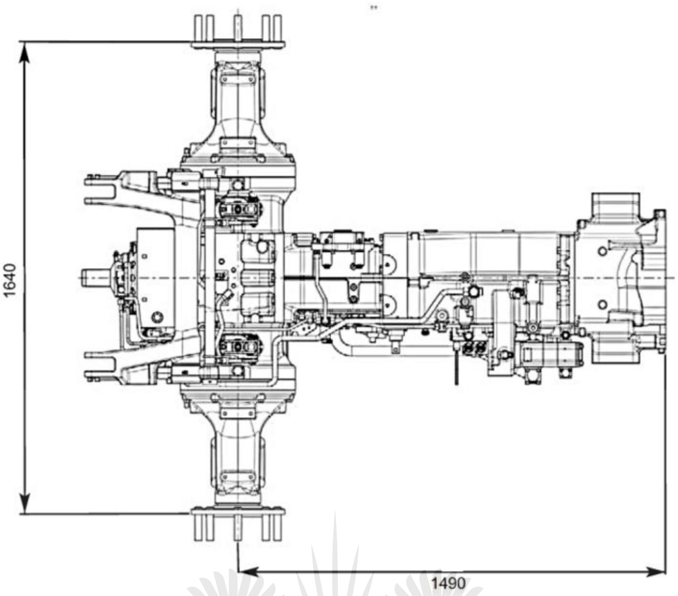
Reverse gears, as well as crawler gears forward and reverse:

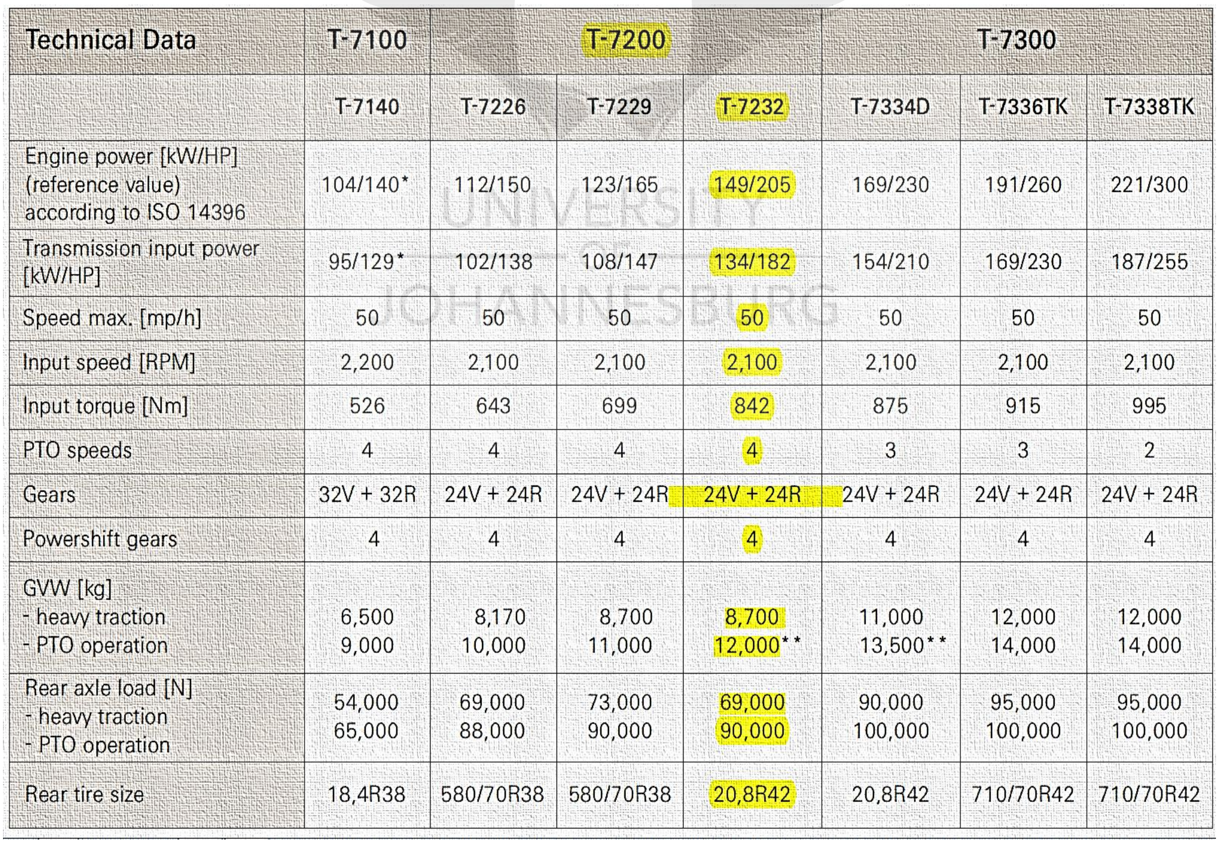
Continuous operation only for implement shaft work (PTO), no towing operations permitted

max. permissible weights, loads, tires:
incl. auxiliaries and/or mounted-on implements

Gross vehicle weight	max. kg	…….	7 000	to	8 200
For heavy towing operations	max. N	…….	55 000	to	64 000
Static rear axle load					
Gross vehicle weight	max. kg	…….	8 500	to	9 500
For implement shaft work (PTO), road transports and brake test	max. N	…….	65 000	to	72 000
Static rear axle load					
Tires – front axle		…….	16.9R24	to	480/70R28
Tires – rear axle		…….	8.4 - R38	to	20.8 - R38
Controlled power lift (operating pressure 200 bar) continuous lifting force on the tractor linkage drawbar	N	…….		68 000	

Driveline and Chassis Technology

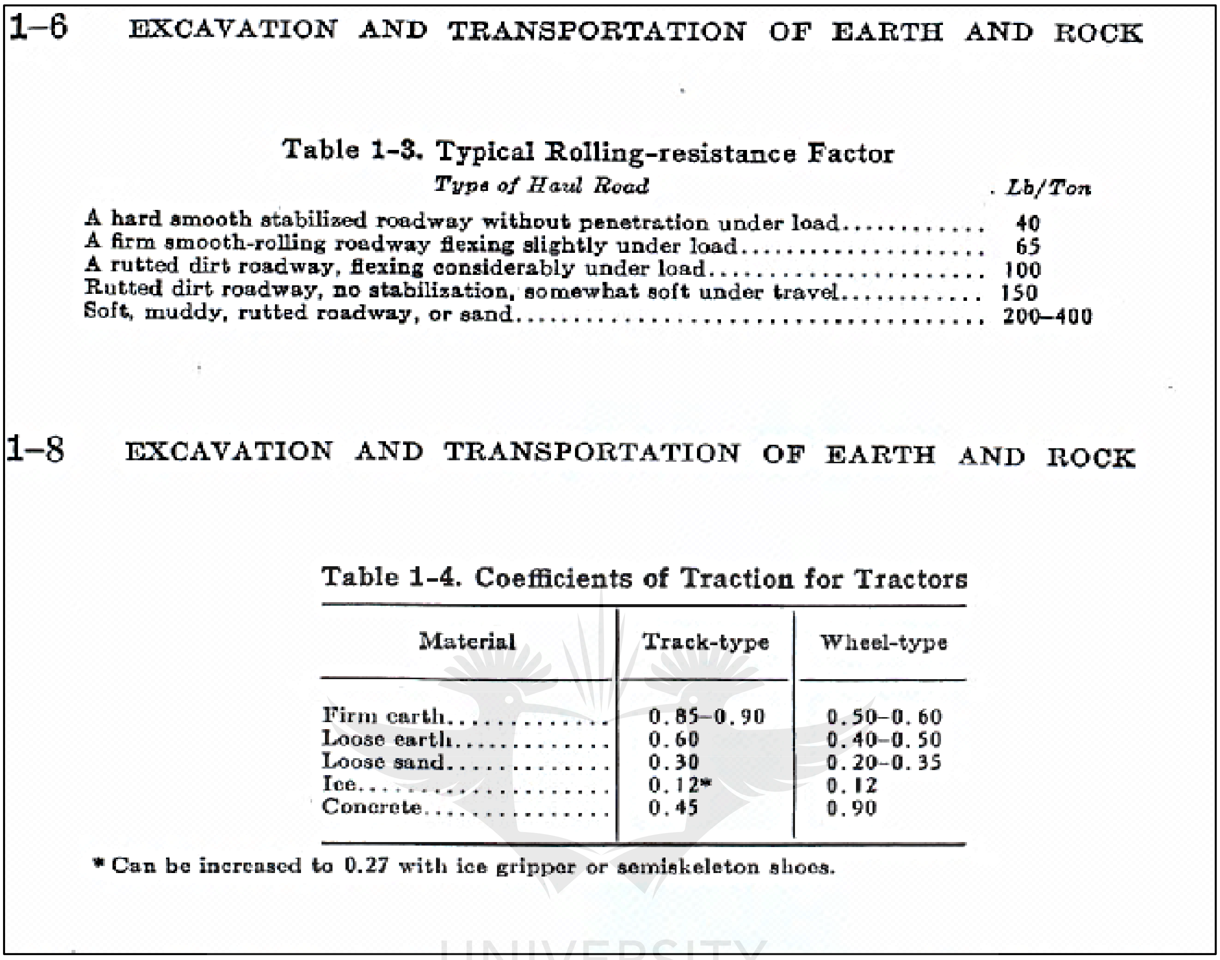




Technical Data	T-7100	T-7200			T-7300		
	T-7140	T-7226	T-7229	T-7232	T-7334D	T-7336TK	T-7338TK
Engine power [kW/HP] (reference value) according to ISO 14396	104/140*	112/150	123/165	149/205	169/230	191/260	221/300
Transmission input power [kW/HP]	95/129*	102/138	108/147	134/182	154/210	169/230	187/255
Speed max. [mp/h]	50	50	50	50	50	50	50
Input speed [RPM]	2,200	2,100	2,100	2,100	2,100	2,100	2,100
Input torque [Nm]	526	643	699	842	875	915	995
PTO speeds	4	4	4	4	3	3	2
Gears	32V + 32R	24V + 24R	24V + 24R	24V + 24R	24V + 24R	24V + 24R	24V + 24R
Powershift gears	4	4	4	4	4	4	4
GVW [kg] - heavy traction - PTO operation	6,500 9,000	8,170 10,000	8,700 11,000	8,700 12,000**	11,000 13,500**	12,000 14,000	12,000 14,000
Rear axle load [N] - heavy traction - PTO operation	54,000 65,000	69,000 88,000	73,000 90,000	69,000 90,000	90,000 100,000	95,000 100,000	95,000 100,000
Rear tire size	18,4R38	580/70R38	580/70R38	20,8R42	20,8R42	710/70R42	710/70R42

Appendix 4: Rolling resistance.

1. FW Stubbs



1–6 EXCAVATION AND TRANSPORTATION OF EARTH AND ROCK

Table 1-3. Typical Rolling-resistance Factor

Type of Haul Road	*Lb/Ton*
A hard smooth stabilized roadway without penetration under load	40
A firm smooth-rolling roadway flexing slightly under load	65
A rutted dirt roadway, flexing considerably under load	100
Rutted dirt roadway, no stabilization, somewhat soft under travel	150
Soft, muddy, rutted roadway, or sand	200–400

1–8 EXCAVATION AND TRANSPORTATION OF EARTH AND ROCK

Table 1-4. Coefficients of Traction for Tractors

Material	Track-type	Wheel-type
Firm earth	0.85–0.90	0.50–0.60
Loose earth	0.60	0.40–0.50
Loose sand	0.30	0.20–0.35
Ice	0.12*	0.12
Concrete	0.45	0.90

* Can be increased to 0.27 with ice gripper or semiskeleton shoes.

2. Wenger, Karl F

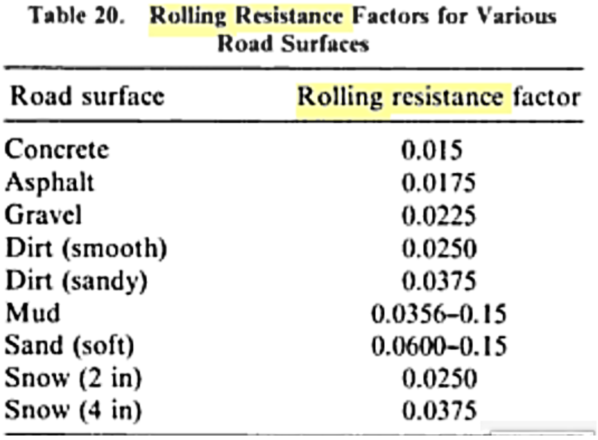
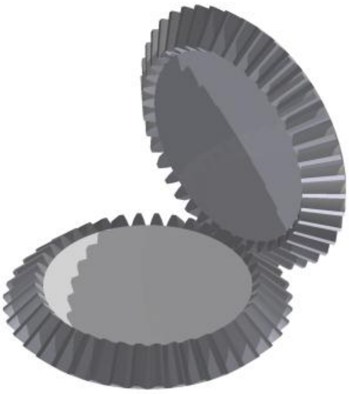
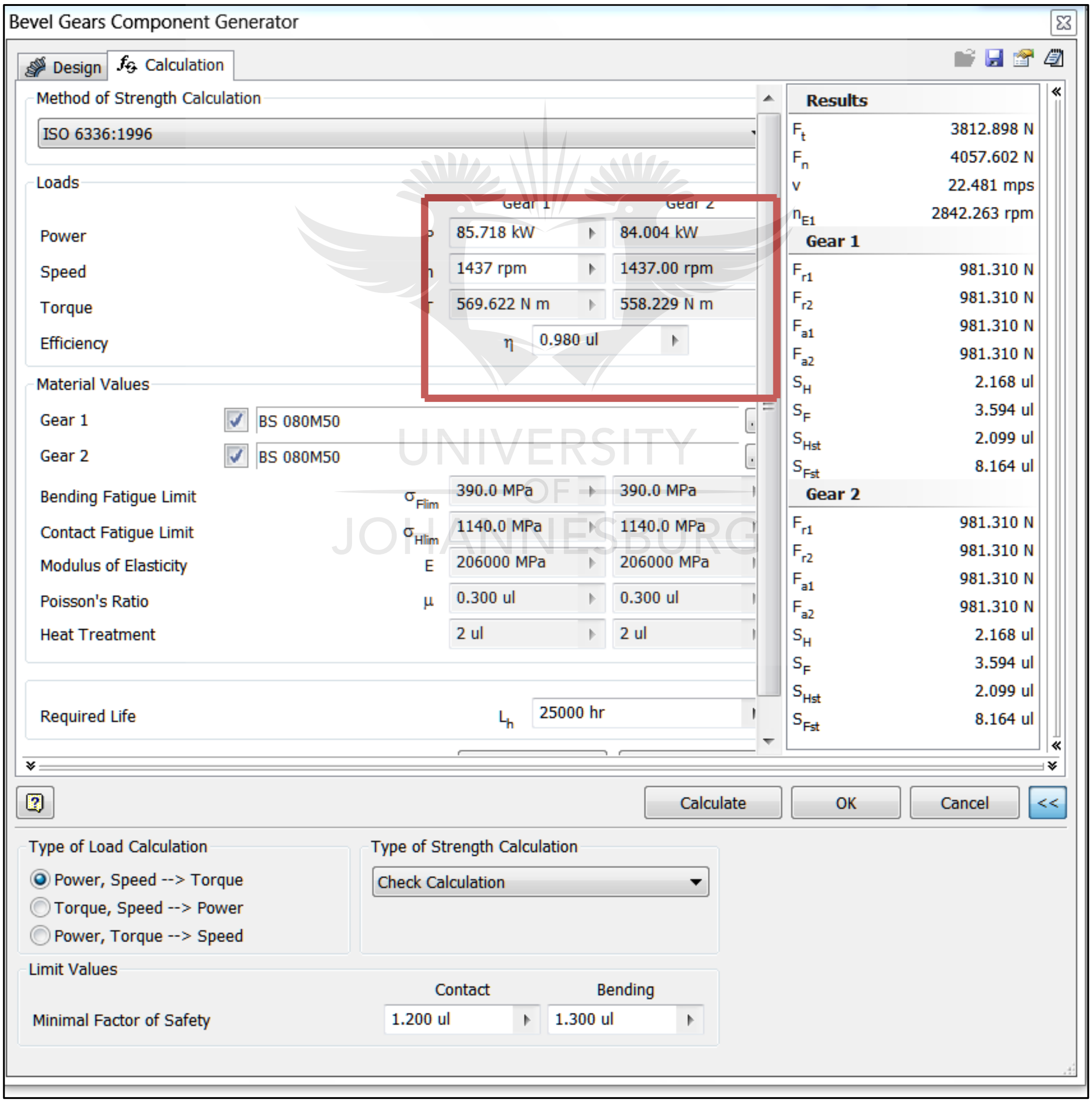


Table 20. Rolling Resistance Factors for Various Road Surfaces

Road surface	Rolling resistance factor
Concrete	0.015
Asphalt	0.0175
Gravel	0.0225
Dirt (smooth)	0.0250
Dirt (sandy)	0.0375
Mud	0.0356–0.15
Sand (soft)	0.0600–0.15
Snow (2 in)	0.0250
Snow (4 in)	0.0375

Appendix 5: Bevel Gear Design Success Inventor.





Method of Strength Calculation: ISO 6336:1996

Loads

		Gear 1	Gear 2
Power	P	85.718 kW	84.004 kW
Speed	n	1437 rpm	1437.00 rpm
Torque	T	569.622 N m	558.229 N m
Efficiency	η	0.980 ul	

Material Values

Gear 1	☑	BS 080M50	
Gear 2	☑	BS 080M50	
Bending Fatigue Limit	σ_{Flim}	390.0 MPa	390.0 MPa
Contact Fatigue Limit	σ_{Hlim}	1140.0 MPa	1140.0 MPa
Modulus of Elasticity	E	206000 MPa	206000 MPa
Poisson's Ratio	μ	0.300 ul	0.300 ul
Heat Treatment		2 ul	2 ul

Required Life L_h 25000 hr

Results

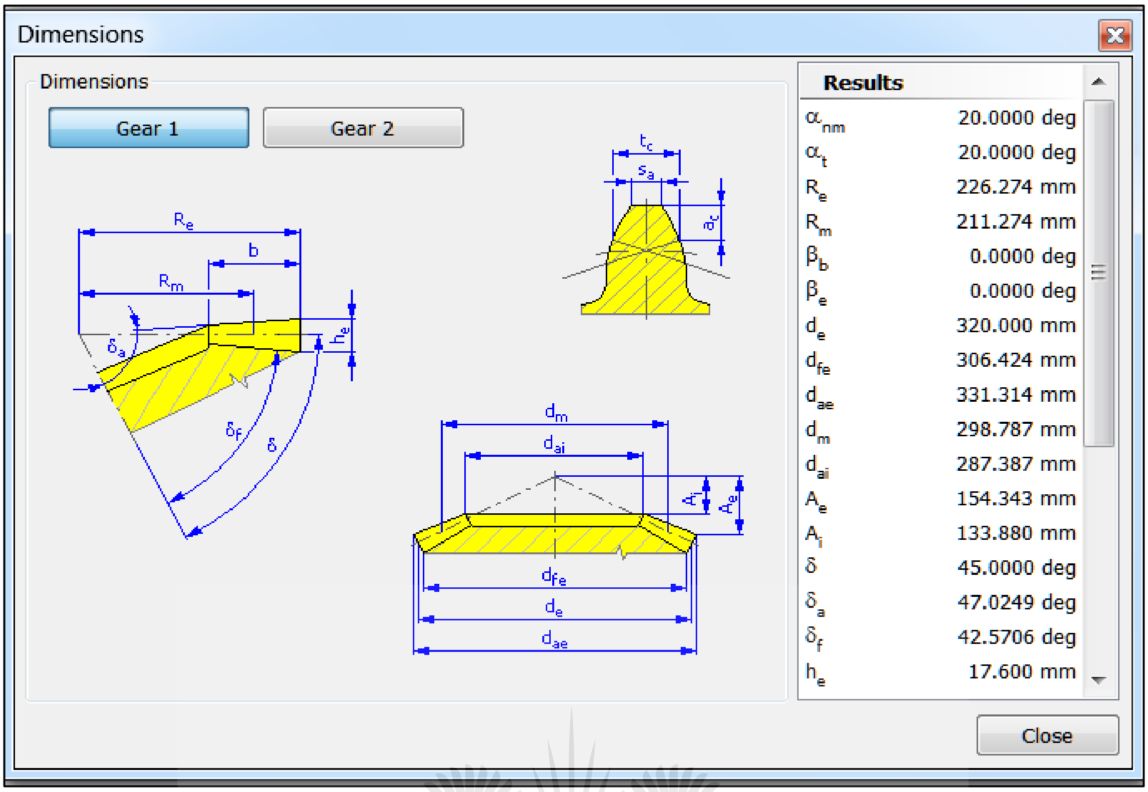
F_t	3812.898 N
F_n	4057.602 N
v	22.481 mps
n_{E1}	2842.263 rpm
Gear 1	
F_{r1}	981.310 N
F_{r2}	981.310 N
F_{a1}	981.310 N
F_{a2}	981.310 N
S_H	2.168 ul
S_F	3.594 ul
S_{Hst}	2.099 ul
S_{Fst}	8.164 ul
Gear 2	
F_{r1}	981.310 N
F_{r2}	981.310 N
F_{a1}	981.310 N
F_{a2}	981.310 N
S_H	2.168 ul
S_F	3.594 ul
S_{Hst}	2.099 ul
S_{Fst}	8.164 ul

Type of Load Calculation
- ● Power, Speed --> Torque
- ○ Torque, Speed --> Power
- ○ Power, Torque --> Speed

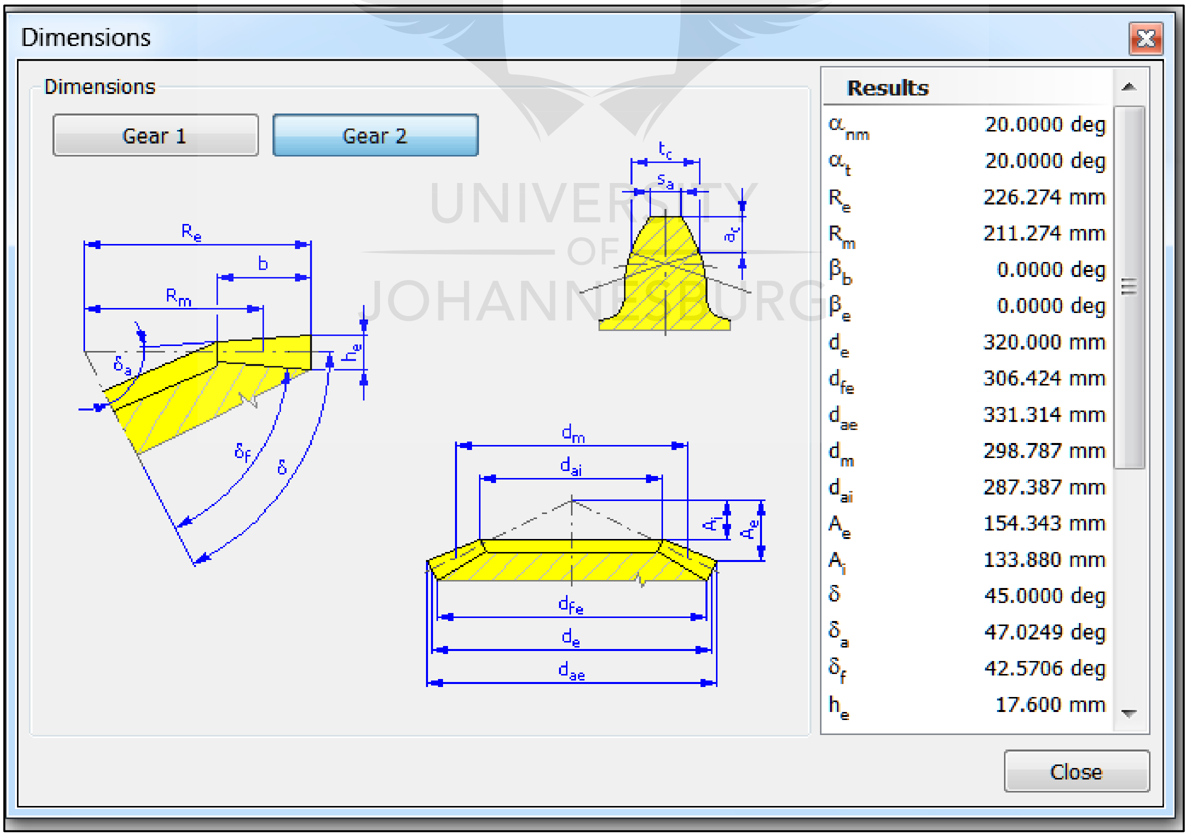
Type of Strength Calculation: Check Calculation

Limit Values

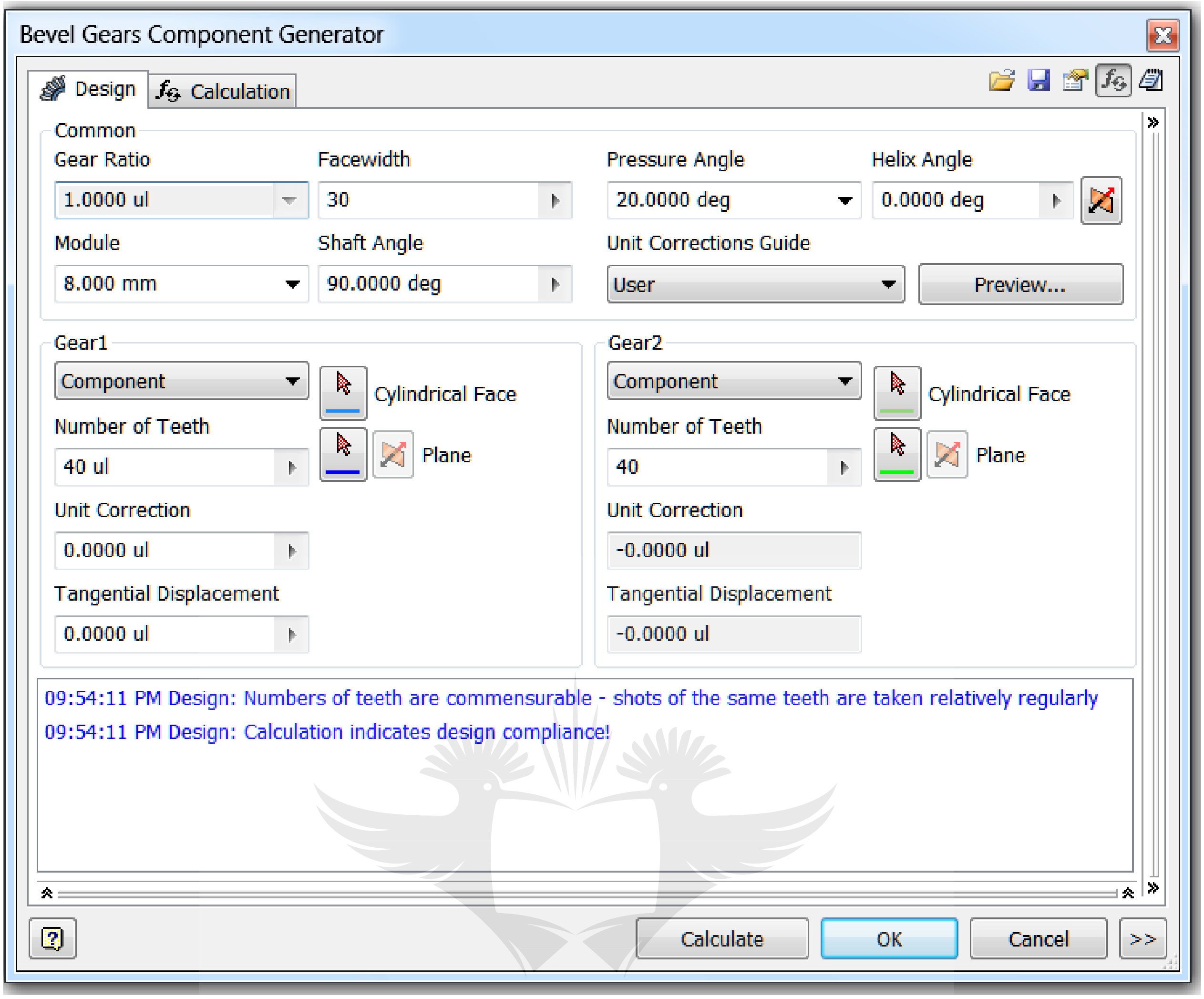
	Contact	Bending
Minimal Factor of Safety	1.200 ul	1.300 ul



Results	
α_{nm}	20.0000 deg
α_t	20.0000 deg
R_e	226.274 mm
R_m	211.274 mm
β_b	0.0000 deg
β_e	0.0000 deg
d_e	320.000 mm
d_{fe}	306.424 mm
d_{ae}	331.314 mm
d_m	298.787 mm
d_{ai}	287.387 mm
A_e	154.343 mm
A_i	133.880 mm
δ	45.0000 deg
δ_a	47.0249 deg
δ_f	42.5706 deg
h_e	17.600 mm



Results	
α_{nm}	20.0000 deg
α_t	20.0000 deg
R_e	226.274 mm
R_m	211.274 mm
β_b	0.0000 deg
β_e	0.0000 deg
d_e	320.000 mm
d_{fe}	306.424 mm
d_{ae}	331.314 mm
d_m	298.787 mm
d_{ai}	287.387 mm
A_e	154.343 mm
A_i	133.880 mm
δ	45.0000 deg
δ_a	47.0249 deg
δ_f	42.5706 deg
h_e	17.600 mm



Bevel Gears Component Generator

Common

Gear Ratio: 1.0000 ul
Facewidth: 30
Pressure Angle: 20.0000 deg
Helix Angle: 0.0000 deg
Module: 8.000 mm
Shaft Angle: 90.0000 deg
Unit Corrections Guide: User

Gear1

Component
Number of Teeth: 40 ul
Unit Correction: 0.0000 ul
Tangential Displacement: 0.0000 ul

Gear2

Component
Number of Teeth: 40
Unit Correction: -0.0000 ul
Tangential Displacement: -0.0000 ul

09:54:11 PM Design: Numbers of teeth are commensurable - shots of the same teeth are taken relatively regularly
09:54:11 PM Design: Calculation indicates design compliance!

Unit Corrections Guide - User

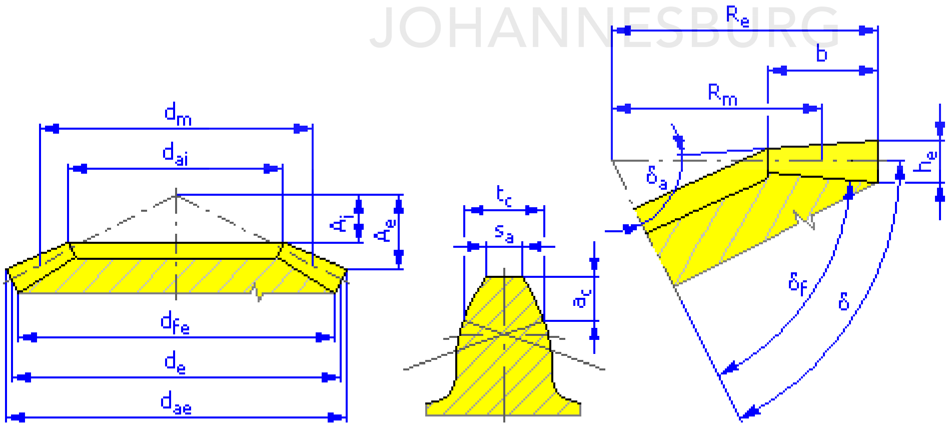
Type of Load Calculation - Torque calculation for the specified power and speed

Type of Strength Calculation - Check Calculation

Method of Strength Calculation - ISO 6336:1996

Common Parameters

Gear Ratio	i	1.0000 ul
Tangential Module	m_{et}	8.000 mm
Helix Angle	β	0.00 deg
Tangential Pressure Angle	α_t	20.0000 deg
Shaft Angle	Σ	90.00 deg
Normal Pressure Angle at End	α_{ne}	20.0000 deg
Contact Ratio	ε	1.7754 ul
Limit Deviation of Axis Parallelism	f_x	0.0150 mm
Limit Deviation of Axis Parallelism	f_y	0.0075 mm
Virtual Gear Ratio	i_v	1.000 ul
Equivalent Centre Distance	a_v	402.548 mm
Virtual Centre Distance	a_n	402.548 mm
Pitch Cone Radius	R_e	226.274 mm
Pitch Cone Radius in Middle Plane	R_m	201.274 mm



Gears Dimensions

		Gear 1	Gear 2
Type of model		Component	Component
Number of Teeth	z	40.000 ul	40.000 ul
Unit Correction	x	0.000 ul	-0.0000 ul
Tangential Displacement	x_t	0.0000 ul	-0.0000 ul
Pitch Diameter at End	d_e	320.000 mm	320.000 mm
Pitch Diameter in Middle Plane	d_m	284.645 mm	284.645 mm
Outside Diameter at End	d_{ae}	331.314 mm	331.314 mm
Outside Diameter at Small End	d_{ai}	258.103 mm	258.103 mm
Root Diameter at End	d_{fe}	306.424 mm	306.424 mm
Vertex Distance	A_e	154.343 mm	154.343 mm
Vertex Distance at Small End	A_i	120.238 mm	120.238 mm
Pitch Cone Angle	δ	45.0000 deg	45.0000 deg
Outside Cone Angle	δ_a	47.0249 deg	47.0249 deg
Root Cone Angle	δ_f	42.5706 deg	42.5706 deg
Face width	b	50.000 mm	
Face width Ratio	b_r	0.2210 ul	
Addendum	a^*	1.0000 ul	1.0000 ul
Clearance	c^*	0.2000 ul	0.2000 ul
Root Fillet	r_f^*	0.3000 ul	0.3000 ul
Whole Depth of Tooth	h_e	17.600 mm	17.600 mm
Tooth Thickness at End	s_e	12.566 mm	12.566 mm
Chordal Thickness	t_c	11.096 mm	11.096 mm
Chordal Addendum	a_c	5.981 mm	5.981 mm
Limit Deviation of Helix Angle	F_β	0.0150 mm	0.0150 mm
Limit Circumferential Run-out	F_r	0.0390 mm	0.0390 mm
Limit Deviation of Axial Pitch	f_{pt}	0.0120 mm	0.0120 mm
Limit Deviation of Basic Pitch	f_{pb}	0.0110 mm	0.0110 mm
Equivalent Number of Teeth	z_v	56.569 ul	56.569 ul
Equivalent Pitch Diameter	d_v	402.548 mm	402.548 mm
Equivalent Outside Diameter	d_{va}	416.781 mm	416.781 mm
Equivalent Base Circle Diameter	d_{vb}	378.272 mm	378.272 mm

Loads

		Gear 1	Gear 2
Power	P	85.718 kW	84.004 kW
Speed	n	1437.00 r/min	1437.00 r/min
Torque	T	569.622 N m	558.229 N m
Efficiency	η	0.980 ul	
Tangential Force	F_t	4002.336 N	
Normal Force	F_n	4259.198 N	
Radial Force (direction 1)	F_{r1}	1030.065 N	1030.065 N
Radial Force (direction 2)	F_{r2}	1030.065 N	1030.065 N
Axial Force (direction 1)	F_{a1}	1030.065 N	1030.065 N
Axial Force (direction 2)	F_{a2}	1030.065 N	1030.065 N
Circumferential Speed	v	21.417 mps	
Resonance Speed	n_{E1}	2983.477 r/min	

Material

		Gear 1	Gear 2
		BS 080M50	BS 080M50
Ultimate Tensile Strength	S_u	640 MPa	640 MPa
Yield Strength	S_y	390 MPa	390 MPa
Modulus of Elasticity	E	206000 MPa	206000 MPa
Poisson's Ratio	μ	0.300 ul	0.300 ul
Bending Fatigue Limit	σ_{Flim}	390.0 MPa	390.0 MPa
Contact Fatigue Limit	σ_{Hlim}	1140.0 MPa	1140.0 MPa
Hardness in Tooth Core	JHV	210 ul	210 ul
Hardness in Tooth Side	VHV	600 ul	600 ul

Strength Calculation

Factors of Additional Load

Application Factor	K_A	1.200 ul	
Dynamic Factor	K_{Hv}	2.020 ul	2.020 ul
Face Load Factor	$K_{H\beta}$	1.274 ul	1.194 ul
Transverse Load Factor	$K_{H\alpha}$	1.135 ul	1.135 ul
One-time Overloading Factor	K_{AS}	1.000 ul	

Factors for Contact

Elasticity Factor	Z_E	189.812 ul	
Zone Factor	Z_H	2.495 ul	
Contact Ratio Factor	Z_ε	0.861 ul	
Bevel Gear Factor	Z_K	0.850 ul	
Single Pair Tooth Contact Factor	Z_B	1.001 ul	1.001 ul
Life Factor	Z_N	1.000 ul	1.000 ul
Lubricant Factor	Z_L	0.962 ul	
Roughness Factor	Z_R	1.000 ul	
Speed Factor	Z_v	1.027 ul	
Helix Angle Factor	Z_β	1.000 ul	
Size Factor	Z_X	1.000 ul	1.000 ul

Factors for Bending

Form Factor	Y_{Fa}	2.309 ul	2.309 ul
Stress Correction Factor	Y_{Sa}	1.802 ul	1.802 ul
Teeth with Grinding Notches Factor	Y_{Sag}	1.000 ul	1.000 ul
Helix Angle Factor	Y_β	1.000 ul	
Contact Ratio Factor	Y_ε	0.672 ul	
Alternating Load Factor	Y_A	1.000 ul	1.000 ul
Production Technology Factor	Y_T	1.000 ul	1.000 ul
Life Factor	Y_N	1.000 ul	1.000 ul
Notch Sensitivity Factor	Y_δ	1.103 ul	1.103 ul
Size Factor	Y_X	1.000 ul	1.000 ul

Results

Factor of Safety from Pitting	S_H	2.313 ul	2.313 ul
Factor of Safety from Tooth Breakage	S_F	4.164 ul	4.164 ul
Static Safety in Contact	S_{Hst}	2.243 ul	2.243 ul
Static Safety in Bending	S_{Fst}	9.437 ul	9.437 ul
Check Calculation		Positive	

Bevel gear manual calculation check.

P= Power transmitted

m = module

T_{gear} = Number of gear teeth

T_{pinion} = Number of pinion teeth

Cd = Cone distance

ϒ = Pitch angle

φ = Pressure angle

d = Pitch diameter

$$module = 1.5 \times Power^{\frac{1}{3}} = 1.5 \times 85.715^{\frac{1}{3}} = 6.614\ mm$$

Stepping up the module to *8 mm*, to accommodate for bearing space within the mitre gearbox

$$Working\ tooth\ depth = 2 \times module = 2 \times 8 = 16\ mm$$

$$d_{pitch} = module \times Number\ of\ teeth = 8 \times 40 = 320\ mm$$

(40 teeth assumed)

$$\gamma = tan^{-1}\left[\frac{T_{pinion}}{T_{gear}}\right] = tan^{-1}[ratio] = tan^{-1}[1] = 45^{\circ}$$ (Pitch angle)

$$A_o = \frac{d}{2 \times \sin\gamma} = \frac{320}{2 \times \sin 45} = 226.274\ mm$$ (Cone distance)

$$Diametral\ pitch = \frac{Number\ of\ teeth}{d} = \frac{40}{320} = 0.125\ per\ mm$$

$$Circular\ pitch = \frac{\pi}{Diametral\ pitch} = \frac{\pi}{.125} = 25.133\ mm$$

$$Face\ width = \frac{A_o}{3} = \frac{226.274}{3} = 75.425\ mm\ \approx 80\ mm$$

$$Face\ width = \frac{module \times Circular\ pitch}{\pi} = \frac{8 \times 25.133}{\pi} = 64\ mm$$

Selecting the greater value of 80 mm between the two face width values.

$$Torque = \frac{30 \times Power}{\pi \times N} = \frac{30 \times 85715}{\pi \times 1437} = 569.6\ N.m$$

$$W_t = \frac{2 \times Torque}{d_{pitch}} = \frac{2 \times 569.6}{.32} = 3560.01\ N \qquad (Tangential\ Force)$$

$$W_a = W_r = W_t \tan \emptyset \sin \gamma$$

$$= 3560.01 \times \tan 20 \times \sin 45 = 916.225\ N \quad (Axial\ \&\ Radial\ Force)$$

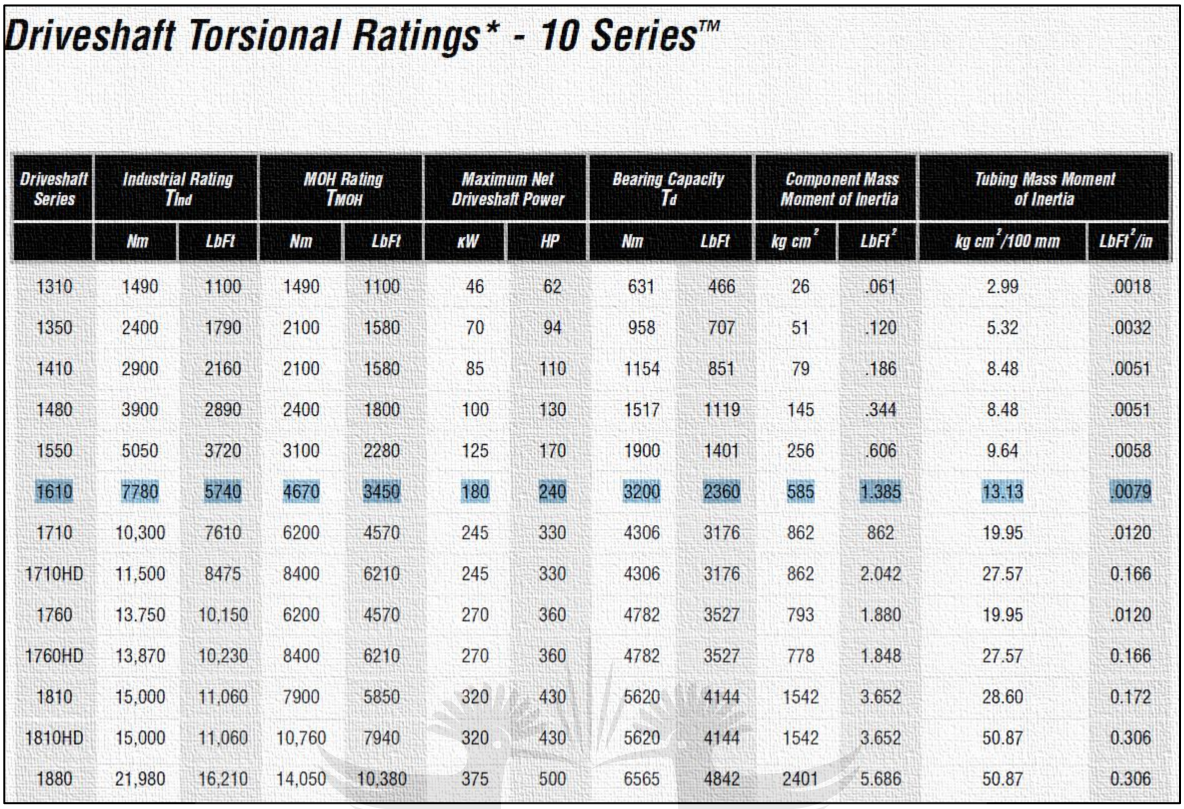
Selecting the kv factor from Mech Eng Design/ JE Shigley , CR Mischke pg 618, figure 15-5.

For 40 teeth, 20º pressure angle, J = 0.25

Heat treatment is advised for gear strengthening.

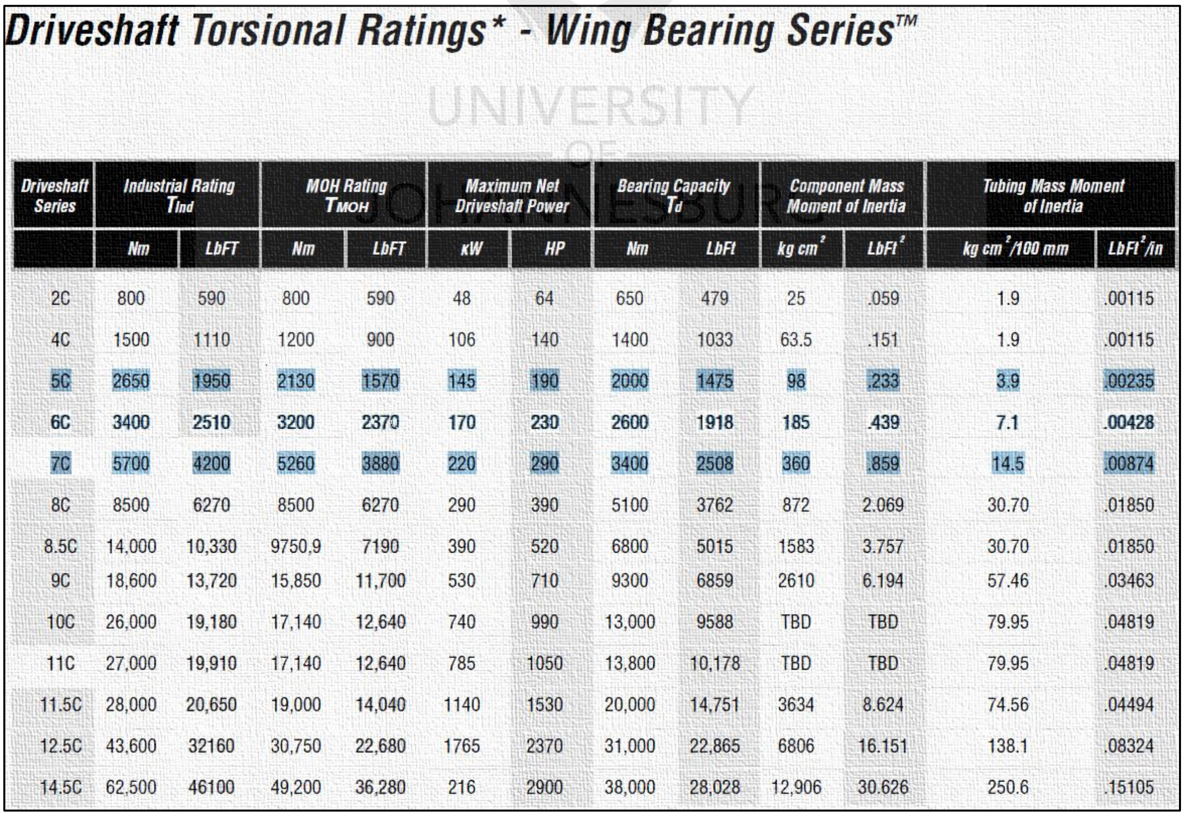
$$v = \frac{\pi \times N \times d}{60} = \frac{\pi \times 1437 \times 0.320}{60} = 24.077\ m/\mathrm{s}$$

Appendix 6: Spicer & Hardy (PTY) LTD Drive Shaft Data.



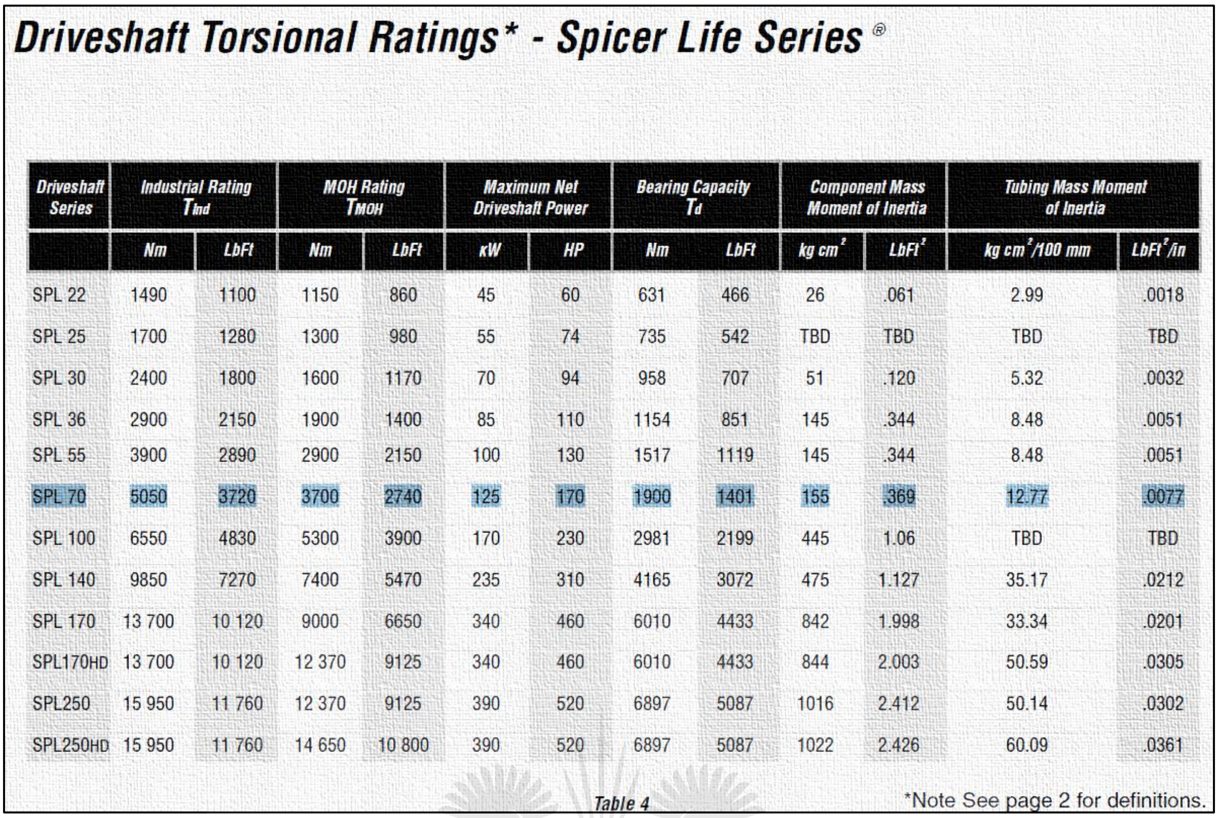
Driveshaft Torsional Ratings* - 10 Series™

Driveshaft Series	Industrial Rating T_{Ind}		MOH Rating T_{MOH}		Maximum Net Driveshaft Power		Bearing Capacity T_d		Component Mass Moment of Inertia		Tubing Mass Moment of Inertia	
	Nm	LbFt	Nm	LbFt	kW	HP	Nm	LbFt	kg cm^2	LbFt^2	kg cm^2/100 mm	LbFt^2/in
1310	1490	1100	1490	1100	46	62	631	466	26	.061	2.99	.0018
1350	2400	1790	2100	1580	70	94	958	707	51	.120	5.32	.0032
1410	2900	2160	2100	1580	85	110	1154	851	79	.186	8.48	.0051
1480	3900	2890	2400	1800	100	130	1517	1119	145	.344	8.48	.0051
1550	5050	3720	3100	2280	125	170	1900	1401	256	.606	9.64	.0058
1610	7780	5740	4670	3450	180	240	3200	2360	585	1.385	13.13	.0079
1710	10,300	7610	6200	4570	245	330	4306	3176	862	862	19.95	.0120
1710HD	11,500	8475	8400	6210	245	330	4306	3176	862	2.042	27.57	0.166
1760	13.750	10,150	6200	4570	270	360	4782	3527	793	1.880	19.95	.0120
1760HD	13,870	10,230	8400	6210	270	360	4782	3527	778	1.848	27.57	0.166
1810	15,000	11,060	7900	5850	320	430	5620	4144	1542	3.652	28.60	0.172
1810HD	15,000	11,060	10,760	7940	320	430	5620	4144	1542	3.652	50.87	0.306
1880	21,980	16,210	14,050	10,380	375	500	6565	4842	2401	5.686	50.87	0.306



Driveshaft Torsional Ratings* - Wing Bearing Series™

Driveshaft Series	Industrial Rating T_{Ind}		MOH Rating T_{MOH}		Maximum Net Driveshaft Power		Bearing Capacity T_d		Component Mass Moment of Inertia		Tubing Mass Moment of Inertia	
	Nm	LbFT	Nm	LbFT	kW	HP	Nm	LbFt	kg cm^2	LbFt^2	kg cm^2/100 mm	LbFt^2/in
2C	800	590	800	590	48	64	650	479	25	.059	1.9	.00115
4C	1500	1110	1200	900	106	140	1400	1033	63.5	.151	1.9	.00115
5C	2650	1950	2130	1570	145	190	2000	1475	98	.233	3.9	.00235
6C	**3400**	**2510**	**3200**	**2370**	**170**	**230**	**2600**	**1918**	**185**	**.439**	**7.1**	**.00428**
7C	5700	4200	5260	3880	220	290	3400	2508	360	.859	14.5	.00874
8C	8500	6270	8500	6270	290	390	5100	3762	872	2.069	30.70	.01850
8.5C	14,000	10,330	9750,9	7190	390	520	6800	5015	1583	3.757	30.70	.01850
9C	18,600	13,720	15,850	11,700	530	710	9300	6859	2610	6.194	57.46	.03463
10C	26,000	19,180	17,140	12,640	740	990	13,000	9588	TBD	TBD	79.95	.04819
11C	27,000	19,910	17,140	12,640	785	1050	13,800	10,178	TBD	TBD	79.95	.04819
11.5C	28,000	20,650	19,000	14,040	1140	1530	20,000	14,751	3634	8.624	74.56	.04494
12.5C	43,600	32160	30,750	22,680	1765	2370	31,000	22,865	6806	16.151	138.1	.08324
14.5C	62,500	46100	49,200	36,280	216	2900	38,000	28,028	12,906	30.626	250.6	.15105

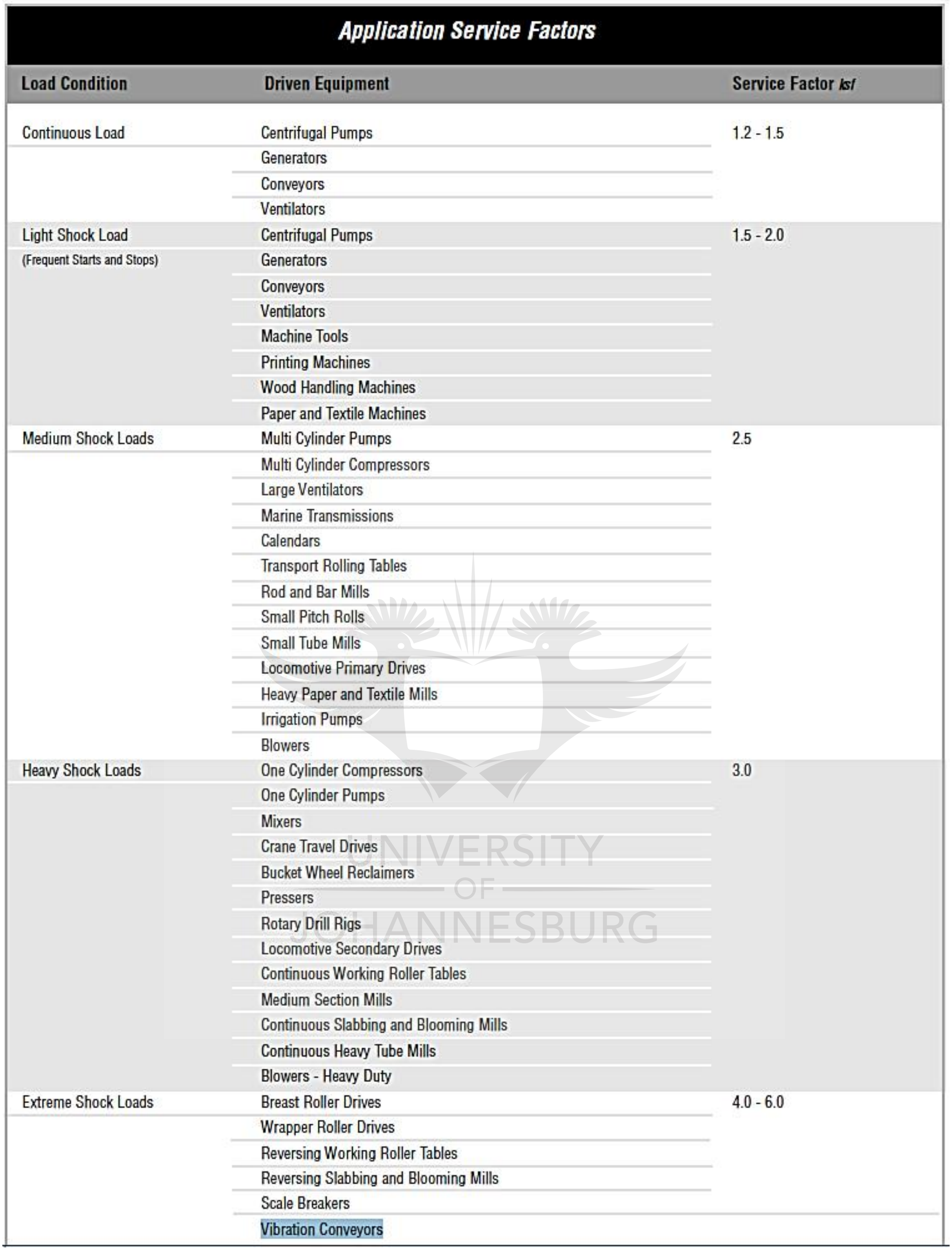


Driveshaft Torsional Ratings* - Spicer Life Series ®

Driveshaft Series	Industrial Rating T_{Ind}		MOH Rating T_{MOH}		Maximum Net Driveshaft Power		Bearing Capacity T_d		Component Mass Moment of Inertia		Tubing Mass Moment of Inertia	
	Nm	LbFt	Nm	LbFt	kW	HP	Nm	LbFt	kg cm²	LbFt²	kg cm²/100 mm	LbFt²/in
SPL 22	1490	1100	1150	860	45	60	631	466	26	.061	2.99	.0018
SPL 25	1700	1280	1300	980	55	74	735	542	TBD	TBD	TBD	TBD
SPL 30	2400	1800	1600	1170	70	94	958	707	51	.120	5.32	.0032
SPL 36	2900	2150	1900	1400	85	110	1154	851	145	.344	8.48	.0051
SPL 55	3900	2890	2900	2150	100	130	1517	1119	145	.344	8.48	.0051
SPL 70	5050	3720	3700	2740	125	170	1900	1401	155	.369	12.77	.0077
SPL 100	6550	4830	5300	3900	170	230	2981	2199	445	1.06	TBD	TBD
SPL 140	9850	7270	7400	5470	235	310	4165	3072	475	1.127	35.17	.0212
SPL 170	13 700	10 120	9000	6650	340	460	6010	4433	842	1.998	33.34	.0201
SPL170HD	13 700	10 120	12 370	9125	340	460	6010	4433	844	2.003	50.59	.0305
SPL250	15 950	11 760	12 370	9125	390	520	6897	5087	1016	2.412	50.14	.0302
SPL250HD	15 950	11 760	14 650	10 800	390	520	6897	5087	1022	2.426	60.09	.0361

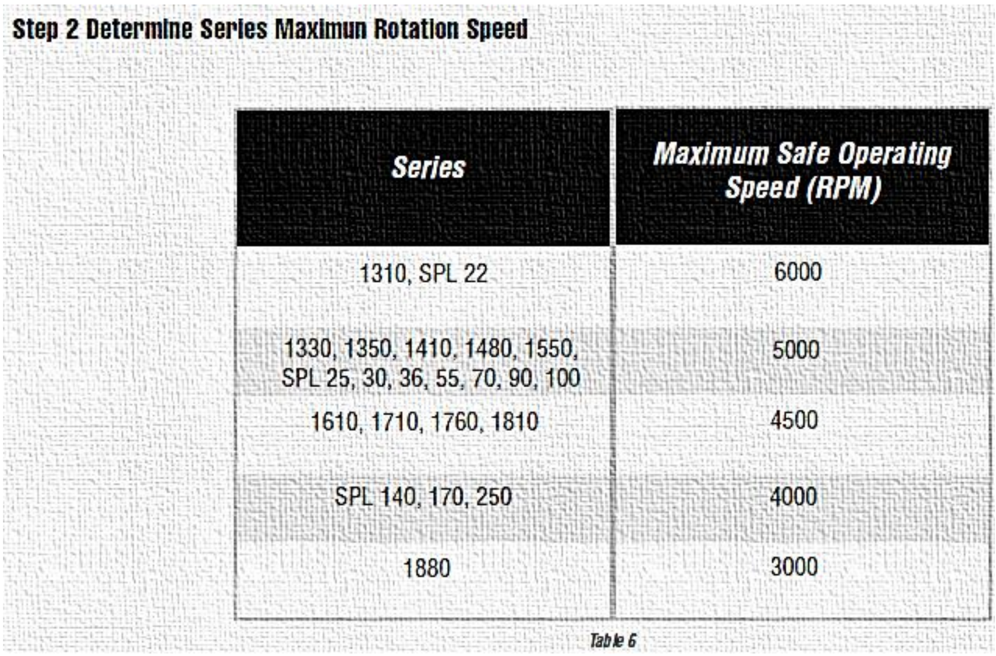
Table 4

*Note See page 2 for definitions.



Application Service Factors

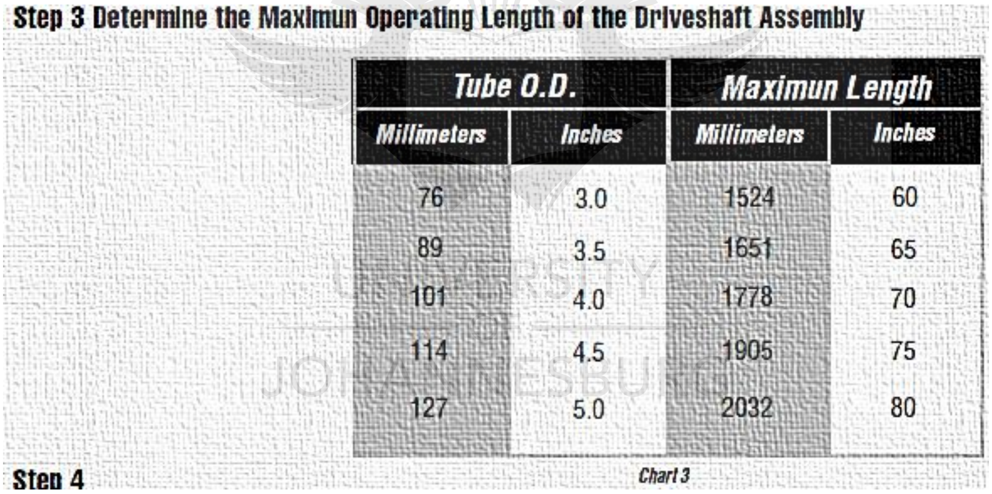
Load Condition	Driven Equipment	Service Factor ksf
Continuous Load	Centrifugal Pumps	1.2 - 1.5
	Generators	
	Conveyors	
	Ventilators	
Light Shock Load (Frequent Starts and Stops)	Centrifugal Pumps	1.5 - 2.0
	Generators	
	Conveyors	
	Ventilators	
	Machine Tools	
	Printing Machines	
	Wood Handling Machines	
	Paper and Textile Machines	
Medium Shock Loads	Multi Cylinder Pumps	2.5
	Multi Cylinder Compressors	
	Large Ventilators	
	Marine Transmissions	
	Calendars	
	Transport Rolling Tables	
	Rod and Bar Mills	
	Small Pitch Rolls	
	Small Tube Mills	
	Locomotive Primary Drives	
	Heavy Paper and Textile Mills	
	Irrigation Pumps	
	Blowers	
Heavy Shock Loads	One Cylinder Compressors	3.0
	One Cylinder Pumps	
	Mixers	
	Crane Travel Drives	
	Bucket Wheel Reclaimers	
	Pressers	
	Rotary Drill Rigs	
	Locomotive Secondary Drives	
	Continuous Working Roller Tables	
	Medium Section Mills	
	Continuous Slabbing and Blooming Mills	
	Continuous Heavy Tube Mills	
	Blowers - Heavy Duty	
Extreme Shock Loads	Breast Roller Drives	4.0 - 6.0
	Wrapper Roller Drives	
	Reversing Working Roller Tables	
	Reversing Slabbing and Blooming Mills	
	Scale Breakers	
	Vibration Conveyors	



Step 2 Determine Series Maximun Rotation Speed

Series	Maximum Safe Operating Speed (RPM)
1310, SPL 22	6000
1330, 1350, 1410, 1480, 1550, SPL 25, 30, 36, 55, 70, 90, 100	5000
1610, 1710, 1760, 1810	4500
SPL 140, 170, 250	4000
1880	3000

Table 6



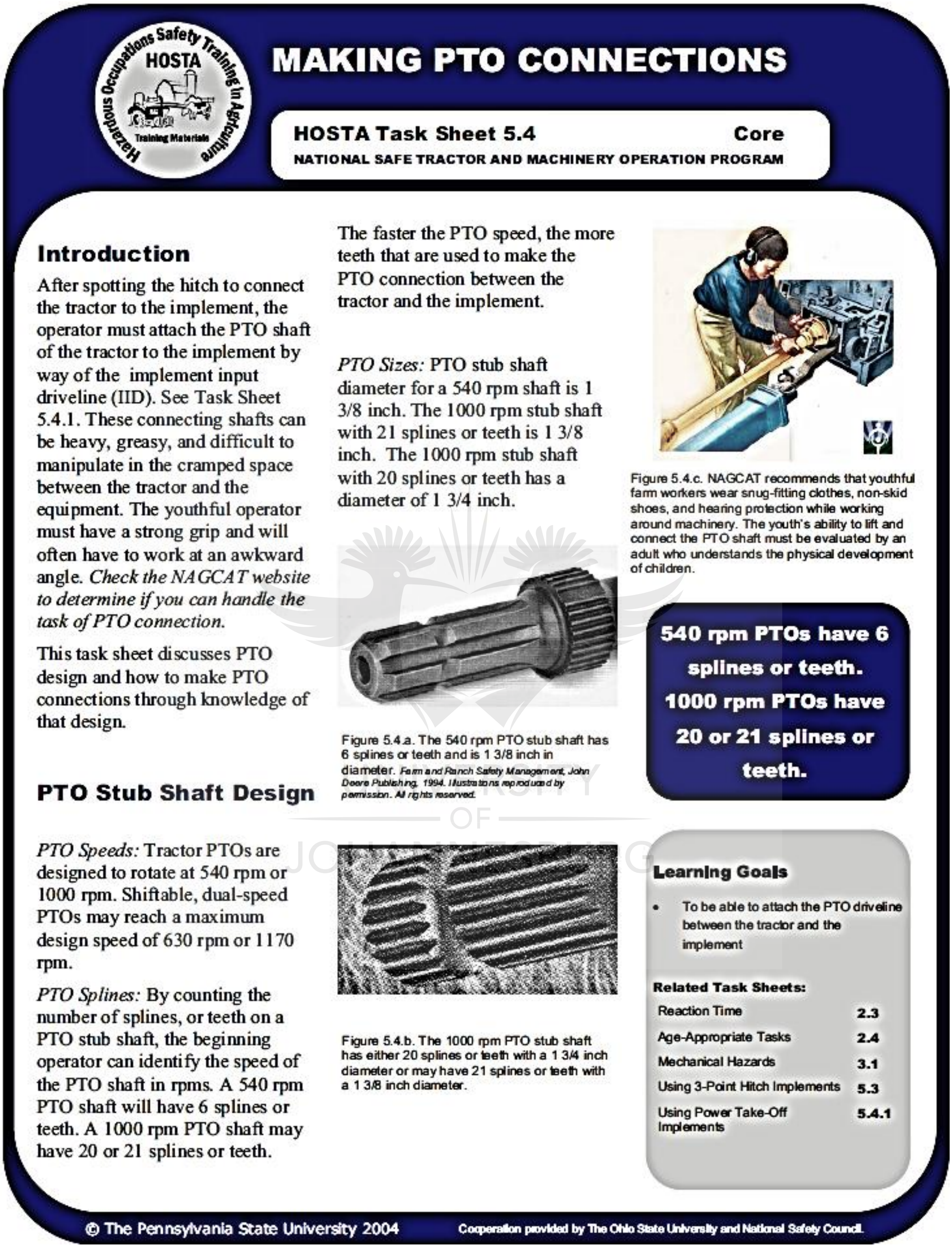
Step 3 Determine the Maximun Operating Length of the Driveshaft Assembly

Tube O.D.		Maximun Length	
Millimeters	Inches	Millimeters	Inches
76	3.0	1524	60
89	3.5	1651	65
101	4.0	1778	70
114	4.5	1905	75
127	5.0	2032	80

Step 4

Chart 3

Appendix 7: Drive shaft installation & maintenance.



Hazardous Occupations Safety Training in Agriculture

HOSTA

Training Materials

MAKING PTO CONNECTIONS

HOSTA Task Sheet 5.4 **Core**

NATIONAL SAFE TRACTOR AND MACHINERY OPERATION PROGRAM

Introduction

After spotting the hitch to connect the tractor to the implement, the operator must attach the PTO shaft of the tractor to the implement by way of the implement input driveline (IID). See Task Sheet 5.4.1. These connecting shafts can be heavy, greasy, and difficult to manipulate in the cramped space between the tractor and the equipment. The youthful operator must have a strong grip and will often have to work at an awkward angle. *Check the NAGCAT website to determine if you can handle the task of PTO connection.*

This task sheet discusses PTO design and how to make PTO connections through knowledge of that design.

PTO Stub Shaft Design

PTO Speeds: Tractor PTOs are designed to rotate at 540 rpm or 1000 rpm. Shiftable, dual-speed PTOs may reach a maximum design speed of 630 rpm or 1170 rpm.

PTO Splines: By counting the number of splines, or teeth on a PTO stub shaft, the beginning operator can identify the speed of the PTO shaft in rpms. A 540 rpm PTO shaft will have 6 splines or teeth. A 1000 rpm PTO shaft may have 20 or 21 splines or teeth. The faster the PTO speed, the more teeth that are used to make the PTO connection between the tractor and the implement.

PTO Sizes: PTO stub shaft diameter for a 540 rpm shaft is 1 3/8 inch. The 1000 rpm stub shaft with 21 splines or teeth is 1 3/8 inch. The 1000 rpm stub shaft with 20 splines or teeth has a diameter of 1 3/4 inch.

Figure 5.4.a. The 540 rpm PTO stub shaft has 6 splines or teeth and is 1 3/8 inch in diameter. *Farm and Ranch Safety Management, John Deere Publishing, 1994. Illustrations reproduced by permission. All rights reserved.*

Figure 5.4.b. The 1000 rpm PTO stub shaft has either 20 splines or teeth with a 1 3/4 inch diameter or may have 21 splines or teeth with a 1 3/8 inch diameter.

Figure 5.4.c. NAGCAT recommends that youthful farm workers wear snug-fitting clothes, non-skid shoes, and hearing protection while working around machinery. The youth's ability to lift and connect the PTO shaft must be evaluated by an adult who understands the physical development of children.

540 rpm PTOs have 6 splines or teeth. 1000 rpm PTOs have 20 or 21 splines or teeth.

Learning Goals

- To be able to attach the PTO driveline between the tractor and the implement

Related Task Sheets:

Reaction Time	2.3
Age-Appropriate Tasks	2.4
Mechanical Hazards	3.1
Using 3-Point Hitch Implements	5.3
Using Power Take-Off Implements	5.4.1

Cooperation provided by The Ohio State University and National Safety Council.

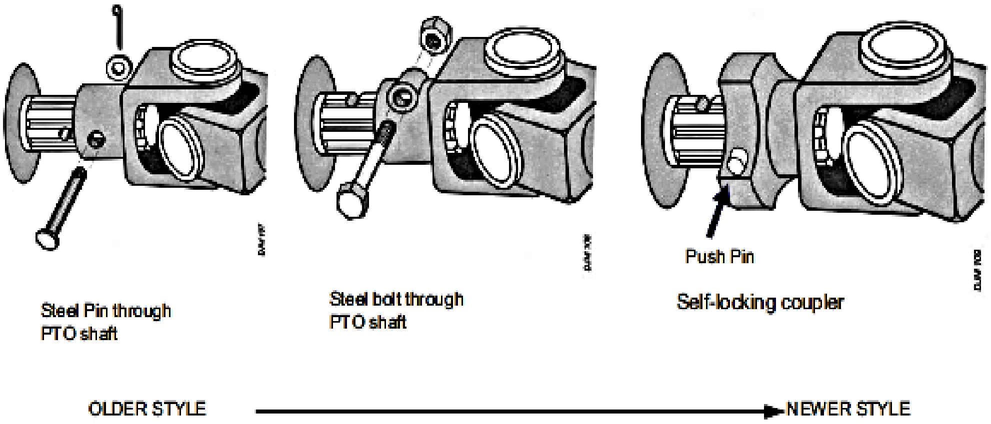


Figure 5.4.d. Various means to secure the PTO shaft to the stub shaft have been used over the years. Besides those connection methods shown above, another popular style is the push pin detent locking type shown in Figure 5.5.e. All types of locking device areas must be guarded as they are wrap points where the operator can become entangled in the PTO.

PTOs must be guarded to prevent an entanglement hazard.

Figure 5.4.e. The push pin detent lock on the PTO driveline has a metal rod which fits in the PTO stub shaft groove to secure it. A firm grip is needed to press the pin. Do you have enough hand strength to push this pin in all the way?

Connecting the PTO

Follow these steps to attach the PTO shaft of a 3-point hitch implement.

1. Connect the tractor to the drawbar or to the 3-point hitch of the implement using the approved steps. See Task Sheets 5.1, 5.2, and 5.3.
2. Attach the PTO shaft of the implement to the PTO stub shaft of the tractor.

Here are some suggestions to make the PTO connection easier.

A. Align the driveline PTO shaft splines with the splines of the stub shaft of the tractor. If the splines will not align, try turning the tractor PTO stub shaft slightly, or use the implement flywheel to move the implement's PTO shaft. Have this procedure shown to you if necessary.

B. Press the detent lock push pin inward (Figure 5.4.e) as you slide the implement shaft onto the tractor stub shaft.

C. Slide the implement shaft forward far enough to make sure the detent pin has snapped into the lock position.

PTO Care and Use

Dirt and grease can make the PTO shaft difficult to grasp and connect. Keep the PTO shaft off the ground. Wipe the excess grease from the PTO shaft with a cloth.

Important: A new PTO shaft has paint inside the splines. This may prevent the shaft from fitting over the PTO stub. The paint must be removed.

PTO Phasing

Older PTO shafts can be separated or pulled apart. The two parts are made so that one part fits into the other. The PTO must be able to telescope in and out to permit machine operation over irregular terrain. If the parts become separated, they must be re-assembled "in phase" to avoid placing extra strain on the universal joints. Many shafts are designed to prevent this from happening.

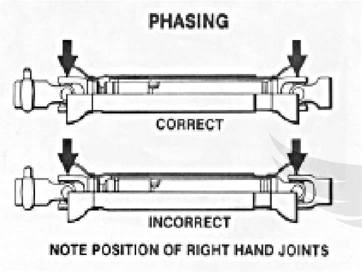


Figure 5.4.f. The upper portion of the drawing illustrates a correctly placed universal joint. You may wish to check the phasing on a PTO shaft.

NAGCAT Guidelines

NAGCAT recommendations for connecting and disconnecting a PTO shaft are shown in this section. These recommendations were developed by a knowledgeable group of safety experts as a means of helping parents to match youthful agricultural workers with the tasks that are appropriate to their development.

The PTO guidelines are presented here.

Adult Responsibilities:

- Be sure implement is in working order.
- Be sure that all safety features are in place.
- Be sure the work area has no hazards.
- Be sure the youth has long hair tied up out of the way, has non-skid shoes, and snug-fitting clothes. Hearing protection is recommended as well.

The adult in charge should also evaluate you using the following questions:

1. Can the youth drive the tractor skillfully?
2. Can the youth hitch and unhitch implements?
3. Does the PTO shaft weigh more than 10-15% of the youth's body weight? To avoid back injury, this should be the maximum weight you should be asked to lift.
4. Can the youth follow a 5-step process?
5. Has the youth been trained in proper lifting techniques?
6. Has an adult demonstrated connecting and disconnecting a PTO?
7. Can the youth do the job 4 or 5 times under direct supervision?
8. Can an adult provide the recommended supervision?

Your experience level may be acceptable to you, but proof of your expertise should be evaluated by a qualified tractor operator.

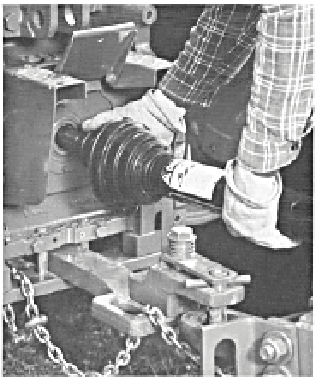


Figure 5.4.g. This is what the task of connecting a PTO looks like. You must lift a heavy object at an awkward angle while squeezing in the lock mechanism detent pin. Watch someone else connect a PTO several times before doing this job. Continue practicing connecting a PTO on your own with supervision.

Connecting a PTO shaft will be easier after practicing the job several times.

Figure 5.4.h. This PTO stub shaft is protected with a tractor master shield and stub shaft cover. To remove the stub shaft cover, grip the cover firmly and turn counterclockwise. Store the stub shaft cover where it will be available to replace when the job is done.

Appendix 8: Bonfiglioli Catalogue.

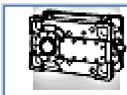
Bonfiglioli
Riduttori

HDO series

Riduttori ad assi ortogonali
Bevel helical gear units
Kegelstirnradgetrieben

Bonfiglioli
power, control and green solutions

1. Service factor and input power.



Nei seguenti casi è necessario prevedere il pre-riscaldamento dell'olio attraverso un'opportuna resistenza elettrica (variante opzionale HE):

- funzionamento a temperature inferiori a 0°C
- avviamento di riduttori lubrificati ad immersione e sbattimento qualora la temperatura ambiente minima non sia superiore di almeno 10°C al punto di scorrimento dell'olio
- avviamento di riduttori con dispositivi di lubrificazione forzata (varianti OP1, OP2, MOP), quando la viscosità dell'olio è superiore a 1800 cst. In funzione del lubrificante utilizzato questo valore si riscontra indicativamente a temperature ambiente comprese fra 10°C e 20°C.

Lubricant must be pre-heated through the appropriate electric resistance (HE option) in the following cases:

- *operation at ambient temperatures lower than 0°C*
- *operation of gear units lubricated by oil immersion and splashing when the minimum ambient temperature exceeds the pour point of lubricant by less than 10°C*
- *upon starting up gear units with forced lubrication systems (options OP1, OP2 or MOP) if the oil viscosity exceeds 1800 cst. Depending of the type of lubricant used, this value may be produced with ambient temperatures between 10°C and 20°C approx.*

In folgenden Fällen muss das Öl mit einem geeigneten elektrischen Heizwiderstand (optionale Variante HE) vorgewärmt werden:

- Betrieb bei Temperaturen unter 0°C
- Anfahren von Getrieben mit Tauch- und Ölspritzschmierung, wenn die niedrigste Umgebungstemperatur mehr als 10° unter dem Fließpunkt des Öls liegt
- Anfahren von Getrieben mit Zwangsschmierung (Varianten OP1, OP2, MOP), wenn die Viskosität des Öls über 1800 cSt liegt. Je nach verwendetem Schmiermittel tritt dieser Wert ungefähr bei Umgebungstemperaturen zwischen 10°C und 20°C auf.

2 - SELEZIONE DEL RIDUTTORE

La selezione ottimale della trasmissione può essere condotta solo previa la piena conoscenza delle condizioni applicative, sia di natura funzionale, che ambientale.

A garanzia di un corretto dimensionamento del prodotto, è vivamente consigliato ricorrere all'esperienza e alla specifica conoscenza del Servizio Tecnico di Bonfiglioli.

2 - SELECTING THE GEAR UNIT

The selection of the drive unit can only be optimized upon knowing both the engineering and the environmental conditions the gearbox will operate into.

For a safe selection it is strongly recommended to rely on the long time experience of the Bonfiglioli Technical Service Dept.

2 - WAHL DES GETRIEBES

Eine optimale Wahl der Uebertragung kann durch eine vollständige Bekanntschaft von allen Anwendungsbedingungen sowohl die zweckmäßige als auch die Umweltbedingungen ausgeführt werden.

Um eine richtige Bemessung zu gewähren, empfehlen wir Sie, an den Dienstleistungsservice von der Bonfiglioli zu wenden.

2.1 - DIMENSIONAMENTO

1. Determinare il rapporto di trasmissione:

2.1 - ENGINEERING SELECTION

1. *First determine the gear ratio:*

2.1 - BEMESSUNG

1. Die Übersetzung ermitteln:

$$i = \frac{n_1}{n_2}$$

2. Calcolare la potenza richiesta P_{r1} all'albero veloce del riduttore:

2. *Calculate the power P_{r1} required at the input shaft:*

2. Benötigte Leistung P_{r1} an der Antriebswelle des Getriebes berechnen:

$$P_{r1} = \frac{M_2 \times n_2}{9550 \times \eta}$$

	η
2x	0.96
3x	0.94
4x	0.92

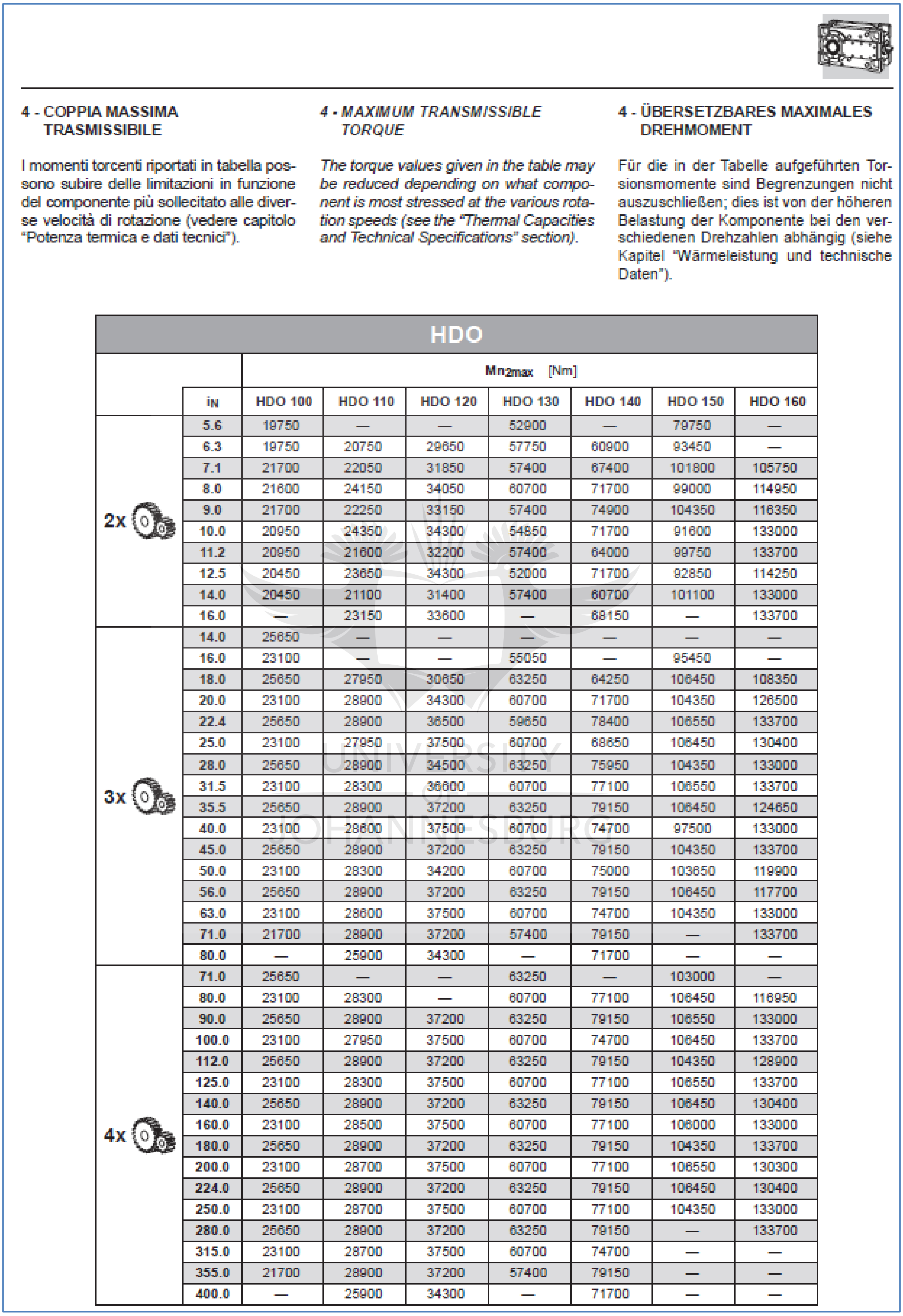
3. Determinare il fattore di servizio f_s applicabile e il fattore correttivo dipendente dal tipo di organo motore f_m:

3. *Determine the applicable service factor f_s and the adjusting factor f_m depending on prime mover.*

3. Bestimmen Sie den geeigneten Betriebsfaktor f_s und den Korrekturfaktor f_m in Abhängigkeit von der Antriebsmaschine.

			f_m
Motore elettrico / Motore idraulico / Turbina	*Electric motor / Hydraulic motor / Turbine*	Elektromotor / Hydraulikmotor / Turbine	1.00
Motore a combustione interna pluricilindrico	*Multi-cylinder internal combustion engine*	Mehrzylinder-Verbrennungsmotor	1.25
Motore a combustione interna monocilindrico	*Single cylinder internal combustion engine*	Einzelzylinder-Verbrennungsmotor	1.50

2. Gear sets as per available ratio.



4 - COPPIA MASSIMA TRASMISSIBILE

I momenti torcenti riportati in tabella possono subire delle limitazioni in funzione del componente più sollecitato alle diverse velocità di rotazione (vedere capitolo "Potenza termica e dati tecnici").

4 - MAXIMUM TRANSMISSIBLE TORQUE

The torque values given in the table may be reduced depending on what component is most stressed at the various rotation speeds (see the "Thermal Capacities and Technical Specifications" section).

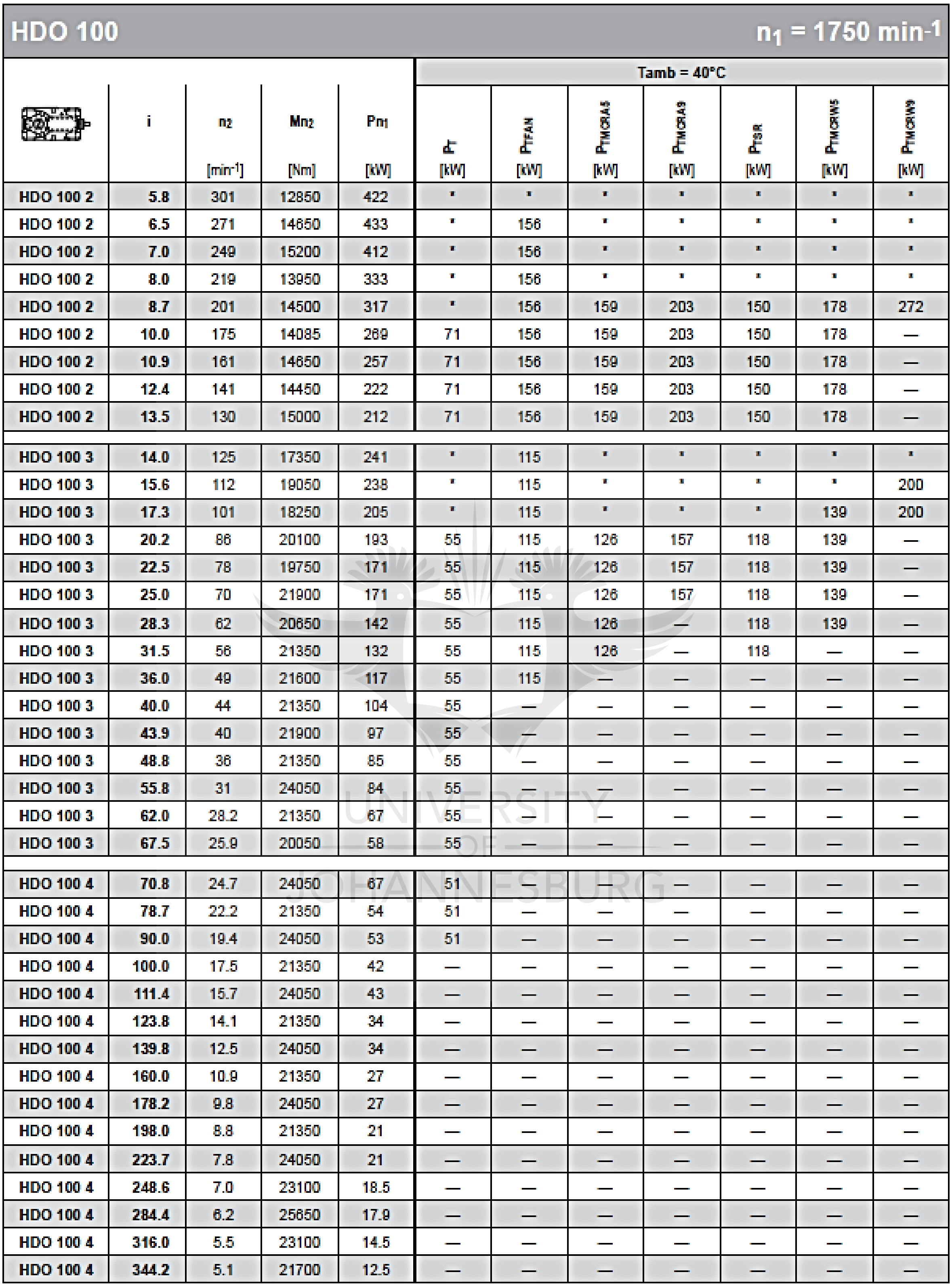
4 - ÜBERSETZBARES MAXIMALES DREHMOMENT

Für die in der Tabelle aufgeführten Torsionsmomente sind Begrenzungen nicht auszuschließen; dies ist von der höheren Belastung der Komponente bei den verschiedenen Drehzahlen abhängig (siehe Kapitel "Wärmeleistung und technische Daten").

HDO

		Mn_{2max} [Nm]						
	i_N	HDO 100	HDO 110	HDO 120	HDO 130	HDO 140	HDO 150	HDO 160
2x	**5.6**	19750	—	—	52900	—	79750	—
	6.3	19750	20750	29650	57750	60900	93450	—
	7.1	21700	22050	31850	57400	67400	101800	105750
	8.0	21600	24150	34050	60700	71700	99000	114950
	9.0	21700	22250	33150	57400	74900	104350	116350
	10.0	20950	24350	34300	54850	71700	91600	133000
	11.2	20950	21600	32200	57400	64000	99750	133700
	12.5	20450	23650	34300	52000	71700	92850	114250
	14.0	20450	21100	31400	57400	60700	101100	133000
	16.0	—	23150	33600	—	68150	—	133700
3x	**14.0**	25650	—	—	—	—	—	—
	16.0	23100	—	—	55050	—	95450	—
	18.0	25650	27950	30650	63250	64250	106450	108350
	20.0	23100	28900	34300	60700	71700	104350	126500
	22.4	25650	28900	36500	59650	78400	106550	133700
	25.0	23100	27950	37500	60700	68650	106450	130400
	28.0	25650	28900	34500	63250	75950	104350	133000
	31.5	23100	28300	36600	60700	77100	106550	133700
	35.5	25650	28900	37200	63250	79150	106450	124650
	40.0	23100	28600	37500	60700	74700	97500	133000
	45.0	25650	28900	37200	63250	79150	104350	133700
	50.0	23100	28300	34200	60700	75000	103650	119900
	56.0	25650	28900	37200	63250	79150	106450	117700
	63.0	23100	28600	37500	60700	74700	104350	133000
	71.0	21700	28900	37200	57400	79150	—	133700
	80.0	—	25900	34300	—	71700	—	—
4x	**71.0**	25650	—	—	63250	—	103000	—
	80.0	23100	28300	—	60700	77100	106450	116950
	90.0	25650	28900	37200	63250	79150	106550	133000
	100.0	23100	27950	37500	60700	74700	106450	133700
	112.0	25650	28900	37200	63250	79150	104350	128900
	125.0	23100	28300	37500	60700	77100	106550	133700
	140.0	25650	28900	37200	63250	79150	106450	130400
	160.0	23100	28500	37500	60700	77100	106000	133000
	180.0	25650	28900	37200	63250	79150	104350	133700
	200.0	23100	28700	37500	60700	77100	106550	130300
	224.0	25650	28900	37200	63250	79150	106450	130400
	250.0	23100	28700	37500	60700	77100	104350	133000
	280.0	25650	28900	37200	63250	79150	—	133700
	315.0	23100	28700	37500	60700	74700	—	—
	355.0	21700	28900	37200	57400	79150	—	—
	400.0	—	25900	34300	—	71700	—	—

3. HDO 100 2 Gearbox Selection table at 1750 r/min.

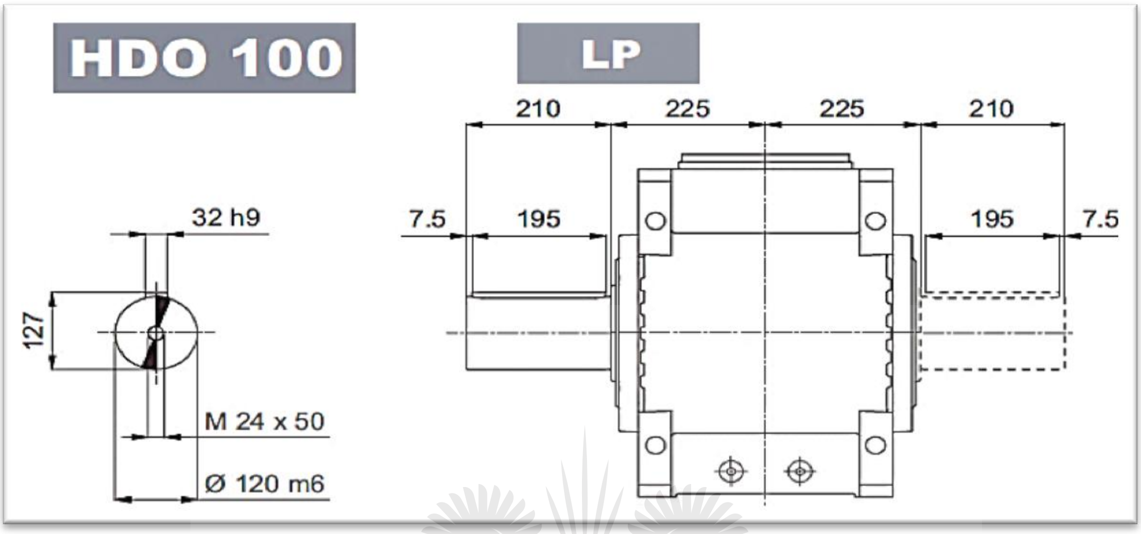


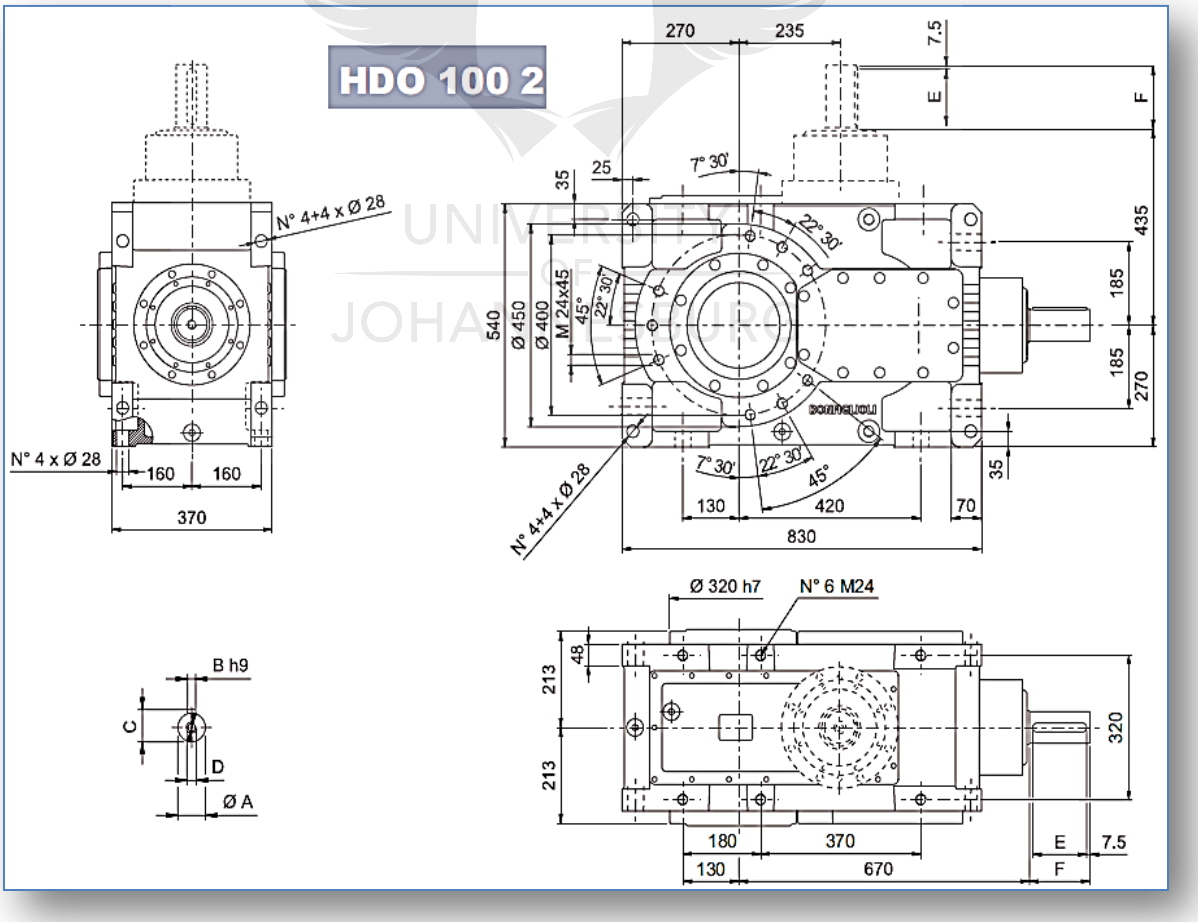
HDO 100 $n_1 = 1750\ min^{-1}$

	i	n_2 [min-1]	M_{n2} [Nm]	P_{n1} [kW]	Tamb = 40°C P_T [kW]	P_{TFAN} [kW]	P_{TMCRA5} [kW]	P_{TMCRA9} [kW]	P_{TSR} [kW]	P_{TMCRW5} [kW]	P_{TMCRW9} [kW]
HDO 100 2	5.8	301	12850	422	*	*	*	*	*	*	*
HDO 100 2	6.5	271	14650	433	*	156	*	*	*	*	*
HDO 100 2	7.0	249	15200	412	*	156	*	*	*	*	*
HDO 100 2	8.0	219	13950	333	*	156	*	*	*	*	*
HDO 100 2	8.7	201	14500	317	*	156	159	203	150	178	272
HDO 100 2	10.0	175	14085	269	71	156	159	203	150	178	—
HDO 100 2	10.9	161	14650	257	71	156	159	203	150	178	—
HDO 100 2	12.4	141	14450	222	71	156	159	203	150	178	—
HDO 100 2	13.5	130	15000	212	71	156	159	203	150	178	—
HDO 100 3	14.0	125	17350	241	*	115	*	*	*	*	*
HDO 100 3	15.6	112	19050	238	*	115	*	*	*	*	200
HDO 100 3	17.3	101	18250	205	*	115	*	*	*	139	200
HDO 100 3	20.2	86	20100	193	55	115	126	157	118	139	—
HDO 100 3	22.5	78	19750	171	55	115	126	157	118	139	—
HDO 100 3	25.0	70	21900	171	55	115	126	157	118	139	—
HDO 100 3	28.3	62	20650	142	55	115	126	—	118	139	—
HDO 100 3	31.5	56	21350	132	55	115	126	—	118	—	—
HDO 100 3	36.0	49	21600	117	55	115	—	—	—	—	—
HDO 100 3	40.0	44	21350	104	55	—	—	—	—	—	—
HDO 100 3	43.9	40	21900	97	55	—	—	—	—	—	—
HDO 100 3	48.8	36	21350	85	55	—	—	—	—	—	—
HDO 100 3	55.8	31	24050	84	55	—	—	—	—	—	—
HDO 100 3	62.0	28.2	21350	67	55	—	—	—	—	—	—
HDO 100 3	67.5	25.9	20050	58	55	—	—	—	—	—	—
HDO 100 4	70.8	24.7	24050	67	51	—	—	—	—	—	—
HDO 100 4	78.7	22.2	21350	54	51	—	—	—	—	—	—
HDO 100 4	90.0	19.4	24050	53	51	—	—	—	—	—	—
HDO 100 4	100.0	17.5	21350	42	—	—	—	—	—	—	—
HDO 100 4	111.4	15.7	24050	43	—	—	—	—	—	—	—
HDO 100 4	123.8	14.1	21350	34	—	—	—	—	—	—	—
HDO 100 4	139.8	12.5	24050	34	—	—	—	—	—	—	—
HDO 100 4	160.0	10.9	21350	27	—	—	—	—	—	—	—
HDO 100 4	178.2	9.8	24050	27	—	—	—	—	—	—	—
HDO 100 4	198.0	8.8	21350	21	—	—	—	—	—	—	—
HDO 100 4	223.7	7.8	24050	21	—	—	—	—	—	—	—
HDO 100 4	248.6	7.0	23100	18.5	—	—	—	—	—	—	—
HDO 100 4	284.4	6.2	25650	17.9	—	—	—	—	—	—	—
HDO 100 4	316.0	5.5	23100	14.5	—	—	—	—	—	—	—
HDO 100 4	344.2	5.1	21700	12.5	—	—	—	—	—	—	—

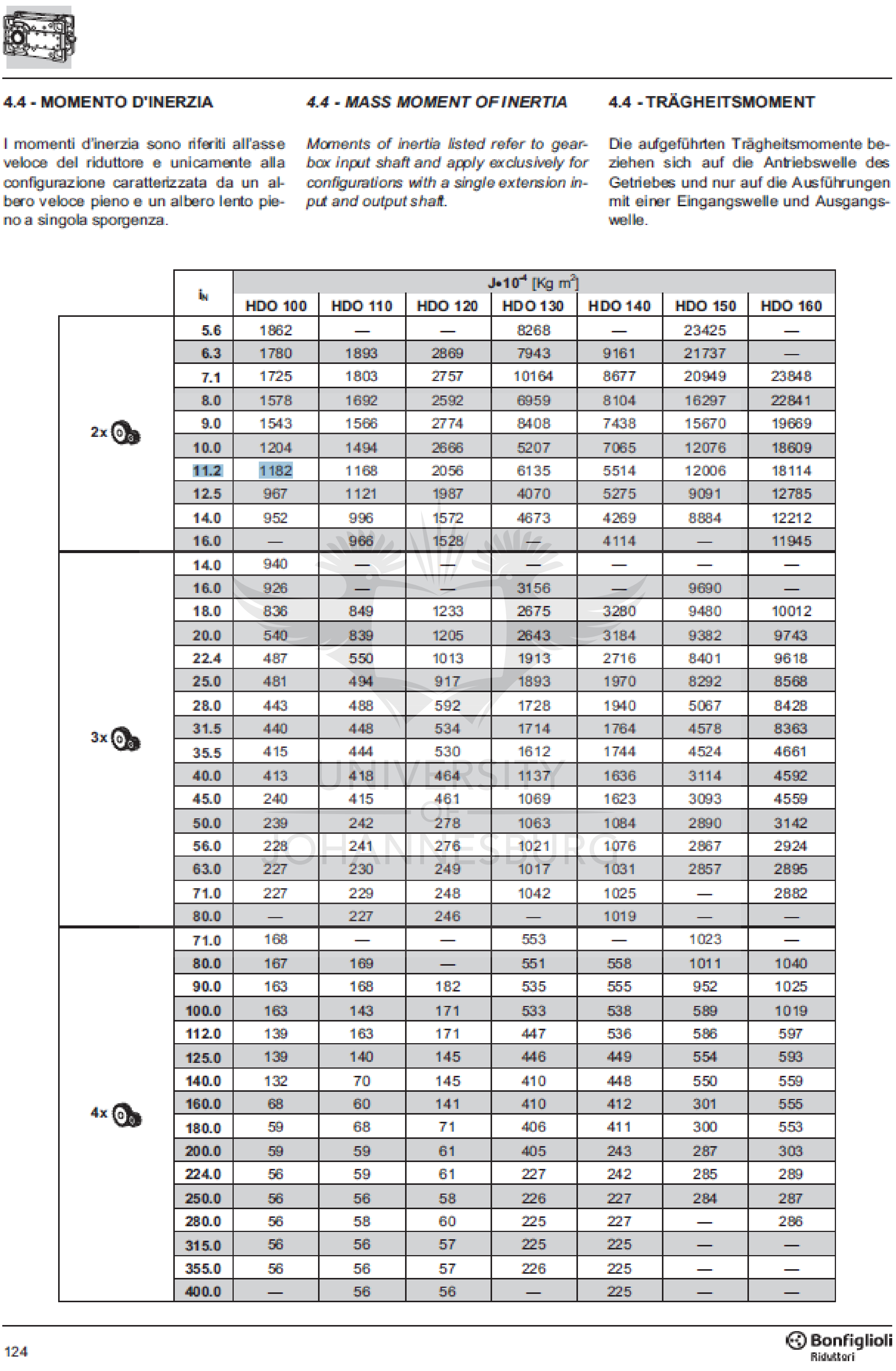
4. HDO 100 Mass & Dimensions

VP	i =	A	B	C	D	E	F	Kg
HDO 100 2	5.8 … 13.5	70 m6	20	74.5	M20x42	125	140	660
HDO 100 3	14 … 17.3	55 m6	16	59	M20x42	100	110	750
HDO 100 3	20.2 … 67.5	45 k6	14	48.5	M16x36	100	110	750
HDO 100 4	70.8 … 139.8	35 k6	10	38	M12x28	70	80	760
HDO 100 4	160 … 344.2	32 k6	10	35	M12x28	70	80	760





5. Mass Moment of Inertia



4.4 - MOMENTO D'INERZIA

I momenti d'inerzia sono riferiti all'asse veloce del riduttore e unicamente alla configurazione caratterizzata da un albero veloce pieno e un albero lento pieno a singola sporgenza.

4.4 - MASS MOMENT OF INERTIA

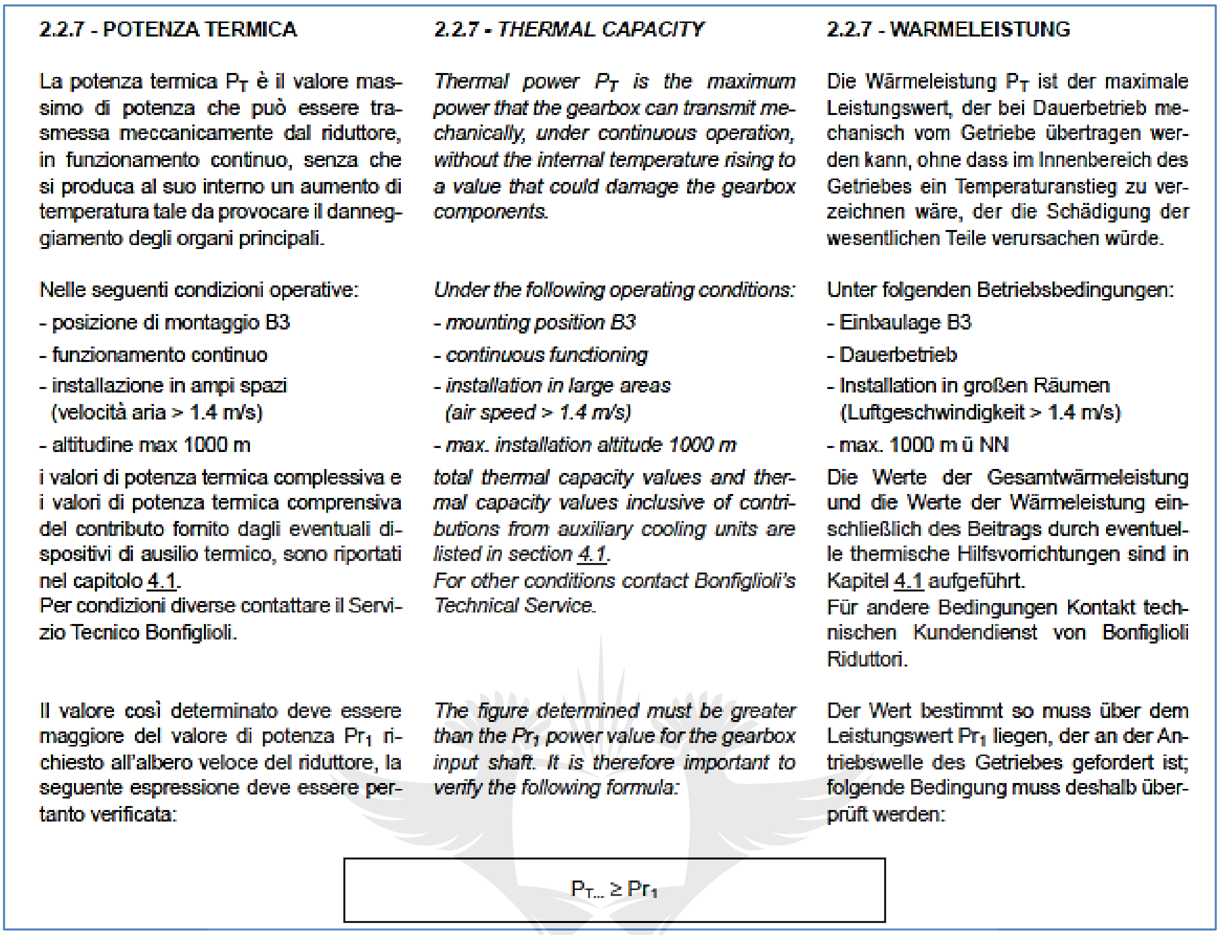
Moments of inertia listed refer to gearbox input shaft and apply exclusively for configurations with a single extension input and output shaft.

4.4 - TRÄGHEITSMOMENT

Die aufgeführten Trägheitsmomente beziehen sich auf die Antriebswelle des Getriebes und nur auf die Ausführungen mit einer Eingangswelle und Ausgangswelle.

	i_N	$J \cdot 10^{-4}$ [Kg m^2]						
		HDO 100	**HDO 110**	**HDO 120**	**HDO 130**	**HDO 140**	**HDO 150**	**HDO 160**
2x	**5.6**	1862	—	—	8268	—	23425	—
	6.3	1780	1893	2869	7943	9161	21737	—
	7.1	1725	1803	2757	10164	8677	20949	23848
	8.0	1578	1692	2592	6959	8104	16297	22841
	9.0	1543	1566	2774	8408	7438	15670	19669
	10.0	1204	1494	2666	5207	7065	12076	18609
	11.2	1182	1168	2056	6135	5514	12006	18114
	12.5	967	1121	1987	4070	5275	9091	12785
	14.0	952	996	1572	4673	4269	8884	12212
	16.0	—	966	1528	—	4114	—	11945
3x	**14.0**	940	—	—	—	—	—	—
	16.0	926	—	—	3156	—	9690	—
	18.0	836	849	1233	2675	3280	9480	10012
	20.0	540	839	1205	2643	3184	9382	9743
	22.4	487	550	1013	1913	2716	8401	9618
	25.0	481	494	917	1893	1970	8292	8568
	28.0	443	488	592	1728	1940	5067	8428
	31.5	440	448	534	1714	1764	4578	8363
	35.5	415	444	530	1612	1744	4524	4661
	40.0	413	418	464	1137	1636	3114	4592
	45.0	240	415	461	1069	1623	3093	4559
	50.0	239	242	278	1063	1084	2890	3142
	56.0	228	241	276	1021	1076	2867	2924
	63.0	227	230	249	1017	1031	2857	2895
	71.0	227	229	248	1042	1025	—	2882
	80.0	—	227	246	—	1019	—	—
4x	**71.0**	168	—	—	553	—	1023	—
	80.0	167	169	—	551	558	1011	1040
	90.0	163	168	182	535	555	952	1025
	100.0	163	143	171	533	538	589	1019
	112.0	139	163	171	447	536	586	597
	125.0	139	140	145	446	449	554	593
	140.0	132	70	145	410	448	550	559
	160.0	68	60	141	410	412	301	555
	180.0	59	68	71	406	411	300	553
	200.0	59	59	61	405	243	287	303
	224.0	56	59	61	227	242	285	289
	250.0	56	56	58	226	227	284	287
	280.0	56	58	60	225	227	—	286
	315.0	56	56	57	225	225	—	—
	355.0	56	56	57	226	225	—	—
	400.0	—	56	56	—	225	—	—

6. Gearbox cooling.



2.2.7 - POTENZA TERMICA

La potenza termica P_T è il valore massimo di potenza che può essere trasmessa meccanicamente dal riduttore, in funzionamento continuo, senza che si produca al suo interno un aumento di temperatura tale da provocare il danneggiamento degli organi principali.

Nelle seguenti condizioni operative:

- posizione di montaggio B3
- funzionamento continuo
- installazione in ampi spazi (velocità aria > 1.4 m/s)
- altitudine max 1000 m

i valori di potenza termica complessiva e i valori di potenza termica comprensiva del contributo fornito dagli eventuali dispositivi di ausilio termico, sono riportati nel capitolo 4.1.
Per condizioni diverse contattare il Servizio Tecnico Bonfiglioli.

Il valore così determinato deve essere maggiore del valore di potenza Pr_1 richiesto all'albero veloce del riduttore, la seguente espressione deve essere pertanto verificata:

2.2.7 - THERMAL CAPACITY

Thermal power P_T is the maximum power that the gearbox can transmit mechanically, under continuous operation, without the internal temperature rising to a value that could damage the gearbox components.

Under the following operating conditions:

- *mounting position B3*
- *continuous functioning*
- *installation in large areas (air speed > 1.4 m/s)*
- *max. installation altitude 1000 m*

total thermal capacity values and thermal capacity values inclusive of contributions from auxiliary cooling units are listed in section 4.1.
For other conditions contact Bonfiglioli's Technical Service.

The figure determined must be greater than the Pr_1 power value for the gearbox input shaft. It is therefore important to verify the following formula:

2.2.7 - WARMELEISTUNG

Die Wärmeleistung P_T ist der maximale Leistungswert, der bei Dauerbetrieb mechanisch vom Getriebe übertragen werden kann, ohne dass im Innenbereich des Getriebes ein Temperaturanstieg zu verzeichnen wäre, der die Schädigung der wesentlichen Teile verursachen würde.

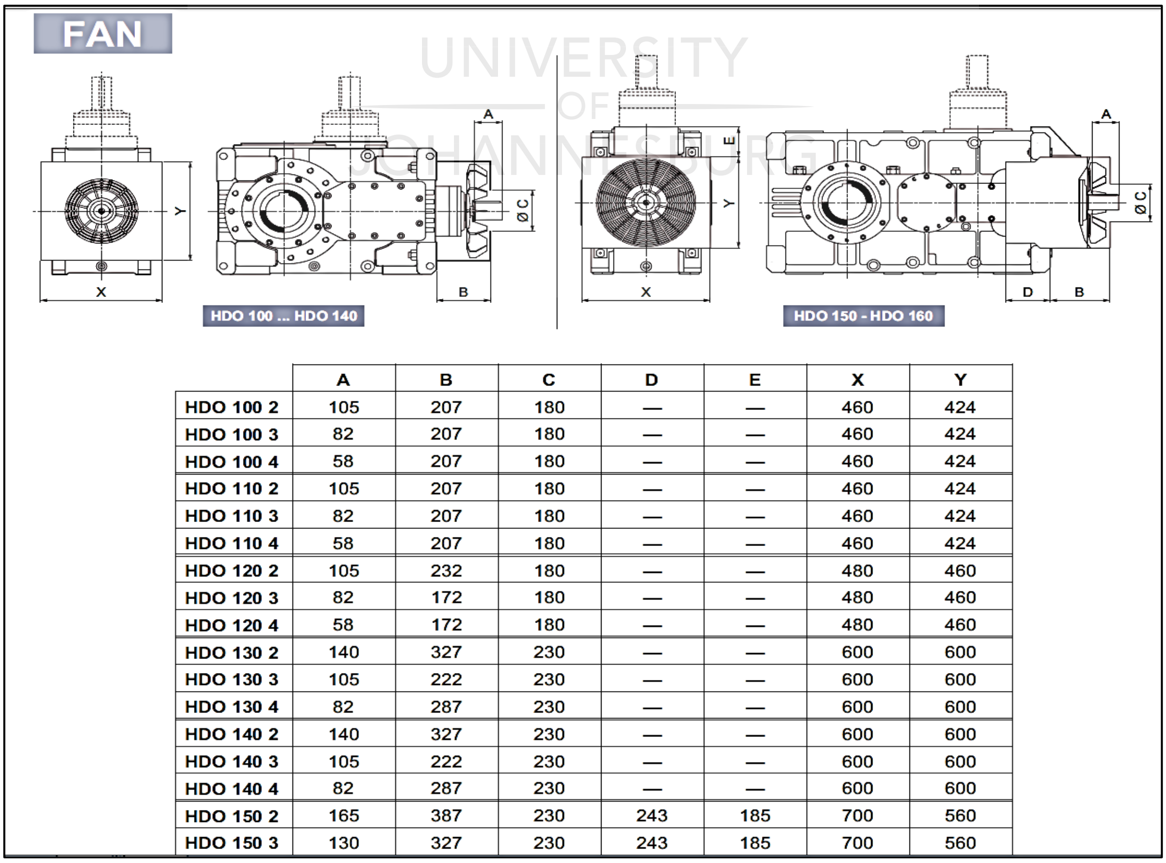
Unter folgenden Betriebsbedingungen:

- Einbaulage B3
- Dauerbetrieb
- Installation in großen Räumen (Luftgeschwindigkeit > 1.4 m/s)
- max. 1000 m ü NN

Die Werte der Gesamtwärmeleistung und die Werte der Wärmeleistung einschließlich des Beitrags durch eventuelle thermische Hilfsvorrichtungen sind in Kapitel 4.1 aufgeführt.
Für andere Bedingungen Kontakt technischen Kundendienst von Bonfiglioli Riduttori.

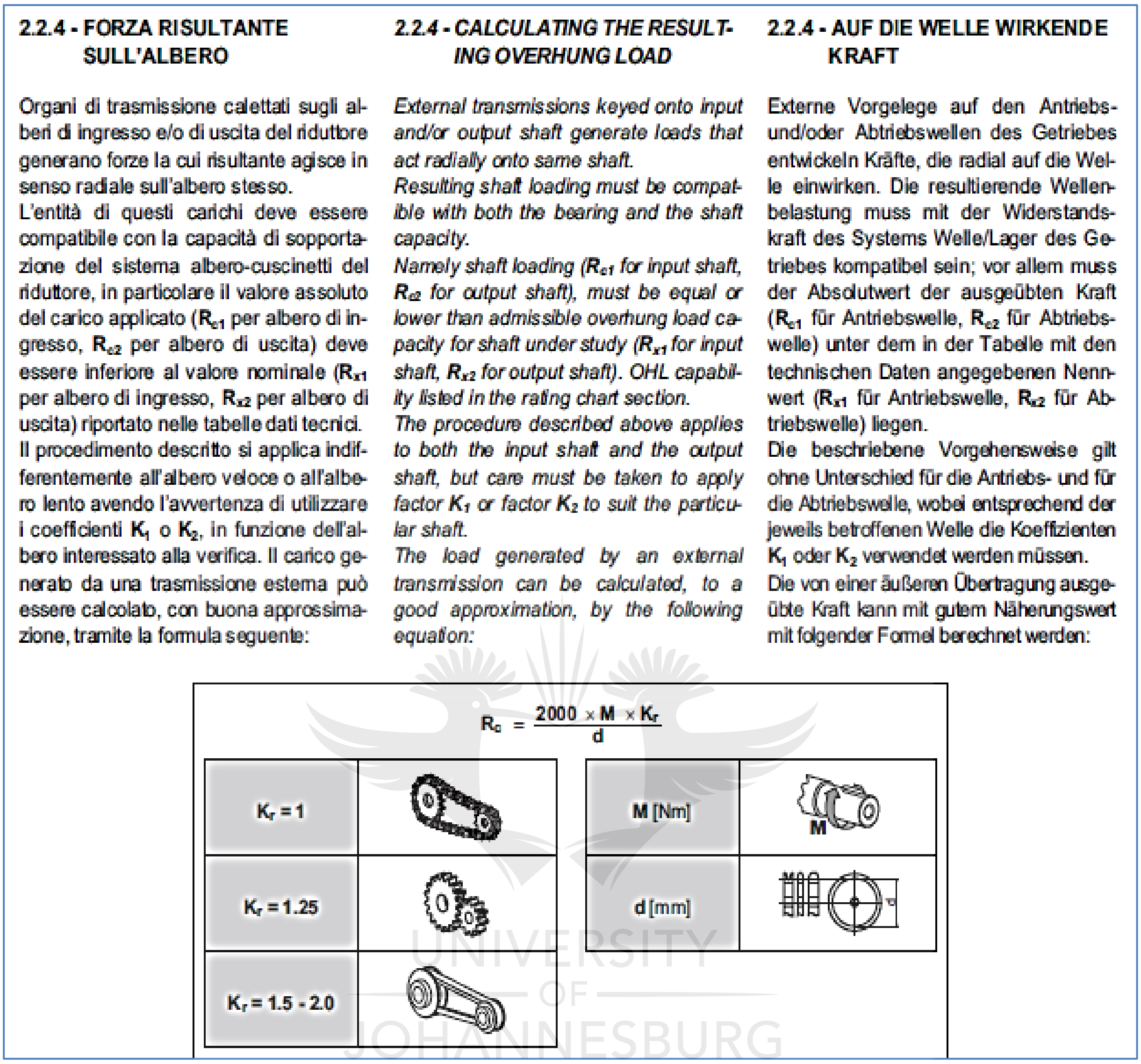
Der Wert bestimmt so muss über dem Leistungswert Pr_1 liegen, der an der Antriebswelle des Getriebes gefordert ist; folgende Bedingung muss deshalb überprüft werden:

$$P_{T...} \geq Pr_1$$



	A	B	C	D	E	X	Y
HDO 100 2	105	207	180	—	—	460	424
HDO 100 3	82	207	180	—	—	460	424
HDO 100 4	58	207	180	—	—	460	424
HDO 110 2	105	207	180	—	—	460	424
HDO 110 3	82	207	180	—	—	460	424
HDO 110 4	58	207	180	—	—	460	424
HDO 120 2	105	232	180	—	—	480	460
HDO 120 3	82	172	180	—	—	480	460
HDO 120 4	58	172	180	—	—	480	460
HDO 130 2	140	327	230	—	—	600	600
HDO 130 3	105	222	230	—	—	600	600
HDO 130 4	82	287	230	—	—	600	600
HDO 140 2	140	327	230	—	—	600	600
HDO 140 3	105	222	230	—	—	600	600
HDO 140 4	82	287	230	—	—	600	600
HDO 150 2	165	387	230	243	185	700	560
HDO 150 3	130	327	230	243	185	700	560

7. Gearbox overhanging load.



2.2.4 - FORZA RISULTANTE SULL'ALBERO

Organi di trasmissione calettati sugli alberi di ingresso e/o di uscita del riduttore generano forze la cui risultante agisce in senso radiale sull'albero stesso.

L'entità di questi carichi deve essere compatibile con la capacità di sopportazione del sistema albero-cuscinetti del riduttore, in particolare il valore assoluto del carico applicato (**R_{c1}** per albero di ingresso, **R_{c2}** per albero di uscita) deve essere inferiore al valore nominale (**R_{x1}** per albero di ingresso, **R_{x2}** per albero di uscita) riportato nelle tabelle dati tecnici.

Il procedimento descritto si applica indifferentemente all'albero veloce o all'albero lento avendo l'avvertenza di utilizzare i coefficienti **K_1** o **K_2**, in funzione dell'albero interessato alla verifica. Il carico generato da una trasmissione esterna può essere calcolato, con buona approssimazione, tramite la formula seguente:

2.2.4 - CALCULATING THE RESULTING OVERHUNG LOAD

External transmissions keyed onto input and/or output shaft generate loads that act radially onto same shaft.

Resulting shaft loading must be compatible with both the bearing and the shaft capacity.

*Namely shaft loading (**R_{c1}** for input shaft, **R_{c2}** for output shaft), must be equal or lower than admissible overhung load capacity for shaft under study (**R_{x1}** for input shaft, **R_{x2}** for output shaft). OHL capability listed in the rating chart section.*

*The procedure described above applies to both the input shaft and the output shaft, but care must be taken to apply factor **K_1** or factor **K_2** to suit the particular shaft.*

The load generated by an external transmission can be calculated, to a good approximation, by the following equation:

2.2.4 - AUF DIE WELLE WIRKENDE KRAFT

Externe Vorgelege auf den Antriebs- und/oder Abtriebswellen des Getriebes entwickeln Kräfte, die radial auf die Welle einwirken. Die resultierende Wellenbelastung muss mit der Widerstandskraft des Systems Welle/Lager des Getriebes kompatibel sein; vor allem muss der Absolutwert der ausgeübten Kraft (**R_{c1}** für Antriebswelle, **R_{c2}** für Abtriebswelle) unter dem in der Tabelle mit den technischen Daten angegebenen Nennwert (**R_{x1}** für Antriebswelle, **R_{x2}** für Abtriebswelle) liegen.

Die beschriebene Vorgehensweise gilt ohne Unterschied für die Antriebs- und für die Abtriebswelle, wobei entsprechend der jeweils betroffenen Welle die Koeffizienten **K_1** oder **K_2** verwendet werden müssen.

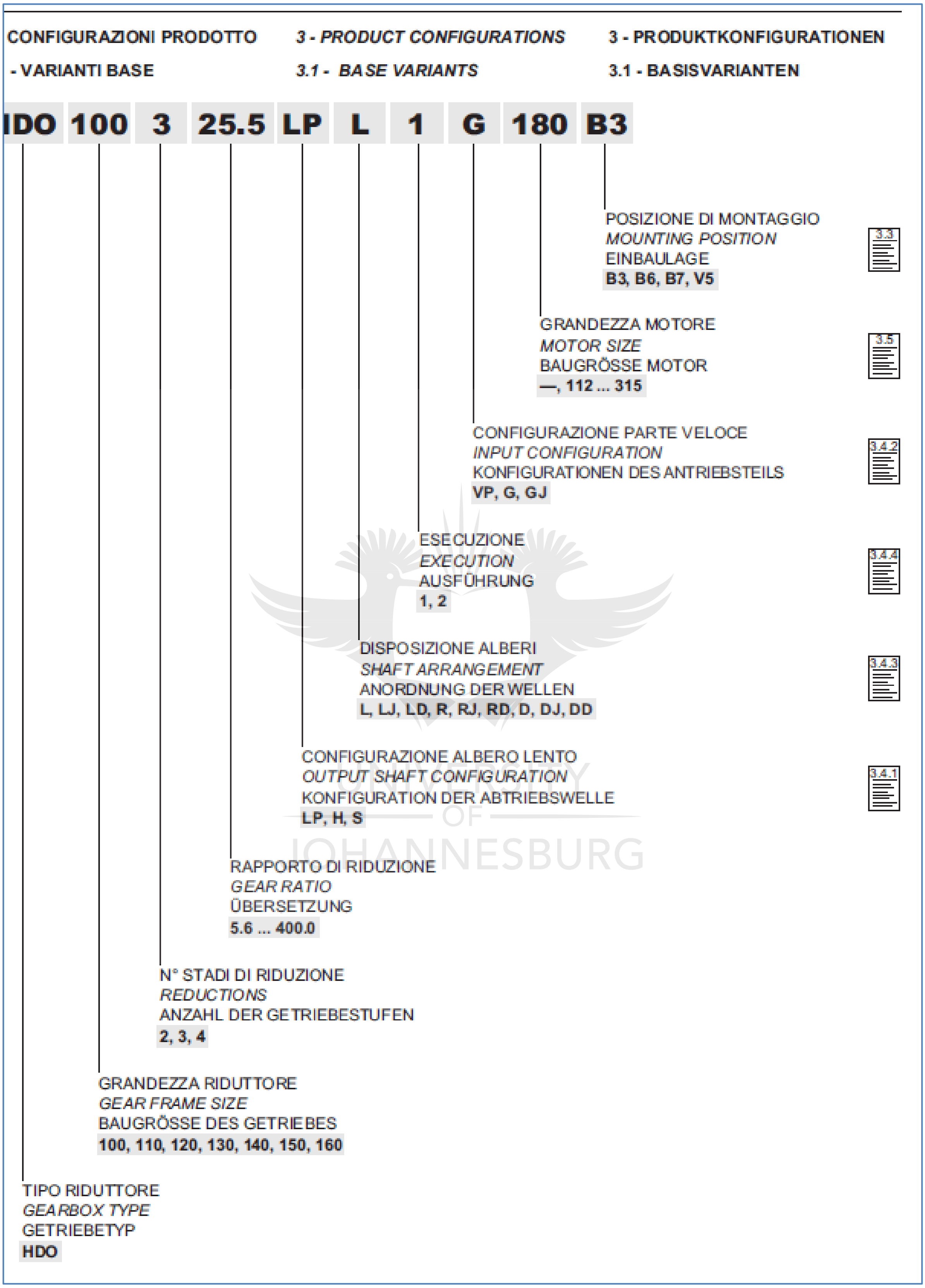
Die von einer äußeren Übertragung ausgeübte Kraft kann mit gutem Näherungswert mit folgender Formel berechnet werden:

$$R_c = \frac{2000 \times M \times K_r}{d}$$

$K_r = 1$	
$K_r = 1.25$	
$K_r = 1.5 - 2.0$	

M [Nm]	
d [mm]	

8. Gearbox specification description.



CONFIGURAZIONI PRODOTTO - VARIANTI BASE

3 - PRODUCT CONFIGURATIONS

3.1 - BASE VARIANTS

3 - PRODUKTKONFIGURATIONEN

3.1 - BASISVARIANTEN

IDO 100 3 25.5 LP L 1 G 180 B3

POSIZIONE DI MONTAGGIO
MOUNTING POSITION
EINBAULAGE
B3, B6, B7, V5
3.3

GRANDEZZA MOTORE
MOTOR SIZE
BAUGRÖSSE MOTOR
—, 112 ... 315
3.5

CONFIGURAZIONE PARTE VELOCE
INPUT CONFIGURATION
KONFIGURATIONEN DES ANTRIEBSTEILS
VP, G, GJ
3.4.2

ESECUZIONE
EXECUTION
AUSFÜHRUNG
1, 2
3.4.4

DISPOSIZIONE ALBERI
SHAFT ARRANGEMENT
ANORDNUNG DER WELLEN
L, LJ, LD, R, RJ, RD, D, DJ, DD
3.4.3

CONFIGURAZIONE ALBERO LENTO
OUTPUT SHAFT CONFIGURATION
KONFIGURATION DER ABTRIEBSWELLE
LP, H, S
3.4.1

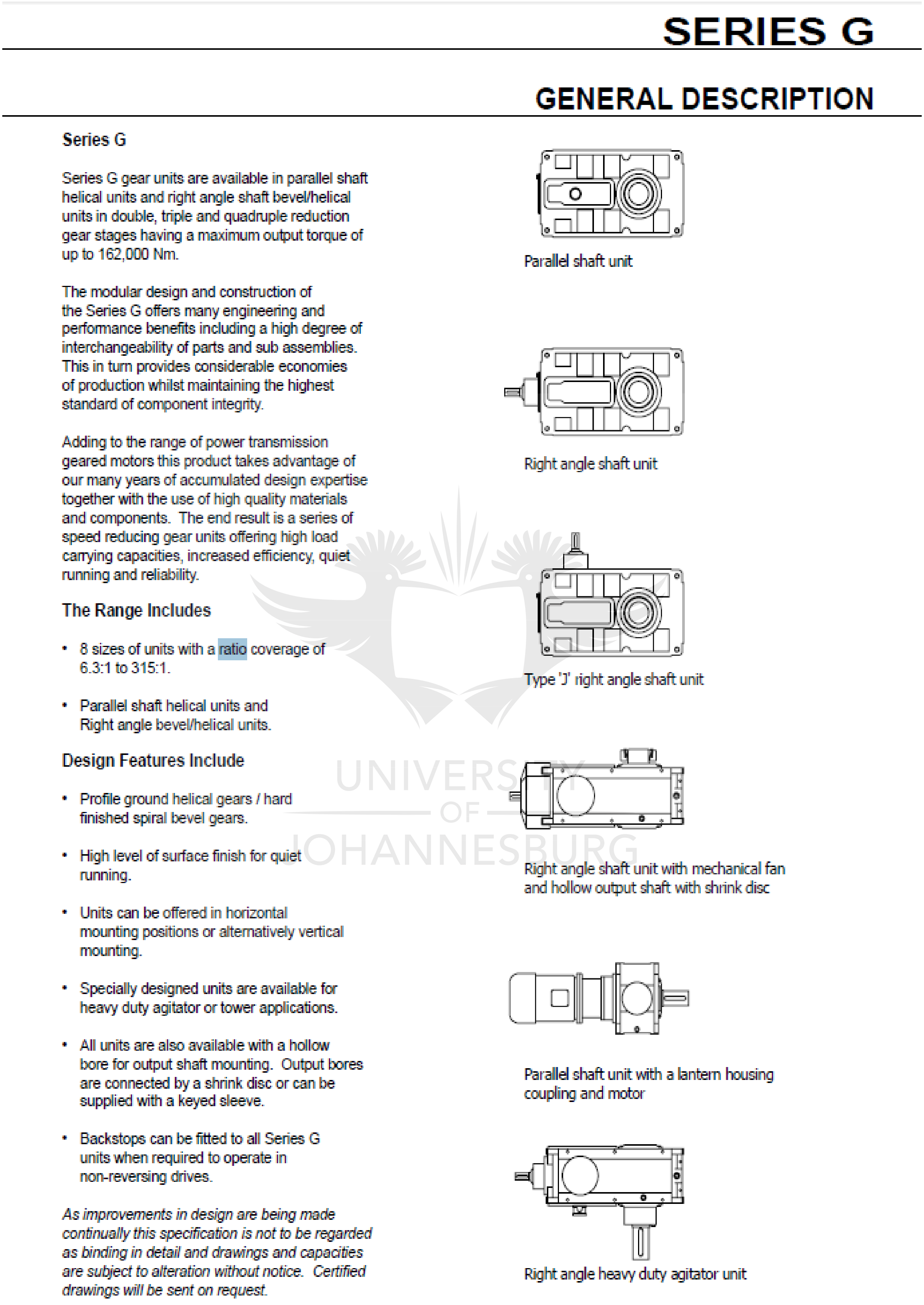
RAPPORTO DI RIDUZIONE
GEAR RATIO
ÜBERSETZUNG
5.6 ... 400.0

N° STADI DI RIDUZIONE
REDUCTIONS
ANZAHL DER GETRIEBESTUFEN
2, 3, 4

GRANDEZZA RIDUTTORE
GEAR FRAME SIZE
BAUGRÖSSE DES GETRIEBES
100, 110, 120, 130, 140, 150, 160

TIPO RIDUTTORE
GEARBOX TYPE
GETRIEBETYP
HDO

9. Redicon gearbox ratios.



SERIES G

GENERAL DESCRIPTION

Series G

Series G gear units are available in parallel shaft helical units and right angle shaft bevel/helical units in double, triple and quadruple reduction gear stages having a maximum output torque of up to 162,000 Nm.

The modular design and construction of the Series G offers many engineering and performance benefits including a high degree of interchangeability of parts and sub assemblies. This in turn provides considerable economies of production whilst maintaining the highest standard of component integrity.

Adding to the range of power transmission geared motors this product takes advantage of our many years of accumulated design expertise together with the use of high quality materials and components. The end result is a series of speed reducing gear units offering high load carrying capacities, increased efficiency, quiet running and reliability.

The Range Includes

- 8 sizes of units with a ratio coverage of 6.3:1 to 315:1.
- Parallel shaft helical units and Right angle bevel/helical units.

Design Features Include

- Profile ground helical gears / hard finished spiral bevel gears.
- High level of surface finish for quiet running.
- Units can be offered in horizontal mounting positions or alternatively vertical mounting.
- Specially designed units are available for heavy duty agitator or tower applications.
- All units are also available with a hollow bore for output shaft mounting. Output bores are connected by a shrink disc or can be supplied with a keyed sleeve.
- Backstops can be fitted to all Series G units when required to operate in non-reversing drives.

As improvements in design are being made continually this specification is not to be regarded as binding in detail and drawings and capacities are subject to alteration without notice. Certified drawings will be sent on request.

Parallel shaft unit

Right angle shaft unit

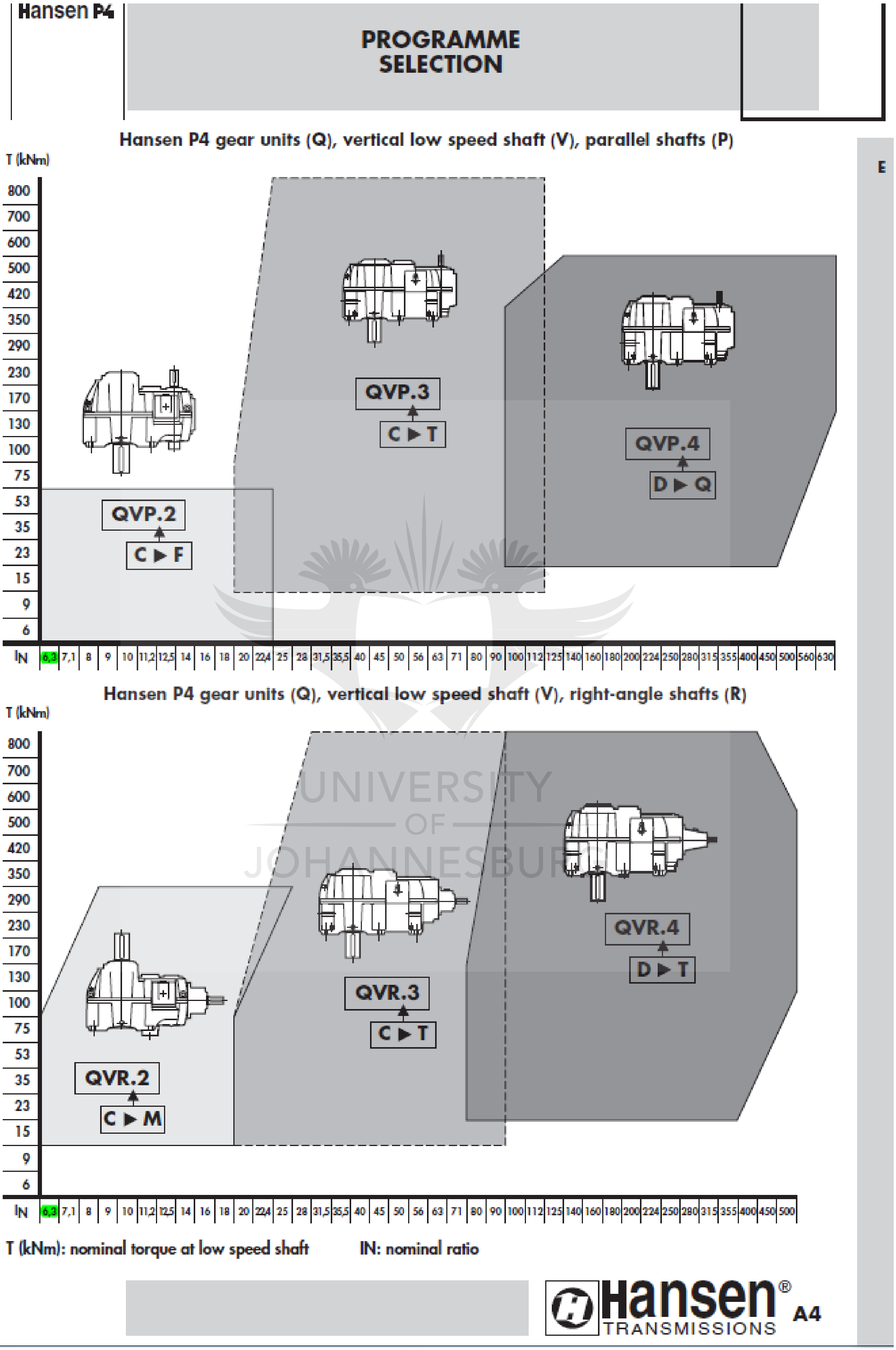
Type 'J' right angle shaft unit

Right angle shaft unit with mechanical fan and hollow output shaft with shrink disc

Parallel shaft unit with a lantern housing coupling and motor

Right angle heavy duty agitator unit

10. Hansen P4 right angle gearbox ratios.



Hansen P4

PROGRAMME SELECTION

E

Hansen P4 gear units (Q), vertical low speed shaft (V), parallel shafts (P)

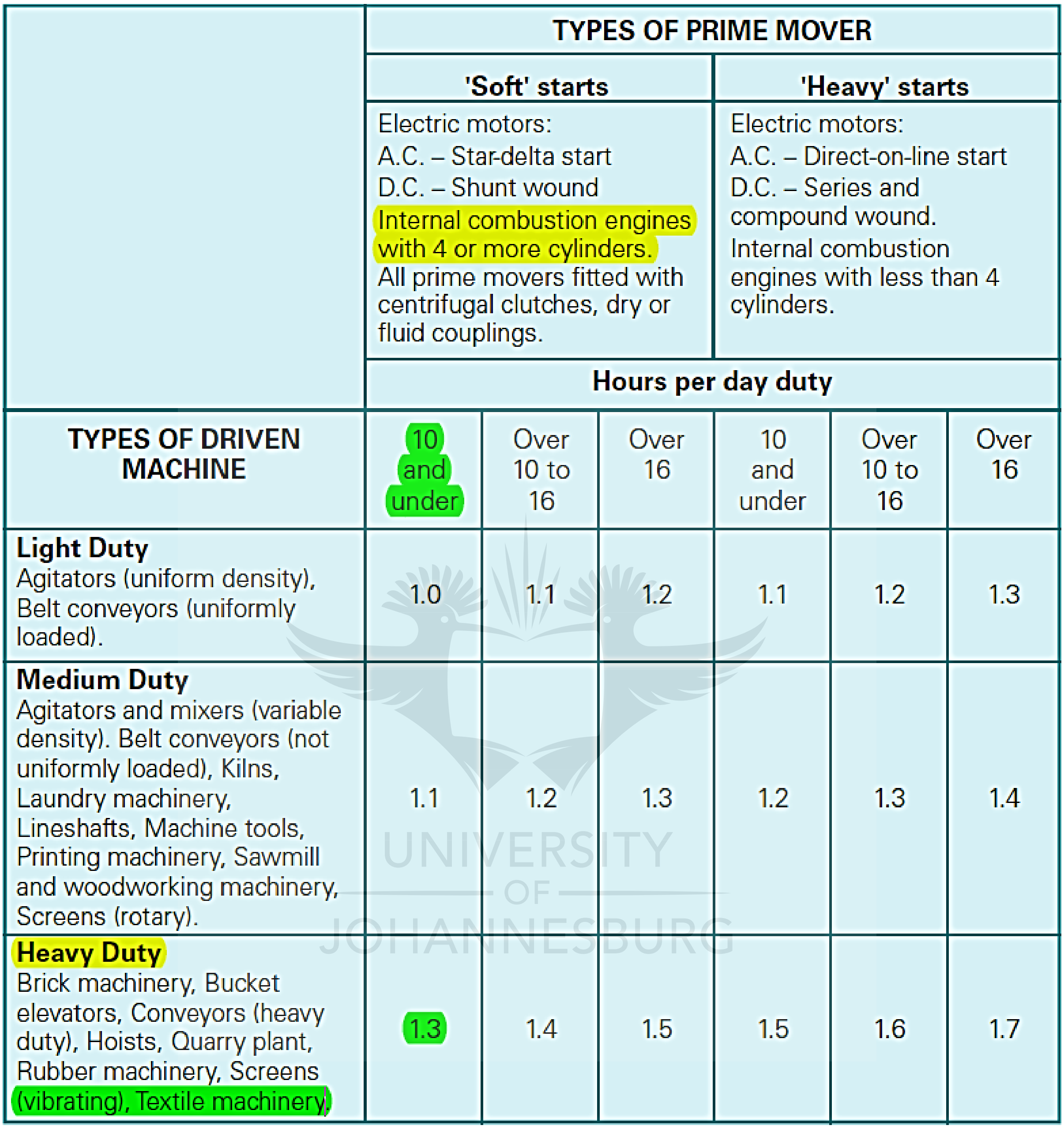
Hansen P4 gear units (Q), vertical low speed shaft (V), right-angle shafts (R)

T (kNm): nominal torque at low speed shaft IN: nominal ratio

Hansen® TRANSMISSIONS A4

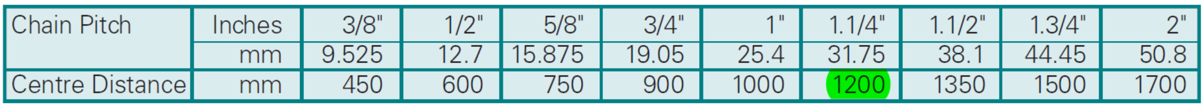
Appendix 9: Fenner Chain Drive

1. Service Factor table.



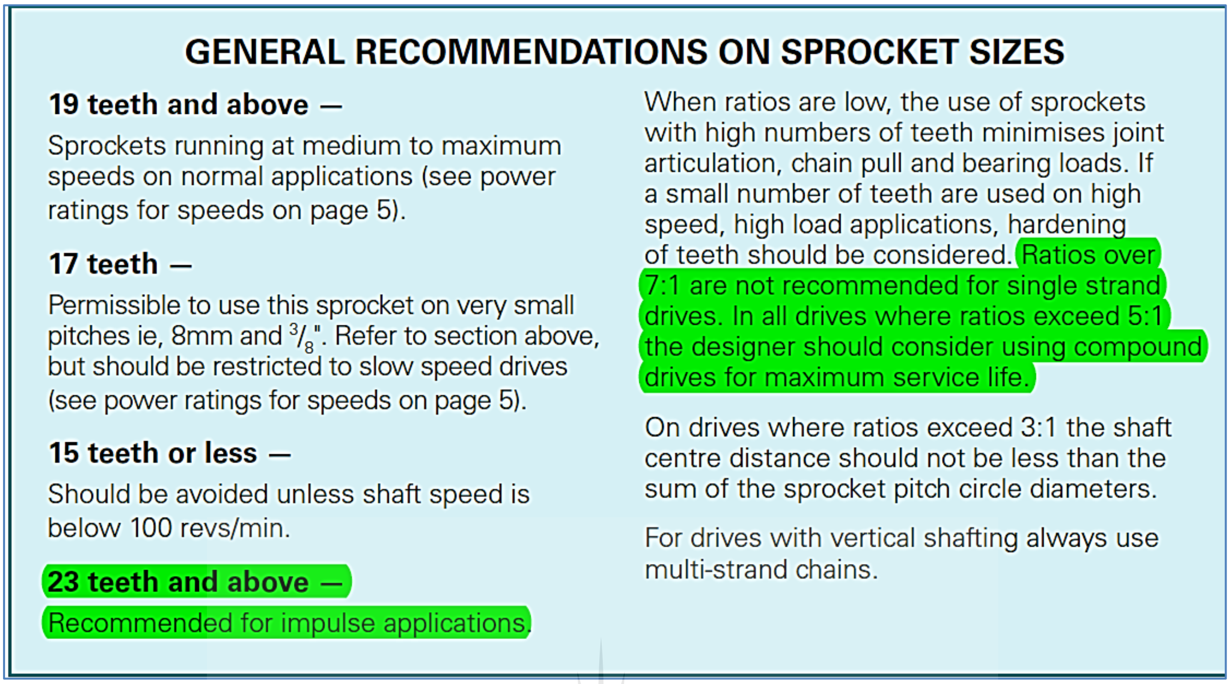
	TYPES OF PRIME MOVER					
	'Soft' starts			**'Heavy' starts**		
	Electric motors: A.C. – Star-delta start D.C. – Shunt wound Internal combustion engines with 4 or more cylinders. All prime movers fitted with centrifugal clutches, dry or fluid couplings.			Electric motors: A.C. – Direct-on-line start D.C. – Series and compound wound. Internal combustion engines with less than 4 cylinders.		
	Hours per day duty					
TYPES OF DRIVEN MACHINE	10 and under	Over 10 to 16	Over 16	10 and under	Over 10 to 16	Over 16
Light Duty Agitators (uniform density), Belt conveyors (uniformly loaded).	1.0	1.1	1.2	1.1	1.2	1.3
Medium Duty Agitators and mixers (variable density). Belt conveyors (not uniformly loaded), Kilns, Laundry machinery, Lineshafts, Machine tools, Printing machinery, Sawmill and woodworking machinery, Screens (rotary).	1.1	1.2	1.3	1.2	1.3	1.4
Heavy Duty Brick machinery, Bucket elevators, Conveyors (heavy duty), Hoists, Quarry plant, Rubber machinery, Screens (vibrating), Textile machinery	1.3	1.4	1.5	1.5	1.6	1.7

2. Centre distance table for 20B chain pitch.



Chain Pitch	Inches	3/8"	1/2"	5/8"	3/4"	1"	1.1/4"	1.1/2"	1.3/4"	2"
	mm	9.525	12.7	15.875	19.05	25.4	31.75	38.1	44.45	50.8
Centre Distance	mm	450	600	750	900	1000	1200	1350	1500	1700

3. Sprocket size recommendations.



GENERAL RECOMMENDATIONS ON SPROCKET SIZES

19 teeth and above —

Sprockets running at medium to maximum speeds on normal applications (see power ratings for speeds on page 5).

17 teeth —

Permissible to use this sprocket on very small pitches ie, 8mm and $^{3}/_{8}$". Refer to section above, but should be restricted to slow speed drives (see power ratings for speeds on page 5).

15 teeth or less —

Should be avoided unless shaft speed is below 100 revs/min.

23 teeth and above —

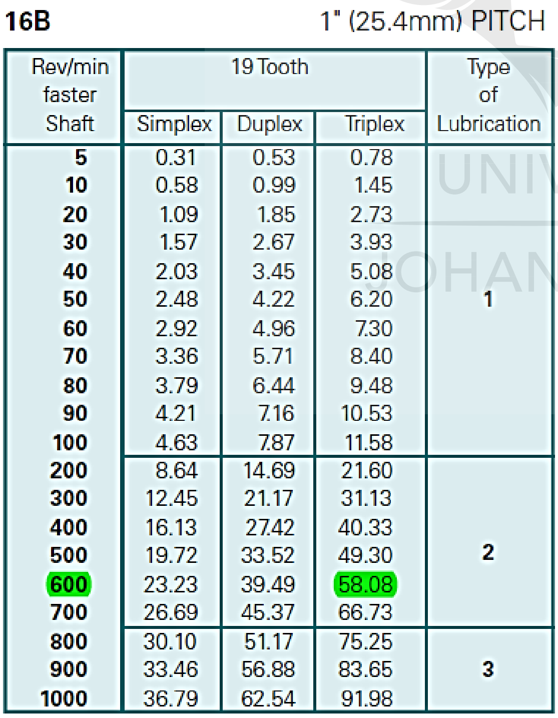
Recommended for impulse applications.

When ratios are low, the use of sprockets with high numbers of teeth minimises joint articulation, chain pull and bearing loads. If a small number of teeth are used on high speed, high load applications, hardening of teeth should be considered. Ratios over 7:1 are not recommended for single strand drives. In all drives where ratios exceed 5:1 the designer should consider using compound drives for maximum service life.

On drives where ratios exceed 3:1 the shaft centre distance should not be less than the sum of the sprocket pitch circle diameters.

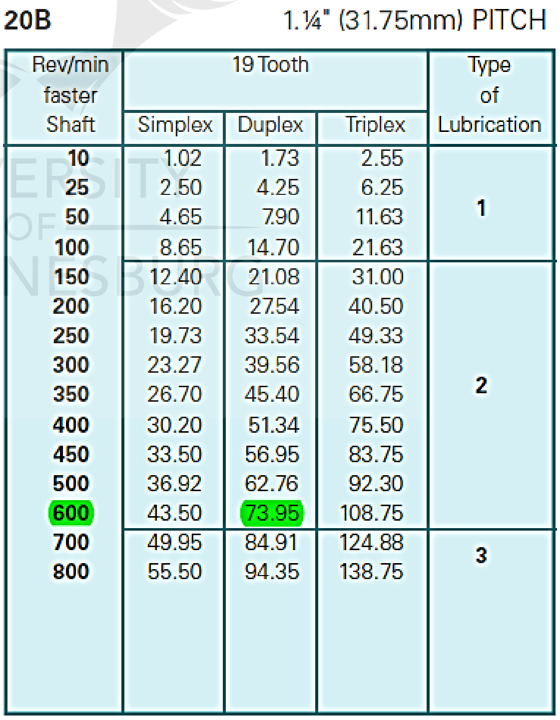
For drives with vertical shafting always use multi-strand chains.

4. Power rating tables & sprocket factor (suitable power ratings).



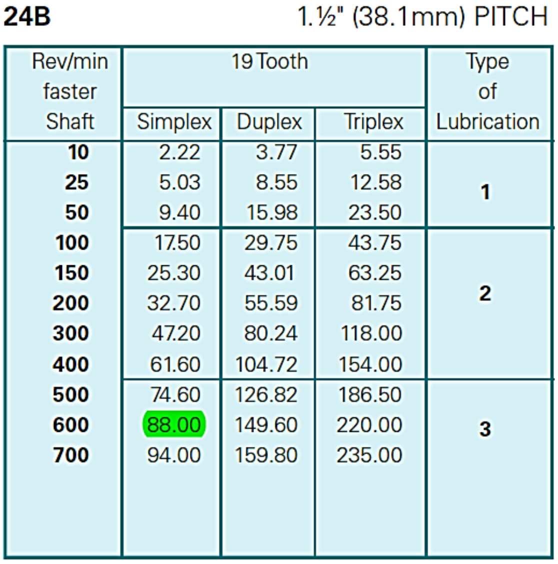
16B 1" (25.4mm) PITCH

Rev/min faster Shaft	19 Tooth			Type of Lubrication
	Simplex	Duplex	Triplex	
5	0.31	0.53	0.78	1
10	0.58	0.99	1.45	
20	1.09	1.85	2.73	
30	1.57	2.67	3.93	
40	2.03	3.45	5.08	
50	2.48	4.22	6.20	
60	2.92	4.96	7.30	
70	3.36	5.71	8.40	
80	3.79	6.44	9.48	
90	4.21	7.16	10.53	
100	4.63	7.87	11.58	
200	8.64	14.69	21.60	2
300	12.45	21.17	31.13	
400	16.13	27.42	40.33	
500	19.72	33.52	49.30	
600	23.23	39.49	58.08	
700	26.69	45.37	66.73	
800	30.10	51.17	75.25	3
900	33.46	56.88	83.65	
1000	36.79	62.54	91.98	



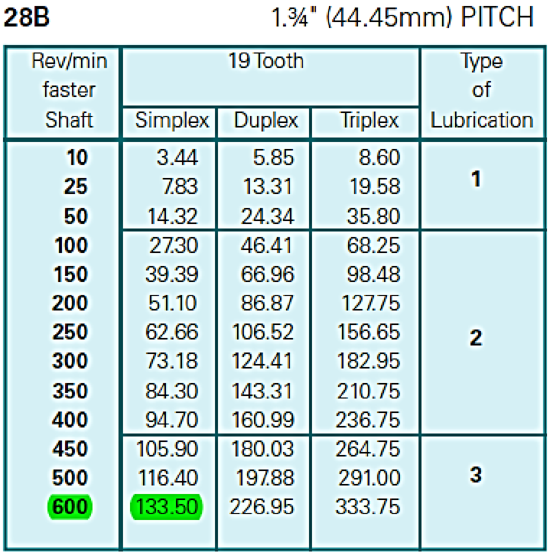
20B 1.¼" (31.75mm) PITCH

Rev/min faster Shaft	19 Tooth			Type of Lubrication
	Simplex	Duplex	Triplex	
10	1.02	1.73	2.55	1
25	2.50	4.25	6.25	
50	4.65	7.90	11.63	
100	8.65	14.70	21.63	
150	12.40	21.08	31.00	2
200	16.20	27.54	40.50	
250	19.73	33.54	49.33	
300	23.27	39.56	58.18	
350	26.70	45.40	66.75	
400	30.20	51.34	75.50	
450	33.50	56.95	83.75	
500	36.92	62.76	92.30	
600	43.50	73.95	108.75	
700	49.95	84.91	124.88	3
800	55.50	94.35	138.75	



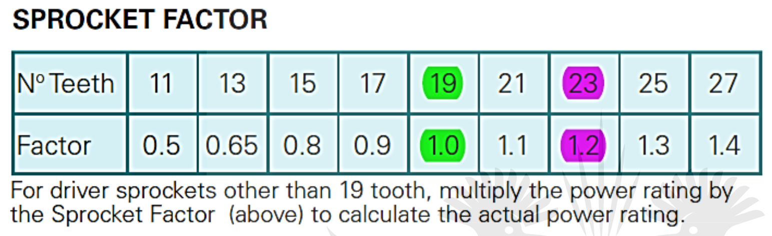
24B 1.½" (38.1mm) PITCH

Rev/min faster Shaft	19 Tooth			Type of Lubrication
	Simplex	Duplex	Triplex	
10	2.22	3.77	5.55	**1**
25	5.03	8.55	12.58	
50	9.40	15.98	23.50	
100	17.50	29.75	43.75	**2**
150	25.30	43.01	63.25	
200	32.70	55.59	81.75	
300	47.20	80.24	118.00	
400	61.60	104.72	154.00	
500	74.60	126.82	186.50	**3**
600	88.00	149.60	220.00	
700	94.00	159.80	235.00	



28B 1.¾" (44.45mm) PITCH

Rev/min faster Shaft	19 Tooth			Type of Lubrication
	Simplex	Duplex	Triplex	
10	3.44	5.85	8.60	**1**
25	7.83	13.31	19.58	
50	14.32	24.34	35.80	
100	27.30	46.41	68.25	**2**
150	39.39	66.96	98.48	
200	51.10	86.87	127.75	
250	62.66	106.52	156.65	
300	73.18	124.41	182.95	
350	84.30	143.31	210.75	
400	94.70	160.99	236.75	
450	105.90	180.03	264.75	**3**
500	116.40	197.88	291.00	
600	133.50	226.95	333.75	



SPROCKET FACTOR

N° Teeth	11	13	15	17	19	21	23	25	27
Factor	0.5	0.65	0.8	0.9	1.0	1.1	1.2	1.3	1.4

For driver sprockets other than 19 tooth, multiply the power rating by the Sprocket Factor (above) to calculate the actual power rating.

5. Speed ratios for 23 teeth and 95 teeth sprockets available.

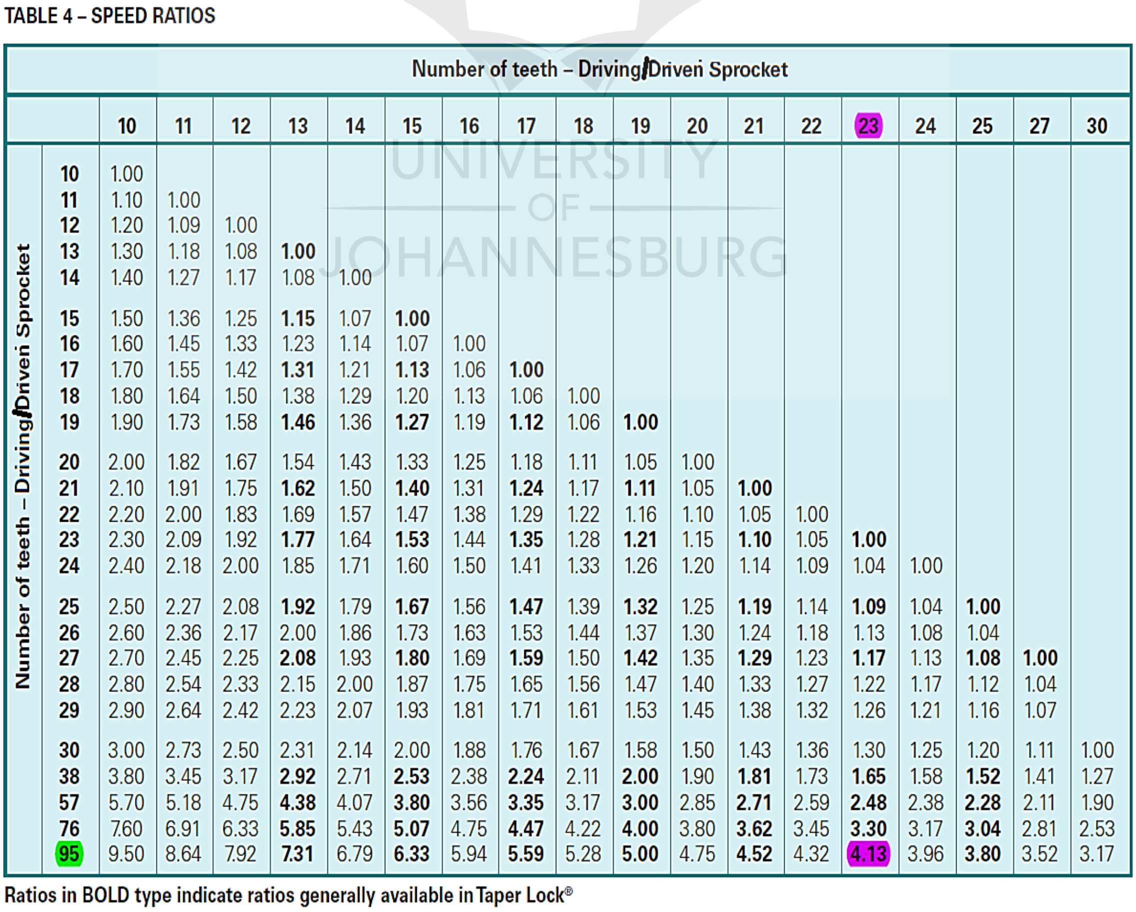
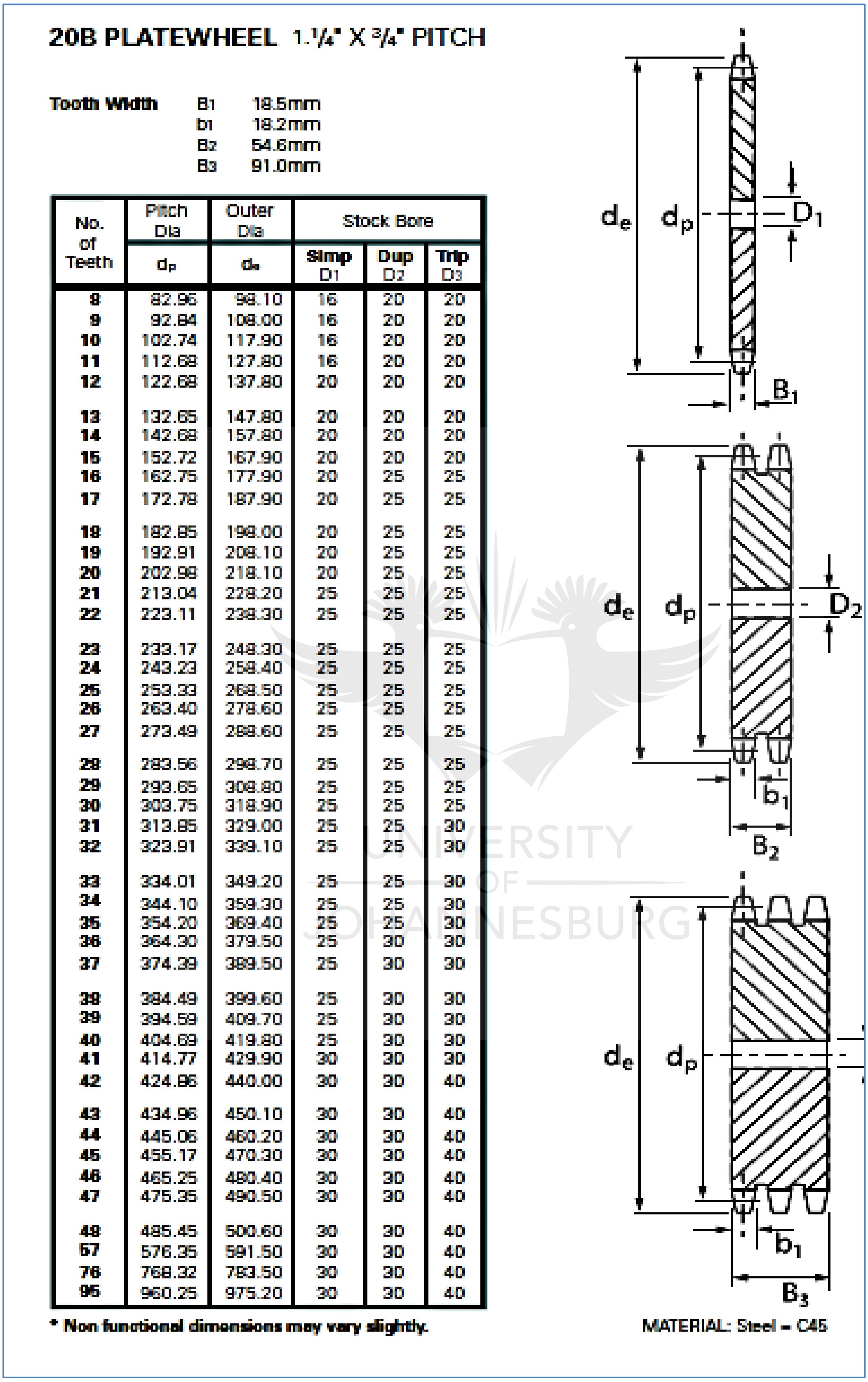


TABLE 4 – SPEED RATIOS

		Number of teeth – Driving/Driven Sprocket																	
		10	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**27**	**30**
Number of teeth – Driving/Driven Sprocket	**10**	1.00																	
	11	1.10	1.00																
	12	1.20	1.09	1.00															
	13	1.30	1.18	1.08	**1.00**														
	14	1.40	1.27	1.17	1.08	1.00													
	15	1.50	1.36	1.25	**1.15**	1.07	**1.00**												
	16	1.60	1.45	1.33	1.23	1.14	1.07	1.00											
	17	1.70	1.55	1.42	**1.31**	1.21	**1.13**	1.06	**1.00**										
	18	1.80	1.64	1.50	1.38	1.29	1.20	1.13	1.06	1.00									
	19	1.90	1.73	1.58	**1.46**	1.36	**1.27**	1.19	**1.12**	1.06	**1.00**								
	20	2.00	1.82	1.67	1.54	1.43	1.33	1.25	1.18	1.11	1.05	1.00							
	21	2.10	1.91	1.75	**1.62**	1.50	**1.40**	1.31	**1.24**	1.17	**1.11**	1.05	**1.00**						
	22	2.20	2.00	1.83	1.69	1.57	1.47	1.38	1.29	1.22	1.16	1.10	1.05	1.00					
	23	2.30	2.09	1.92	**1.77**	1.64	**1.53**	1.44	**1.35**	1.28	**1.21**	1.15	**1.10**	1.05	**1.00**				
	24	2.40	2.18	2.00	1.85	1.71	1.60	1.50	1.41	1.33	1.26	1.20	1.14	1.09	1.04	1.00			
	25	2.50	2.27	2.08	**1.92**	1.79	**1.67**	1.56	**1.47**	1.39	**1.32**	1.25	**1.19**	1.14	**1.09**	1.04	**1.00**		
	26	2.60	2.36	2.17	2.00	1.86	1.73	1.63	1.53	1.44	1.37	1.30	1.24	1.18	1.13	1.08	1.04		
	27	2.70	2.45	2.25	**2.08**	1.93	**1.80**	1.69	**1.59**	1.50	**1.42**	1.35	**1.29**	1.23	**1.17**	1.13	**1.08**	**1.00**	
	28	2.80	2.54	2.33	2.15	2.00	1.87	1.75	1.65	1.56	1.47	1.40	1.33	1.27	1.22	1.17	1.12	1.04	
	29	2.90	2.64	2.42	2.23	2.07	1.93	1.81	1.71	1.61	1.53	1.45	1.38	1.32	1.26	1.21	1.16	1.07	
	30	3.00	2.73	2.50	2.31	2.14	2.00	1.88	1.76	1.67	1.58	1.50	1.43	1.36	1.30	1.25	1.20	1.11	1.00
	38	3.80	3.45	3.17	**2.92**	2.71	**2.53**	2.38	**2.24**	2.11	**2.00**	1.90	**1.81**	1.73	**1.65**	1.58	**1.52**	1.41	1.27
	57	5.70	5.18	4.75	**4.38**	4.07	**3.80**	3.56	**3.35**	3.17	**3.00**	2.85	**2.71**	2.59	**2.48**	2.38	**2.28**	2.11	1.90
	76	7.60	6.91	6.33	**5.85**	5.43	**5.07**	4.75	**4.47**	4.22	**4.00**	3.80	**3.62**	3.45	**3.30**	3.17	**3.04**	2.81	2.53
	95	9.50	8.64	7.92	**7.31**	6.79	**6.33**	5.94	**5.59**	5.28	**5.00**	4.75	**4.52**	4.32	**4.13**	3.96	**3.80**	3.52	3.17

Ratios in BOLD type indicate ratios generally available in Taper Lock®

6. Plate Wheel Sprocket Type (23 teeth and 95 teeth selected).



20B PLATEWHEEL 1.¼" X ¾" PITCH

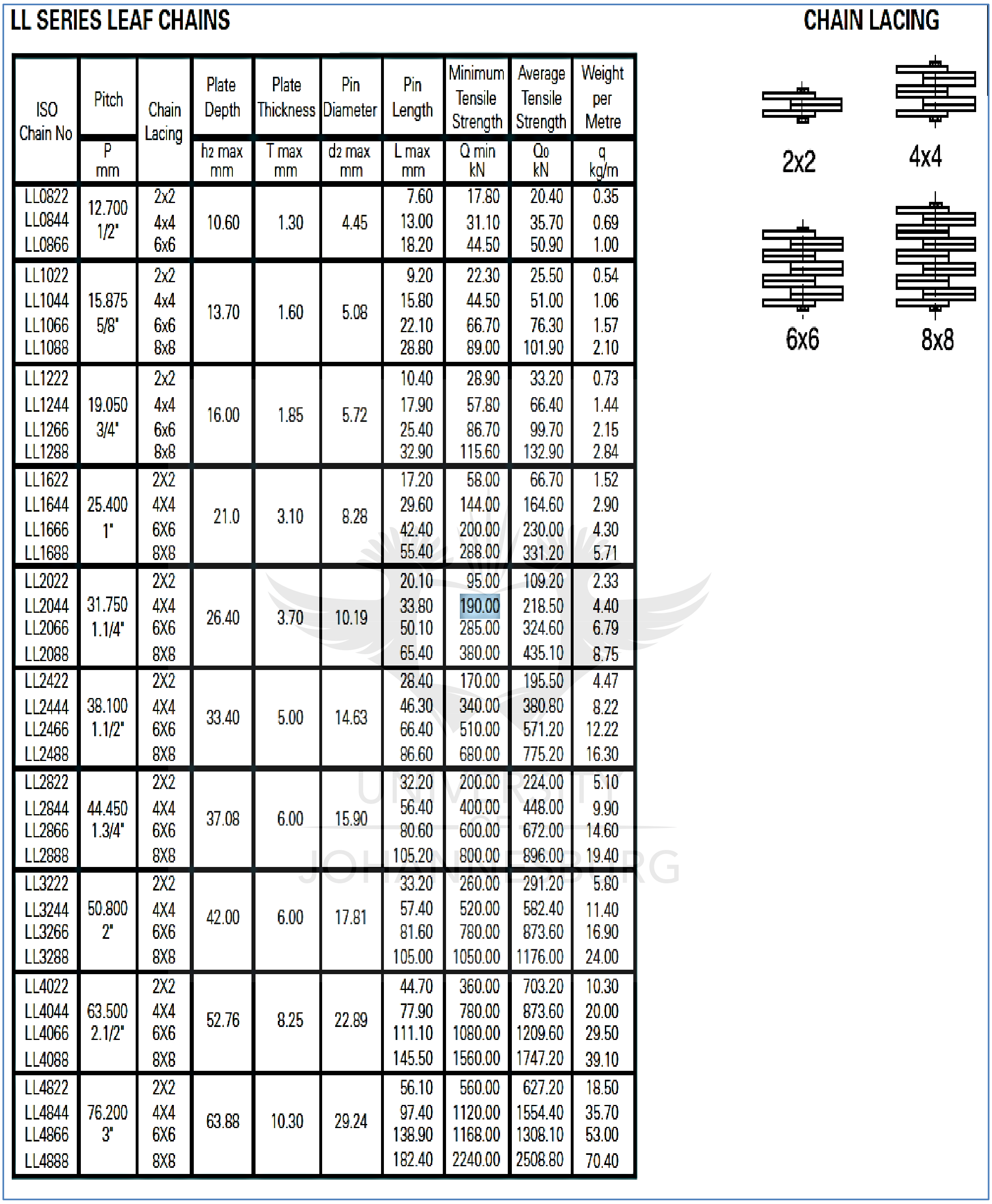
Tooth Width		
	B1	18.5mm
	b1	18.2mm
	B2	54.6mm
	B3	91.0mm

No. of Teeth	Pitch Dia	Outer Dia	Stock Bore		
	d_p	d_a	**Simp** D1	**Dup** D2	**Trip** D3
8	82.96	98.10	16	20	20
9	92.84	108.00	16	20	20
10	102.74	117.90	16	20	20
11	112.68	127.80	16	20	20
12	122.68	137.80	20	20	20
13	132.65	147.80	20	20	20
14	142.68	157.80	20	20	20
15	152.72	167.90	20	20	20
16	162.75	177.90	20	25	25
17	172.78	187.90	20	25	25
18	182.85	198.00	20	25	25
19	192.91	208.10	20	25	25
20	202.98	218.10	20	25	25
21	213.04	228.20	25	25	25
22	223.11	238.30	25	25	25
23	233.17	248.30	25	25	25
24	243.23	258.40	25	25	25
25	253.33	268.50	25	25	25
26	263.40	278.60	25	25	25
27	273.49	288.60	25	25	25
28	283.56	298.70	25	25	25
29	293.65	308.80	25	25	25
30	303.75	318.90	25	25	25
31	313.85	329.00	25	25	30
32	323.91	339.10	25	25	30
33	334.01	349.20	25	25	30
34	344.10	359.30	25	25	30
35	354.20	369.40	25	25	30
36	364.30	379.50	25	30	30
37	374.39	389.50	25	30	30
38	384.49	399.60	25	30	30
39	394.59	409.70	25	30	30
40	404.69	419.80	25	30	30
41	414.77	429.90	30	30	30
42	424.86	440.00	30	30	40
43	434.96	450.10	30	30	40
44	445.06	460.20	30	30	40
45	455.17	470.30	30	30	40
46	465.25	480.40	30	30	40
47	475.35	490.50	30	30	40
48	485.45	500.60	30	30	40
57	576.35	591.50	30	30	40
76	768.32	783.50	30	30	40
95	960.25	975.20	30	30	40

*** Non functional dimensions may vary slightly.**

MATERIAL: Steel – C45

7. Leaf Chain selection for pitch 31.75 mm (20B) at a minimum tensile strength of 190 kN.



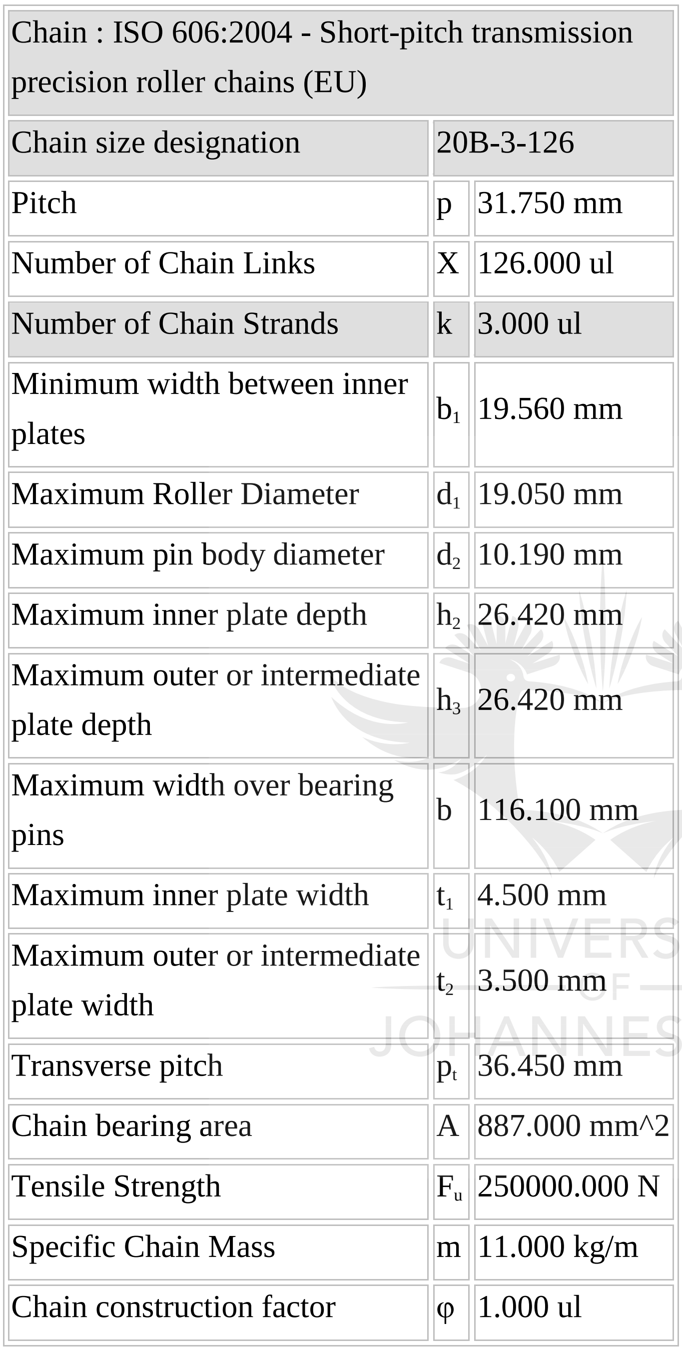
LL SERIES LEAF CHAINS

ISO Chain No	Pitch	Chain Lacing	Plate Depth	Plate Thickness	Pin Diameter	Pin Length	Minimum Tensile Strength	Average Tensile Strength	Weight per Metre
	P mm		h2 max mm	T max mm	d2 max mm	L max mm	Q min kN	Q0 kN	q kg/m
LL0822	12.700 1/2'	2x2	10.60	1.30	4.45	7.60	17.80	20.40	0.35
LL0844		4x4				13.00	31.10	35.70	0.69
LL0866		6x6				18.20	44.50	50.90	1.00
LL1022	15.875 5/8'	2x2	13.70	1.60	5.08	9.20	22.30	25.50	0.54
LL1044		4x4				15.80	44.50	51.00	1.06
LL1066		6x6				22.10	66.70	76.30	1.57
LL1088		8x8				28.80	89.00	101.90	2.10
LL1222	19.050 3/4'	2x2	16.00	1.85	5.72	10.40	28.90	33.20	0.73
LL1244		4x4				17.90	57.80	66.40	1.44
LL1266		6x6				25.40	86.70	99.70	2.15
LL1288		8x8				32.90	115.60	132.90	2.84
LL1622	25.400 1'	2X2	21.0	3.10	8.28	17.20	58.00	66.70	1.52
LL1644		4X4				29.60	144.00	164.60	2.90
LL1666		6X6				42.40	200.00	230.00	4.30
LL1688		8X8				55.40	288.00	331.20	5.71
LL2022	31.750 1.1/4'	2X2	26.40	3.70	10.19	20.10	95.00	109.20	2.33
LL2044		4X4				33.80	190.00	218.50	4.40
LL2066		6X6				50.10	285.00	324.60	6.79
LL2088		8X8				65.40	380.00	435.10	8.75
LL2422	38.100 1.1/2'	2X2	33.40	5.00	14.63	28.40	170.00	195.50	4.47
LL2444		4X4				46.30	340.00	380.80	8.22
LL2466		6X6				66.40	510.00	571.20	12.22
LL2488		8X8				86.60	680.00	775.20	16.30
LL2822	44.450 1.3/4'	2X2	37.08	6.00	15.90	32.20	200.00	224.00	5.10
LL2844		4X4				56.40	400.00	448.00	9.90
LL2866		6X6				80.60	600.00	672.00	14.60
LL2888		8X8				105.20	800.00	896.00	19.40
LL3222	50.800 2'	2X2	42.00	6.00	17.81	33.20	260.00	291.20	5.80
LL3244		4X4				57.40	520.00	582.40	11.40
LL3266		6X6				81.60	780.00	873.60	16.90
LL3288		8X8				105.00	1050.00	1176.00	24.00
LL4022	63.500 2.1/2'	2X2	52.76	8.25	22.89	44.70	360.00	703.20	10.30
LL4044		4X4				77.90	780.00	873.60	20.00
LL4066		6X6				111.10	1080.00	1209.60	29.50
LL4088		8X8				145.50	1560.00	1747.20	39.10
LL4822	76.200 3'	2X2	63.88	10.30	29.24	56.10	560.00	627.20	18.50
LL4844		4X4				97.40	1120.00	1554.40	35.70
LL4866		6X6				138.90	1168.00	1308.10	53.00
LL4888		8X8				182.40	2240.00	2508.80	70.40

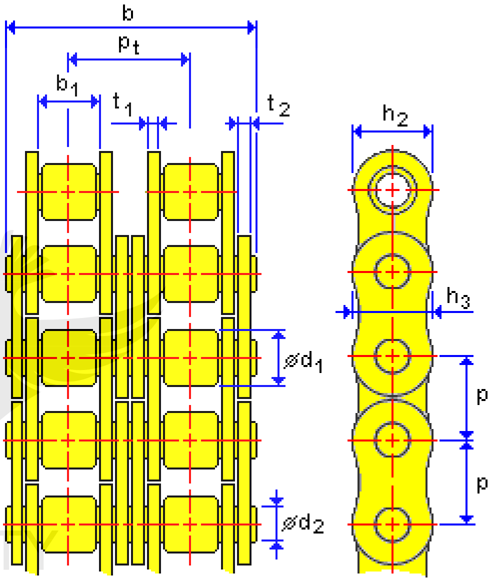
CHAIN LACING

8. Roller Chains Generator (Inventor Version: 2013).

Chain 20B (Triplex) properties

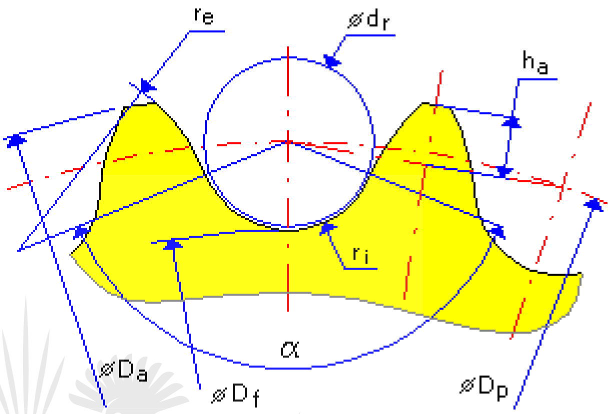


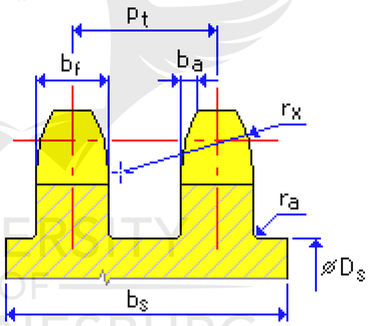
Chain : ISO 606:2004 - Short-pitch transmission precision roller chains (EU)		
Chain size designation	20B-3-126	
Pitch	p	31.750 mm
Number of Chain Links	X	126.000 ul
Number of Chain Strands	k	3.000 ul
Minimum width between inner plates	b_1	19.560 mm
Maximum Roller Diameter	d_1	19.050 mm
Maximum pin body diameter	d_2	10.190 mm
Maximum inner plate depth	h_2	26.420 mm
Maximum outer or intermediate plate depth	h_3	26.420 mm
Maximum width over bearing pins	b	116.100 mm
Maximum inner plate width	t_1	4.500 mm
Maximum outer or intermediate plate width	t_2	3.500 mm
Transverse pitch	p_t	36.450 mm
Chain bearing area	A	887.000 mm^2
Tensile Strength	F_u	250000.000 N
Specific Chain Mass	m	11.000 kg/m
Chain construction factor	φ	1.000 ul

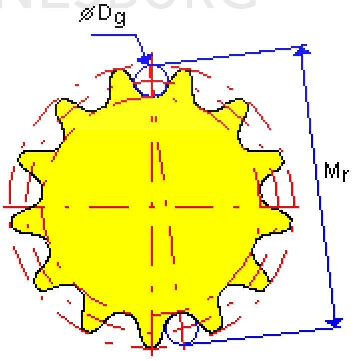


Sprocket 2 properties: Toothed sprocket

Type	Driven sprocket	
Number of Teeth	z	23.000 ul
Number of Teeth in Contact	z_c	9.000 ul
Pitch Diameter	D_p	233.170 mm
Number of strands	k	3.000 ul
Transverse pitch	p_t	36.450 mm
Seating clearance	SC	0.095 mm
Tooth width	b_f	18.191 mm
Tooth side relief	b_a	4.128 mm
Tooth side radius	r_x	31.750 mm
Shroud diameter	D_s	193.382 mm
Sprocket shroud width	b_s	91.091 mm
Height of tooth above pitch polygon	h_a	9.525 mm
Roller-seating radius	r_i	9.620 mm
Tooth-flank radius	r_e	57.150 mm
Roller-seating angle	α	136.09 deg
Shroud fillet radius	r_a	1.270 mm
Sprocket tip diameter	D_a	250.048 mm
Sprocket root diameter	D_f	213.930 mm
Measuring pin diameter	D_g	19.050 mm
Measurement over pins	M_R	251.677 mm
Span Length	L_f	927.746 mm
Power Ratio	P_x	1.000 ul
Power	P	46.124 kW
Torque	T	808.168 N m
Speed	n	545.000 r/min
Arc of contact	β	137.20 deg

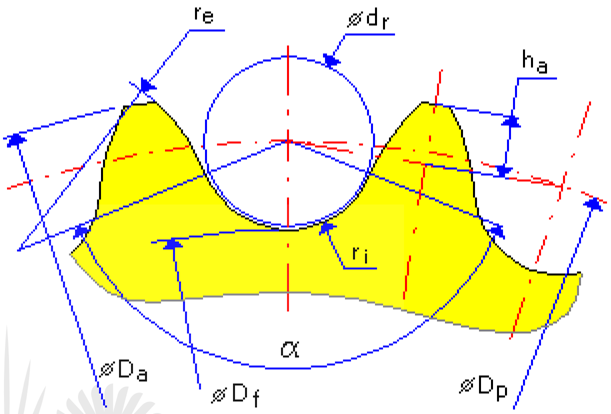


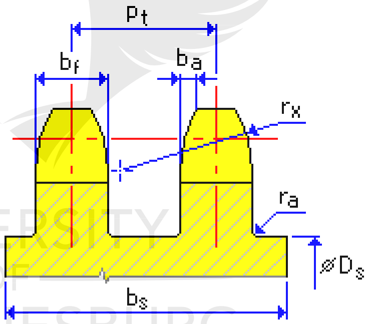


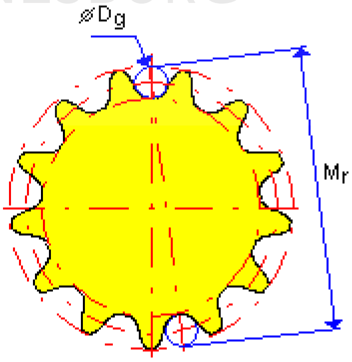


Sprocket 1 properties: Toothed sprocket

Type	Driver sprocket	
Number of Teeth	z	95.000 ul
Number of Teeth in Contact	z_c	59.000 ul
Pitch Diameter	D_p	960.277 mm
Number of strands	k	3.000 ul
Transverse pitch	p_t	36.450 mm
Seating clearance	SC	0.095 mm
Tooth width	b_f	18.191 mm
Tooth side radius	r_x	31.750 mm
Shroud diameter	D_s	922.135 mm
Sprocket shroud width	b_s	91.091 mm
Height of tooth above pitch polygon	h_a	9.525 mm
Roller-seating radius	r_i	9.620 mm
Tootk-flank radius	r_e	221.742 mm
Roller-seating angle	α	139.05 deg
Shroud fillet radius	r_a	1.270 mm
Sprocket tip diameter	D_a	978.802 mm
Sprocket root diameter	D_f	941.037 mm
Measuring pin diameter	D_g	19.050 mm
Measurement over pins	M_R	979.196 mm
Centre Distance	C	996.435 mm
Span Length	L_f	927.746 mm
Power Ratio	P_x	1.000 ul
Power	P	47.046 kW
Torque	T	3271.325 N m
Speed	n	131.947 r/min
Transmission Ratio	i	4.130 ul
Arc of contact	β	222.80 deg



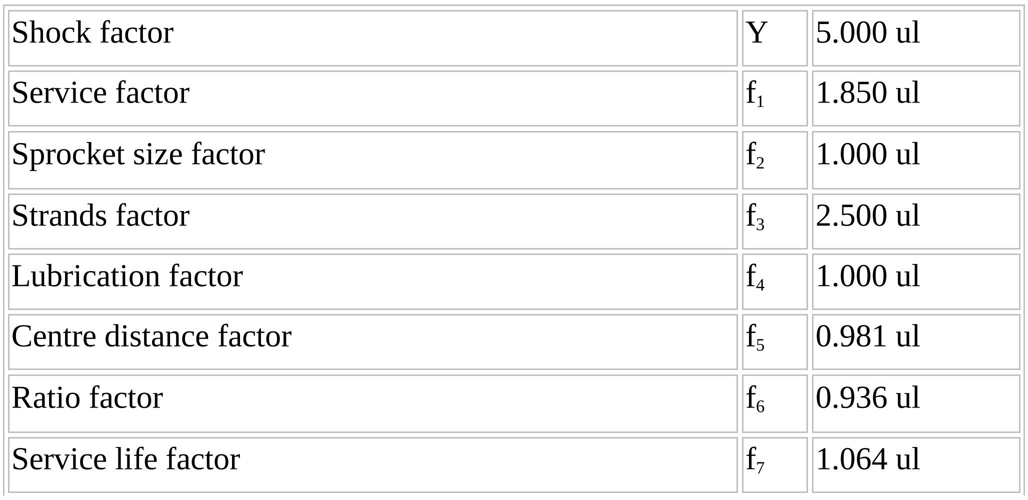




Working conditions

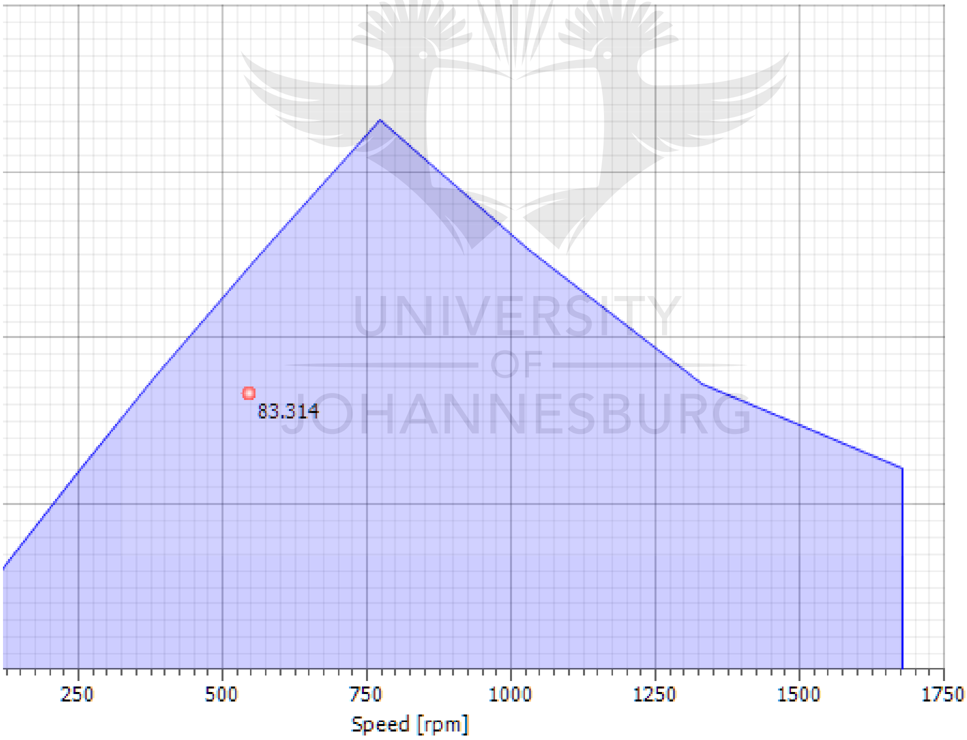
Power	P	46.124 kW
Torque	T	808.168 N m
Speed	n	545.000 r/min
Efficiency	η	0.980 ul
Required service life	L_h	17500.000 hr
Maximum chain elongation	ΔL_{max}	0.030 ul
Application	Heavy shocks with moderate overloads	
Environment	Soiled	
Lubrication	Recommended (see notes below)	

Power correction factors



Shock factor	Y	5.000 ul
Service factor	f_1	1.850 ul
Sprocket size factor	f_2	1.000 ul
Strands factor	f_3	2.500 ul
Lubrication factor	f_4	1.000 ul
Centre distance factor	f_5	0.981 ul
Ratio factor	f_6	0.936 ul
Service life factor	f_7	1.064 ul

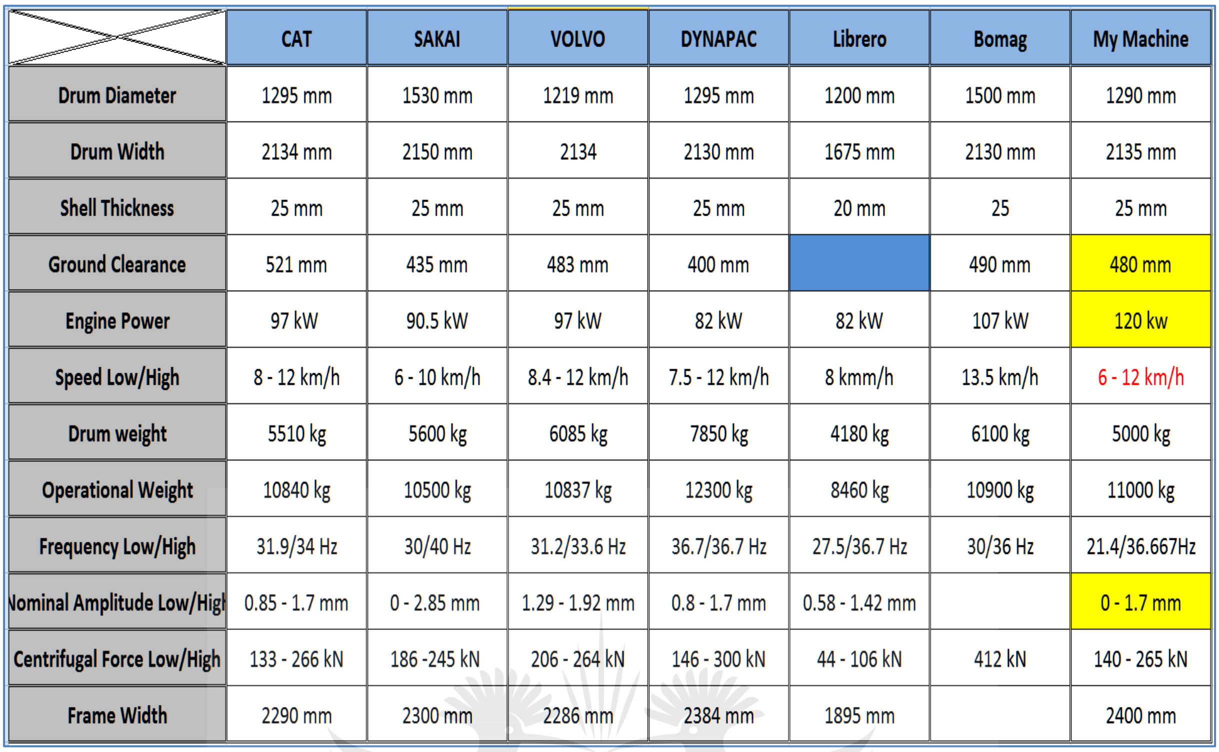
Chain power rating



Results

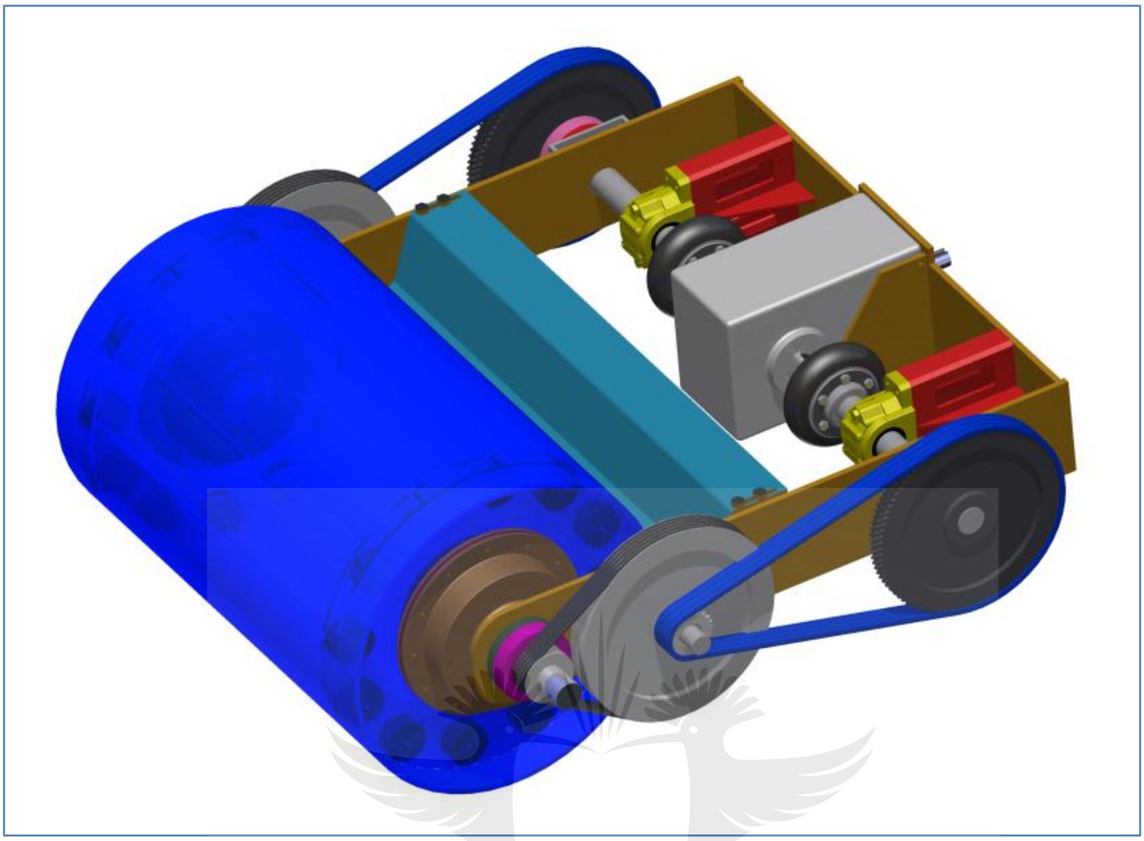
Chain Speed	v	6.654 mps
Effective pull	F_p	6932.005 N
Centrifugal force	F_C	487.000 N
Maximum tension in chain span	F_{Tmax}	7419.005 N
Static safety factor	$S_S > S_{Smin}$	33.697 ul > 7.000 ul
Dynamic safety factor	$S_D > S_{Dmin}$	6.739 ul > 5.000 ul
Bearing pressure	$p_B < p_0 * \lambda$	8.364 MPa
Permissible bearing pressure	p_0	17.609 MPa
Specific friction factor	λ	0.554 ul
Design power	$P_D < P_R$	83.314 kW
Chain power rating	P_R	121.073 kW
Chain service life for specified elongation	$t_h > L_h$	85990 hr
Chain link plates service life	$t_{hL} > L_h$	2777778 hr
Roller and bushing service life	$t_{hr} > L_h$	61661 hr

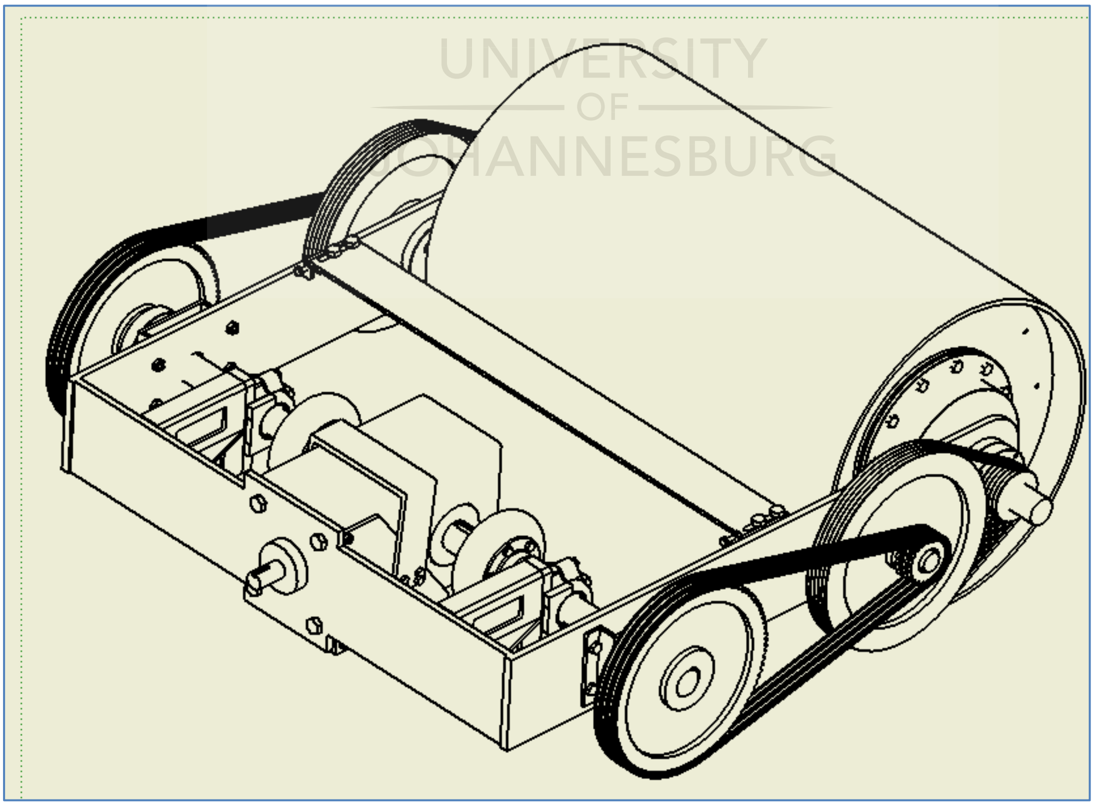
Appendix 10: Road roller specification summary table.



	CAT	SAKAI	VOLVO	DYNAPAC	Librero	Bomag	My Machine
Drum Diameter	1295 mm	1530 mm	1219 mm	1295 mm	1200 mm	1500 mm	1290 mm
Drum Width	2134 mm	2150 mm	2134	2130 mm	1675 mm	2130 mm	2135 mm
Shell Thickness	25 mm	25 mm	25 mm	25 mm	20 mm	25	25 mm
Ground Clearance	521 mm	435 mm	483 mm	400 mm		490 mm	480 mm
Engine Power	97 kW	90.5 kW	97 kW	82 kW	82 kW	107 kW	120 kw
Speed Low/High	8 - 12 km/h	6 - 10 km/h	8.4 - 12 km/h	7.5 - 12 km/h	8 kmm/h	13.5 km/h	6 - 12 km/h
Drum weight	5510 kg	5600 kg	6085 kg	7850 kg	4180 kg	6100 kg	5000 kg
Operational Weight	10840 kg	10500 kg	10837 kg	12300 kg	8460 kg	10900 kg	11000 kg
Frequency Low/High	31.9/34 Hz	30/40 Hz	31.2/33.6 Hz	36.7/36.7 Hz	27.5/36.7 Hz	30/36 Hz	21.4/36.667Hz
ominal Amplitude Low/Hig	0.85 - 1.7 mm	0 - 2.85 mm	1.29 - 1.92 mm	0.8 - 1.7 mm	0.58 - 1.42 mm		0 - 1.7 mm
Centrifugal Force Low/High	133 - 266 kN	186 -245 kN	206 - 264 kN	146 - 300 kN	44 - 106 kN	412 kN	140 - 265 kN
Frame Width	2290 mm	2300 mm	2286 mm	2384 mm	1895 mm		2400 mm

Appendix 11: Overall final machine design in 3D.





Appendix 12: Fenner friction wedge belt drive tables.

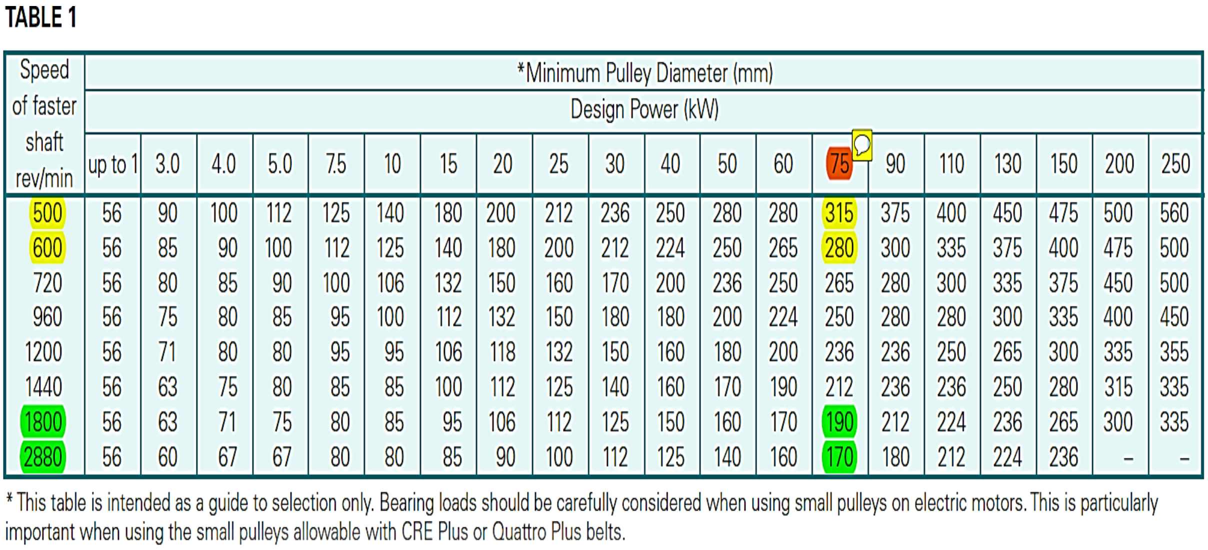


TABLE 1

Speed of faster shaft rev/min	*Minimum Pulley Diameter (mm)																			
	Design Power (kW)																			
	up to 1	3.0	4.0	5.0	7.5	10	15	20	25	30	40	50	60	75	90	110	130	150	200	250
500	56	90	100	112	125	140	180	200	212	236	250	280	280	315	375	400	450	475	500	560
600	56	85	90	100	112	125	140	180	200	212	224	250	265	280	300	335	375	400	475	500
720	56	80	85	90	100	106	132	150	160	170	200	236	250	265	280	300	335	375	450	500
960	56	75	80	85	95	100	112	132	150	180	180	200	224	250	280	280	300	335	400	450
1200	56	71	80	80	95	95	106	118	132	150	160	180	200	236	236	250	265	300	335	355
1440	56	63	75	80	85	85	100	112	125	140	160	170	190	212	236	236	250	280	315	335
1800	56	63	71	75	80	85	95	106	112	125	150	160	170	190	212	224	236	265	300	335
2880	56	60	67	67	80	80	85	90	100	112	125	140	160	170	180	212	224	236	–	–

* This table is intended as a guide to selection only. Bearing loads should be carefully considered when using small pulleys on electric motors. This is particularly important when using the small pulleys allowable with CRE Plus or Quattro Plus belts.

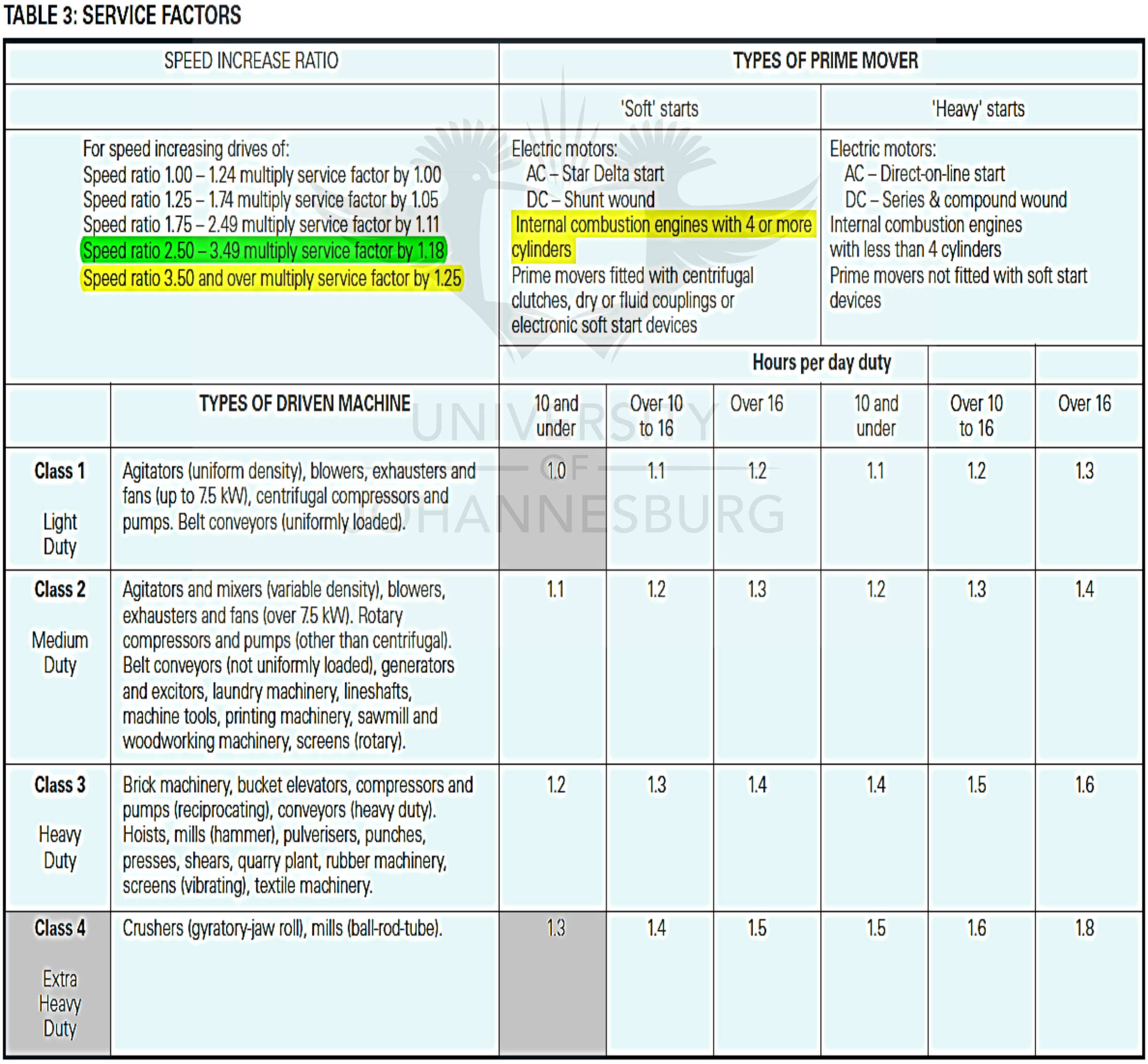


TABLE 3: SERVICE FACTORS

SPEED INCREASE RATIO		TYPES OF PRIME MOVER					
		'Soft' starts			'Heavy' starts		
For speed increasing drives of: Speed ratio 1.00 – 1.24 multiply service factor by 1.00 Speed ratio 1.25 – 1.74 multiply service factor by 1.05 Speed ratio 1.75 – 2.49 multiply service factor by 1.11 Speed ratio 2.50 – 3.49 multiply service factor by 1.18 Speed ratio 3.50 and over multiply service factor by 1.25		Electric motors: AC – Star Delta start DC – Shunt wound Internal combustion engines with 4 or more cylinders Prime movers fitted with centrifugal clutches, dry or fluid couplings or electronic soft start devices			Electric motors: AC – Direct-on-line start DC – Series & compound wound Internal combustion engines with less than 4 cylinders Prime movers not fitted with soft start devices		
		Hours per day duty					
	TYPES OF DRIVEN MACHINE	10 and under	Over 10 to 16	Over 16	10 and under	Over 10 to 16	Over 16
Class 1 Light Duty	Agitators (uniform density), blowers, exhausters and fans (up to 7.5 kW), centrifugal compressors and pumps. Belt conveyors (uniformly loaded).	1.0	1.1	1.2	1.1	1.2	1.3
Class 2 Medium Duty	Agitators and mixers (variable density), blowers, exhausters and fans (over 7.5 kW). Rotary compressors and pumps (other than centrifugal). Belt conveyors (not uniformly loaded), generators and excitors, laundry machinery, lineshafts, machine tools, printing machinery, sawmill and woodworking machinery, screens (rotary).	1.1	1.2	1.3	1.2	1.3	1.4
Class 3 Heavy Duty	Brick machinery, bucket elevators, compressors and pumps (reciprocating), conveyors (heavy duty). Hoists, mills (hammer), pulverisers, punches, presses, shears, quarry plant, rubber machinery, screens (vibrating), textile machinery.	1.2	1.3	1.4	1.4	1.5	1.6
Class 4 Extra Heavy Duty	Crushers (gyratory-jaw roll), mills (ball-rod-tube).	1.3	1.4	1.5	1.5	1.6	1.8

Centre Distance SPB, XPB, & QXPB Wedge Belt Drives

Combined Arc and Belt Length Correction Factor					0.80				0.85		0.90		0.95		1.00		1.05		1.10		
Speed Ratio	Pitch Diameter of Pulleys		Power per SPB Belt (kW)		BELT LENGTH									SPB & QXPB ONLY			SPB ONLY				Speed Ratio
	Driver	Driven	1440 rev/min	960 rev/min	1250	1400	1800	2000	2240	2500	2800	3150	3550	4000	4500	5000	5600	6300	7100	8000	
4.24	**236**	**1000**	17.86	12.82	–	–	–	–	–	–	–	–	700	953	1219	1480	1788	2145	2551	3005	4.24
4.44	**180**	**800**	12.31	8.80	–	–	–	–	–	–	542	740	955	1190	1447	1702	2006	2360	2763	3215	4.44
4.46	**224**	**1000**	16.71	11.97	–	–	–	–	–	–	–	–	707	960	1227	1488	1797	2154	2559	3014	4.46
4.50	**140**	**630**	8.11	5.82	–	–	–	–	449	595	756	938	1144	1373	1627	1879	2181	2533	2935	3386	4.50
4.71	**170**	**800**	11.28	8.06	–	–	–	–	–	–	548	747	962	1197	1454	1709	2014	2367	2770	3223	4.71
4.72	**212**	**1000**	15.53	11.11	–	–	–	–	–	–	–	–	714	968	1235	1496	1805	2162	2568	3022	4.72
5.00	**160**	**800**	10.23	7.32	–	–	–	–	–	–	554	753	968	1203	1461	1716	2021	2374	2778	3230	5.00
5.00	**200**	**1000**	14.34	10.25	–	–	–	–	–	–	–	–	722	976	1243	1504	1813	2171	2576	3031	5.00
5.26	**190**	**1000**	13.33	9.53	–	–	–	–	–	–	–	–	728	982	1250	1511	1820	2178	2584	3038	5.26
5.33	**150**	**800**	9.18	6.57	–	–	–	–	–	–	–	759	975	1210	1468	1723	2028	2382	2785	3238	5.33
5.56	**180**	**1000**	12.31	8.80	–	–	–	–	–	–	–	–	734	988	1256	1518	1827	2185	2591	3046	5.56
5.71	**140**	**800**	8.11	5.82	–	–	–	–	–	–	–	766	981	1217	1475	1730	2035	2389	2792	3245	5.71
5.88	**170**	**1000**	11.28	8.06	–	–	–	–	–	–	–	–	740	994	1263	1525	1834	2192	2598	3053	5.88

The above drives are based on the ISO belt length designations, other belt lengths and pulley combinations are available - consult your local Authorised Distributor.

Rev/min of faster shaft	RATED POWER (kW) PER BELT FOR SMALL PULLEY PITCH DIA. (mm)																			
	XPZ							XPA							XPB					
	56	60	63	67	71	75	80	80	85	90	95	100	106	112	112	118	125	132	140	150
100	0.10	0.11	0.13	0.14	0.16	0.17	0.19	0.22	0.25	0.28	0.32	0.35	0.39	0.43	0.51	0.57	0.64	0.71	0.79	0.89
200	0.18	0.21	0.23	0.26	0.29	0.32	0.35	0.38	0.45	0.52	0.58	0.65	0.72	0.80	0.93	1.04	1.17	1.30	1.46	1.64
300	0.25	0.29	0.33	0.37	0.41	0.45	0.51	0.54	0.63	0.73	0.82	0.92	1.03	1.14	1.30	1.46	1.66	1.85	2.07	2.34
400	0.32	0.37	0.42	0.47	0.53	0.58	0.65	0.68	0.80	0.93	1.05	1.17	1.32	1.47	1.64	1.86	2.11	2.37	2.65	3.01
500	0.38	0.45	0.50	0.57	0.64	0.71	0.79	0.81	0.96	1.11	1.27	1.42	1.60	1.78	1.97	2.24	2.55	2.86	3.21	3.65
600	0.44	0.53	0.59	0.67	0.75	0.83	0.93	0.93	1.11	1.29	1.48	1.66	1.87	2.08	2.28	2.60	2.97	3.33	3.75	4.26
700	0.50	0.60	0.67	0.76	0.85	0.95	1.06	1.05	1.26	1.47	1.68	1.88	2.13	2.38	2.58	2.95	3.37	3.79	4.27	4.86
720	**0.51**	**0.61**	**0.68**	**0.78**	**0.87**	**0.97**	**1.09**	**1.07**	**1.29**	**1.50**	**1.72**	**1.93**	**2.18**	**2.44**	**2.64**	**3.01**	**3.45**	**3.88**	**4.37**	**4.98**
800	0.56	0.67	0.75	0.85	0.96	1.06	1.19	1.16	1.40	1.64	1.87	2.11	2.39	2.67	2.87	3.28	3.76	4.23	4.77	5.44
900	0.61	0.73	0.82	0.94	1.06	1.17	1.32	1.27	1.53	1.80	2.06	2.32	2.64	2.95	3.14	3.60	4.13	4.66	5.26	6.00
960	**0.65**	**0.77**	**0.87**	**0.99**	**1.12**	**1.24**	**1.39**	**1.33**	**1.61**	**1.89**	**2.17**	**2.45**	**2.78**	**3.11**	**3.30**	**3.79**	**4.35**	**4.91**	**5.55**	**6.34**
1000	0.67	0.80	0.90	1.03	1.15	1.28	1.44	1.37	1.67	1.96	2.25	2.53	2.88	3.22	3.41	3.91	4.50	5.08	5.74	6.55
1100	0.72	0.86	0.97	1.11	1.25	1.39	1.56	1.47	1.79	2.11	2.43	2.74	3.11	3.49	3.67	4.21	4.85	5.48	6.20	7.09
1200	0.77	0.92	1.04	1.19	1.34	1.50	1.68	1.57	1.92	2.26	2.60	2.94	3.35	3.75	3.91	4.51	5.19	5.87	6.65	7.61
1300	0.82	0.99	1.11	1.27	1.44	1.60	1.80	1.66	2.04	2.40	2.77	3.14	3.57	4.01	4.15	4.79	5.53	6.26	7.09	8.12
1400	0.87	1.04	1.18	1.35	1.53	1.70	1.92	1.75	2.15	2.55	2.94	3.33	3.79	4.26	4.38	5.06	5.85	6.63	7.52	8.61
1440	**0.89**	**1.07**	**1.20**	**1.39**	**1.56**	**1.74**	**1.97**	**1.79**	**2.20**	**2.60**	**3.00**	**3.40**	**3.88**	**4.35**	**4.47**	**5.17**	**5.98**	**6.78**	**7.68**	**8.81**
1500	0.91	1.10	1.24	1.43	1.62	1.80	2.03	1.84	2.26	2.68	3.10	3.52	4.01	4.50	4.61	5.33	6.16	6.99	7.93	9.09
1600	0.96	1.16	1.31	1.51	1.71	1.90	2.15	1.92	2.37	2.82	3.26	3.70	4.22	4.74	4.82	5.59	6.47	7.34	8.34	9.56
1700	1.00	1.22	1.38	1.58	1.79	2.00	2.26	2.01	2.48	2.95	3.41	3.88	4.43	4.98	5.03	5.84	6.76	7.69	8.73	10.02
1800	1.05	1.27	1.44	1.66	1.88	2.10	2.37	2.08	2.58	3.07	3.57	4.05	4.63	5.21	5.23	6.08	7.05	8.02	9.11	10.46
1900	1.09	1.33	1.50	1.73	1.96	2.19	2.48	2.16	2.68	3.20	3.71	4.22	4.83	5.43	5.43	6.31	7.33	8.34	9.48	10.89
2000	1.13	1.38	1.56	1.81	2.05	2.29	2.58	2.23	2.78	3.32	3.86	4.39	5.02	5.65	5.61	6.53	7.60	8.65	9.84	11.30
2100	1.18	1.43	1.62	1.88	2.13	2.38	2.69	2.30	2.87	3.44	4.00	4.55	5.21	5.87	5.79	6.75	7.86	8.95	10.18	11.70
2200	1.22	1.48	1.68	1.95	2.21	2.47	2.79	2.37	2.96	3.55	4.13	4.71	5.40	6.08	5.96	6.96	8.11	9.24	10.52	12.09
2300	1.26	1.53	1.74	2.02	2.29	2.56	2.90	2.44	3.05	3.66	4.27	4.87	5.58	6.28	6.13	7.16	8.35	9.52	10.84	12.46
2400	1.29	1.58	1.80	2.08	2.37	2.65	3.00	2.50	3.14	3.77	4.40	5.02	5.76	6.48	6.28	7.35	8.58	9.79	11.15	12.82
2500	1.33	1.63	1.86	2.15	2.44	2.74	3.10	2.56	3.22	3.87	4.52	5.16	5.93	6.68	6.43	7.53	8.80	10.05	11.45	13.17
2600	1.37	1.68	1.91	2.22	2.52	2.82	3.20	2.61	3.30	3.97	4.64	5.31	6.09	6.87	6.58	7.71	9.01	10.29	11.73	13.49
2700	1.41	1.73	1.97	2.28	2.60	2.91	3.29	2.67	3.37	4.07	4.76	5.45	6.25	7.05	6.71	7.88	9.21	10.53	12.01	13.81
2800	1.44	1.77	2.02	2.35	2.67	2.99	3.39	2.72	3.45	4.17	4.88	5.58	6.41	7.23	6.84	8.03	9.41	10.76	12.27	14.11
2880	**1.47**	**1.81**	**2.06**	**2.40**	**2.73**	**3.06**	**3.47**	**2.76**	**3.50**	**4.24**	**4.97**	**5.68**	**6.53**	**7.37**	**6.93**	**8.15**	**9.55**	**10.93**	**12.47**	**14.34**
2900	1.48	1.82	2.07	2.41	2.74	3.07	3.48	2.77	3.52	4.26	4.99	5.71	6.56	7.41	6.96	8.18	9.59	10.97	12.51	14.39
3000	1.51	1.86	2.13	2.47	2.82	3.16	3.58	2.82	3.59	4.35	5.10	5.84	6.71	7.57	7.07	8.32	9.76	11.17	12.75	14.66
3100	1.55	1.91	2.18	2.53	2.89	3.24	3.67	2.86	3.65	4.43	5.20	5.96	6.86	7.74	7.17	8.45	9.92	11.36	12.97	14.91
3200	1.58	1.95	2.23	2.59	2.96	3.32	3.76	2.90	3.71	4.51	5.30	6.08	6.99	7.89	7.27	8.58	10.08	11.54	13.17	15.14
3300	1.61	1.99	2.28	2.65	3.03	3.39	3.85	2.94	3.77	4.59	5.40	6.19	7.13	8.05	7.35	8.69	10.22	11.71	13.36	15.36
3400	1.64	2.03	2.33	2.71	3.09	3.47	3.94	2.98	3.83	4.66	5.49	6.30	7.25	8.19	7.43	8.79	10.35	11.86	13.54	15.56
3500	1.67	2.08	2.37	2.77	3.16	3.55	4.02	3.01	3.88	4.74	5.58	6.40	7.38	8.33	7.50	8.89	10.47	12.00	13.70	15.74
3600	1.70	2.12	2.42	2.83	3.23	3.62	4.11	3.05	3.93	4.80	5.66	6.50	7.49	8.46	7.56	8.97	10.57	12.13	13.85	15.90
3700	1.73	2.15	2.47	2.88	3.29	3.69	4.19	3.07	3.98	4.87	5.74	6.60	7.61	8.59	7.62	9.05	10.67	12.24	13.98	16.05
3800	1.76	2.19	2.51	2.94	3.35	3.77	4.28	3.10	4.02	4.93	5.82	6.69	7.71	8.71	7.66	9.11	10.76	12.35	14.09	16.18
3900	1.79	2.23	2.56	2.99	3.42	3.84	4.36	3.12	4.06	4.98	5.89	6.77	7.81	8.83	7.70	9.17	10.83	12.43	14.20	16.28
4000	1.82	2.27	2.60	3.04	3.48	3.91	4.44	3.14	4.10	5.04	5.96	6.86	7.91	8.94	7.73	9.21	10.89	12.51	14.28	16.37
4100	1.85	2.30	2.65	3.09	3.54	3.98	4.52	3.16	4.13	5.09	6.02	6.93	8.00	9.04						
4200	1.87	2.34	2.69	3.15	3.60	4.04	4.59	3.18	4.16	5.13	6.08	7.00	8.09	9.13						
4300	1.90	2.38	2.73	3.20	3.66	4.11	4.67	3.19	4.19	5.17	6.13	7.07	8.16	9.22						
4400	1.92	2.41	2.77	3.24	3.71	4.17	4.74	3.20	4.22	5.21	6.18	7.13	8.24	9.30						
4500	1.95	2.44	2.81	3.29	3.77	4.24	4.81	3.20	4.24	5.25	6.23	7.19	8.30	9.38						
4600	1.97	2.48	2.85	3.34	3.82	4.30	4.89	3.21	4.25	5.28	6.27	7.24	8.36	9.45						
4700	1.99	2.51	2.89	3.39	3.88	4.36	4.96	3.21	4.27	5.30	6.31	7.28	8.42	9.51						
4800	2.02	2.54	2.92	3.43	3.93	4.42	5.02	3.21	4.28	5.32	6.34	7.33	8.47	9.56						
4900	2.04	2.57	2.96	3.48	3.98	4.48	5.09	3.20	4.29	5.34	6.37	7.36	8.51	9.61						
5000	2.06	2.60	3.00	3.52	4.03	4.54	5.16	3.19	4.29	5.36	6.39	7.39	8.54	9.65						
5100	2.08	2.63	3.03	3.56	4.08	4.59	5.22													
5200	2.10	2.66	3.06	3.60	4.13	4.65	5.28													
5300	2.12	2.68	3.10	3.64	4.18	4.70	5.34													
5400	2.14	2.71	3.13	3.68	4.22	4.76	5.40													
5500	2.16	2.73	3.16	3.72	4.27	4.81	5.46													
5600	2.17	2.76	3.19	3.76	4.31	4.86	5.52													
5700	2.19	2.78	3.22	3.79	4.36	4.90	5.57													
5800	2.21	2.81	3.25	3.83	4.40	4.95	5.63													
5900	2.22	2.83	3.28	3.86	4.44	5.00	5.68													
6000	2.24	2.85	3.30	3.90	4.48	5.04	5.73													

Power Ratings
CRE Plus Wedge Belts

Use the additional power per belt for speed ratio from pages 54–56.

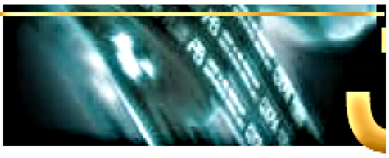
Power Ratings - SPB Wedge Belts

Rev/min of faster shaft	RATED POWER (kW) PER BELT FOR SMALL PULLEY PITCH DIA (mm)													Belt Speed (m/s)
	140	**150**	**160**	**170**	**180**	**190**	**200**	**212**	**224**	**236**	**250**	**280**	**315**	
100	0.73	0.82	0.92	1.01	1.10	1.20	1.29	1.40	1.51	1.62	1.74	2.01	2.33	
200	1.33	1.51	1.69	1.87	2.05	2.22	2.40	2.61	2.82	3.02	3.26	3.78	4.37	
300	1.89	2.15	2.41	2.67	2.93	3.18	3.44	3.74	4.04	4.35	4.70	5.44	6.30	
400	2.42	2.76	3.09	3.43	3.77	4.10	4.43	4.83	5.22	5.61	6.07	7.04	8.15	
500	2.92	3.33	3.75	4.16	4.57	4.98	5.39	5.87	6.36	6.84	7.39	8.58	9.94	
600	3.40	3.89	4.38	4.87	5.35	5.83	6.31	6.89	7.45	8.02	8.67	10.06	11.66	
700	3.86	4.43	4.99	5.55	6.11	6.66	7.21	7.87	8.52	9.17	9.92	11.50	13.32	10
720	**3.95**	**4.53**	**5.11**	**5.69**	**6.26**	**6.82**	**7.39**	**8.06**	**8.73**	**9.39**	**10.16**	**11.79**	**13.65**	
800	4.31	4.95	5.59	6.22	6.84	7.47	8.08	8.82	9.55	10.28	11.12	12.90	14.93	
900	4.75	5.46	6.16	6.86	7.56	8.25	8.93	9.75	10.56	11.36	12.29	14.25	16.47	
960	**5.00**	**5.75**	**6.50**	**7.24**	**7.98**	**8.71**	**9.43**	**10.29**	**11.15**	**11.99**	**12.97**	**15.03**	**17.37**	
1000	5.17	5.95	6.72	7.49	8.25	9.01	9.76	10.65	11.53	12.41	13.42	15.55	17.96	
1100	5.58	6.42	7.27	8.10	8.93	9.75	10.56	11.52	12.48	13.43	14.52	16.80	19.39	
1200	5.97	6.89	7.79	8.69	9.58	10.46	11.34	12.37	13.40	14.41	15.57	18.01	20.75	
1300	6.36	7.34	8.31	9.27	10.22	11.16	12.09	13.19	14.28	15.36	16.59	19.17	22.05	20
1400	6.73	7.77	8.81	9.83	10.84	11.84	12.82	13.99	15.14	16.27	17.57	20.28	23.28	
1440	**6.88**	**7.95**	**9.00**	**10.05**	**11.08**	**12.10**	**13.11**	**14.30**	**15.47**	**16.63**	**17.96**	**20.70**	**23.75**	
1500	7.09	8.20	9.29	10.37	11.44	12.49	13.53	14.76	15.97	17.15	18.51	21.33	24.43	
1600	7.44	8.61	9.76	10.90	12.02	13.12	14.21	15.50	16.76	18.00	19.41	22.33	25.51	
1700	7.78	9.01	10.21	11.40	12.58	13.73	14.87	16.21	17.52	18.81	20.27	23.27	26.51	
1800	8.11	9.39	10.65	11.90	13.12	14.32	15.50	16.89	18.25	19.58	21.08	24.15	27.43	
1900	8.43	9.76	11.08	12.37	13.64	14.88	16.11	17.54	18.94	20.31	21.85	24.97	28.27	30
2000	8.73	10.12	11.48	12.82	14.14	15.43	16.69	18.16	19.60	20.99	22.57	25.72	29.01	
2100	9.02	10.46	11.88	13.26	14.62	15.94	17.24	18.75	20.22	21.64	23.23	26.41	29.67	
2200	9.31	10.79	12.25	13.68	15.07	16.44	17.76	19.31	20.80	22.24	23.85	27.03	30.22	
2300	9.57	11.11	12.61	14.08	15.51	16.90	18.26	19.83	21.35	22.80	24.42	27.57	30.69	
2400	9.83	11.41	12.95	14.46	15.92	17.34	18.72	20.32	21.85	23.31	24.93	28.05	31.04	
2500	10.08	11.70	13.28	14.82	16.31	17.76	19.16	20.77	22.31	23.78	25.38	28.44	–	40
2600	10.31	11.97	13.59	15.16	16.68	18.14	19.56	21.19	22.73	24.19	25.78	28.76	–	
2700	10.53	12.23	13.88	15.47	17.02	18.50	19.93	21.56	23.11	24.56	26.12	28.99	–	
2800	10.73	12.47	14.15	15.77	17.33	18.83	20.27	21.90	23.44	24.87	26.40	–	–	
2880	**10.89**	**12.65**	**14.35**	**15.99**	**17.57**	**19.07**	**20.51**	**22.14**	**23.67**	**25.08**	**26.57**	–	–	
2900	10.93	12.69	14.40	16.04	17.62	19.13	20.57	22.20	23.72	25.12	26.61	–	–	
3000	11.10	12.90	14.63	16.30	17.89	19.40	20.84	22.46	23.96	25.33	26.76	–	–	

Rev/min of faster shaft	ADDITIONAL POWER (kW) PER BELT FOR SPEED RATIO									
	1.00 to 1.01	1.02 to 1.05	1.06 to 1.11	1.12 to 1.19	1.19 to 1.26	1.27 to 1.38	1.39 to 1.57	1.58 to 1.94	1.95 to 3.38	3.39 and over
100	0.00	0.01	0.02	0.04	0.04	0.06	0.07	0.07	0.08	0.08
200	0.00	0.01	0.04	0.07	0.09	0.11	0.13	0.15	0.16	0.17
300	0.00	0.02	0.06	0.10	0.14	0.17	0.20	0.22	0.24	0.25
400	0.00	0.03	0.07	0.13	0.19	0.22	0.26	0.29	0.32	0.34
500	0.00	0.04	0.09	0.17	0.23	0.28	0.33	0.37	0.40	0.43
600	0.00	0.04	0.12	0.20	0.28	0.34	0.40	0.45	0.48	0.51
700	0.00	0.05	0.13	0.24	0.33	0.39	0.46	0.52	0.57	0.59
720	**0.00**	**0.05**	**0.14**	**0.25**	**0.33**	**0.41**	**0.48**	**0.54**	**0.59**	**0.62**
800	0.00	0.06	0.16	0.28	0.37	0.45	0.53	0.60	0.65	0.69
900	0.00	0.07	0.18	0.31	0.42	0.51	0.60	0.66	0.72	0.77
960	**0.00**	**0.07**	**0.19**	**0.32**	**0.44**	**0.54**	**0.62**	**0.70**	**0.77**	**0.81**
1000	0.00	0.07	0.19	0.34	0.46	0.56	0.66	0.74	0.81	0.86
1100	0.00	0.08	0.22	0.37	0.51	0.62	0.72	0.81	0.89	0.94
1200	0.00	0.09	0.23	0.41	0.56	0.68	0.79	0.89	0.97	1.03
1300	0.00	0.09	0.25	0.44	0.60	0.73	0.86	0.96	1.05	1.11
1400	0.00	0.10	0.28	0.49	0.65	0.79	0.93	1.04	1.13	1.20
1440	**0.00**	**0.10**	**0.28**	**0.48**	**0.66**	**0.79**	**0.94**	**1.06**	**1.15**	**1.21**
1500	0.00	0.10	0.29	0.51	0.69	0.84	0.99	1.11	1.21	1.28
1600	0.00	0.11	0.31	0.54	0.75	0.90	1.05	1.19	1.29	1.37
1700	0.00	0.12	0.34	0.58	0.79	0.95	1.12	1.26	1.37	1.45
1800	0.00	0.13	0.35	0.61	0.84	1.01	1.19	1.34	1.45	1.54
1900	0.00	0.13	0.37	0.65	0.88	1.07	1.25	1.41	1.54	1.63
2000	0.00	0.14	0.39	0.68	0.93	1.13	1.32	1.48	1.62	1.71
2100	0.00	0.15	0.41	0.72	0.98	1.18	1.39	1.56	1.69	1.79
2200	0.00	0.16	0.43	0.75	1.02	1.24	1.45	1.63	1.78	1.88
2300	0.00	0.16	0.45	0.78	1.07	1.29	1.51	1.71	1.86	1.97
2400	0.00	0.17	0.47	0.82	1.11	1.35	1.58	1.78	1.94	2.05
2500	0.00	0.18	0.49	0.85	1.16	1.41	1.65	1.86	2.02	2.14
2600	0.00	0.19	0.51	0.89	1.21	1.46	1.72	1.92	2.10	2.22
2700	0.00	0.19	0.53	0.92	1.25	1.52	1.78	1.99	2.18	2.31
2800	0.00	0.20	0.54	0.95	1.29	1.57	1.84	2.07	2.26	2.39
2880	**0.00**	**0.20**	**0.56**	**0.97**	**1.32**	**1.60**	**1.88**	**2.11**	**2.31**	**2.44**
2900	0.00	0.21	0.57	0.99	1.34	1.63	1.91	2.15	2.34	2.48
3000	0.00	0.22	0.59	1.02	1.39	1.69	1.98	2.23	2.42	2.57

NOTE: Only Fenner brand pulleys should be used where belt speed falls between 30 and 40 m/s.

Power Ratings - QXPB Quattro Plus Wedge Belts



Rev/min of faster shaft	RATED POWER (kW) PER BELT FOR SMALL PULLEY PITCH DIA (mm)																			Belt Speed (m/s)
	112	**118**	**125**	**132**	**140**	**150**	**160**	**170**	**180**	**190**	**200**	**212**	**224**	**236**	**250**	**265**	**280**	**300**	**315**	
100	0.57	0.63	0.70	0.77	0.85	0.95	1.05	1.16	1.26	1.36	1.46	1.58	1.69	1.81	1.95	2.10	2.25	2.44	2.59	
200	1.05	1.17	1.31	1.45	1.60	1.80	1.99	2.19	2.38	2.57	2.77	3.00	3.23	3.45	3.72	4.00	4.29	4.66	4.94	
300	1.50	1.68	1.88	2.08	2.31	2.60	2.89	3.17	3.45	3.74	4.02	4.36	4.69	5.03	5.42	5.83	6.25	6.80	7.21	
400	1.94	2.17	2.43	2.70	3.00	3.37	3.75	4.12	4.49	4.86	5.23	5.67	6.11	6.55	7.06	7.61	8.15	8.86	9.40	
500	2.35	2.64	2.96	3.29	3.66	4.13	4.59	5.05	5.51	5.96	6.42	6.96	7.50	8.04	8.67	9.33	10.00	10.88	11.54	
600	2.76	3.09	3.48	3.87	4.31	4.86	5.41	5.96	6.50	7.04	7.58	8.22	8.86	9.50	10.24	11.03	11.81	12.85	13.62	
700	3.15	3.54	3.99	4.44	4.95	5.58	6.21	6.84	7.47	8.09	8.71	9.45	10.19	10.92	11.77	12.68	13.58	14.77	15.66	
720	**3.23**	**3.63**	**4.09**	**4.55**	**5.07**	**5.72**	**6.37**	**7.02**	**7.66**	**8.30**	**8.94**	**9.70**	**10.45**	**11.20**	**12.08**	**13.01**	**13.93**	**15.15**	**16.06**	
800	3.54	3.98	4.49	4.99	5.57	6.29	7.00	7.71	8.42	9.12	9.83	10.66	11.49	12.32	13.28	14.30	15.31	16.65	17.65	
900	3.91	4.40	4.97	5.54	6.18	6.98	7.78	8.57	9.36	10.14	10.92	11.85	12.77	13.69	14.75	15.88	17.00	18.48	19.58	
960	**4.14**	**4.66**	**5.26**	**5.86**	**6.54**	**7.39**	**8.23**	**9.07**	**9.91**	**10.74**	**11.56**	**12.55**	**13.53**	**14.50**	**15.62**	**16.81**	**18.00**	**19.56**	**20.72**	
1000	4.28	4.82	5.45	6.07	6.78	7.66	8.54	9.41	10.27	11.13	11.99	13.01	14.02	15.03	16.19	17.43	18.65	20.27	21.46	
1100	4.64	5.23	5.91	6.59	7.37	8.33	9.28	10.23	11.18	12.11	13.04	14.15	15.25	16.34	17.60	18.94	20.26	22.00	23.29	
1200	5.00	5.63	6.37	7.11	7.95	8.99	10.02	11.04	12.06	13.07	14.07	15.27	16.45	17.62	18.98	20.41	21.83	23.69	25.06	
1300	5.34	6.03	6.82	7.61	8.51	9.63	10.74	11.84	12.93	14.01	15.09	16.36	17.63	18.88	20.32	21.85	23.35	25.33	26.78	
1400	5.68	6.41	7.27	8.11	9.07	10.26	11.45	12.62	13.78	14.93	16.08	17.43	18.78	20.10	21.63	23.24	24.83	26.91	28.43	
1440	**5.82**	**6.57**	**7.44**	**8.31**	**9.29**	**10.52**	**11.73**	**12.93**	**14.12**	**15.30**	**16.47**	**17.86**	**19.23**	**20.58**	**22.15**	**23.79**	**25.41**	**27.52**	**29.07**	
1500	6.01	6.79	7.70	8.60	9.62	10.89	12.14	13.39	14.62	15.84	17.05	18.48	19.90	21.30	22.91	24.60	26.26	28.43	30.02	
1600	6.34	7.17	8.13	9.08	10.16	11.50	12.82	14.14	15.44	16.72	18.00	19.50	20.99	22.46	24.14	25.91	27.64	29.89	31.54	
1700	6.66	7.53	8.54	9.55	10.69	12.10	13.49	14.87	16.24	17.59	18.92	20.50	22.06	23.59	25.34	27.18	28.97	31.30	32.99	
1800	6.97	7.89	8.96	10.01	11.21	12.69	14.15	15.60	17.02	18.43	19.83	21.47	23.09	24.69	26.50	28.40	30.25	32.63	34.36	
1900	7.28	8.24	9.36	10.46	11.72	13.26	14.79	16.30	17.79	19.26	20.71	22.42	24.10	25.74	27.61	29.57	31.47	33.90	35.66	
2000	7.58	8.59	9.75	10.91	12.21	13.83	15.42	16.99	18.54	20.06	21.57	23.33	25.07	26.76	28.69	30.69	32.63	35.10	36.88	
2100	7.88	8.93	10.14	11.34	12.70	14.38	16.03	17.66	19.27	20.85	22.40	24.22	26.01	27.75	29.72	31.76	33.73	36.23	38.01	
2200	8.17	9.26	10.52	11.77	13.18	14.92	16.63	18.32	19.98	21.61	23.21	25.08	26.91	28.69	30.70	32.78	34.77	37.28	39.06	
2300	8.45	9.58	10.89	12.18	13.65	15.45	17.22	18.96	20.67	22.35	23.99	25.91	27.78	29.60	31.64	33.74	35.75	38.26	40.02	
2400	8.72	9.90	11.25	12.59	14.10	15.96	17.79	19.58	21.34	23.06	24.74	26.71	28.61	30.46	32.53	34.65	36.65	39.15	40.88	40
2500	8.99	10.21	11.61	12.99	14.55	16.47	18.35	20.19	21.99	23.75	25.47	27.47	29.41	31.28	33.37	35.49	37.49	39.95	41.64	
2600	9.26	10.51	11.95	13.38	14.98	16.95	18.89	20.77	22.62	24.42	26.17	28.21	30.17	32.05	34.15	36.27	38.26	40.67	42.31	
2700	9.51	10.80	12.29	13.75	15.41	17.43	19.41	21.34	23.22	25.06	26.84	28.90	30.89	32.78	34.88	36.99	38.95	41.30	42.87	
2800	9.76	11.09	12.62	14.12	15.82	17.89	19.92	21.89	23.81	25.67	27.48	29.57	31.56	33.46	35.56	37.64	39.56	41.84	43.32	
2880	**9.96**	**11.31**	**12.87**	**14.41**	**16.14**	**18.25**	**20.31**	**22.31**	**24.26**	**26.14**	**27.97**	**30.07**	**32.07**	**33.97**	**36.06**	**38.12**	**39.99**	**42.19**	**43.60**	
2900	10.00	11.37	12.94	14.48	16.22	18.34	20.41	22.42	24.37	26.26	28.09	30.19	32.20	34.10	36.18	38.23	40.10	42.27	43.66	
3000	10.24	11.64	13.25	14.83	16.60	18.77	20.88	22.92	24.91	26.82	28.67	30.78	32.79	34.68	36.73	38.75	40.55	42.61	43.89	
3100	10.47	11.90	13.55	15.16	16.98	19.19	21.33	23.41	25.42	27.35	29.21	31.33	33.34	35.21	37.23	39.19	40.92	42.85	44.00	
3200	10.69	12.15	13.84	15.49	17.34	19.59	21.77	23.88	25.91	27.86	29.72	31.85	33.84	35.69	37.67	39.56	41.20	42.98		
3300	10.90	12.40	14.12	15.81	17.69	19.98	22.19	24.32	26.37	28.33	30.20	32.32	34.29	36.11	38.04	39.85	41.39	43.00		
3400	11.11	12.64	14.39	16.11	18.02	20.35	22.59	24.74	26.81	28.77	30.64	32.75	34.70	36.48	38.34	40.07	41.49			
3500	11.31	12.87	14.65	16.40	18.35	20.70	22.97	25.14	27.22	29.19	31.05	33.14	35.05	36.79	38.58	40.20	41.50			
3600	11.50	13.09	14.91	16.68	18.66	21.04	23.33	25.52	27.60	29.57	31.42	33.48	35.36	37.04	38.75	40.25				
3700	11.68	13.30	15.15	16.95	18.95	21.36	23.67	25.87	27.95	29.92	31.75	33.78	35.61	37.23	38.84					
3800	11.86	13.50	15.38	17.21	19.23	21.67	23.99	26.20	28.28	30.23	32.05	34.04	35.81	37.36	38.87					
3900	12.03	13.70	15.60	17.45	19.50	21.96	24.29	26.50	28.58	30.51	32.30	34.25	35.96	37.43						
4000	12.19	13.88	15.81	17.68	19.75	22.23	24.57	26.78	28.84	30.76	32.52	34.41	36.05							
4100	12.34	14.06	16.01	17.90	19.99	22.48	24.83	27.03	29.08	30.97	32.69	34.52	36.08							
4200	12.48	14.22	16.20	18.11	20.21	22.71	25.06	27.26	29.29	31.14	32.82	34.58								
4300	12.62	14.38	16.37	18.30	20.41	22.92	25.27	27.45	29.46	31.28	32.91									
4400	12.74	14.52	16.54	18.48	20.60	23.12	25.46	27.62	29.60	31.38	32.95									
4500	12.86	14.66	16.69	18.64	20.78	23.29	25.62	27.77	29.71	31.44										
4600	12.97	14.78	16.83	18.79	20.93	23.45	25.76	27.88	29.78	31.46										
4700	13.07	14.90	16.96	18.93	21.07	23.58	25.88	27.97	29.82											
4800	13.16	15.00	17.07	19.05	21.20	23.69	25.97	28.02	29.83											
4900	13.24	15.10	17.18	19.16	21.30	23.78	26.04	28.05												
5000	13.31	15.18	17.27	19.25	21.39	23.86	26.08													

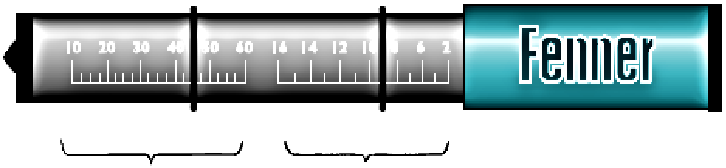
NOTES: Only Fenner pulleys should be used for belt speeds between 30 and 40 m/s.
Belt speeds over 40 m/sec require special pulleys

1400	5.68	6.41	7.27	8.11	9.07	10.26	11.45	12.62	13.78	14.93	16.08	17.43	18.78	20.10	21.63	23.24	24.83	26.91	28.43	40
1440	**5.82**	**6.57**	**7.44**	**8.31**	**9.29**	**10.52**	**11.73**	**12.93**	**14.12**	**15.30**	**16.47**	**17.86**	**19.23**	**20.58**	**22.15**	**23.79**	**25.41**	**27.52**	**29.07**	
1500	6.01	6.79	7.70	8.60	9.62	10.89	12.14	13.39	14.62	15.84	17.05	18.48	19.90	21.30	22.91	24.60	26.26	28.43	30.02	
1600	6.34	7.17	8.13	9.08	10.16	11.50	12.82	14.14	15.44	16.72	18.00	19.50	20.99	22.46	24.14	25.91	27.64	29.89	31.54	
1700	6.66	7.53	8.54	9.55	10.69	12.10	13.49	14.87	16.24	17.59	18.92	20.50	22.06	23.59	25.34	27.18	28.97	31.30	32.99	
1800	6.97	7.89	8.96	10.01	11.21	12.69	14.15	15.60	17.02	18.43	19.83	21.47	23.09	24.68	26.50	28.40	30.25	32.63	34.36	
1900	7.28	8.24	9.36	10.46	11.72	13.26	14.79	16.30	17.79	19.26	20.71	22.42	24.10	25.74	27.61	29.57	31.47	33.90	35.66	
2000	7.58	8.59	9.75	10.91	12.21	13.83	15.42	16.99	18.54	20.06	21.57	23.33	25.07	26.76	28.69	30.69	32.63	35.10	36.88	
2100	7.88	8.93	10.14	11.34	12.70	14.38	16.03	17.66	19.27	20.85	22.40	24.22	26.01	27.75	29.72	31.76	33.73	36.23	38.01	
2200	8.17	9.26	10.52	11.77	13.18	14.92	16.63	18.32	19.98	21.61	23.21	25.08	26.91	28.69	30.70	32.78	34.77	37.28	39.06	
2300	8.45	9.58	10.89	12.18	13.65	15.45	17.22	18.96	20.67	22.35	23.99	25.91	27.78	29.60	31.64	33.74	35.75	38.26	40.02	
2400	8.72	9.90	11.25	12.59	14.10	15.96	17.79	19.58	21.34	23.06	24.74	26.71	28.61	30.46	32.53	34.65	36.65	39.15	40.88	
2500	8.99	10.21	11.61	12.99	14.55	16.47	18.35	20.19	21.99	23.75	25.47	27.47	29.41	31.28	33.37	35.49	37.49	39.95	41.64	
2600	9.26	10.51	11.95	13.38	14.98	16.95	18.89	20.77	22.62	24.42	26.17	28.21	30.17	32.05	34.15	36.27	38.26	40.67	42.31	
2700	9.51	10.80	12.29	13.75	15.41	17.43	19.41	21.34	23.22	25.06	26.84	28.90	30.89	32.78	34.88	36.99	38.95	41.30	42.87	
2800	9.76	11.09	12.62	14.12	15.82	17.89	19.92	21.89	23.81	25.67	27.48	29.57	31.56	33.46	35.56	37.64	39.56	41.84	43.32	
2880	**9.96**	**11.31**	**12.87**	**14.41**	**16.14**	**18.25**	**20.31**	**22.31**	**24.26**	**26.14**	**27.97**	**30.07**	**32.07**	**33.97**	**36.06**	**38.12**	**39.99**	**42.19**	**43.60**	
2900	10.00	11.37	12.94	14.48	16.22	18.34	20.41	22.42	24.37	26.26	28.09	30.19	32.20	34.10	36.18	38.23	40.10	42.27	43.66	
3000	10.24	11.64	13.25	14.83	16.60	18.77	20.88	22.92	24.91	26.82	28.67	30.78	32.79	34.68	36.73	38.75	40.55	42.61	43.89	
3100	10.47	11.90	13.55	15.16	16.98	19.19	21.33	23.41	25.42	27.35	29.21	31.33	33.34	35.21	37.23	39.19	40.92	42.85	44.00	
3200	10.69	12.15	13.84	15.49	17.34	19.59	21.77	23.88	25.91	27.86	29.72	31.85	33.84	35.69	37.67	39.56	41.20	42.98		
3300	10.90	12.40	14.12	15.81	17.69	19.98	22.19	24.32	26.37	28.33	30.20	32.32	34.29	36.11	38.04	39.85	41.39	43.00		
3400	11.11	12.64	14.39	16.11	18.02	20.35	22.59	24.74	26.81	28.77	30.64	32.75	34.70	36.48	38.34	40.07	41.49			
3500	11.31	12.87	14.65	16.40	18.35	20.70	22.97	25.14	27.22	29.19	31.05	33.14	35.05	36.79	38.58	40.20	41.50			
3600	11.50	13.09	14.91	16.68	18.66	21.04	23.33	25.52	27.60	29.57	31.42	33.48	35.36	37.04	38.75	40.25				
3700	11.68	13.30	15.15	16.95	18.95	21.36	23.67	25.87	27.95	29.92	31.75	33.78	35.61	37.23	38.84					
3800	11.86	13.50	15.38	17.21	19.23	21.67	23.99	26.20	28.28	30.23	32.05	34.04	35.81	37.36	38.87					
3900	12.03	13.70	15.60	17.45	19.50	21.96	24.29	26.50	28.58	30.51	32.30	34.25	35.96	37.43						
4000	12.19	13.88	15.81	17.68	19.75	22.23	24.57	26.78	28.84	30.76	32.52	34.41	36.05							
4100	12.34	14.06	16.01	17.90	19.99	22.48	24.83	27.03	29.08	30.97	32.69	34.52	36.08							
4200	12.48	14.22	16.20	18.11	20.21	22.71	25.06	27.26	29.29	31.14	32.82	34.58								
4300	12.62	14.38	16.37	18.30	20.41	22.92	25.27	27.45	29.46	31.28	32.91									
4400	12.74	14.52	16.54	18.48	20.60	23.12	25.46	27.62	29.60	31.38	32.95									
4500	12.86	14.66	16.69	18.64	20.78	23.29	25.62	27.77	29.71	31.44										
4600	12.97	14.78	16.83	18.79	20.93	23.45	25.76	27.88	29.78	31.46										
4700	13.07	14.90	16.96	18.93	21.07	23.58	25.88	27.97	29.82											
4800	13.16	15.00	17.07	19.05	21.20	23.69	25.97	28.02	29.83											
4900	13.24	15.10	17.18	19.16	21.30	23.78	26.04	28.05												
5000	13.31	15.18	17.27	19.25	21.39	23.86	26.08													

NOTES: Only Fenner pulleys should be used for belt speeds between 30 and 40 m/s.
Belt speeds over 40 m/sec require special pulleys
Use the additional power per belt for speed ratio from page 54

The high performance and efficiency of Fenner Precision Built belts requires correct tension. We recommend using the Fenner Belt Tension Indicator.

FENNER BELTTENSION INDICATOR



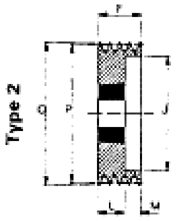
Deflection in mm

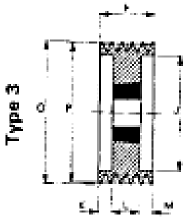
Setting force in kgf

Product Code 230A0000

Appendix 13: Pulley selection.

Taper Lock Pulleys for B, SPB, XPB, QXPB & USPB Belts

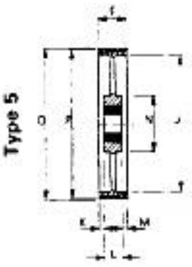


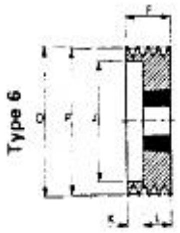


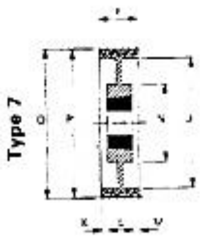
Catalogue Code	Pitch Dia (P)	No. of Grooves	Bush No.	Max. Bore		Pulley Type	F	J	K	L	M	N	Outside Dia (O)
				Metric	Inch								
031B0162	***112***	*2*	*2012*	*50*	*2*	*2*	*44*	*72*	*25*	*19.0*	–	–	*119*
031B0163	***112***	*3*	*2012*	*50*	*2*	*2*	*63*	*72*	–	*25*	*37.0*	–	*119*
031B0172	***118***	*2*	*2012*	*50*	*2*	*2*	*44*	*78*	–	*25*	*19.0*	–	*125*
031B0173	***118***	*3*	*2012*	*50*	*2*	*2*	*63*	*78*	–	*25*	*37.0*	–	*125*
031B0182	***125***	*2*	*2012*	*50*	*2*	*2*	*44*	*82*	–	*32*	*12.0*	–	*132*
031B0183	***125***	*3*	*2012*	*50*	*2*	*2*	*63*	*89*	–	*32*	*31.0*	–	*132*
031B0184	***125***	*4*	*2012*	*50*	*2*	*3*	*82*	*82*	*25.0*	*32*	*25.0*	–	*132*
*031B0185**	***125***	*5*	*2012*	*50*	*2*	*6*	*101*	*87*	*69.0*	*32*	–	–	*132*
031B0192	***132***	*2*	*2012*	*50*	*2*	*2*	*44*	*89*	–	*32*	*12.0*	–	*139*
031B0193	***132***	*3*	*2012*	*50*	*2*	*2*	*63*	*89*	–	*32*	*31.0*	–	*139*
031B0194	***132***	*4*	*2012*	*50*	*2*	*3*	*82*	*89*	*25.0*	*32*	*25.0*	–	*139*
*031B0195**	***132***	*5*	*2517*	*60*	*2½*	*6*	*101*	*94*	*56.0*	*45*	–	–	*139*
031B0202	**140**	2	2012	50	2	2	44	97	–	32	12.0	–	147
031B0203	**140**	3	2012	50	2	2	63	97	–	32	31.0	–	147
031B0204	**140**	4	2517	60	2½	3	82	100	18.5	45	18.5	–	147
031B0205	**140**	5	2517	60	2½	3	101	97	28.0	45	28.0	–	147
031B0206	**140**	6	2517	60	2½	3	120	100	37.5	45	37.5	–	147
031B0212	**150**	2	2012	50	2	2	44	107	–	32	12.0	–	157
031B0213	**150**	3	2517	60	2½	2	63	107	–	45	18.0	–	157
031B0214	**150**	4	2517	60	2½	3	82	107	18.5	45	18.5	–	157
031B0215	**150**	5	2517	60	2½	3	101	107	28.0	45	28.0	–	157
031B0216	**150**	6	2517	60	2½	3	120	107	37.5	45	37.5	–	157
031B0222	**160**	2	2012	50	2	2	44	117	–	32	12.0	–	167
031B0223	**160**	3	2517	60	2½	2	63	117	–	45	18.0	–	167
031B0224	**160**	4	2517	60	2½	3	82	117	18.5	45	18.5	–	167
031B0225	**160**	5	2517	60	2½	3	101	117	28.0	45	28.0	–	167
031B0226	**160**	6	3020	75	3	3	120	117	34.5	51	34.5	–	167
031B0232	**170**	2	2012	50	2	2	44	127	–	32	12.0	–	177
031B0233	**170**	3	2517	60	2½	2	63	127	–	45	18.0	–	177
031B0234	**170**	4	2517	60	2½	3	82	127	18.5	45	18.5	–	177
031B0235	**170**	5	3020	75	3	3	101	127	25.0	51	25.0	–	177
031B0236	**170**	6	3020	75	3	3	120	127	34.5	51	34.5	–	177
031B0242	**180**	2	2517	60	2½	1	44	–	–	45	1.0	117	187
031B0243	**180**	3	2517	60	2½	2	63	137	–	45	18.0	–	187
031B0244	**180**	4	2517	60	2½	3	82	137	18.5	45	18.5	–	187
031B0245	**180**	5	3020	75	3	3	101	137	25.0	51	25.0	–	187
031B0246	**180**	6	3020	75	3	3	120	137	34.5	51	34.5	–	187
031B0248*	**180**	8	3020	75	3	3	158	137	53.5	51	53.5	–	187
031B0252	**190**	2	2517	60	2½	1	44	–	–	45	1.0	117	197
031B0253	**190**	3	2517	60	2½	2	63	147	–	45	18.0	–	197
031B0254	**190**	4	2517	60	2½	3	82	147	18.5	45	18.5	–	197
031B0255	**190**	5	3020	75	3	3	101	147	25.0	51	25.0	–	197
031B0256	**190**	6	3020	75	3	3	120	147	34.5	51	34.5	–	197
031B0258*	**190**	8	3020	75	3	3	158	147	53.5	51	53.5	–	197
031B0262	**200**	2	2517	60	2½	1	44	–	–	45	1.0	117	207
031B0263	**200**	3	2517	60	2½	7	63	157	–	45	18.0	117	207
031B0264	**200**	4	3020	75	3	3	82	157	15.5	51	15.5	–	207
031B0265	**200**	5	3020	75	3	3	101	157	25.0	51	25.0	–	207
031B0266	**200**	6	3020	75	3	3	120	157	34.5	51	34.5	–	207
031B0268*	**200**	8	3525	100	4	3	158	157	46.5	65	46.5	–	207
031B0272	**212**	2	2517	60	2½	1	44	–	–	45	1.0	117	219
031B0273	**212**	3	2517	60	2½	7	63	169	–	45	18.0	117	219
031B0274	**212**	4	3020	75	3	3	82	169	15.5	51	15.5	–	219
031B0275	**212**	5	3020	75	3	3	101	169	25.0	51	25.0	–	219
031B0276	**212**	6	3525	100	4	3	120	169	28.0	65	28.0	–	219
031B0278*	**212**	8	3525	100	4	3	158	169	46.5	65	46.5	–	219
031B0282	**224**	2	2517	60	2½	8	44	181	1.0	45	-	117	231
031B0283	**224**	3	2517	60	2½	7	63	181	–	45	18.0	117	231
031B0284	**224**	4	3020	75	3	3	82	181	15.5	51	15.5	–	231
031B0285	**224**	5	3020	75	3	3	101	181	25.0	51	25.0	–	231
031B0286	**224**	6	3525	100	4	3	120	181	28.0	65	28.0	–	231
031B0288*	**224**	8	3525	100	4	3	158	181	46.5	65	46.5	–	231

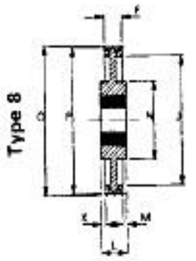
Dimensions in millimetres unless otherwise stated. *Non-preferred pulley sizes. Pitch diameters in italic type indicate pulleys to be used with B V- belts, XPB & QXPB wedge belts only.
All envelope, prime functional and Taper Lock bush dimensions are correct at the time of publication. Non-functional dimensions may vary slightly.
Non-functional dimensions may vary slightly.
These pulleys are designed to operate at rim speeds upto 40m/sec, for higher speeds contact your local authorised distributor.

Taper Lock Pulleys for B, SPB, XPB, QXPB & USPB Belts





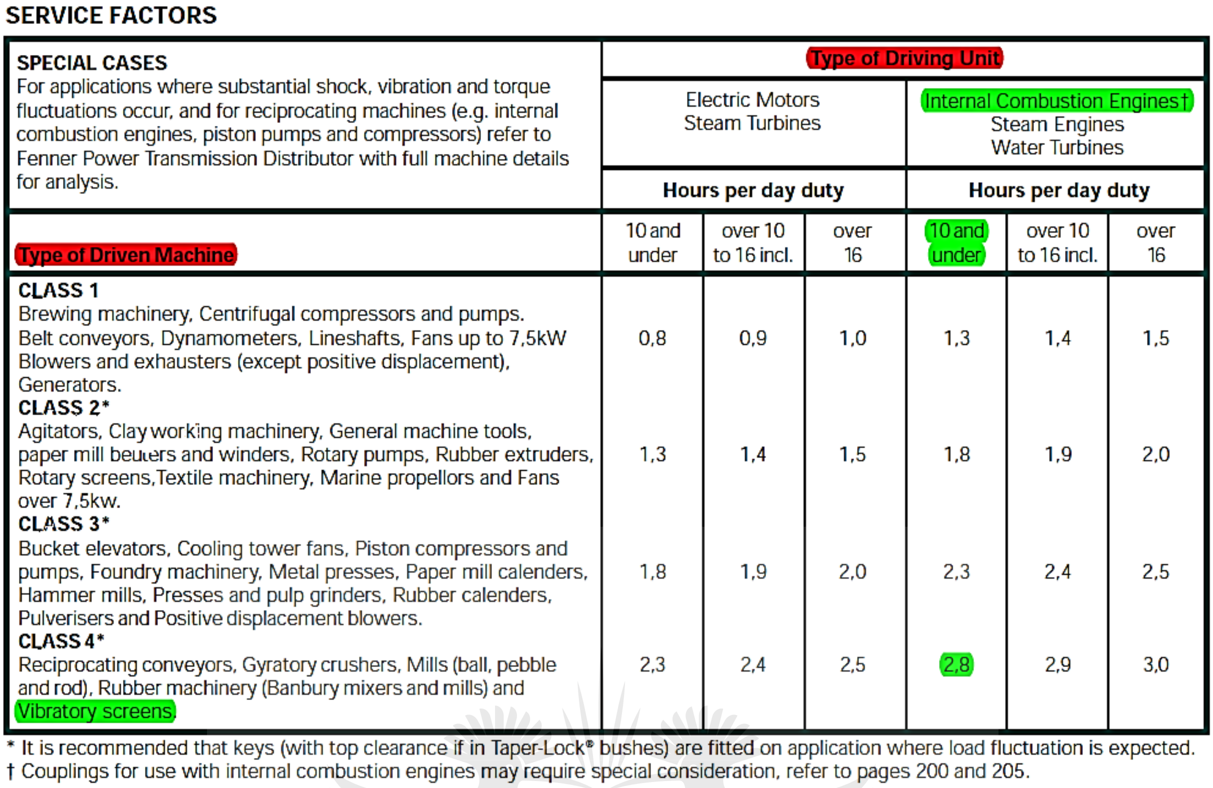




Catalogue Code	Pitch Dia (P)	No. of Grooves	Bush No.	Max. Bore		Pulley Type	F	J	K	L	M	N	Outside Dia (D)
				Metric	Inch								
031B0292	236	2	2517	60	2 1/2	8	44	193	1.0	45	–	117	243
031B0293	236	3	2517	60	2 1/2	7	63	193	–	45	18.0	117	243
031B0294	236	4	3020	75	3	3	82	193	15.5	51	15.5	–	243
031B0295	236	5	3525	100	4	3	101	193	18.0	65	18.0	–	243
031B0296	236	6	3525	100	4	3	120	193	28.0	65	28.0	–	243
031B0298*	236	8	3525	100	4	3	158	193	46.5	65	46.5	–	243
031B0302	250	2	2517	60	2 1/2	8	44	207	1.0	45	1.0	117	257
031B0303	250	3	3020	75	3	7	63	207	–	51	12.0	144	257
031B0304	250	4	3020	75	3	7	82	207	15.5	51	15.5	144	257
031B0305	250	5	3525	100	4	3	101	207	18.0	65	18.0	–	257
031B0306	250	6	3525	100	4	3	120	207	28.0	65	28.0	–	257
031B0308*	250	8	3525	100	4	3	158	207	46.5	65	46.5	–	257
031B0322	280	2	2517	60	2 1/2	8	44	237	1.0	45	–	117	287
031B0323	280	3	3020	75	3	7	63	237	6.0	51	6.0	144	287
031B0324	280	4	3020	75	3	7	82	237	15.5	51	15.5	144	287
031B0325	280	5	3525	100	4	7	101	237	18.0	65	18.0	187	287
031B0326	280	6	3525	100	4	7	120	237	27.5	65	27.5	187	287
031B0328*	280	8	3525	100	4	7	158	237	46.5	65	46.5	187	287
031B0332	315	2	2517	60	2 1/2	8	44	272	1.0	45	–	117	322
031B0333	315	3	3020	75	3	7	63	272	6.0	51	6.0	144	322
031B0334	315	4	3525	100	4	7	82	272	3.5	65	3.5	187	322
031B0335	315	5	3525	100	4	7	101	272	18.0	65	18.0	187	322
031B0336	315	6	3525	100	4	7	120	272	27.5	65	27.5	187	322
031B0338*	315	8	3525	100	4	7	158	272	46.5	65	46.5	187	322
031B0342	355	2	3020	75	3	5	44	312	3.5	51	3.5	144	362
031B0343	355	3	3020	75	3	8	63	312	6.0	51	6.0	144	362
031B0344	355	4	3525	100	4	8	82	312	3.5	89	3.5	187	362
031B0345	355	5	3525	100	4	5	101	312	18.0	65	18.0	187	362
031B0346	355	6	3525	100	4	5	120	312	27.5	65	27.5	187	362
031B0348*	355	8	3525	100	4	7	158	312	46.5	65	46.5	187	362
031B0352	400	2	3020	75	3	4	44	357	3.5	51	3.5	144	407
031B0353	400	3	3525	100	4	8	63	357	1.0	65	1.0	187	407
031B0354	400	4	3525	100	4	4	82	357	8.5	65	8.5	187	407
031B0355	400	5	3525	100	4	7	101	357	18.0	65	18.0	187	407
031B0356	400	6	3525	100	4	5	120	357	27.5	65	27.5	187	407
031B0358*	400	8	3525	100	4	5	158	357	46.5	65	46.5	200	407
014B0362	450	2	3020	75	3	4	44	407	3.5	51	3.5	144	457
014B0363	450	3	3525	100	4	4	63	407	1.0	65	1.0	187	457
014B0364	450	4	3525	100	4	4	82	407	8.5	65	8.5	187	457
014B0365	450	5	3525	100	4	5	101	407	18.0	65	18.0	187	457
014B0366	450	6	3525	100	4	5	120	407	27.5	65	27.5	216	457
014B0368*	450	8	3525	100	4	5	158	407	46.5	65	46.5	216	457
031B0372	500	2	3020	75	3	4	44	457	3.5	51	3.5	144	507
031B0373	500	3	3525	100	4	4	63	457	1.0	65	1.0	187	507
031B0374	500	4	3525	100	4	4	82	457	8.5	65	8.5	187	507
031B0375	500	5	3525	100	4	5	101	457	18.0	65	18.0	187	507
031B0376	500	6	3525	100	4	5	120	457	27.5	65	27.5	216	507
031B0378*	500	8	3525	100	4	5	158	457	46.5	65	46.5	216	507
014B0382	560	2	3020	75	3	4	44	517	4.0	76	4.0	144	567
014B0383	560	3	3525	100	4	4	63	517	1.0	65	1.0	187	567
014B0384	560	4	3525	100	4	4	82	517	8.5	65	8.5	187	567
014B0385	560	5	3525	100	4	4	101	517	18.0	65	18.0	216	567
014B0386	560	6	3525	100	4	5	120	517	27.5	65	27.5	187	567
014B0388*	560	8	4030	115	4 1/2	5	158	517	41.0	76	41.0	242	567
031B0392	630	2	3020	75	3	4	44	587	3.5	51	3.5	144	637
031B0393	630	3	3525	100	4	4	63	587	1.0	65	1.0	187	637
031B0394	630	4	3525	100	4	4	82	587	8.5	65	8.5	187	637
031B0395	630	5	3525	100	4	4	101	587	18.0	65	18.0	216	637
031B0396	630	6	3525	100	4	5	120	587	27.5	65	27.5	216	637
031B0398*	630	8	4030	115	4 1/2	5	158	587	41.0	76	41.0	242	637
031B0413	800	3	3525	100	4	4	63	754	1.0	65	1.0	187	807
031B0414	800	4	4030	115	4 1/2	4	82	754	3.0	76	3.0	216	807
031B0415	800	5	4030	115	4 1/2	4	101	754	12.5	76	12.5	216	807
031B0416	800	6	4535	125	5	5	120	754	15.5	89	15.5	242	807
031B0418*	800	8	4535	125	5	5	158	754	34.5	89	34.5	242	807
031B0433*	1000	3	4030	115	4 1/2	4	63	954	6.5	76	6.5	216	1007
031B0434	1000	4	4030	115	4 1/2	4	82	954	3.0	76	3.0	216	1007
031B0435	1000	5	4535	125	5	4	101	954	6.0	89	6.0	242	1007
031B0436	1000	6	4535	125	5	5	120	954	15.5	89	15.5	242	1007
031B0438*	1000	8	4535	125	5	5	158	954	34.5	89	34.5	242	1007

Dimensions in millimetres unless otherwise stated. * Non-preferred pulley sizes. Intermediate diameters available on a non-stock basis, see page 70.
All envelope, prime functional and Taper Lock bush dimensions are correct at the time of publication. Non-functional dimensions may vary slightly.
These pulleys are designed to operate at rim speeds upto 40m/sec, for higher speeds contact your local authorised distributor.

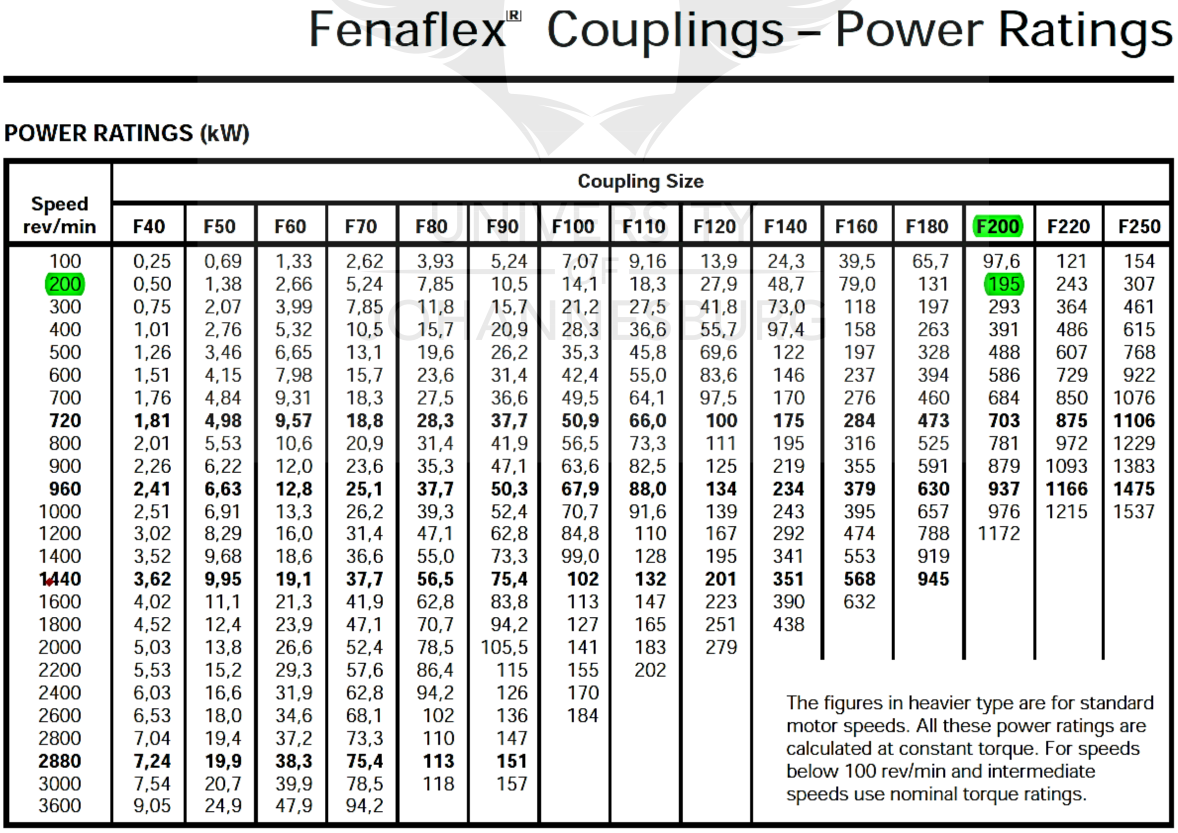
Appendix 14: Fenner flex tyre coupling tables.



SERVICE FACTORS

SPECIAL CASES For applications where substantial shock, vibration and torque fluctuations occur, and for reciprocating machines (e.g. internal combustion engines, piston pumps and compressors) refer to Fenner Power Transmission Distributor with full machine details for analysis.	Type of Driving Unit					
	Electric Motors Steam Turbines			Internal Combustion Engines† Steam Engines Water Turbines		
	Hours per day duty			**Hours per day duty**		
Type of Driven Machine	10 and under	over 10 to 16 incl.	over 16	10 and under	over 10 to 16 incl.	over 16
CLASS 1 Brewing machinery, Centrifugal compressors and pumps. Belt conveyors, Dynamometers, Lineshafts, Fans up to 7,5kW Blowers and exhausters (except positive displacement), Generators.	0,8	0,9	1,0	1,3	1,4	1,5
CLASS 2* Agitators, Clay working machinery, General machine tools, paper mill beaters and winders, Rotary pumps, Rubber extruders, Rotary screens, Textile machinery, Marine propellors and Fans over 7,5kw.	1,3	1,4	1,5	1,8	1,9	2,0
CLASS 3* Bucket elevators, Cooling tower fans, Piston compressors and pumps, Foundry machinery, Metal presses, Paper mill calenders, Hammer mills, Presses and pulp grinders, Rubber calenders, Pulverisers and Positive displacement blowers.	1,8	1,9	2,0	2,3	2,4	2,5
CLASS 4* Reciprocating conveyors, Gyratory crushers, Mills (ball, pebble and rod), Rubber machinery (Banbury mixers and mills) and Vibratory screens.	2,3	2,4	2,5	2,8	2,9	3,0

* It is recommended that keys (with top clearance if in Taper-Lock® bushes) are fitted on application where load fluctuation is expected.
† Couplings for use with internal combustion engines may require special consideration, refer to pages 200 and 205.



Fenaflex® Couplings – Power Ratings

POWER RATINGS (kW)

Speed rev/min	Coupling Size														
	F40	F50	F60	F70	F80	F90	F100	F110	F120	F140	F160	F180	F200	F220	F250
100	0,25	0,69	1,33	2,62	3,93	5,24	7,07	9,16	13,9	24,3	39,5	65,7	97,6	121	154
200	0,50	1,38	2,66	5,24	7,85	10,5	14,1	18,3	27,9	48,7	79,0	131	195	243	307
300	0,75	2,07	3,99	7,85	11,8	15,7	21,2	27,5	41,8	73,0	118	197	293	364	461
400	1,01	2,76	5,32	10,5	15,7	20,9	28,3	36,6	55,7	97,4	158	263	391	486	615
500	1,26	3,46	6,65	13,1	19,6	26,2	35,3	45,8	69,6	122	197	328	488	607	768
600	1,51	4,15	7,98	15,7	23,6	31,4	42,4	55,0	83,6	146	237	394	586	729	922
700	1,76	4,84	9,31	18,3	27,5	36,6	49,5	64,1	97,5	170	276	460	684	850	1076
720	**1,81**	**4,98**	**9,57**	**18,8**	**28,3**	**37,7**	**50,9**	**66,0**	**100**	**175**	**284**	**473**	**703**	**875**	**1106**
800	2,01	5,53	10,6	20,9	31,4	41,9	56,5	73,3	111	195	316	525	781	972	1229
900	2,26	6,22	12,0	23,6	35,3	47,1	63,6	82,5	125	219	355	591	879	1093	1383
960	**2,41**	**6,63**	**12,8**	**25,1**	**37,7**	**50,3**	**67,9**	**88,0**	**134**	**234**	**379**	**630**	**937**	**1166**	**1475**
1000	2,51	6,91	13,3	26,2	39,3	52,4	70,7	91,6	139	243	395	657	976	1215	1537
1200	3,02	8,29	16,0	31,4	47,1	62,8	84,8	110	167	292	474	788	1172		
1400	3,52	9,68	18,6	36,6	55,0	73,3	99,0	128	195	341	553	919			
1440	**3,62**	**9,95**	**19,1**	**37,7**	**56,5**	**75,4**	**102**	**132**	**201**	**351**	**568**	**945**			
1600	4,02	11,1	21,3	41,9	62,8	83,8	113	147	223	390	632				
1800	4,52	12,4	23,9	47,1	70,7	94,2	127	165	251	438					
2000	5,03	13,8	26,6	52,4	78,5	105,5	141	183	279						
2200	5,53	15,2	29,3	57,6	86,4	115	155	202							
2400	6,03	16,6	31,9	62,8	94,2	126	170								
2600	6,53	18,0	34,6	68,1	102	136	184								
2800	7,04	19,4	37,2	73,3	110	147									
2880	**7,24**	**19,9**	**38,3**	**75,4**	**113**	**151**									
3000	7,54	20,7	39,9	78,5	118	157									
3600	9,05	24,9	47,9	94,2											

The figures in heavier type are for standard motor speeds. All these power ratings are calculated at constant torque. For speeds below 100 rev/min and intermediate speeds use nominal torque ratings.

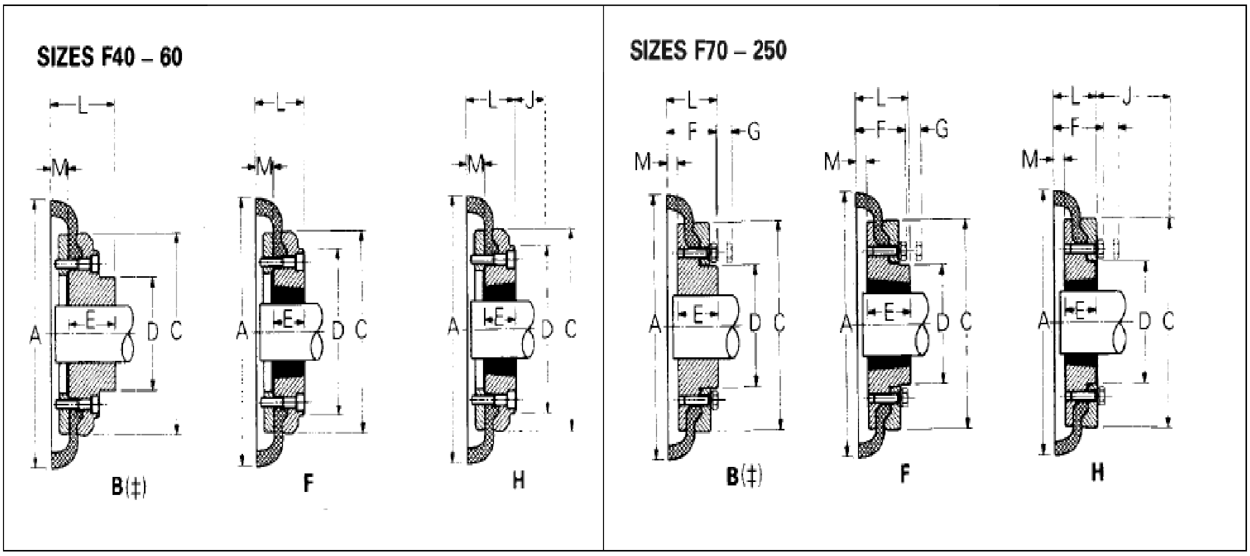
PHYSICAL CHARACTERISTICS – FLEXIBLE TYRES

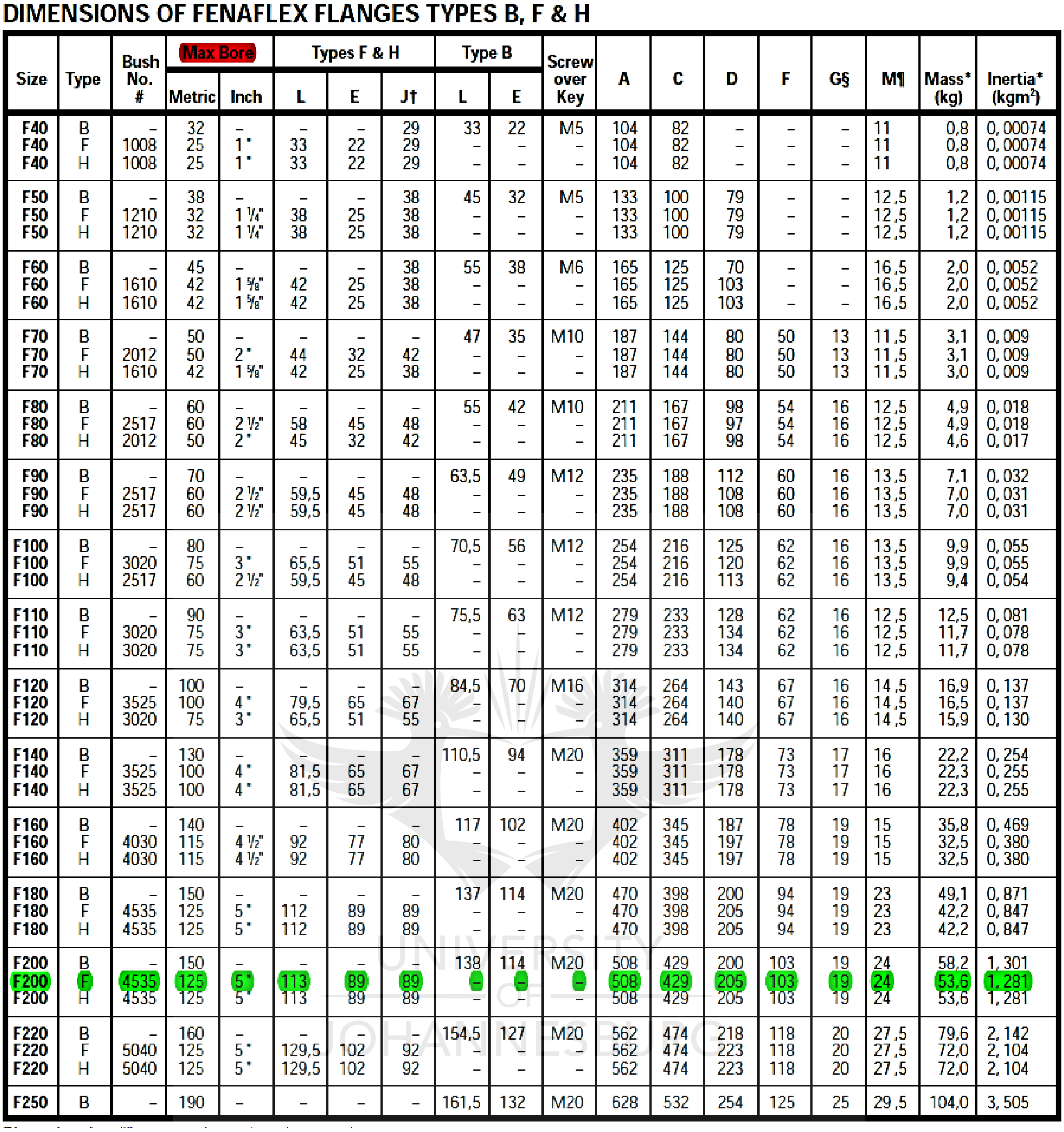
	Coupling Size														
Characteristics	**F40**	**F50**	**F60**	**F70**	**F80**	**F90**	**F100**	**F110**	**F120**	**F140**	**F160**	**F180**	**F200**	**F220**	**F250**
Maximum speed rev/min	4500	4500	4000	3600	3100	3000	2600	2300	2050	1800	1600	1500	1300	1100	1000
Nominal Torque Nm T_{KN}	24	66	127	250	375	500	675	875	1330	2325	3770	6270	9325	11600	14675
Maximum Torque Nm $T_{K\,MAX}$	64	160	318	487	759	1096	1517	2137	3547	5642	9339	16455	23508	33125	42740
Torsional Stiffness Nm/^{O}	5	13	26	41	63	91	126	178	296	470	778	1371	1959	2760	3562
Max, parallel misalignment mm	1,1	1,3	1,6	1,9	2,1	2,4	2,6	2,9	3,2	3,7	4,2	4,8	5,3	5,8	6,6
Maximum end float mm ±	1,3	1,7	2,0	2,3	2,6	3,0	3,3	3,7	4,0	4,6	5,3	6,0	6,6	7,3	8,2
Approximate mass, kg	0,1	0,3	0,5	0,7	1,0	1,1	1,1	1,4	2,3	2,6	3,4	7,7	8,0	10	15
Alternating Torque ± Nm @ 10Hz T_{KW}	11	26	53	81	127	183	252	356	591	940	1556	2742	3918	5521	7124
Resonance Factor V_R	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7
Damping Coefficient Ψ	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9	0,9

Coupling Size	**M Dimension (mm)**	**Gap Between Tyre Ends (mm)**	**Clamping Screw Torque (Nm)**	**Screw Size**
F40*	11	2	15	M6
F50*	12,5	2	15	M6
F60*	16,5	2	15	M6
F70	11,5	3	24	M8
F80	12,5	3	24	M8
F90	13,5	3	40	M10
F100	13,5	3	40	M10
F110	12,5	3	40	M10
F120	14,5	3	50	M12
F140	16	5	55	M12
F160	15	5	80	M16
F180	23	6	105	M16
F200	24	6	120	M16
F220	27,5	6	165	M20
F250	29,5	6	165	M20

*Hexagonal socket caphead clamping screws on these sizes.

FLANGES





DIMENSIONS OF FENAFLEX FLANGES TYPES B, F & H

Size	Type	Bush No. #	Max Bore		Types F & H			Type B		Screw over Key	A	C	D	F	G§	M¶	Mass* (kg)	Inertia* (kgm²)
			Metric	Inch	L	E	J†	L	E									
F40	B	–	32	–	–	–	29	33	22	M5	104	82	–	–	–	11	0,8	0, 00074
F40	F	1008	25	1"	33	22	29	–	–	–	104	82	–	–	–	11	0,8	0, 00074
F40	H	1008	25	1"	33	22	29	–	–	–	104	82	–	–	–	11	0,8	0, 00074
F50	B	–	38	–	–	–	38	45	32	M5	133	100	79	–	–	12 ,5	1,2	0, 00115
F50	F	1210	32	1 ¼"	38	25	38	–	–	–	133	100	79	–	–	12 ,5	1,2	0, 00115
F50	H	1210	32	1 ¼"	38	25	38	–	–	–	133	100	79	–	–	12 ,5	1,2	0, 00115
F60	B	–	45	–	–	–	38	55	38	M6	165	125	70	–	–	16 ,5	2,0	0, 0052
F60	F	1610	42	1 ⅝"	42	25	38	–	–	–	165	125	103	–	–	16 ,5	2,0	0, 0052
F60	H	1610	42	1 ⅝"	42	25	38	–	–	–	165	125	103	–	–	16 ,5	2,0	0, 0052
F70	B	–	50	–	–	–	–	47	35	M10	187	144	80	50	13	11 ,5	3,1	0, 009
F70	F	2012	50	2"	44	32	42	–	–	–	187	144	80	50	13	11 ,5	3,1	0, 009
F70	H	1610	42	1 ⅝"	42	25	38	–	–	–	187	144	80	50	13	11 ,5	3,0	0, 009
F80	B	–	60	–	–	–	–	55	42	M10	211	167	98	54	16	12 ,5	4,9	0, 018
F80	F	2517	60	2 ½"	58	45	48	–	–	–	211	167	97	54	16	12 ,5	4,9	0, 018
F80	H	2012	50	2"	45	32	42	–	–	–	211	167	98	54	16	12 ,5	4,6	0, 017
F90	B	–	70	–	–	–	–	63,5	49	M12	235	188	112	60	16	13 ,5	7,1	0, 032
F90	F	2517	60	2 ½"	59,5	45	48	–	–	–	235	188	108	60	16	13 ,5	7,0	0, 031
F90	H	2517	60	2 ½"	59,5	45	48	–	–	–	235	188	108	60	16	13 ,5	7,0	0, 031
F100	B	–	80	–	–	–	–	70,5	56	M12	254	216	125	62	16	13 ,5	9,9	0, 055
F100	F	3020	75	3"	65,5	51	55	–	–	–	254	216	120	62	16	13 ,5	9,9	0, 055
F100	H	2517	60	2 ½"	59,5	45	48	–	–	–	254	216	113	62	16	13 ,5	9,4	0, 054
F110	B	–	90	–	–	–	–	75,5	63	M12	279	233	128	62	16	12 ,5	12,5	0, 081
F110	F	3020	75	3"	63,5	51	55	–	–	–	279	233	134	62	16	12 ,5	11,7	0, 078
F110	H	3020	75	3"	63,5	51	55	–	–	–	279	233	134	62	16	12 ,5	11,7	0, 078
F120	B	–	100	–	–	–	–	84,5	70	M16	314	264	143	67	16	14 ,5	16,9	0, 137
F120	F	3525	100	4"	79,5	65	67	–	–	–	314	264	140	67	16	14 ,5	16,5	0, 137
F120	H	3020	75	3"	65,5	51	55	–	–	–	314	264	140	67	16	14 ,5	15,9	0, 130
F140	B	–	130	–	–	–	–	110,5	94	M20	359	311	178	73	17	16	22,2	0, 254
F140	F	3525	100	4"	81,5	65	67	–	–	–	359	311	178	73	17	16	22,3	0, 255
F140	H	3525	100	4"	81,5	65	67	–	–	–	359	311	178	73	17	16	22,3	0, 255
F160	B	–	140	–	–	–	–	117	102	M20	402	345	187	78	19	15	35,8	0, 469
F160	F	4030	115	4 ½"	92	77	80	–	–	–	402	345	197	78	19	15	32,5	0, 380
F160	H	4030	115	4 ½"	92	77	80	–	–	–	402	345	197	78	19	15	32,5	0, 380
F180	B	–	150	–	–	–	–	137	114	M20	470	398	200	94	19	23	49,1	0, 871
F180	F	4535	125	5"	112	89	89	–	–	–	470	398	205	94	19	23	42,2	0, 847
F180	H	4535	125	5"	112	89	89	–	–	–	470	398	205	94	19	23	42,2	0, 847
F200	B	–	150	–	–	–	–	138	114	M20	508	429	200	103	19	24	58,2	1, 301
F200	F	4535	125	5"	113	89	89	–	–	–	508	429	205	103	19	24	53,6	1, 281
F200	H	4535	125	5"	113	89	89	–	–	–	508	429	205	103	19	24	53,6	1, 281
F220	B	–	160	–	–	–	–	154,5	127	M20	562	474	218	118	20	27 ,5	79,6	2, 142
F220	F	5040	125	5"	129,5	102	92	–	–	–	562	474	223	118	20	27 ,5	72,0	2, 104
F220	H	5040	125	5"	129,5	102	92	–	–	–	562	474	223	118	20	27 ,5	72,0	2, 104
F250	B	–	190	–	–	–	–	161,5	132	M20	628	532	254	125	25	29 ,5	104,0	3, 505

Appendix 15: Caterpillar Catalogue.

CS-533E CP-533E

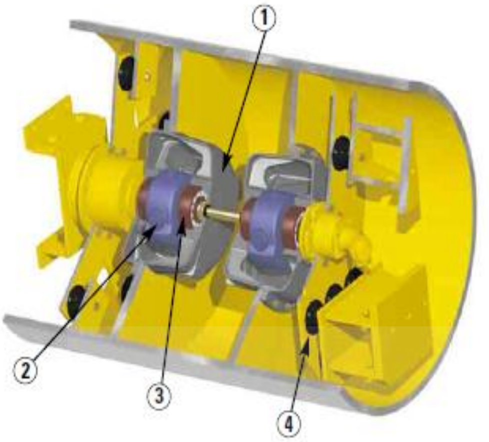
Vibratory Soil Compactors

CAT®

Cat® 3054C Turbocharged Diesel Engine	
Gross Power	97 kW/130 hp
Drum Width	2134 mm
Operating Weight (with ROPS/FOPS cab)	
CS-533E	10 840 kg
CS-533E with Heavy Weight option	12 360 kg
CP-533E	11 680 kg

Vibratory System

The pod-style vibratory system, delivers superior compactive force while offering serviceability advantages.



1 Pod-style Weight Housings
2 Patented Eccentric Weights
3 Heavy-duty Bearings
4 Isolation Mounts

Pod-style weight housings. They are assembled and sealed at the factory to ensure cleanliness, longer bearing life and easier field exchange or service.

Dual amplitude. Dual amplitude works efficiently in a wider range of applications. High or low amplitude is selected from the operator's station.

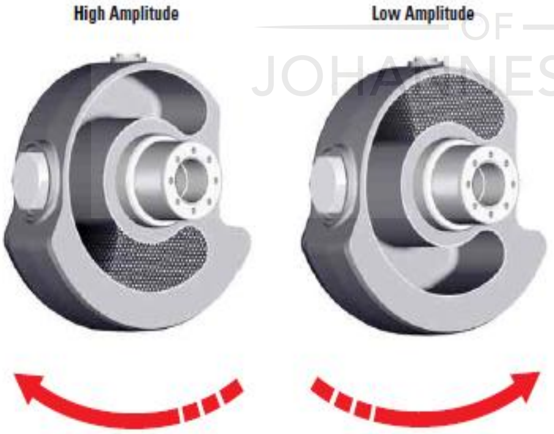
Heavy-duty bearings. Large heavy-duty bearings for the eccentric weight shaft designed for high compactive forces.

Service interval. 3 year/3000 hour vibratory bearing lube service interval for reduced maintenance.

Heavy duty isolation mounts. Improved isolation mounts allow more force to be transmitted to the ground and less vibration to the drum yoke.

Patented Eccentric Weights

Reliable dual amplitude selection and innovative design ensure precise performance.



Amplitude selection. Positive amplitude selection is accomplished when the steel shot is repositioned inside the hollow eccentric weight. Direction of weight shaft rotation determines amplitude level.

High reliability. High reliability since there is no chance of the high-strength spherical steel shot wedging together. System reliability is superior to swinging mechanical weights and is also quieter during starts and stops.

Simplified control. Simplified control from the operator's station with a selection switch on the operator's console.

Longer service life. No heavy weights to slam together, no metal fragments to contaminate the bearing lubrication system.

Engine

Four-stroke, four cylinder Caterpillar 3054C turbocharged diesel engine. Meets EU directive 97/68/EC Stage II emission requirements.

Ratings at 2200 rpm	kW	hp
Gross power	97	130
Net power		
EEC 80/1269	93	125
ISO 9249	93	125

Net power advertised is the power available at the flywheel when the engine is equipped with fan, air cleaner, muffler and alternator. No derating required up to 2500 m altitude.

Dimensions

Bore	105 mm
Stroke	127 mm
Displacement	4.4 liters

Dual-element, dry-type air cleaner with visual restriction indicator, glow plug starting aid and fuel/water separator are standard.

Transmission

Two variable displacement piston pumps supply pressurized flow to two dual displacement piston motors. One pump and motor drives the drum propel system while the other pump and motor drives the rear wheels. The dual pump system ensures equal flow to the drive motors regardless of the operating conditions. In case the drum or wheels lose traction, the other motor can still build additional pressure to provide added torque.

The drive motors have two swashplate positions allowing operation at either maximum torque for compaction and gradeability or greater speed for moving around the job site. A rocker switch at the operator's console triggers an electric over hydraulic control to change speed ranges.

Speeds (forward and reverse)

Low Range	8.0 km/h
High Range	12.0 km/h

Gradeability with or without vibration (subject to underfoot conditions) 50%

Brakes

Service brake features

Closed-loop hydrostatic drive system provides dynamic braking during operation.

Secondary brake features*

Spring-applied/hydraulically-released multiple disc type brake mounted on the drum drive gear reducer. Secondary brakes are activated by: a button on the operator's console; loss of hydraulic pressure in the brake circuit; or when the engine is shut down. A brake interlock system helps prevent driving through the secondary brake.

* All machines sold within European Union are equipped with a brake release pump which allows the manual release of the secondary brake system for towing the machine.

Braking system meets EN 500.

Steering

A priority-demand hydraulic power-assist steering system provides smooth low-effort steering. The system always receives the power it needs regardless of other hydraulic functions.

Minimum turning radius:

Inside	3680 mm
Outside	5810 mm

Steering angle

(each direction) ± 34°

Oscillation angle

(each direction) ± 15°

Hydraulic system

Two 76 mm bore, double-acting cylinders powered by a gear-type pump.

Final Drives and Axle

Final drive is hydrostatic with gear reducer to the drum and hydrostatic with differential and planetary gear reduction to each wheel.

Axle

Heavy-duty fixed rear axle with a limited slip differential for smooth and quiet torque transfer.

Tires

587 mm x 660 mm (23.1" x 26")

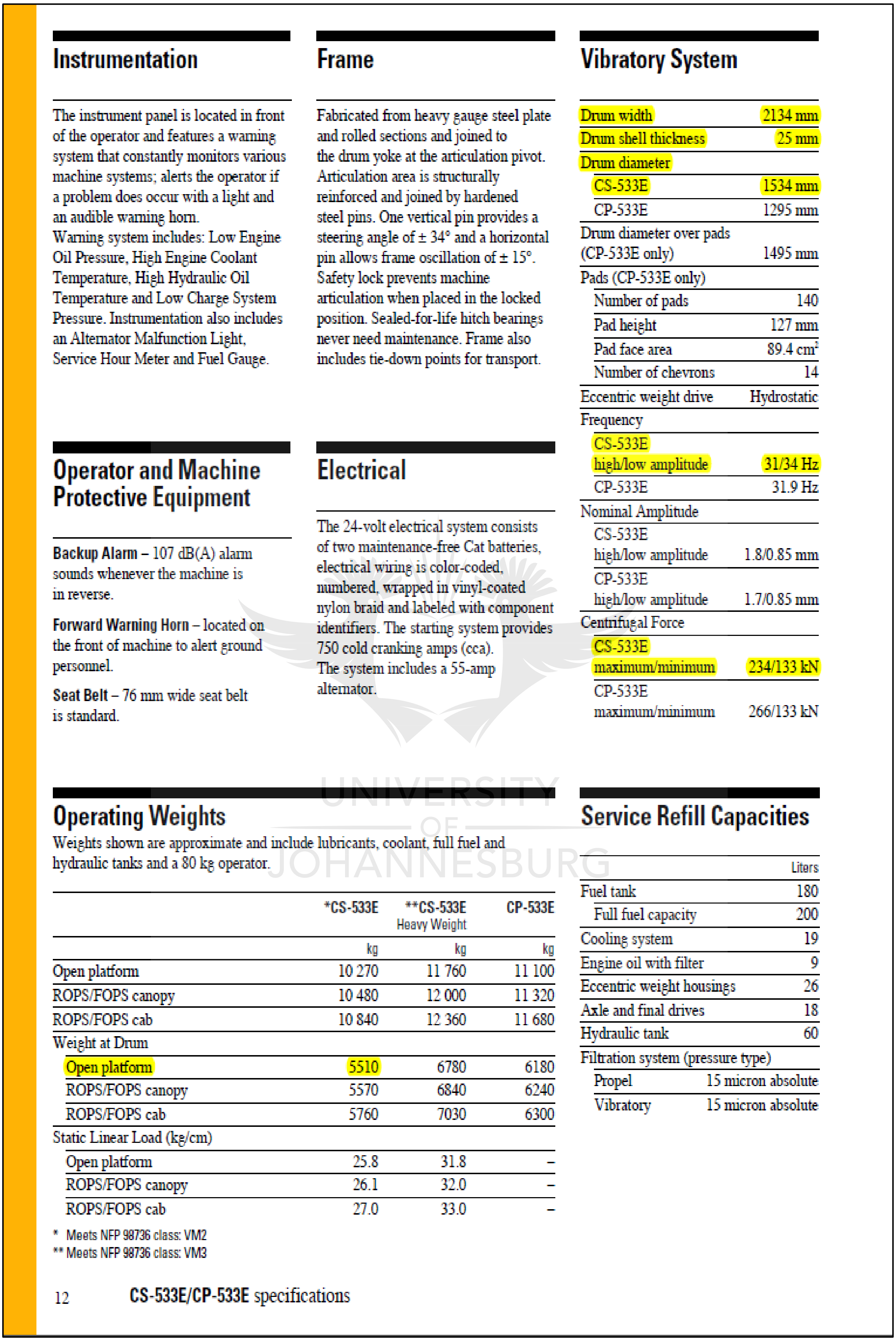
CS-533E	10-ply flotation
CP-533E	14-ply traction

Ballasted with 30-35% calcium chloride/water solution, approximately 430 liters per tire.

Sound

Operator Sound. The operator sound level measured according to the procedures specified in ISO6394 is 77 dB(A), for cab offered by Caterpillar, when properly installed and maintained and tested with the doors and windows closed.

Exterior Sound. The labeled spectator sound power level measured according to the test procedures and conditions specified in 2000/14/EC is 111 dB(A).



Instrumentation

The instrument panel is located in front of the operator and features a warning system that constantly monitors various machine systems; alerts the operator if a problem does occur with a light and an audible warning horn.
Warning system includes: Low Engine Oil Pressure, High Engine Coolant Temperature, High Hydraulic Oil Temperature and Low Charge System Pressure. Instrumentation also includes an Alternator Malfunction Light, Service Hour Meter and Fuel Gauge.

Operator and Machine Protective Equipment

Backup Alarm – 107 dB(A) alarm sounds whenever the machine is in reverse.

Forward Warning Horn – located on the front of machine to alert ground personnel.

Seat Belt – 76 mm wide seat belt is standard.

Frame

Fabricated from heavy gauge steel plate and rolled sections and joined to the drum yoke at the articulation pivot. Articulation area is structurally reinforced and joined by hardened steel pins. One vertical pin provides a steering angle of ± 34° and a horizontal pin allows frame oscillation of ± 15°. Safety lock prevents machine articulation when placed in the locked position. Sealed-for-life hitch bearings never need maintenance. Frame also includes tie-down points for transport.

Electrical

The 24-volt electrical system consists of two maintenance-free Cat batteries, electrical wiring is color-coded, numbered, wrapped in vinyl-coated nylon braid and labeled with component identifiers. The starting system provides 750 cold cranking amps (cca). The system includes a 55-amp alternator.

Vibratory System

Drum width	2134 mm
Drum shell thickness	25 mm
Drum diameter	
CS-533E	1534 mm
CP-533E	1295 mm
Drum diameter over pads (CP-533E only)	1495 mm
Pads (CP-533E only)	
Number of pads	140
Pad height	127 mm
Pad face area	89.4 cm²
Number of chevrons	14
Eccentric weight drive	Hydrostatic
Frequency	
CS-533E high/low amplitude	31/34 Hz
CP-533E	31.9 Hz
Nominal Amplitude	
CS-533E high/low amplitude	1.8/0.85 mm
CP-533E high/low amplitude	1.7/0.85 mm
Centrifugal Force	
CS-533E maximum/minimum	234/133 kN
CP-533E maximum/minimum	266/133 kN

Operating Weights

Weights shown are approximate and include lubricants, coolant, full fuel and hydraulic tanks and a 80 kg operator.

	*CS-533E	**CS-533E Heavy Weight	CP-533E
	kg	kg	kg
Open platform	10 270	11 760	11 100
ROPS/FOPS canopy	10 480	12 000	11 320
ROPS/FOPS cab	10 840	12 360	11 680
Weight at Drum			
Open platform	5510	6780	6180
ROPS/FOPS canopy	5570	6840	6240
ROPS/FOPS cab	5760	7030	6300
Static Linear Load (kg/cm)			
Open platform	25.8	31.8	–
ROPS/FOPS canopy	26.1	32.0	–
ROPS/FOPS cab	27.0	33.0	–

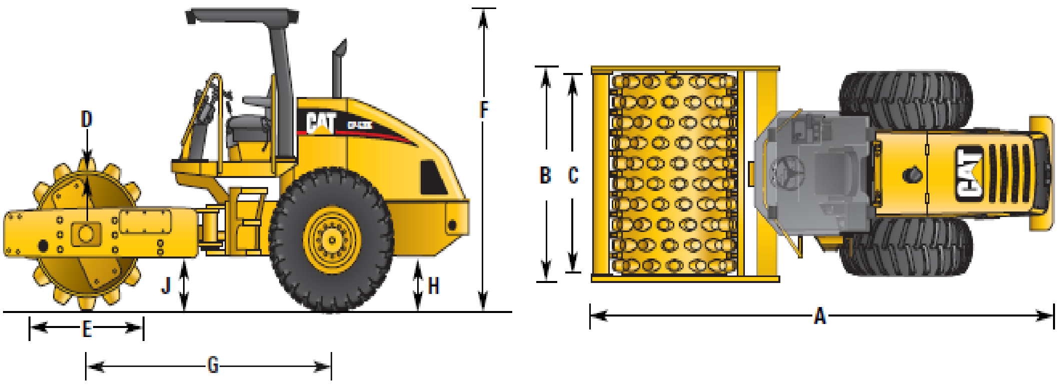
* Meets NFP 98736 class: VM2
** Meets NFP 98736 class: VM3

Service Refill Capacities

	Liters
Fuel tank	180
Full fuel capacity	200
Cooling system	19
Engine oil with filter	9
Eccentric weight housings	26
Axle and final drives	18
Hydraulic tank	60
Filtration system (pressure type)	
Propel	15 micron absolute
Vibratory	15 micron absolute

Dimensions

All dimensions are approximate.



		CS-533E	CP-533E
		mm	mm
A	Overall length	5510	5510
B	Overall width	2290	2290
	with Heavy Weight option	2360	–
C	Drum width	2134	2134
D	Drum shell thickness	25	25
E	Drum diameter	1534	1295
	Drum diameter over pads	–	1549

		CS-533E	CP-533E
		mm	mm
F	Height at ROPS/FOPS canopy	3060	3070
	Height at ROPS/FOPS cab	3070	3070
G	Wheelbase	2900	2900
H	Ground clearance	543	543
J	Curb clearance	521	521
	Inside turning radius	3680	3680
	Outside turning radius	5810	5810

Total Customer Support System

Service capability. Most dedicated dealer support system to ensure fast service whether at the dealer's shop or in the field by trained technicians using the latest tools and technology.

Parts availability. Most parts on dealer's shelf when you need them. Computer-controlled, emergency search system backup.

Parts stock lists. Dealer helps you plan on-site parts stock to minimize your parts investment while maximizing machine availability.

Literature support. Easy-to-use parts books, operation and maintenance manuals and service manuals to help you get maximum value from your Caterpillar equipment.

Remanufactured parts. Pumps and motors, pod-style weight housings, engines, fuel system and charging system components available from dealer at a fraction of new part cost.

Machine management services. Effective preventive maintenance programs, cost-effective repair options, customer meetings, operator and mechanic training.

Flexible financing. Your dealer can arrange attractive financing on the entire line of Caterpillar equipment. Terms structured to meet cash flow requirements. See how easy it is to own, lease or rent Cat equipment.

Estimated Production

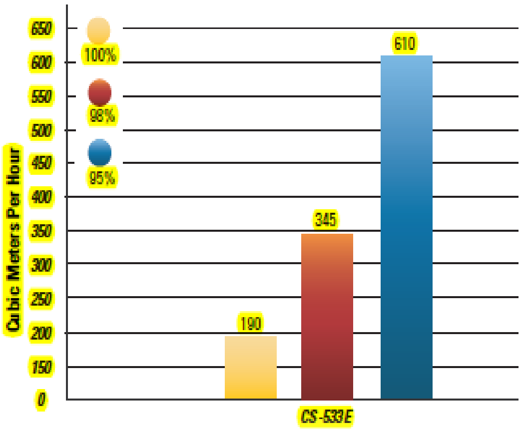
	Depth mm	Density %	150 mm Layer
Surface	0-500	>98	190 m³/h
Shallow	500-3000	95-98	345 m³/h
Deep	>3000	90-95	610 m³/h

Based on depth of fill below final grade (surface)

Based on final compacted thickness of layer

Density spec. is based on Standard Proctor Test

Productivity Comparisons



% Compaction of Standard Proctor

150 mm Crushed Limestone Base

Results are similar for padfoot drums.

Results may vary for different applications.

Machine Selection

Application	Layer Thickness mm	Smooth Drum CS-533E	Padfoot Drum CP-533E
Sand, Clayey or Silty Sand, Mine Tailing	150-300 300-450	● ▲	❐ ❐
Clay, Sandy or Silty Clay, Stabilized Clay	150-300 300-450	▲ ❐	● ▲
Silt, Sandy or Clayey Silt, Coal, Ash, Solid Waste	150-300 300-450	▲ ▲	● ▲
Base Aggregate, Gravel, Crushed Rock, Stabilized Base	150-300 300-450	▲ ❐	❐ ❐

❐ Good ▲ Better ● Best

The CS-533E and CP-533E vibratory soil compactors provide high compaction performance. Ideal for medium to large construction projects with low to moderate grades.

Shell Kit Performance

	Padfoot Drum	Shell Kit	Shell Kit Performance
Number of Pads	140	120	Less Kneading better for silt
Pad Height	127 mm	90 mm	Less Penetration better for silt and sandy clay
Weight at Drum	6240 kg	6990 kg	Higher Ground Pressure better for sandy clay
Max. Amplitude	1.7 mm	1.2 mm	Smaller Drum Movement better for silt and clay

Performance Ranking	Padfoot Drum	Shell Kit
Heavy Clay	●	❐
Sandy Clay	❐	●
Silt with Clay	❐	●
Slopes/Trenches	●	❐
Thick Layers	●	❐

Appendix 16: Sakai

SV512 Series

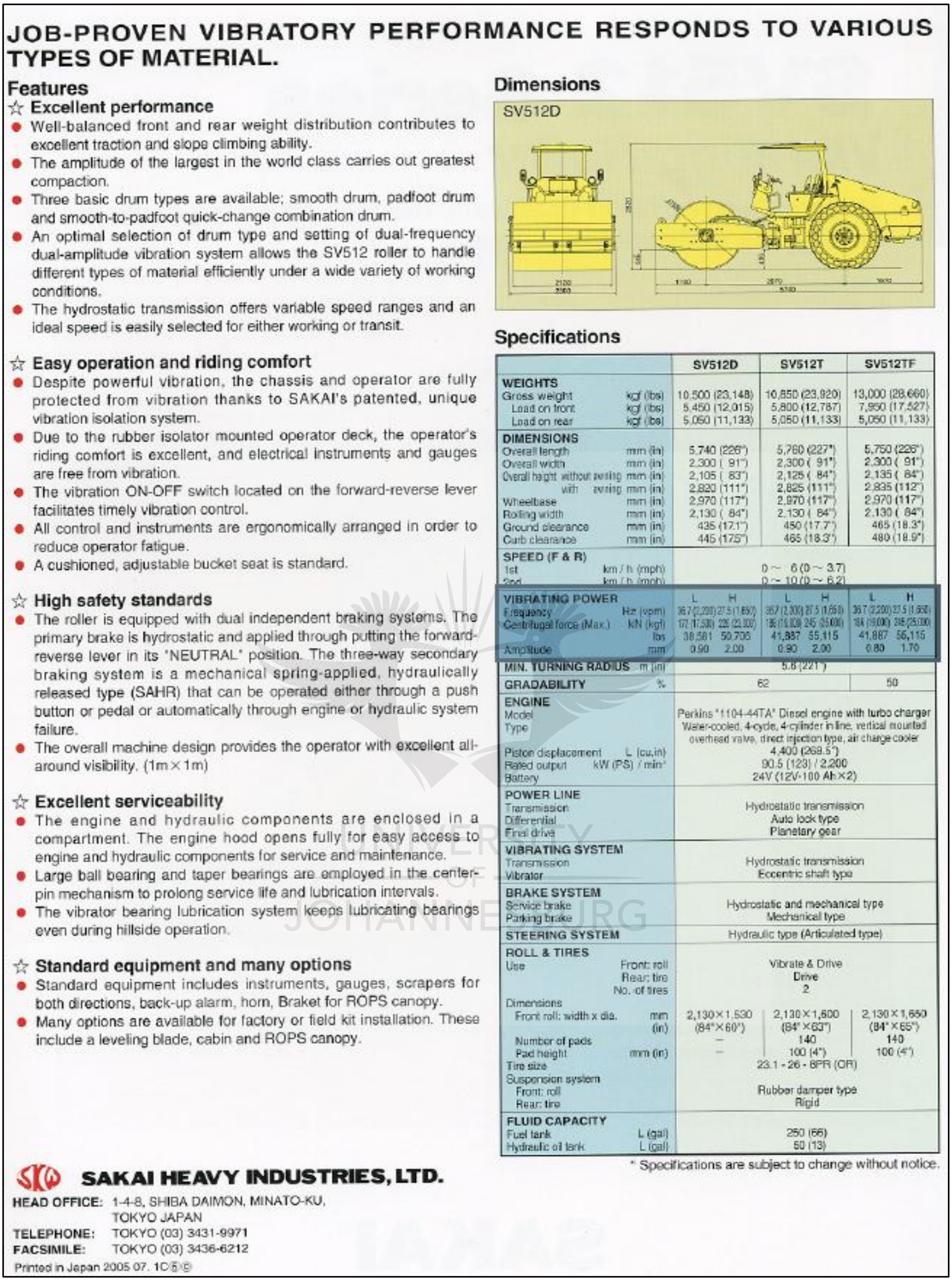
Vibrating Roller

Mighty vibrating roller drastically reduces operating costs in large scale earth-moving projects

SV512D
Smooth drum
Gross weights 10.5 ton
(23,150 lbs)

SV512TF
Padfoot drum with
removable smooth drum shell
Gross weights 13 ton
(28,660 lbs)

SAKAI®



JOB-PROVEN VIBRATORY PERFORMANCE RESPONDS TO VARIOUS TYPES OF MATERIAL.

Features

☆ Excellent performance

- Well-balanced front and rear weight distribution contributes to excellent traction and slope climbing ability.
- The amplitude of the largest in the world class carries out greatest compaction.
- Three basic drum types are available; smooth drum, padfoot drum and smooth-to-padfoot quick-change combination drum.
- An optimal selection of drum type and setting of dual-frequency dual-amplitude vibration system allows the SV512 roller to handle different types of material efficiently under a wide variety of working conditions.
- The hydrostatic transmission offers variable speed ranges and an ideal speed is easily selected for either working or transit.

☆ Easy operation and riding comfort

- Despite powerful vibration, the chassis and operator are fully protected from vibration thanks to SAKAI's patented, unique vibration isolation system.
- Due to the rubber isolator mounted operator deck, the operator's riding comfort is excellent, and electrical instruments and gauges are free from vibration.
- The vibration ON-OFF switch located on the forward-reverse lever facilitates timely vibration control.
- All control and instruments are ergonomically arranged in order to reduce operator fatigue.
- A cushioned, adjustable bucket seat is standard.

☆ High safety standards

- The roller is equipped with dual independent braking systems. The primary brake is hydrostatic and applied through putting the forward-reverse lever in its "NEUTRAL" position. The three-way secondary braking system is a mechanical spring-applied, hydraulically released type (SAHR) that can be operated either through a push button or pedal or automatically through engine or hydraulic system failure.
- The overall machine design provides the operator with excellent all-around visibility. (1m × 1m)

☆ Excellent serviceability

- The engine and hydraulic components are enclosed in a compartment. The engine hood opens fully for easy access to engine and hydraulic components for service and maintenance.
- Large ball bearing and taper bearings are employed in the center-pin mechanism to prolong service life and lubrication intervals.
- The vibrator bearing lubrication system keeps lubricating bearings even during hillside operation.

☆ Standard equipment and many options

- Standard equipment includes instruments, gauges, scrapers for both directions, back-up alarm, horn, Braket for ROPS canopy.
- Many options are available for factory or field kit installation. These include a leveling blade, cabin and ROPS canopy.

Dimensions

SV512D

Specifications

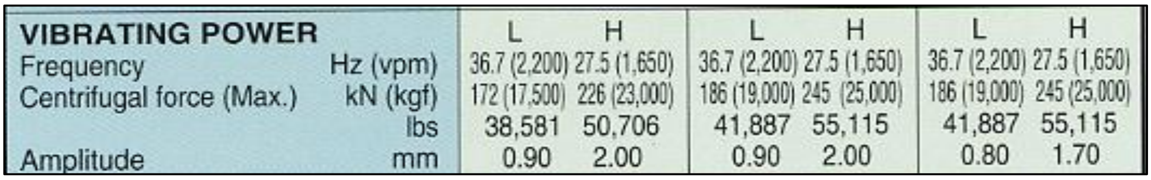
		SV512D	SV512T	SV512TF
WEIGHTS				
Gross weight	kgf (lbs)	10,500 (23,148)	10,850 (23,920)	13,000 (28,660)
Load on front	kgf (lbs)	5,450 (12,015)	5,800 (12,787)	7,950 (17,527)
Load on rear	kgf (lbs)	5,050 (11,133)	5,050 (11,133)	5,050 (11,133)
DIMENSIONS				
Overall length	mm (in)	5,740 (226")	5,760 (227")	5,750 (226")
Overall width	mm (in)	2,300 (91")	2,300 (91")	2,300 (91")
Overall height without awning	mm (in)	2,105 (83")	2,125 (84")	2,135 (84")
with awning	mm (in)	2,820 (111")	2,825 (111")	2,835 (112")
Wheelbase	mm (in)	2,970 (117")	2,970 (117")	2,970 (117")
Rolling width	mm (in)	2,130 (84")	2,130 (84")	2,130 (84")
Ground clearance	mm (in)	435 (17.1")	450 (17.7")	465 (18.3")
Curb clearance	mm (in)	445 (17.5")	465 (18.3")	480 (18.9")
SPEED (F & R)				
1st	km / h (mph)	0 ~ 6 (0 ~ 3.7)		
2nd	km / h (mph)	0 ~ 10 (0 ~ 6.2)		
VIBRATING POWER		L H	L H	L H
Frequency	Hz (vpm)	36.7 (2,200) 27.5 (1,650)	36.7 (2,200) 27.5 (1,650)	36.7 (2,200) 27.5 (1,650)
Centrifugal force (Max.)	kN (kgf)	172 (17,500) 226 (23,000)	186 (19,000) 245 (25,000)	186 (19,000) 245 (25,000)
	lbs	38,581 50,706	41,887 55,115	41,887 55,115
Amplitude	mm	0.90 2.00	0.90 2.00	0.80 1.70
MIN. TURNING RADIUS	m (in)	5.6 (221")		
GRADABILITY	%	62		50
ENGINE				
Model		Perkins "1104-44TA" Diesel engine with turbo charger		
Type		Water-cooled, 4-cycle, 4-cylinder in line, vertical mounted overhead valve, direct injection type, air charge cooler		
Piston displacement	L (cu,in)	4,400 (269.5")		
Rated output	kW (PS) / min⁻¹	90.5 (123) / 2,200		
Battery		24V (12V-100 Ah × 2)		
POWER LINE				
Transmission		Hydrostatic transmission		
Differential		Auto lock type		
Final drive		Planetary gear		
VIBRATING SYSTEM				
Transmission		Hydrostatic transmission		
Vibrator		Eccentric shaft type		
BRAKE SYSTEM				
Service brake		Hydrostatic and mechanical type		
Parking brake		Mechanical type		
STEERING SYSTEM		Hydraulic type (Articulated type)		
ROLL & TIRES				
Use	Front: roll	Vibrate & Drive		
	Rear: tire	Drive		
	No. of tires	2		
Dimensions				
Front roll: width x dia.	mm	2,130 × 1,530	2,130 × 1,600	2,130 × 1,650
	(in)	(84" × 60")	(84" × 63")	(84" × 65")
Number of pads		–	140	140
Pad height	mm (in)	–	100 (4")	100 (4")
Tire size		23.1 - 26 - 8PR (OR)		
Suspension system				
Front: roll		Rubber damper type		
Rear: tire		Rigid		
FLUID CAPACITY				
Fuel tank	L (gal)	250 (66)		
Hydraulic oil tank	L (gal)	50 (13)		

* Specifications are subject to change without notice.

SAKAI HEAVY INDUSTRIES, LTD.

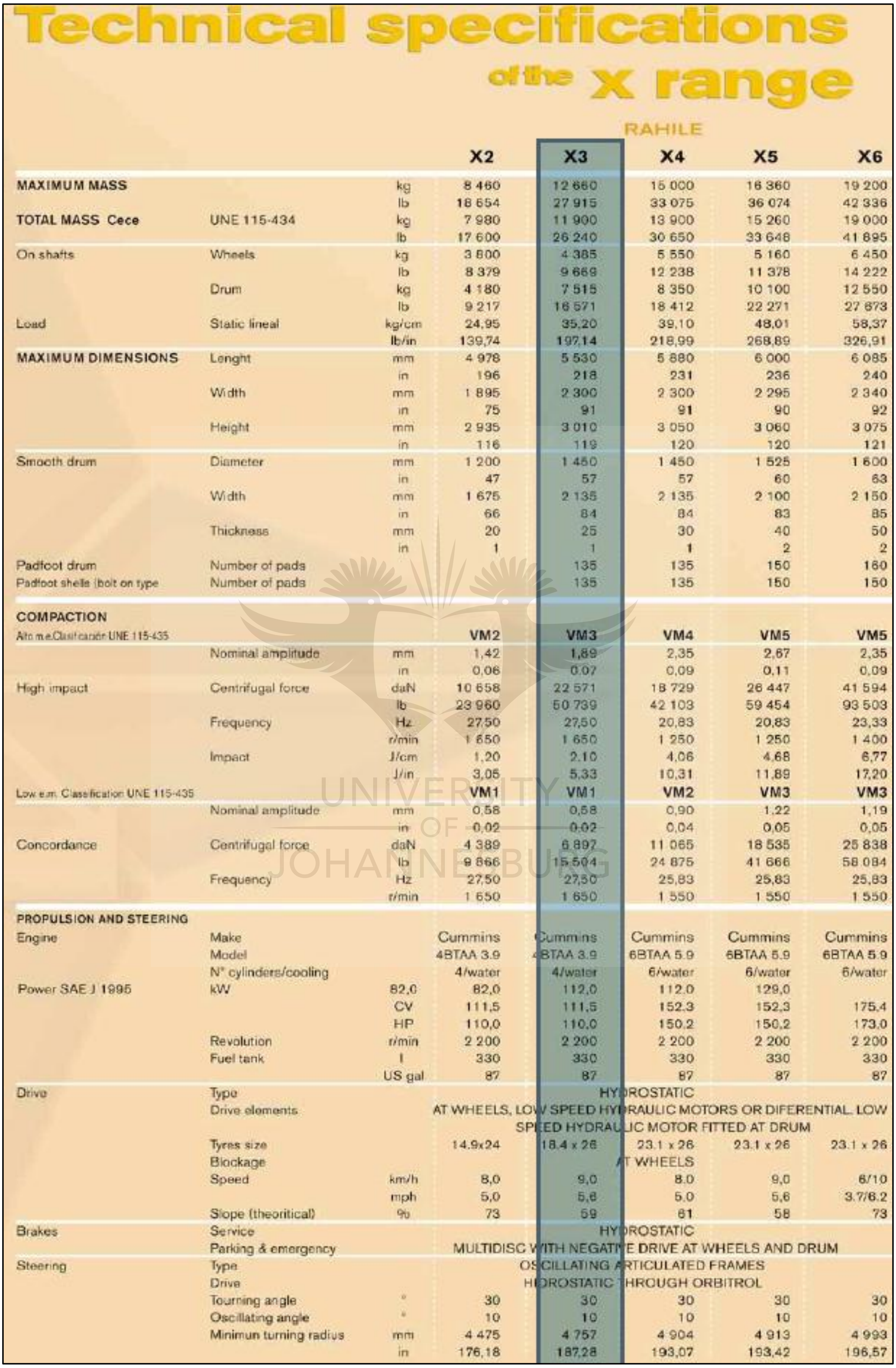
HEAD OFFICE: 1-4-8, SHIBA DAIMON, MINATO-KU,
TOKYO JAPAN
TELEPHONE: TOKYO (03) 3431-9971
FACSIMILE: TOKYO (03) 3436-6212

Printed in Japan 2005 07. 1C⑤⑤



VIBRATING POWER		L	H	L	H	L	H
Frequency	Hz (vpm)	36.7 (2,200)	27.5 (1,650)	36.7 (2,200)	27.5 (1,650)	36.7 (2,200)	27.5 (1,650)
Centrifugal force (Max.)	kN (kgf)	172 (17,500)	226 (23,000)	186 (19,000)	245 (25,000)	186 (19,000)	245 (25,000)
	lbs	38,581	50,706	41,887	55,115	41,887	55,115
Amplitude	mm	0.90	2.00	0.90	2.00	0.80	1.70

Appendix 17: Librero



Technical specifications

of the X range

RAHILE

			X2	X3	X4	X5	X6
MAXIMUM MASS		kg	8 460	12 660	15 000	16 360	19 200
		lb	18 654	27 915	33 075	36 074	42 336
TOTAL MASS Cece	UNE 115-434	kg	7 980	11 900	13 900	15 260	19 000
		lb	17 600	26 240	30 650	33 648	41 895
On shafts	Wheels	kg	3 800	4 385	5 550	5 160	6 450
		lb	8 379	9 669	12 238	11 378	14 222
	Drum	kg	4 180	7 515	8 350	10 100	12 550
		lb	9 217	16 571	18 412	22 271	27 673
Load	Static lineal	kg/cm	24,95	35,20	39,10	48,01	58,37
		lb/in	139,74	197,14	218,99	268,89	326,91
MAXIMUM DIMENSIONS	Lenght	mm	4 978	5 530	5 880	6 000	6 085
		in	196	218	231	236	240
	Width	mm	1 895	2 300	2 300	2 295	2 340
		in	75	91	91	90	92
	Height	mm	2 935	3 010	3 050	3 060	3 075
		in	116	119	120	120	121
Smooth drum	Diameter	mm	1 200	1 450	1 450	1 525	1 600
		in	47	57	57	60	63
	Width	mm	1 675	2 135	2 135	2 100	2 150
		in	66	84	84	83	85
	Thickness	mm	20	25	30	40	50
		in	1	1	1	2	2
Padfoot drum	Number of pads			135	135	150	160
Padfoot shells (bolt on type	Number of pads			135	135	150	150
COMPACTION							
Alto m.e.Clasificación UNE 115-435			**VM2**	**VM3**	**VM4**	**VM5**	**VM5**
	Nominal amplitude	mm	1,42	1,89	2,35	2,67	2,35
		in	0,06	0.07	0,09	0,11	0,09
High impact	Centrifugal force	daN	10 658	22 571	18 729	26 447	41 594
		lb	23 960	50 739	42 103	59 454	93 503
	Frequency	Hz	27,50	27,50	20,83	20,83	23,33
		r/min	1 650	1 650	1 250	1 250	1 400
	Impact	J/cm	1,20	2.10	4,06	4,68	6,77
		J/in	3,05	5,33	10,31	11,89	17,20
Low e.m. Classification UNE 115-435			**VM1**	**VM1**	**VM2**	**VM3**	**VM3**
	Nominal amplitude	mm	0,58	0,58	0,90	1,22	1,19
		in	0,02	0,02	0,04	0,05	0,05
Concordance	Centrifugal force	daN	4 389	6 897	11 065	18 535	25 838
		lb	9 866	15 504	24 875	41 666	58 084
	Frequency	Hz	27,50	27,50	25,83	25,83	25,83
		r/min	1 650	1 650	1 550	1 550	1 550
PROPULSION AND STEERING							
Engine	Make		Cummins	Cummins	Cummins	Cummins	Cummins
	Model		4BTAA 3.9	4BTAA 3.9	6BTAA 5.9	6BTAA 5.9	6BTAA 5.9
	N° cylinders/cooling		4/water	4/water	6/water	6/water	6/water
Power SAE J 1995	kW	82,0	82,0	112,0	112.0	129,0	
		CV	111,5	111,5	152.3	152,3	175.4
		HP	110,0	110.0	150.2	150,2	173.0
	Revolution	r/min	2 200	2 200	2 200	2 200	2 200
	Fuel tank	l	330	330	330	330	330
		US gal	87	87	87	87	87
Drive	Type		HYDROSTATIC				
	Drive elements		AT WHEELS, LOW SPEED HYDRAULIC MOTORS OR DIFERENTIAL. LOW SPEED HYDRAULIC MOTOR FITTED AT DRUM				
	Tyres size		14.9x24	18.4 x 26	23.1 x 26	23.1 x 26	23.1 x 26
	Blockage		AT WHEELS				
	Speed	km/h	8,0	9,0	8.0	9,0	6/10
		mph	5,0	5,6	5.0	5,6	3.7/6.2
	Slope (theoritical)	%	73	59	61	58	73
Brakes	Service		HYDROSTATIC				
	Parking & emergency		MULTIDISC WITH NEGATIVE DRIVE AT WHEELS AND DRUM				
Steering	Type		OSCILLATING ARTICULATED FRAMES				
	Drive		HIDROSTATIC THROUGH ORBITROL				
	Tourning angle	°	30	30	30	30	30
	Oscillating angle	°	10	10	10	10	10
	Minimun turning radius	mm	4 475	4 757	4 904	4 913	4 993
		in	176,18	187,28	193,07	193,42	196,57

Appendix 18: Ingersoll Rand.

Ingersoll Rand

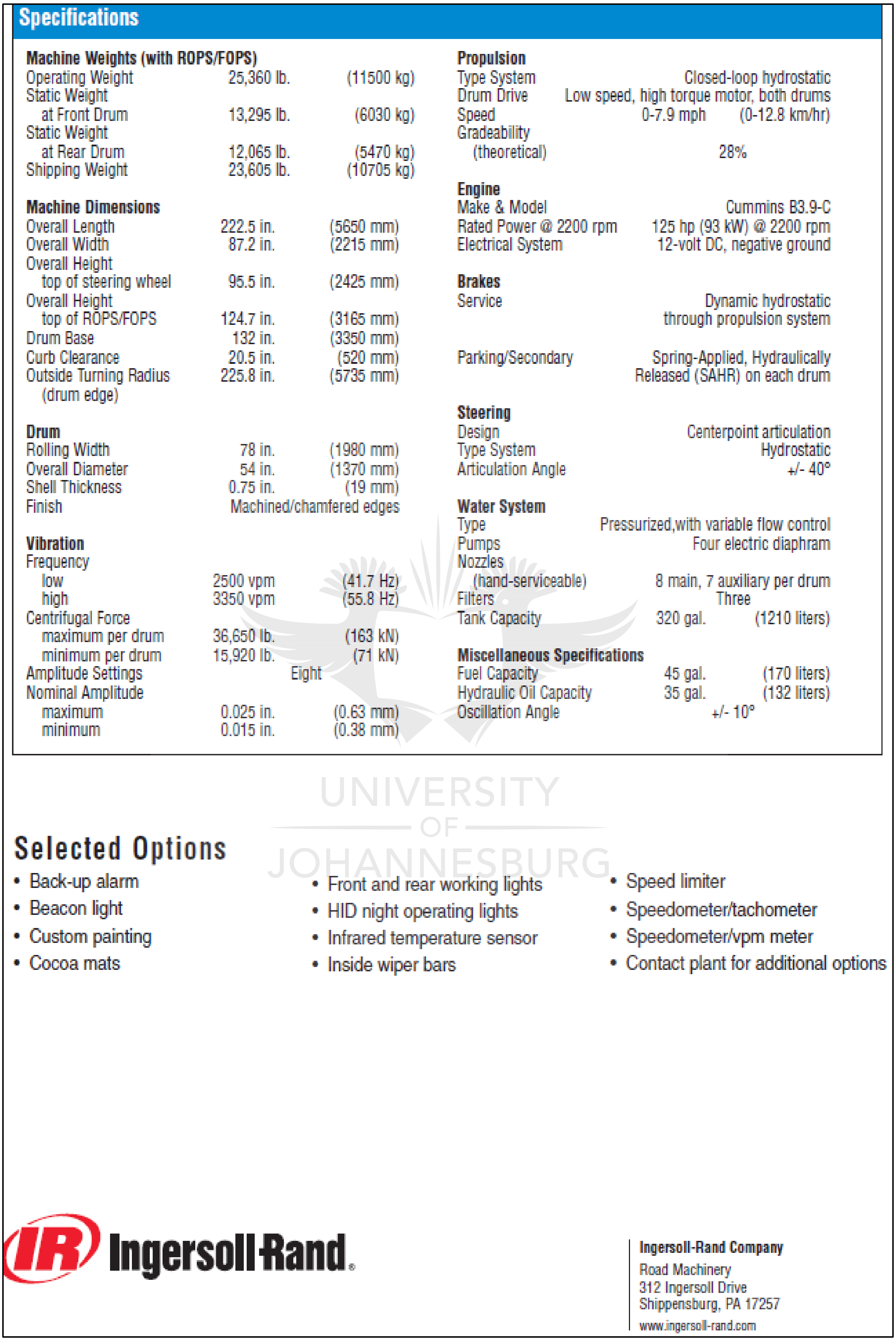
DD-110HF Vibratory Asphalt Compactor

The DD-110HF features dual 78 inch (1980mm) wide drums with 3350 vpm (55.8 Hz) frequency for fast rolling speeds and smooth surface finish. The unit provides high performance capacity for highway and other large paving jobs. Twelve and one-half tons (11.5 metric tonnes) of static weight meet D.O.T. static rolling requirements for increased machine utilization. The DD-110HF has a nominal capcity up to 380 tons (350 metric tonnes) of HMA per hour. A rolling width of 78 inches (1980 mm) permits coverage of 12 feet (3.66 meters) wide paved panels in two passes side-by-side. Drum diameter of 54 inches (1370 mm) reduces rolling resistance for superior performance on thin lifts, tender mixes or thick lifts of harsh mixes.

Standard Features

- Adjustable vibration on/off speeds of 0.5, 1.0, 1.5, 2.0 mph (0.8, 1.6, 2.4, 3.2 km/hr)
- Automatic reversing eccentric rotation
- Dual vibration frequency of 3350 vpm (55.8 Hz) and 2500 vpm (47.5 Hz)
- Eight-position variable amplitude settings
- Five-position rotating operator's module with adjustable seat
- Four-cylinder turbocharged and aftercooled diesel engine
- Machined drums, chamfered drum edges
- Polyethylene water tanks, 320 gallon (1210 liter) capacity
- Patented Impact Meter to provide consistent spacing of drum vibration impacts for uniform compaction of HMA
- Standard ROPS/FOPS and seat belt
- Two complete, independent water systems with triple water filtration, four water pumps four spray bars and variable water flow; automatic water control; water level indicator

Ingersoll Rand



Specifications

Machine Weights (with ROPS/FOPS)

Operating Weight	25,360 lb.	(11500 kg)
Static Weight at Front Drum	13,295 lb.	(6030 kg)
Static Weight at Rear Drum	12,065 lb.	(5470 kg)
Shipping Weight	23,605 lb.	(10705 kg)

Machine Dimensions

Overall Length	222.5 in.	(5650 mm)
Overall Width	87.2 in.	(2215 mm)
Overall Height top of steering wheel	95.5 in.	(2425 mm)
Overall Height top of ROPS/FOPS	124.7 in.	(3165 mm)
Drum Base	132 in.	(3350 mm)
Curb Clearance	20.5 in.	(520 mm)
Outside Turning Radius (drum edge)	225.8 in.	(5735 mm)

Drum

Rolling Width	78 in.	(1980 mm)
Overall Diameter	54 in.	(1370 mm)
Shell Thickness	0.75 in.	(19 mm)
Finish	Machined/chamfered edges	

Vibration

Frequency low	2500 vpm	(41.7 Hz)
Frequency high	3350 vpm	(55.8 Hz)
Centrifugal Force maximum per drum	36,650 lb.	(163 kN)
Centrifugal Force minimum per drum	15,920 lb.	(71 kN)
Amplitude Settings	Eight	
Nominal Amplitude maximum	0.025 in.	(0.63 mm)
Nominal Amplitude minimum	0.015 in.	(0.38 mm)

Propulsion

Type System	Closed-loop hydrostatic
Drum Drive	Low speed, high torque motor, both drums
Speed	0-7.9 mph (0-12.8 km/hr)
Gradeability (theoretical)	28%

Engine

Make & Model	Cummins B3.9-C
Rated Power @ 2200 rpm	125 hp (93 kW) @ 2200 rpm
Electrical System	12-volt DC, negative ground

Brakes

Service	Dynamic hydrostatic through propulsion system
Parking/Secondary	Spring-Applied, Hydraulically Released (SAHR) on each drum

Steering

Design	Centerpoint articulation
Type System	Hydrostatic
Articulation Angle	+/- 40°

Water System

Type	Pressurized,with variable flow control
Pumps	Four electric diaphram
Nozzles (hand-serviceable)	8 main, 7 auxiliary per drum
Filters	Three
Tank Capacity	320 gal. (1210 liters)

Miscellaneous Specifications

Fuel Capacity	45 gal.	(170 liters)
Hydraulic Oil Capacity	35 gal.	(132 liters)
Oscillation Angle	+/- 10°	

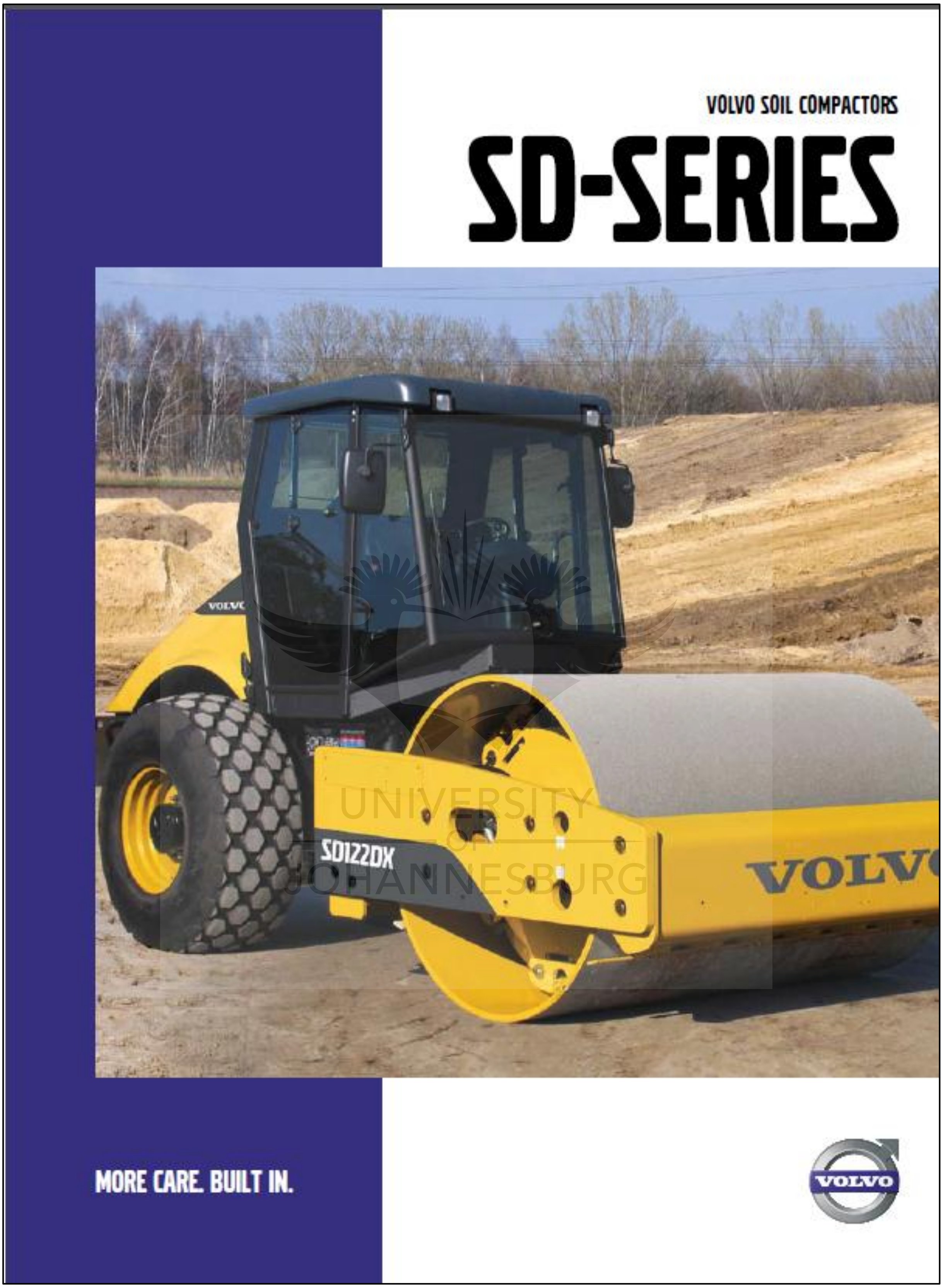
Selected Options

- Back-up alarm
- Beacon light
- Custom painting
- Cocoa mats
- Front and rear working lights
- HID night operating lights
- Infrared temperature sensor
- Inside wiper bars
- Speed limiter
- Speedometer/tachometer
- Speedometer/vpm meter
- Contact plant for additional options

Ingersoll Rand

Ingersoll-Rand Company
Road Machinery
312 Ingersoll Drive
Shippensburg, PA 17257
www.ingersoll-rand.com

Appendix 19: Volvo



VOLVO SOIL COMPACTORS

SD-SERIES

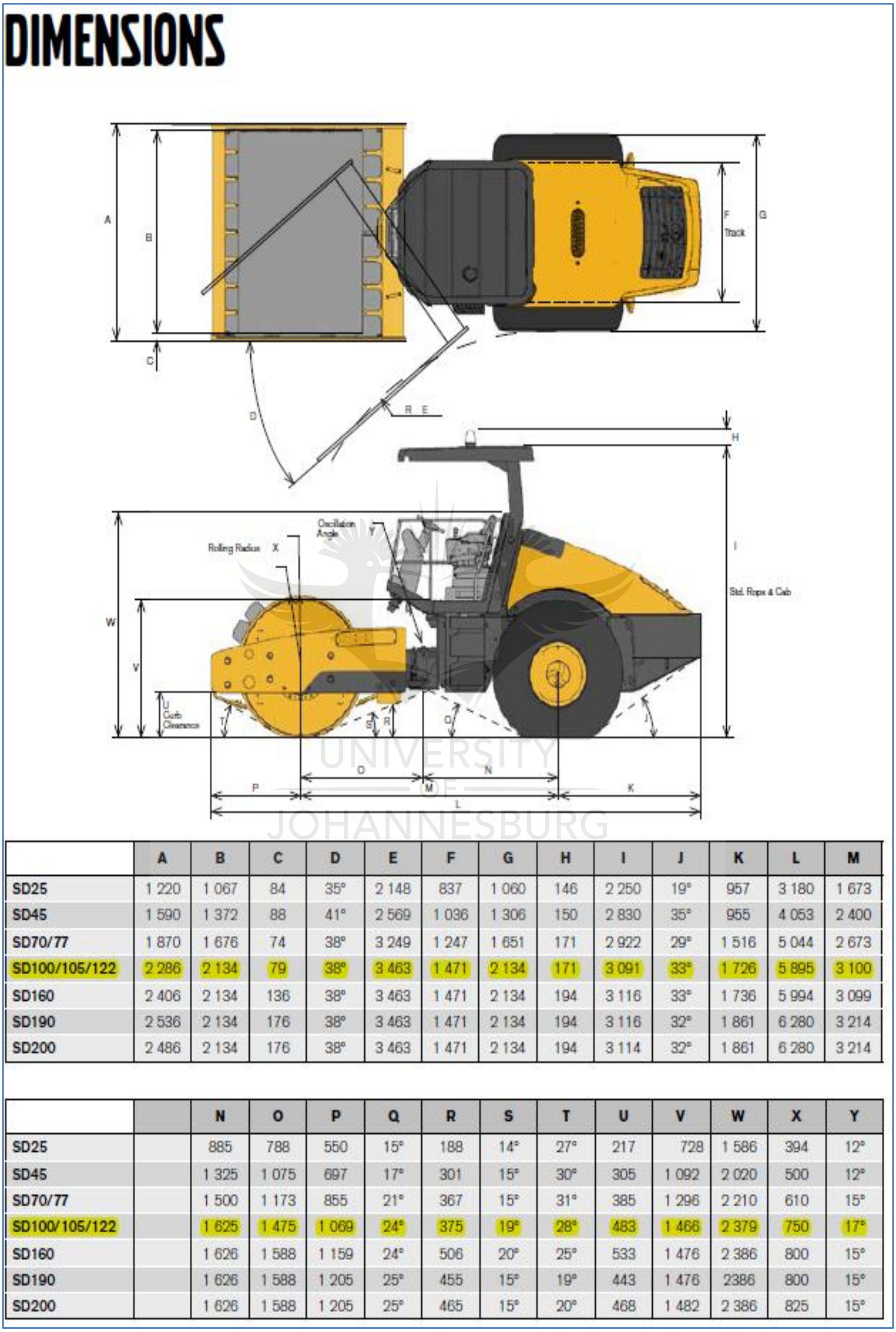
MORE CARE. BUILT IN.

Specifications

Model	Operating Weight (CECE)	Static Weight at Drum	Shipping Weight	Vibration frequeny	Centrifugal force	Nominal Amplitude	Engine Model	Engine Power	Travel Speed
SD25D	2 562 kg	1 292 kg	2 478 kg	37 Hz	62 kN	1,55 mm	Kubota V2203-M	32,8 kW at 2 450 rpm	0 - 9,0 km/h
SD25F	2 685 kg	1 415 kg	2 600 kg	37 Hz	56 kN	1,40 mm	Kubota V2203-M	32,8 kW at 2 450 rpm	0 - 9,6 km/h
SD45D	4 807 kg	2 449 kg	4 712 kg	32 Hz	99 kN	1,98 mm	Cummins 4BT 3.3	59,7 kW at 2 200 rpm	0 - 7,4 km/h
SD45F	5 147 kg	2 789 kg	4 893 kg	32 Hz	99 kN	1,71 mm	Cummins 4BT 3.3	59,7 kW at 2 200 rpm	0 - 7,7 km/h
SD70D	7 536 kg	3 740 kg	7 423 kg	30,8/33,8 Hz	143/104 kN	1,98/1,20 mm	Cummins 4BT	70,8 kW at 2 200 rpm	0 - 13,0 km/h 0 - 8,4 km/h
SD77DX	7 803 kg	4 012 kg	7 696 kg	20,4-33,8 Hz	143/104 kN	1,98/1,20 mm	Cummins 4BT	70,8 kW at 2 200 rpm	0 - 13,0 km/h 0 - 8,4 km/h
SD77F	8 456 kg	4 660 kg	8 344 kg	20,4-33,8 Hz	171/135 kN	1,98/1,33 mm	Cummins 4BT	70,8 kW at 2 200 rpm	0 - 11,3 km/h 0 - 7,7 km/h
SD100D	10 837 kg	6 085 kg	10 708 kg	31,2/33,6 Hz	264/206 kN	1,92/1,29 mm	Cummins QSB 4.5 - COM III	97 kW at 2 200 rpm	0 - 12,0 km/h 0 - 8,4 km/h
SD105DX	11 109 kg	6 357 kg	10 980 kg	20,4-33,6 Hz	264/206 kN	1,92/1,29 mm	Cummins QSB 4.5 - COM III	97 kW at 2 200 rpm	0 - 12,0 km/h 0 - 8,4 km/h
SD105F	11 744 kg	6 992 kg	11 615 kg	20,4-33,6 Hz	347/262 kN	2,14/1,39 mm	Cummins QSB 4.5 - COM III	97 kW at 2 200 rpm	0 - 11,3 km/h 0 - 8,1 km/h
SD122D	11 973 kg	6 939 kg	11 771 kg	30,8/33,7 Hz	281/206 kN	1,90/1,17 mm	Cummins QSB 4.5 - COM III	119 kW at 2 200 rpm	0 - 11,1 km/h 0 - 4,6 km/h
SD122DX	12 086 kg	6 927 kg	11 884 kg	20,4-33,7 Hz	281/206 kN	1,90/1,17 mm	Cummins QSB 4.5 - COM III	119 kW at 2 200 rpm	0 - 11,1 km/h 0 - 4,6 km/h
SD122F	12 857 kg	7 698 kg	12 655 kg	20,4-33,7 Hz	293/264 kN	2,03/1,31 mm	Cummins QSB 4.5 - COM III	119 kW at 2 200 rpm	0 - 11,8 km/h 0 - 4,8 km/h
SD160DX	16 391 kg	11 146 kg	16 255 kg	20,8-33,8 Hz	339/264 kN	1,97/1,28 mm	Cummins QSB 6.7 - COM III	129 kW at 2 200 rpm	0 - 12,4 km/h 0 - 4,5 km/h
SD160F	16 935 kg	11 690 kg	16 799 kg	20,8-33,8 Hz	339/257 kN	2,21/1,68 mm	Cummins QSB 6.7 - COM III	129 kW at 2 200 rpm	0 - 11,8 km/h 0 - 4,8 km/h
SD190DX	19 765 kg	12 749 kg	19 629 kg	20,8-30,8 Hz	339/221 kN	1,97/1,28 mm	Cummins QSB 6.7 - COM III	153 kW at 2 000 rpm	0 - 12,4 km/h 0 - 4,6 km/h
SD200DX	20 763 kg	13 747 kg	20 626 kg	20,8-30,8 Hz	368/239 kN	1,76/1,14 mm	Cummins QSB 6.7 - COM III	151 kW at 2 000 rpm	0 - 11,5 km/h 0 - 4,0 km/h
SD200F	20 536 kg	13 520 kg	20 400 kg	20,8-28,3 Hz	359/273 kN	1,99/1,51 mm	Cummins QSB 6.7 - COM III	151 kW at 2 000 rpm	0 - 11,9 km/h 0 - 4,1 km/h

Model	Drum Width	Drum Diameter	Shell Thickness	Diameter over Pad feet	Number of Pad feet	Pad Height	Pad Tip Area	Gradeability (Theor.)	Fuel Capacity
SD25D	1 067 mm	787 mm	12 mm	-	-	-	-	60 %	45 l
SD25F	1 067 mm	787 mm	12 mm	914 mm	45	64 mm	55 cm^2	50 %	45 l
SD45D	1 372 mm	1 000 mm	20 mm	-	-	-	-	54 %	98 l
SD45F	1 372 mm	1 000 mm	20 mm	1 153 mm	63	76 mm	125 cm^2	49 %	98 l
SD70D	1 676 mm	1 219 mm	22 mm	-	-	-	-	63 %	178 l
SD77DX	1 676 mm	1 219 mm	22 mm	-	-	-	-	70 %	178 l
SD77F	1 676 mm	1 219 mm	22 mm	1 372 mm	84	76 mm	125 cm^2	70 %	178 l
SD100D	2 134 mm	1 499 mm	25 mm	-	-	-	-	55 %	267 l
SD105DX	2 134 mm	1 499 mm	25 mm	-	-	-	-	62 %	267 l
SD105F	2 134 mm	1 499 mm	25 mm	1 702 mm	120	102 mm	125 cm^2	60 %	267 l
SD122D	2 134 mm	1 499 mm	30 mm	-	-	-	-	68 %	267 l
SD122DX	2 134 mm	1 499 mm	30 mm	-	-	-	-	75 %	267 l
SD122F	2 134 mm	1 499 mm	30 mm	1 753 mm	120	127 mm	125 cm^2	71 %	267 l
SD160DX	2 134 mm	1 600 mm	35 mm	-	-	-	-	65 %	267 l
SD160F	2 134 mm	1 600 mm	30 mm	1 854 mm	120	127 mm	125 cm^2	55 %	267 l
SD190DX	2 134 mm	1 600 mm	35 mm	-	-	-	-	60 %	257 l
SD200DX	2 134 mm	1 651 mm	40 mm	-	-	-	-	59 %	267 l
SD200F	2 134 mm	1 651 mm	25 mm	1 875 mm	136	127 mm	125 cm^2	49 %	267 l

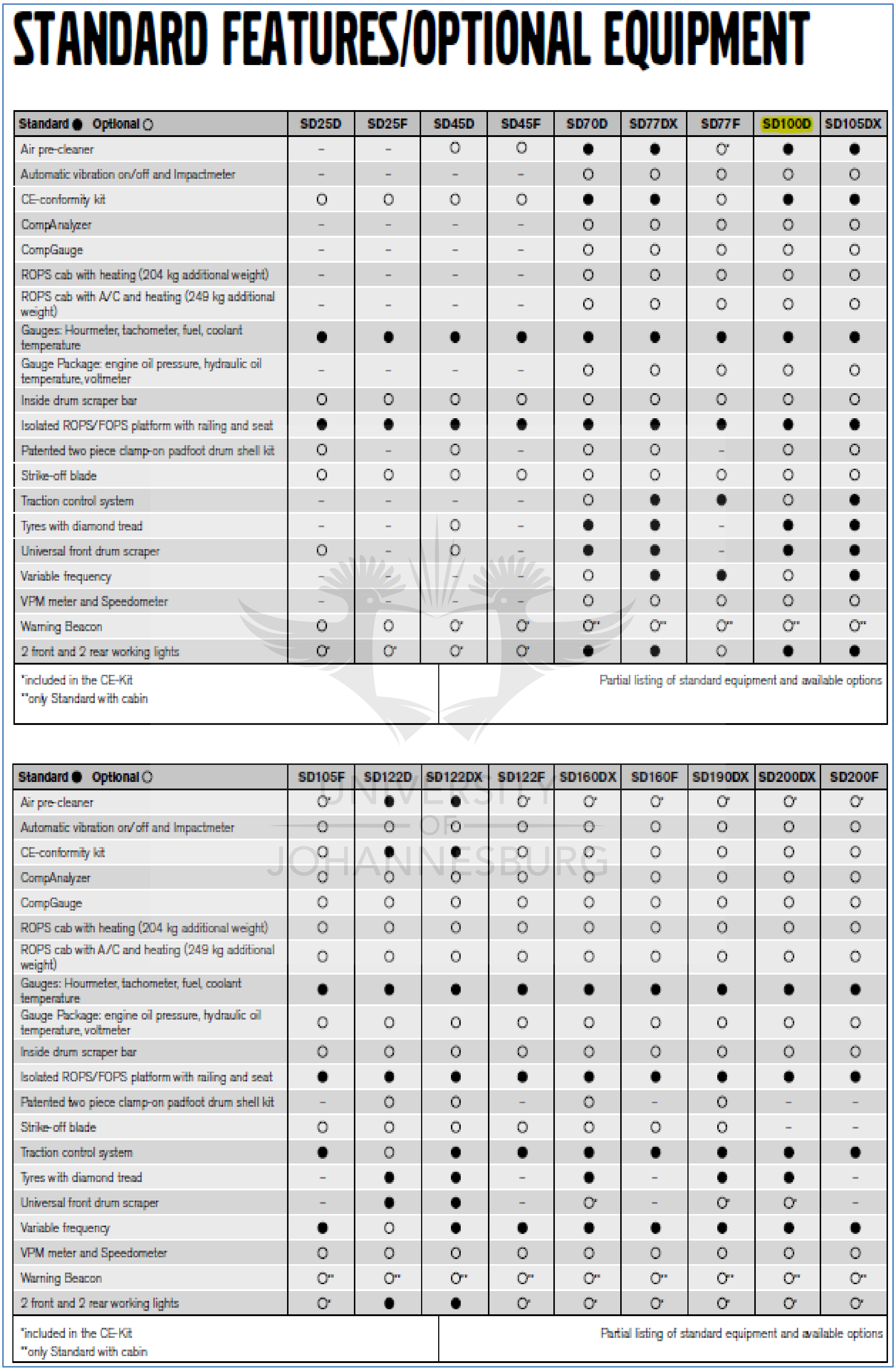
Product improvement is a continuing goal at Volvo. Designs and specifications are subject to change without notice or obligation.



DIMENSIONS

	A	B	C	D	E	F	G	H	I	J	K	L	M
SD25	1 220	1 067	84	35°	2 148	837	1 060	146	2 250	19°	957	3 180	1 673
SD45	1 590	1 372	88	41°	2 569	1 036	1 306	150	2 830	35°	955	4 053	2 400
SD70/77	1 870	1 676	74	38°	3 249	1 247	1 651	171	2 922	29°	1 516	5 044	2 673
SD100/105/122	2 286	2 134	79	38°	3 463	1 471	2 134	171	3 091	33°	1 726	5 895	3 100
SD160	2 406	2 134	136	38°	3 463	1 471	2 134	194	3 116	33°	1 736	5 994	3 099
SD190	2 536	2 134	176	38°	3 463	1 471	2 134	194	3 116	32°	1 861	6 280	3 214
SD200	2 486	2 134	176	38°	3 463	1 471	2 134	194	3 114	32°	1 861	6 280	3 214

		N	O	P	Q	R	S	T	U	V	W	X	Y
SD25		885	788	550	15°	188	14°	27°	217	728	1 586	394	12°
SD45		1 325	1 075	697	17°	301	15°	30°	305	1 092	2 020	500	12°
SD70/77		1 500	1 173	855	21°	367	15°	31°	385	1 296	2 210	610	15°
SD100/105/122		1 625	1 475	1 069	24°	375	19°	28°	483	1 466	2 379	750	17°
SD160		1 626	1 588	1 159	24°	506	20°	25°	533	1 476	2 386	800	15°
SD190		1 626	1 588	1 205	25°	455	15°	19°	443	1 476	2386	800	15°
SD200		1 626	1 588	1 205	25°	465	15°	20°	468	1 482	2 386	825	15°



STANDARD FEATURES/OPTIONAL EQUIPMENT

Standard ● Optional ○	SD25D	SD25F	SD45D	SD45F	SD70D	SD77DX	SD77F	SD100D	SD105DX
Air pre-cleaner	–	–	○	○	●	●	○*	●	●
Automatic vibration on/off and Impactmeter	–	–	–	–	○	○	○	○	○
CE-conformity kit	○	○	○	○	●	●	○	●	●
CompAnalyzer	–	–	–	–	○	○	○	○	○
CompGauge	–	–	–	–	○	○	○	○	○
ROPS cab with heating (204 kg additional weight)	–	–	–	–	○	○	○	○	○
ROPS cab with A/C and heating (249 kg additional weight)	–	–	–	–	○	○	○	○	○
Gauges: Hourmeter, tachometer, fuel, coolant temperature	●	●	●	●	●	●	●	●	●
Gauge Package: engine oil pressure, hydraulic oil temperature, voltmeter	–	–	–	–	○	○	○	○	○
Inside drum scraper bar	○	○	○	○	○	○	○	○	○
Isolated ROPS/FOPS platform with railing and seat	●	●	●	●	●	●	●	●	●
Patented two piece clamp-on padfoot drum shell kit	○	–	○	–	○	○	–	○	○
Strike-off blade	○	○	○	○	○	○	○	○	○
Traction control system	–	–	–	–	○	●	●	○	●
Tyres with diamond tread	–	–	○	–	●	●	–	●	●
Universal front drum scraper	○	–	○	–	●	●	–	●	●
Variable frequency	–	–	–	–	○	●	●	○	●
VPM meter and Speedometer	–	–	–	–	○	○	○	○	○
Warning Beacon	○	○	○*	○*	○**	○**	○**	○**	○**
2 front and 2 rear working lights	○*	○*	○*	○*	●	●	○	●	●

*included in the CE-Kit
**only Standard with cabin

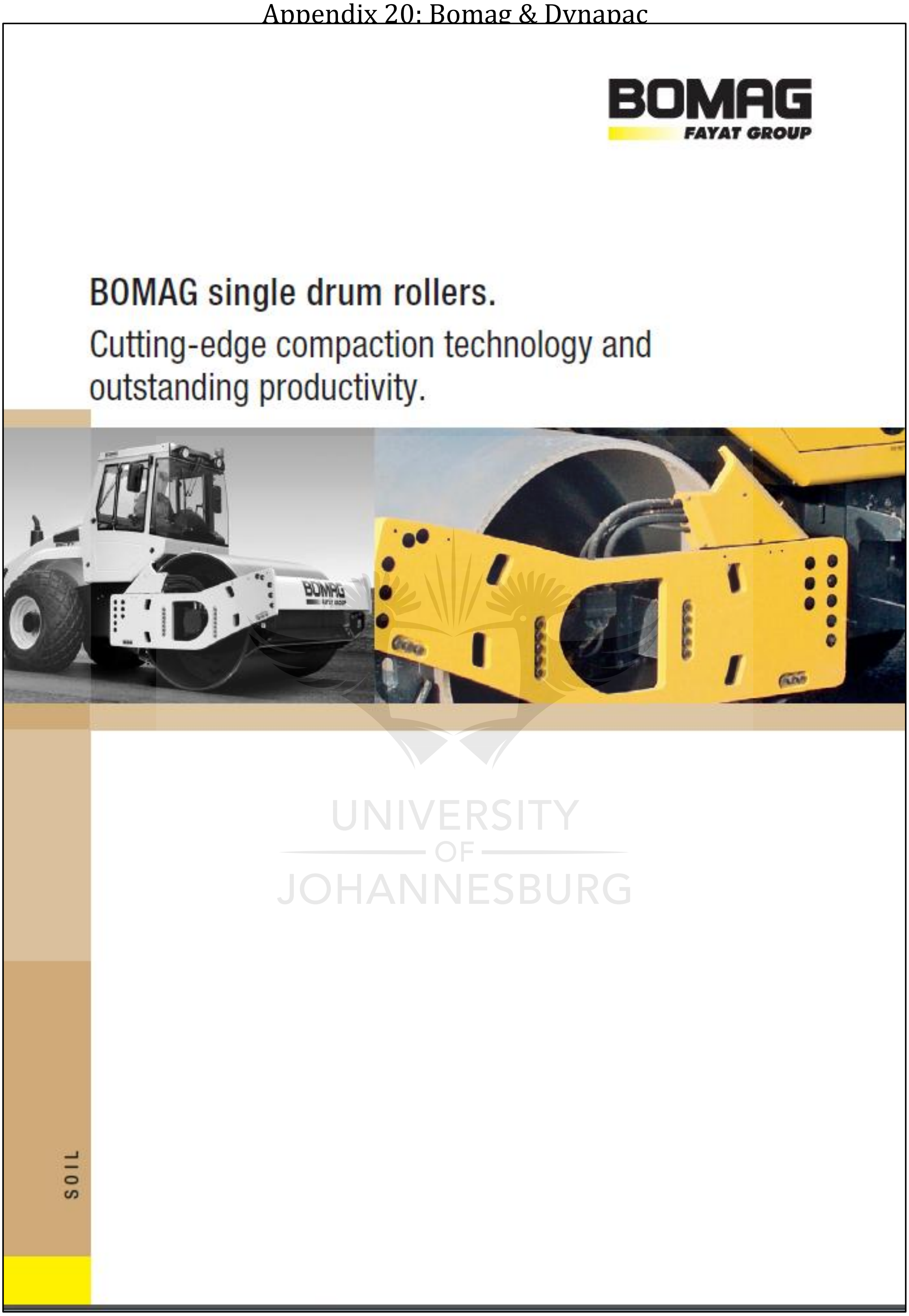
Partial listing of standard equipment and available options

Standard ● Optional ○	SD105F	SD122D	SD122DX	SD122F	SD160DX	SD160F	SD190DX	SD200DX	SD200F
Air pre-cleaner	○*	●	●	○*	○*	○*	○*	○*	○*
Automatic vibration on/ off and Impactmeter	○	○	○	○	○	○	○	○	○
CE-conformity kit	○	●	●	○	○	○	○	○	○
CompAnalyzer	○	○	○	○	○	○	○	○	○
CompGauge	○	○	○	○	○	○	○	○	○
ROPS cab with heating (204 kg additional weight)	○	○	○	○	○	○	○	○	○
ROPS cab with A/C and heating (249 kg additional weight)	○	○	○	○	○	○	○	○	○
Gauges: Hourmeter, tachometer, fuel, coolant temperature	●	●	●	●	●	●	●	●	●
Gauge Package: engine oil pressure, hydraulic oil temperature, voltmeter	○	○	○	○	○	○	○	○	○
Inside drum scraper bar	○	○	○	○	○	○	○	○	○
Isolated ROPS/FOPS platform with railing and seat	●	●	●	●	●	●	●	●	●
Patented two piece clamp-on padfoot drum shell kit	–	○	○	–	○	–	○	–	–
Strike-off blade	○	○	○	○	○	○	○	–	–
Traction control system	●	○	●	●	●	●	●	●	●
Tyres with diamond tread	–	●	●	–	●	–	●	●	–
Universal front drum scraper	–	●	●	–	○*	–	○*	○*	–
Variable frequency	●	○	●	●	●	●	●	●	●
VPM meter and Speedometer	○	○	○	○	○	○	○	○	○
Warning Beacon	○**	○**	○**	○**	○**	○**	○**	○**	○**
2 front and 2 rear working lights	○*	●	●	○*	○*	○*	○*	○*	○*

*included in the CE-Kit
**only Standard with cabin

Partial listing of standard equipment and available options

Appendix 20: Bomag & Dynapac



BOMAG
FAYAT GROUP

BOMAG single drum rollers.

Cutting-edge compaction technology and outstanding productivity.

SOIL

Giving you the competitive edge!

Single drum rollers play a key role on site. They create a firm base, quite literally, for subsequent progress. There's no question of compromise – BOMAG has met in full the demands of site managers using the latest technology for the BW 177 to BW 332 models in the -4 series.

- **Profitable:** Lowest cost per cubic metre is the key measure of success. BOMAG single drum rollers demonstrate how efficient soil compaction is done.
- **Powerful.** Maximum compaction using high linear loads and amplitudes.
- **Cost-effective.** BOMAG ECOMODE is synonymous with the lowest fuel consumption.
- **Intelligent.** Automatic compaction control with VARIOCONTROL.
- **Innovative.** Unique E_{VIB} measuring technology, integrated vibratory plates and special drums: BOMAG sets the pace.
- **Smart designs.** Comfortable workplace and easy servicing. BOMAG single drum rollers were developed together with people on site.
- **Flexible.** A range weighing from 7 to 32 tonnes with the widest choice of options.
- **Best resale values.** Quality speaks loudest: years of hard work and still in demand.

Single Drum Soil Compactors

Full range

DYNAPAC
Part of the Atlas Copco Group

Based on experience

Dynapac is the world's most specialized and experienced manufacturer of compaction and paving equipment. This solid base of expertise is one of the reasons for our numerous successful innovations. To put it simply, we know this business and we've got the power to transform groundbreaking ideas into cost-efficient solutions and reliable machines.

That is why Dynapac is a winner when you compare overall profitability and life-cycle cost. Within our lean and goal-oriented organization there are very short and straight paths between development, manufacturing and our world-wide service network. Your benefit is quality throughout - in products, maintenance,service and overall performance.

In this booklet we describe the basic characteristics of our single-drum soil compactors. A full range of highly efficient vibratory rollers that will make your project more profitable, and strengthen your reputation as a trustworthy working partner.

Welcome to explore the world of Dynapac.

▶ Expertise in every aspect

Experience is the base of excellence. Dynapac will always help you to reach perfection.

Get to know the CA family

STD	Smooth drum without drum drive
D	Smooth drum with drum drive
P	Padfoot drum without drum drive
PD	Padfoot drum with drum drive

CA134D/PD
CA144D/PD

The CA134 - 144 are vibratory rollers designed for compaction operations in pipe trenches, on road shoulders and in cramped spaces in connection with refilling work. The rollers are also suitable for repair work. The D version gives good maneuverability even on very steep slopes. The PD version, equipped with pads and drum drive, is especially suitable for the compaction of silt and clayey soils.

	Drum width, mm	Operating mass (incl.ROPS), kg	Static linear load, kg/cm	Amplitude, mm	Frequency, Hz
CA134D	1 370	4 550	13 (15)*	1,7	35
CA134PD	1 370	4 750	-	1,5	35
CA144D	1 676	4 800	12,2	1,5	35
CA144PD	1 676	5 000	-	1,3	35

** with heavy front beam*

CA150/D/P/PD

This is a vibratory roller designed for compacting roads, streets, parking lots and pipe trenches. Due to the small size and exceptional maneuverability, this roller is also well suited for compaction on large building foundations and industrial construction sites.

	Drum width, mm	Operating mass (incl.ROPS), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA150	1 676	7 000	20,9	1,7/0,8	31/43
CA150D	1 676	7 200	22,1	1,7/0,8	31/43
CA150P	1 676	7 400	-	1,7/0,9	31/43
CA150PD	1 676	7 500	-	1,7/0,9	31/43

CA152/D/PD
CA182D/PD

Designed for lighter operations on streets, roads and parking lots. The small dimensions and excellent manoeuvrability makes it very suitable for compaction work on large building foundations and industrial premises. All types of supporting and reinforcement courses can be compacted.

	Drum width, mm	Operating mass (incl.cab), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA152	1 676	7 300	21,8	1,7/0,8	31/43
CA152D	1 676	7 550	22,4	1,7/0,8	31/43
CA152PD	1 676	7 900	-	1,7/0,9	31/43
CA182D	1 676	8 900	30	1,9/0,9	31/31
CA182PD	1 676	9 000	-	1,8/0,9	31/31

CA250/D/P/PD
CA260/D/P/PD
CA280/D
CA300/D

CA250 and CA260 are typical utility machines, designed for long working days in tough applications. They are utilized for compaction of most types of soil. Typical applications are road building, airfields, dams, harbors and industrial sites.

CA280 and CA300 are medium heavy vibratory soil compactors. All types of base courses and reinforcement courses can be compacted to considerable depth.

	Drum width, mm	Operating mass (incl.ROPS), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA250 /-II	2 130	10 000	25,4	1,7/0,8	33/33
CA250D / -II	2 130	10 200	26,3	1,7/0,8	33/33
CA250P/ -II	2 130	11 400	-	1,6/0,8	33/33
CA250PD/ -II	2 130	11 600	-	1,6/0,8	33/33
CA260 / -II	2 130	10 700	26,3	1,7/0,8	33/33
CA260D / -II	2 130	10 700	26,3	1,7/0,8	33/33
CA260P / -II	2 130	10 900	-	1,6/0,8	33/33
CA260PD / -II	2 130	12 100	-	1,6/0,8	33/33
CA280	2 130	12 300	31,9	1,7/0,8	33/33
CA280D / -II	2 130	12 500	32,9	1,7/0,8	33/33
CA300	2 130	12 300	36,8	1,7/0,8	33/33
CA300D	2 130	12 550	38	1,7/0,8	33/33

TECHNICAL DATA FOR SINGLE DRUM SOIL COMPACTORS

CA252/D
CA262D/PD
CA282/D
CA302D/PD
CA362D/PD
CA402D

Dynapac's medium-range vibratory soil compactors. The padfoot version has it's major range of application on cohesive material and disintegrated rock. All types of base courses and subbase courses can be compacted.

The CA302/362 is ideal for a wide range of applications. The 35 mm hitch drum ensures excellent resistance to wear - even in compaction operations on rockfill. A pad-drum version is available for fine-grained material such as silt, clay or mixed soils.

	Drum width, mm	Operating mass (incl.cab), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA252	2 130	10 250	27,4	1,7 / 0,8	33/33
CA252D	2 130	10 450	28,4	1,7 / 0,8	33/33
CA262D	2 130	10 700	26,3	1,7 / 0,8	33/33
CA262PD	2 130	12 100	-	1,6 / 0,8	33/33
CA282D	2 130	12 200	32	1,7 / 0,8	33/33
CA302D	2 130	12 700	38	1,7 / 0,8	33/33
CA302PD	2 130	12 650	-	1,6 / 0,8	33/33
CA362D	2 130	13 200	38	1,7 / 0,8	33/33
CA362PD	2 130	13 100	-	1,6 / 0,8	33/33
CA402D	2 130	13 800	43,7	1,7 / 0,8	33/33

CA500D/PD
CA600D/PD
CA610D/PD

Three of the heavy machines in the Dynapac vibratory soil compactor range, providing full rock fill capacity. These machines cope with the most heavy-duty type of compaction work on blasted rock and most types of soil and clay masses. Typical applications are road construction, airfields, dams, and major harbor installations. Drum drive comes as standard.

The D-versions are designed for compacting rock fill. The main range of application for the PD versions is on cohesive material and weathered stone material. All types of base courses and subbase courses can be compacted deeper and the interchangeable drums, D to PD, and vice versa, facilitate even greater variety in the range of application.

	Drum width, mm	Operating mass (incl.ROPS), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA500D	2 130	15 600	49,3	1,8/1,1	29/33
CA500PD	2 130	15 800	-	1,7/1,0	29/33
CA600D	2 130	18 300	56,3	1,8/1,1	29/33
CA600PD	2 130	18 500	-	1,7/1,0	29/33
CA610D	2 130	20 600	66	1,8/1,1	29/33
CA610PD	2 130	20 600	-	1,8/1,1	29/33

CA5000D/PD
CA6000D/PD
CA6500D/PD

The CA5000, CA6000 and CA6500 are heavy rollers designed for the toughest compaction applications. Rockfill can be compacted in 2-meter thick layers, in which the size of the rocks can be up to 1 meter in diameter. The smooth drum shell thickness is 43 (CA512) and 48 mm (CA 6000 and CA6500), which gives a long productive lifetime for compaction of rockfill, gravel and sand. Pad-foot drum is available for compaction of silt and clay materials. These rollers are a great investment for the bigger projects as compaction performance and capacity are outstanding.

	Drum width, mm	Operating mass (incl.cab), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA5000D	2 130	16 000	50	2,1/0,8	29/30
CA5000PD	2 130	16 300	-	1,9/1,0	29/30
CA6000D	2 130	19 300	60	2,1/0,8	29/30
CA6000PD	2 130	19 100	-	2,1/0,8	29/30
CA6500D	2 130	20 700	65	2,1/0,8	29/30
CA6500PD	2 130	20 600	-	2,1/0,8	29/30

CA702D/PD

The CA702 is Dynapac's heaviest vibratory soil compaction roller. The machine has been specially developed for the heaviest large-scale compaction work on earth, rockfill and most types of soils and clays. Typical applications include dams, airfields, harbors and major railway and road projects.

	Drum width, mm	Operating mass (incl.cab), kg	Static linear load, kg/cm	Amplitude, Hi/low, mm	Frequency, Hi/low, Hz
CA702D	2 130	26 900	81,2	2,0/1,3	28/30
CA702PD	2 130	26 900	-	2,0/1,3	28/30

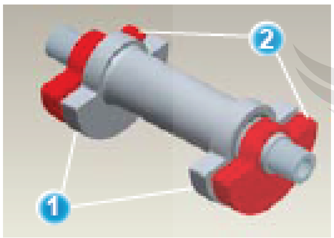
Traditionally innovative

Optimize instead of maximize

We all know that the whole idea with compaction is to reach the correct set of parameters for the type of work in question. There is no point in overdoing anything – it only costs time and fuel, without improving the final result.

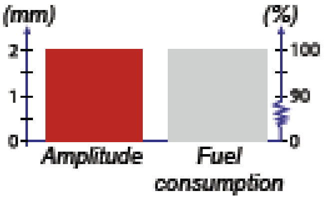
Dynapac Compaction Optimizer, DCO, is an innovative system based on the well-proven compaction meter. The stiffness of the ground constitutes the input value for the setting of amplitude of the vibratory drum. The operator gets full control and the project benefits from this in every respect.

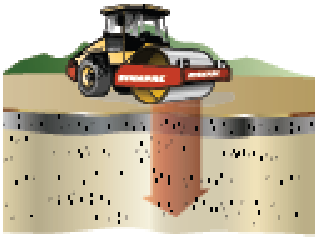
Soft ground



When the adjustable weight (2) is rotated so that it goes to the position shown in the picture above there is an amplitude of 100%.

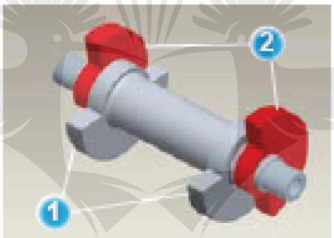
Amplitude (%)





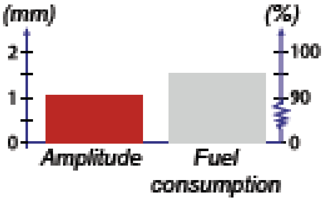
All the dynamic energy from the drum can be absorbed by the ground.

Hard ground



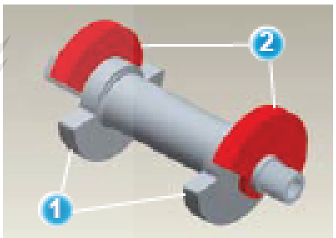
When the adjustable weight (2) is rotated so that it is in the position shown in the picture above there is an amplitude of 50%.

Amplitude (%)



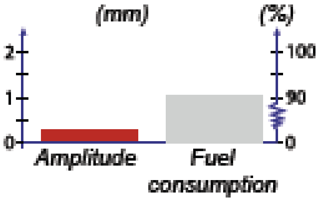
The drum readjusts so that less dynamic energy is supplied to the ground from the drum.

Ready compacted ground



When the adjustable weight (2) is rotated so that it is in the position shown in the picture above there is a minimum amplitude so that the forces almost balance each other out.

Amplitude (%)

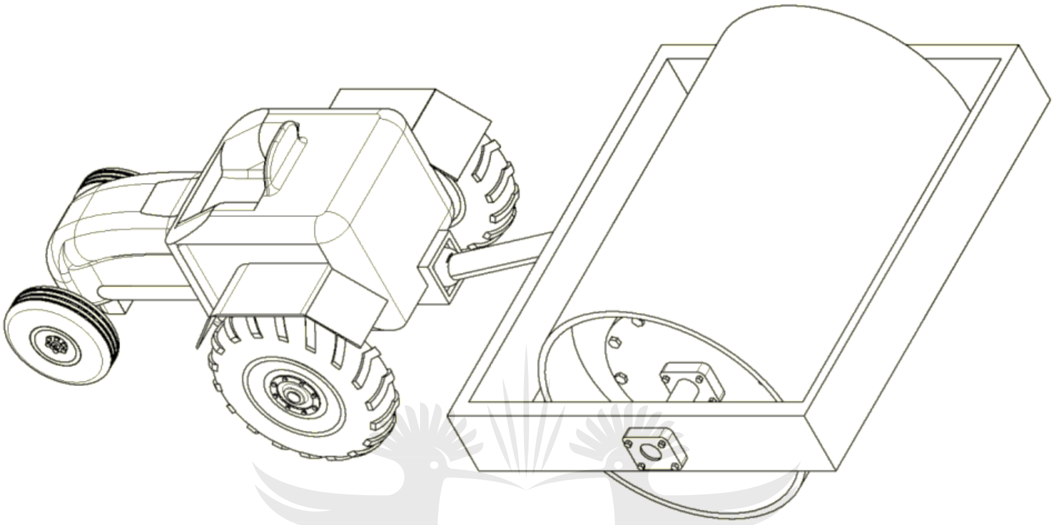


The drum is reset so that there is only a very small amount of dynamic energy supplied to the ground.

Appendix 21: Machine concepts.

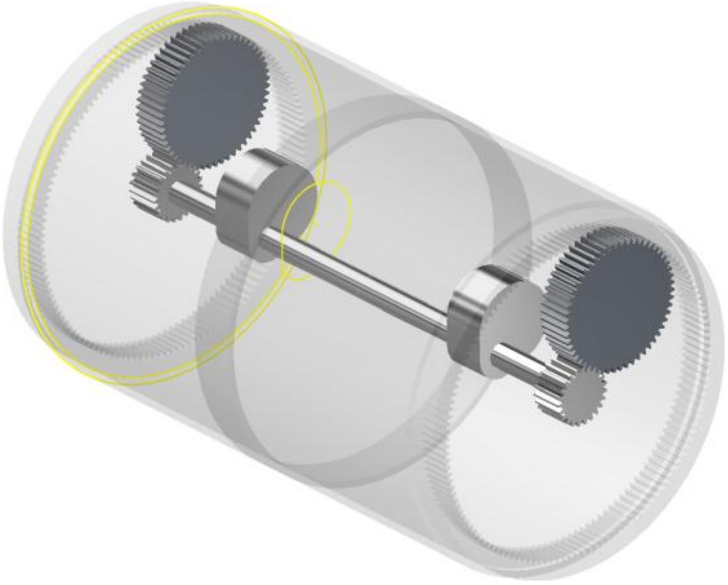
Single Drum Road Roller Concept

The single drum concept is the actual concept that has been perfected in this design project. Due to the simplicity of this concept, this type of design was made to work by overcoming all obstacles mentioned in this dissertation.



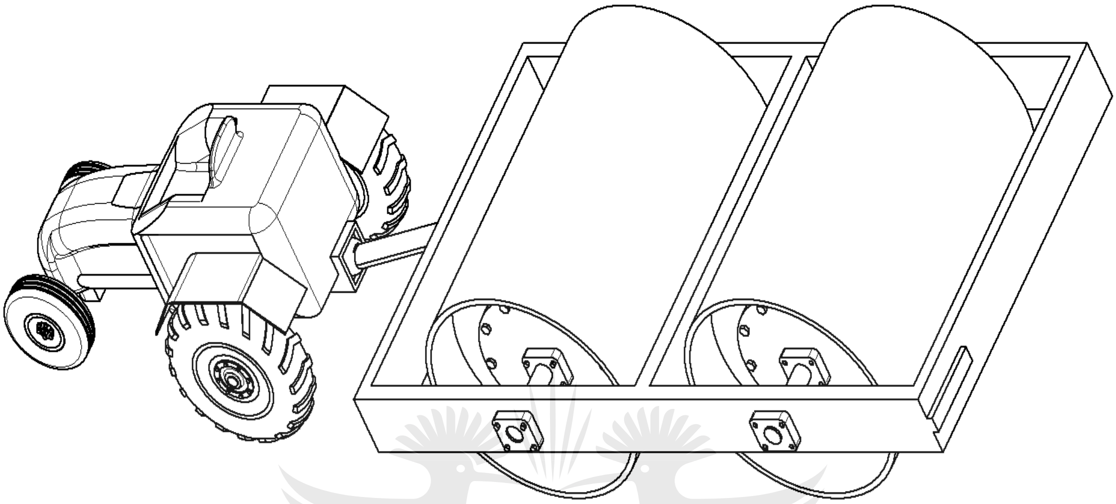
Single Drum Traction Excitation

This concept of the machine needs to have the circumference of the compacting drum continuously in contact with the road surface to promote traction. Since the machine is most of the time in the air and it touches the road surface for a very short time, adequate traction cannot be achieved and therefore will not work. The drum has to touch the road surface all the time to produce torque to excite the centrifugal weights on the MS.



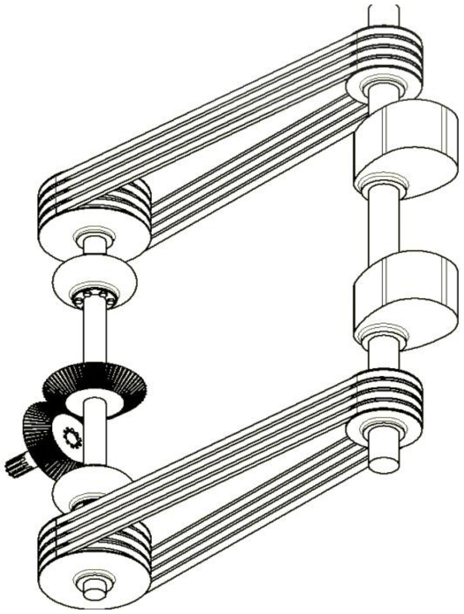
Double Drum Road Roller Concept

The double drum concept was to be used with the traction concept. By having two drums that can alternate during operation, the production efficiency can basically be doubled. While one drum is in the air, the other will be on the road surface, but at a certain point, both drums will be passing one another in the air and no traction can be generated at this stage. The concept was then abandoned.



Ideal Concept line diagram.

The Ideal concept is explained in detail in 9.6.1. Due to readily available gearboxes that are stepped down to very low ratios, this idea could not be implemented without designing a 1:1 ratio gearbox from scratch. Appendix 5 shows all relevant calculation for such a gearbox.



Appendix 22: Zeromax reversible speed drives.

Unidirectional Drives

E Models
1, 2, 41 or 42
Torque Rating 12in.lbs.
Speed Range 0-400.
Normal Input 1/4-1/3 H.P.

JK Models
1, 2, 41 or 42
Torque Rating 25in.lbs.
Speed Range 0-400.
Normal Input 1/4-1/3 H.P.

Y Models
1, 2, 41, or 42
Torque Rating 60in.lbs.
Speed Range 0-400.
Normal Input 1/2 H.P.

QX Models
1, 2, 41 or 42
Torque Rating 100in.lbs.
Speed Range 0-400.
Normal Input 3/4 H.P.

ZX Models
1, 2, 41 or 42
Torque Rating 200in.lbs.
Speed Range 0-400.
Normal Input 1-1/2 H.P.

Reversible Drives

E Model 3
Torque Rating 12in.lbs.
Speed Range 400-0-400.
Normal Input 1/4-1/3 H.P.

JK Model 3
Torque Rating 25in.lbs.
Speed Range 400-0-400.
Normal Input 1/4-1/3 H.P.

Y Model 3
Torque Rating 60in.lbs.
Speed Range 0-400.
Normal Input 1/2 H.P.

Gearhead

Right angle gearheads
available for E and JK Models.
Right Angle - 4 Models
W1 4:1 W2 10:1
W3 20:1 W4 40:1

Motors

Many popular voltage, Hz,
phase and enclosures are
available for use with drive.
E Models 1, 2, 3/ JK
Models 1, 2 and 3

Appendix 23: Stress Analysis Report for Caterpillar eccentric weight.

Analyzed File:	Caterpillar Eccentric weight.ipt
Autodesk Inventor Version:	2012 (Build 160160000, 160)
Creation Date:	2013/02/09, 03:09 AM
Simulation Author:	Terence Miller
Summary:	

Summary

Title	Compactor
Author	tmiller
Company	UJ

Project

Part Number	Eccentric weight
Designer	tmiller
Date Created	2013/02/09

Status

Design Status	Work In Progress

Custom

d0	122.000 mm
d1	166.364 mm
d2	299.455 mm

Physical

Material	Steel, Mild
Density	7.86 g/cm^3
Mass	109.367 kg
Area	712716 mm^2
Volume	13914400 mm^3
Centre of Gravity	x=1.7723 mm y=-46.5359 mm z=96.4909 mm

Simulation:1

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2013/02/09, 03:06 AM
Detect and Eliminate Rigid Body Modes	No

Advanced settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

Name	Steel, Mild	
General	Mass Density	7.86 g/cm^3
	Yield Strength	207 MPa
	Ultimate Tensile Strength	345 MPa
Stress	Young's Modulus	220 GPa
	Poisson's Ratio	0.275 ul
	Shear Modulus	86.2745 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	56 W/(m K)
	Specific Heat	460 J/(kg c)
Part Name(s)	Eccentric weight	

Body Loads

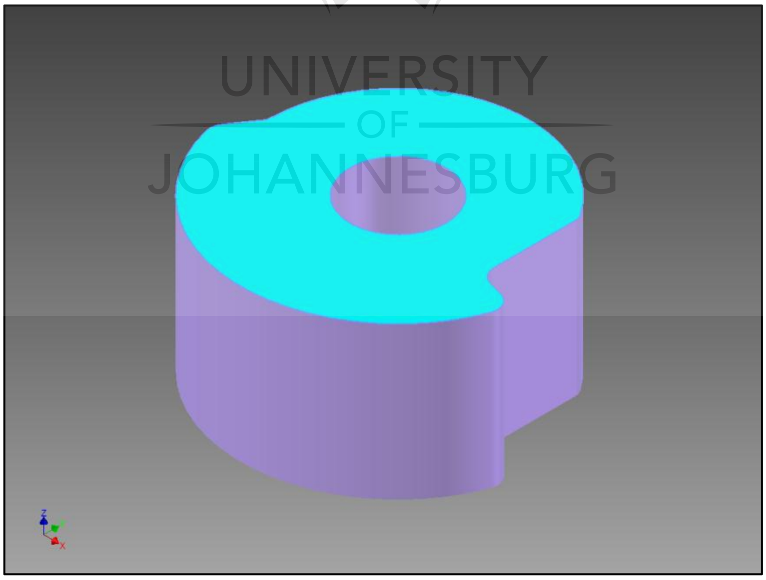
Load Type	Body Loads
Angular Velocity	
Magnitude	230.4 rad/s
Vector X	0.000 deg/s
Vector Y	0.000 deg/s
Angular Acceleration	
Magnitude	0.000 deg/s^2

Selected Face(s)

Fixed Constraint:1

Constraint Type	Fixed Constraint
Vector X	0.000 mm
Vector Y	0.000 mm

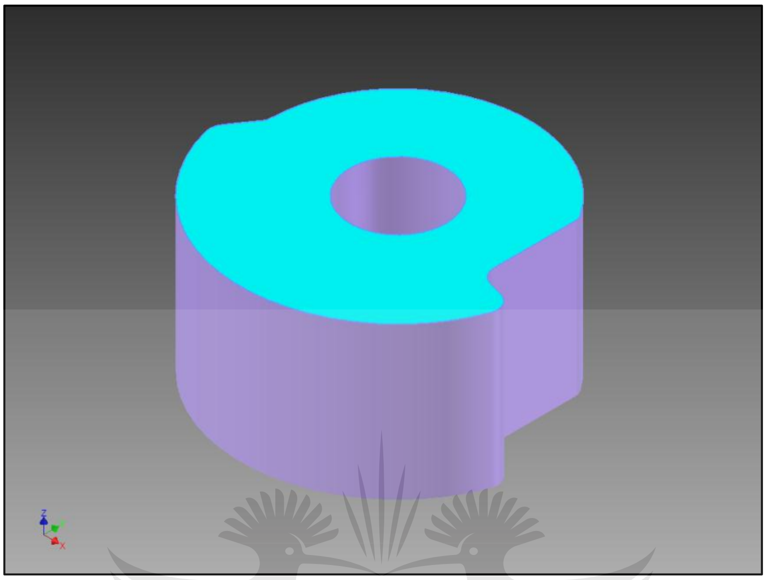
Selected Face(s)



Frictionless Constraint: 1

Constraint Type	Frictionless Constraint

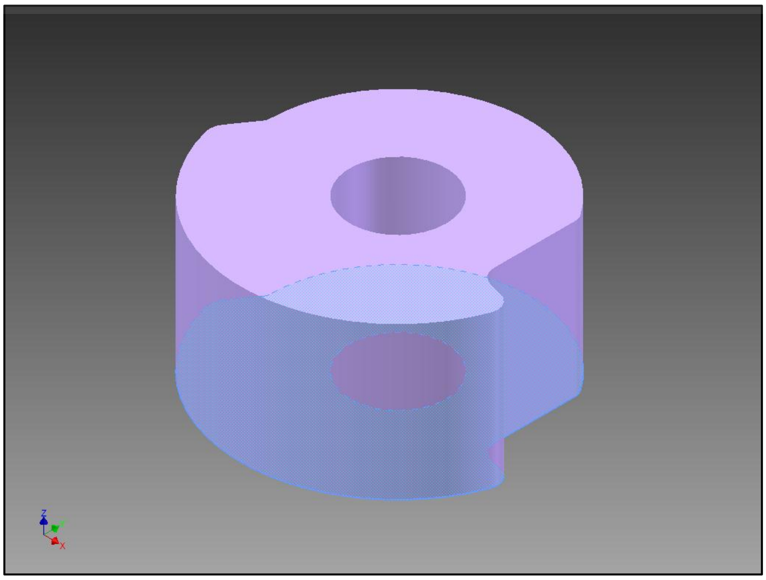
Selected Face(s)



Fixed Constraint: 2

Constraint Type	Fixed Constraint
Vector X	0.000 mm
Vector Y	0.000 mm

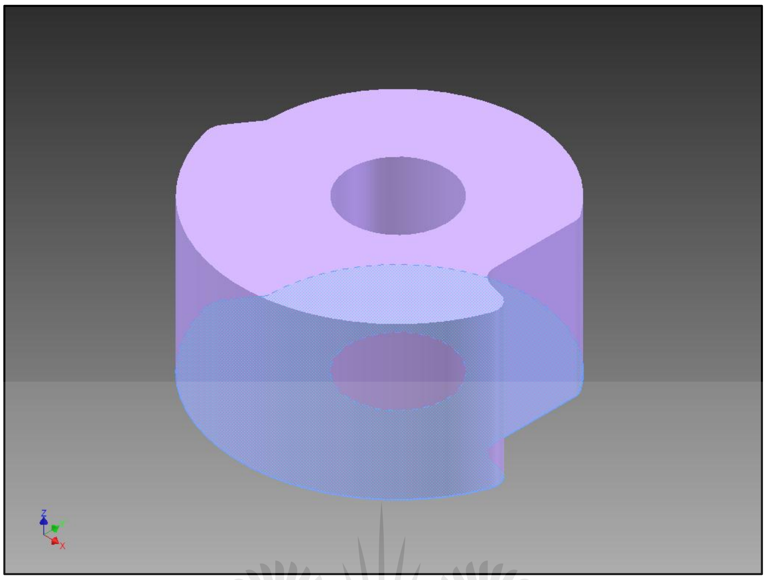
Selected Face(s)



Fixed Constraint: 3

Constraint Type	Fixed Constraint

Selected Face(s)



Results

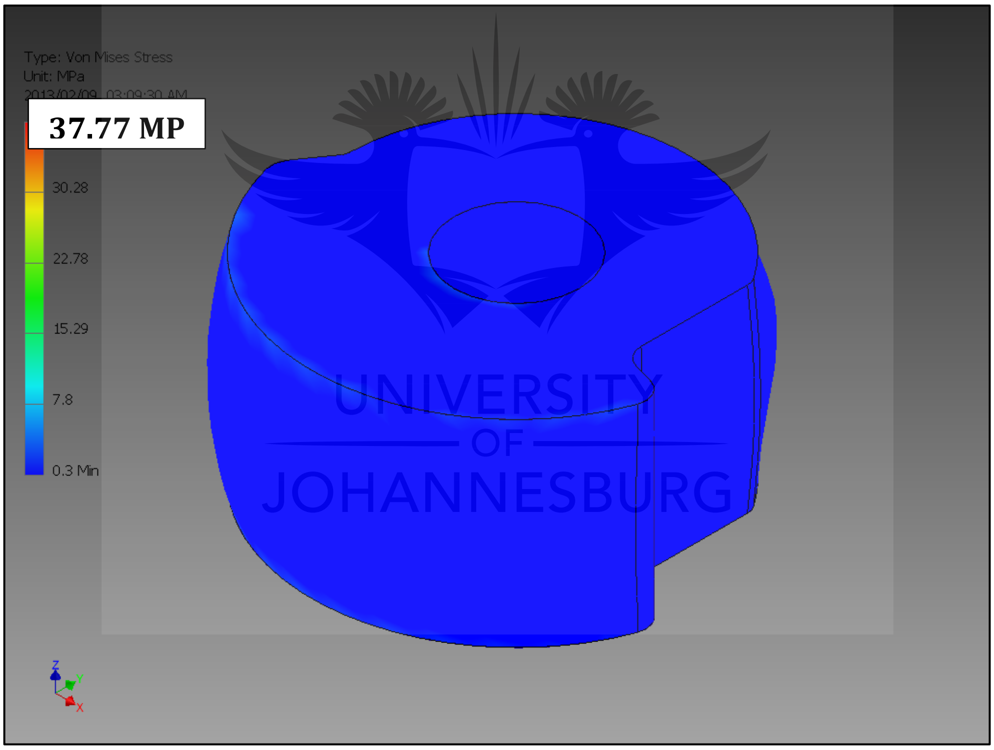
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	63578.8 N	-2219.19 N	2077.84 N m	2038.64 N m
		58870.8 N		9.85214 N m
		23907.3 N		401.6 N m
Frictionless Constraint:1	63578.8 N	-2219.19 N	2077.84 N m	2038.64 N m
		58870.8 N		9.85214 N m
		23907.3 N		401.6 N m
Fixed Constraint:2	63478.4 N	-2262.25 N	2067.21 N m	-2027.74 N m
		58760.5 N		-14.213 N m
		-23907.9 N		401.784 N m
Fixed Constraint:3	63478.4 N	-2262.25 N	2067.21 N m	-2027.74 N m
		58760.5 N		-14.213 N m
		-23907.9 N		401.784 N m

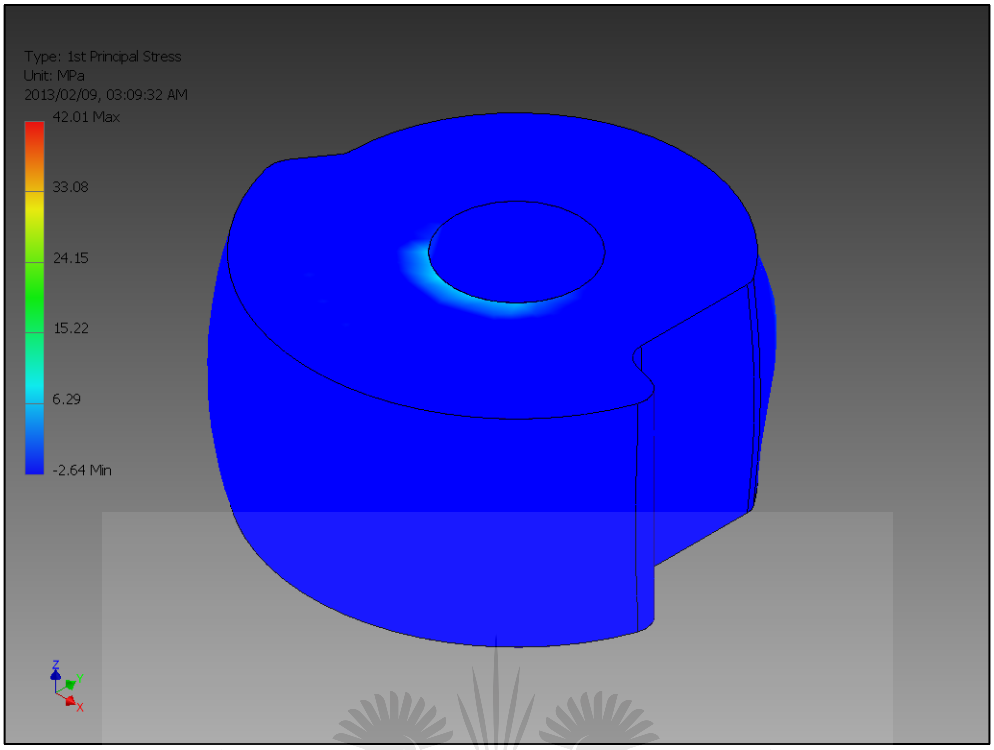
Result Summary

Name	Minimum	Maximum
Volume	13912800 mm^3	
Mass	109.355 kg	
Von Mises Stress	0.302752 MPa	37.7708 MPa
1st Principal Stress	-2.64176 MPa	42.0095 MPa
3rd Principal Stress	-16.1093 MPa	9.67617 MPa
Displacement	0 mm	0.00579428 mm
Safety Factor	5.48043 ul	15 ul

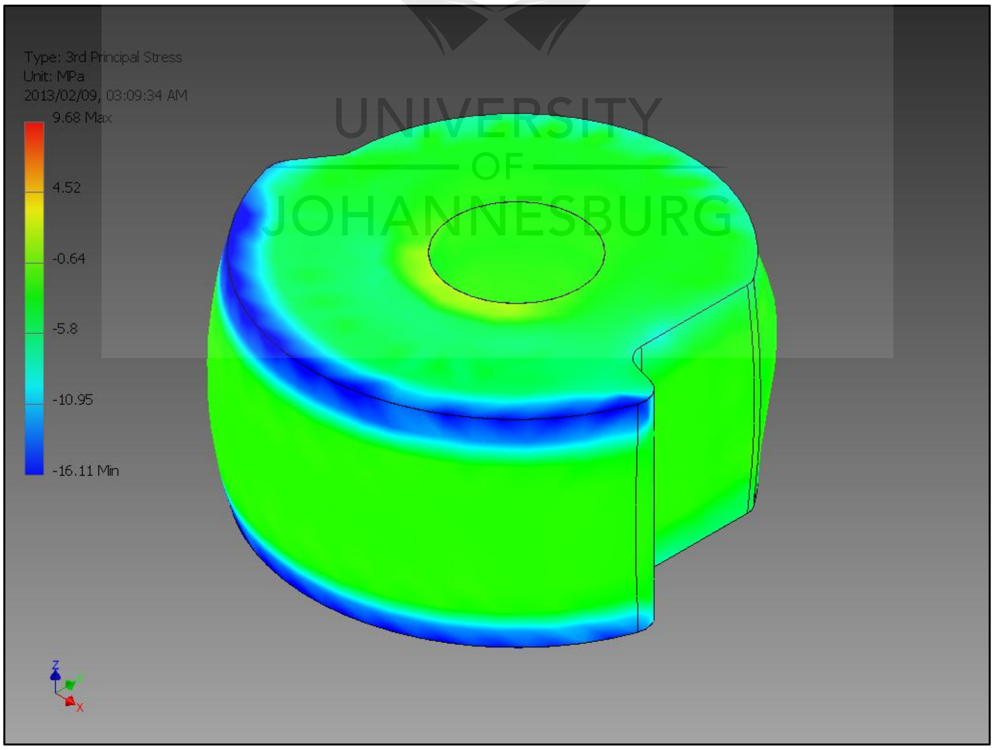
Von Mises Stress



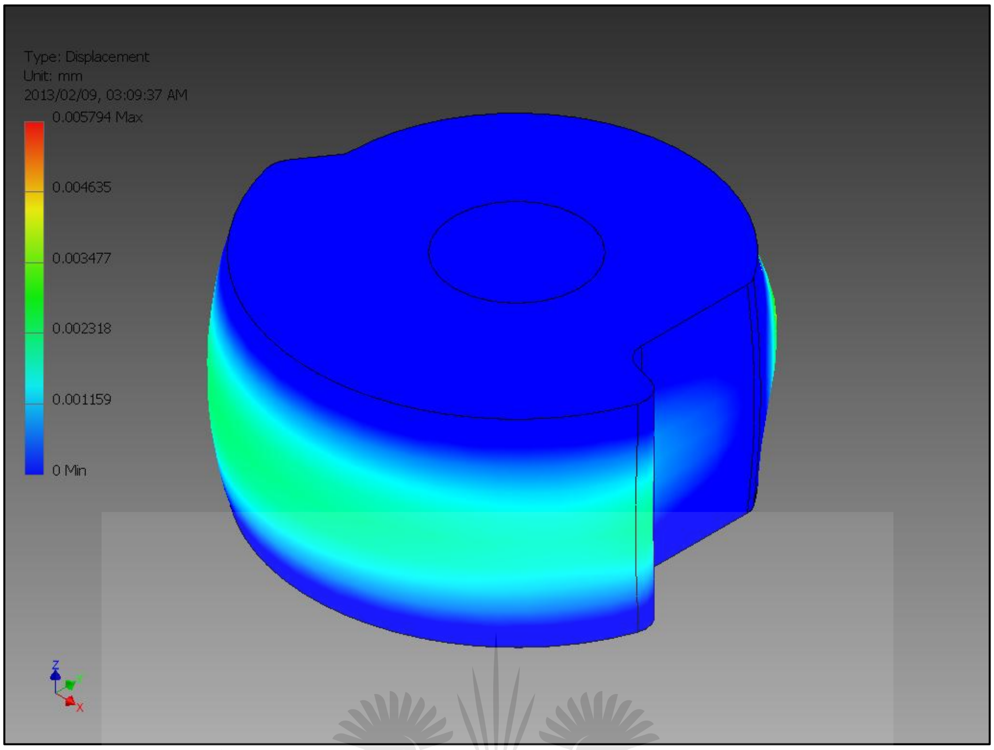
1st Principal Stress



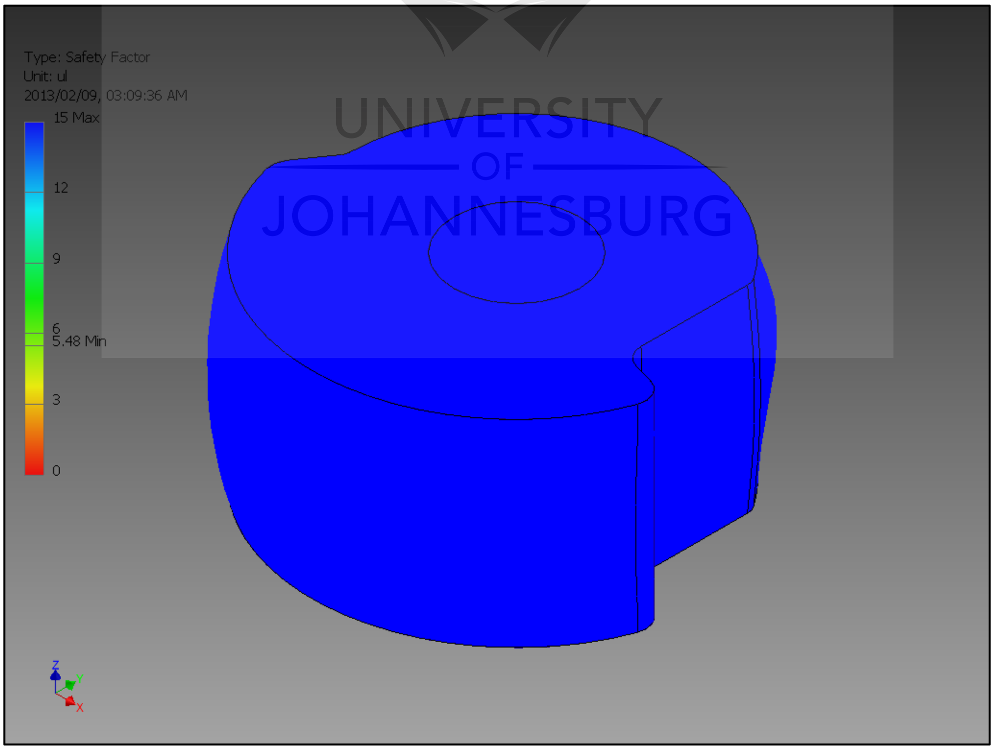
3rd Principal Stress



Displacement

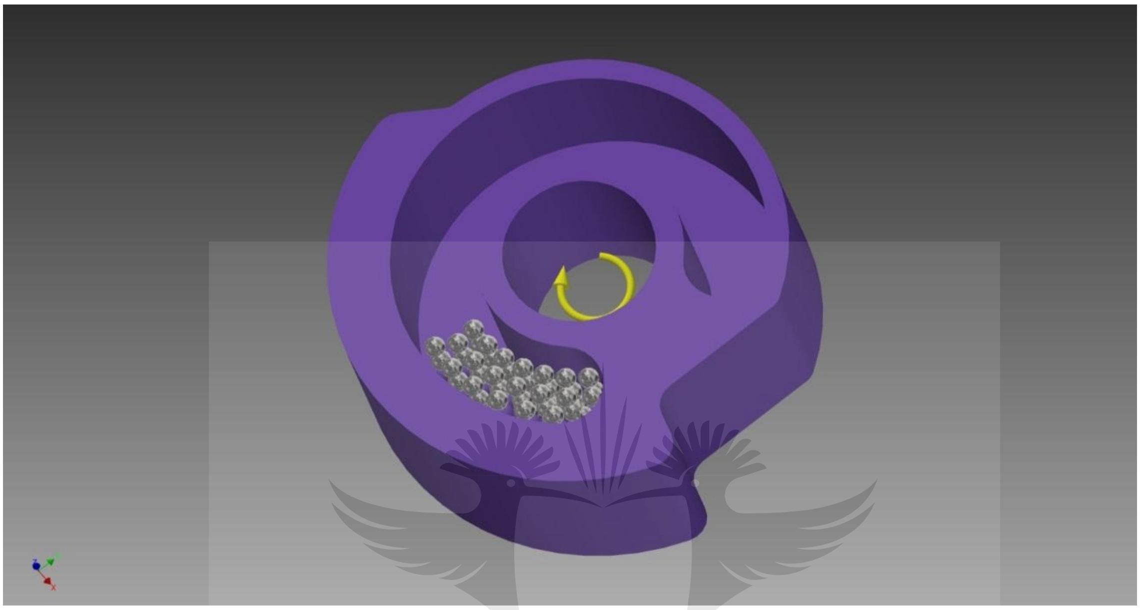


Safety Factor

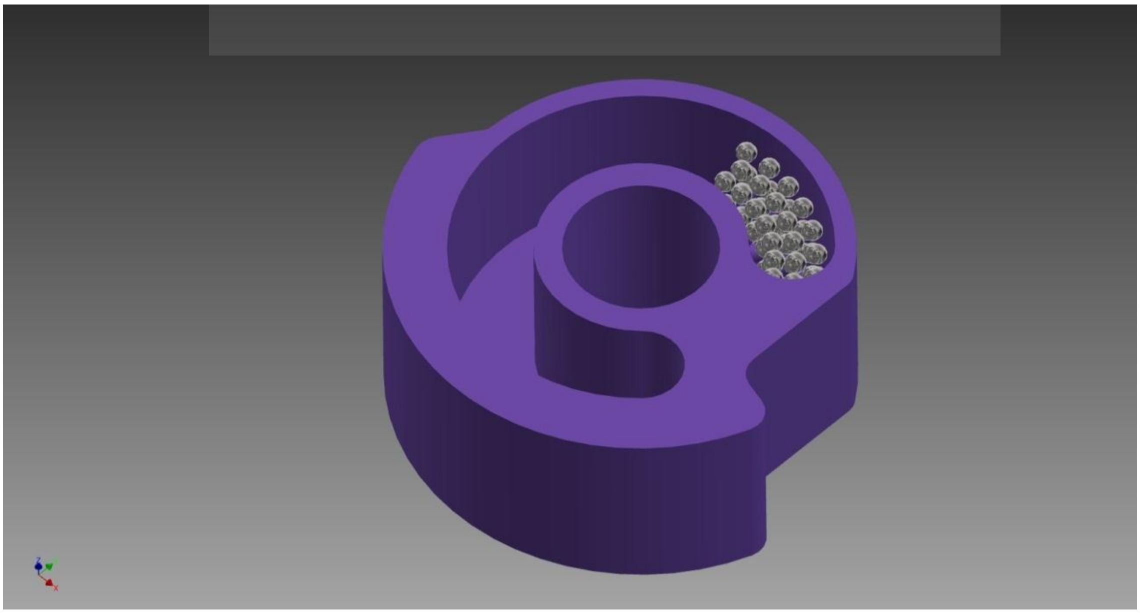


Appendix 24: Eccentric weight concepts on the inside.

Clockwise Rotation: This eccentric weight is the actual Caterpillar single drum vibratory compactor design and works in the following manner: for clockwise rotation, the eccentric moment changes due to balls rolling to one side of the eccentric weight as seen in the figure below.



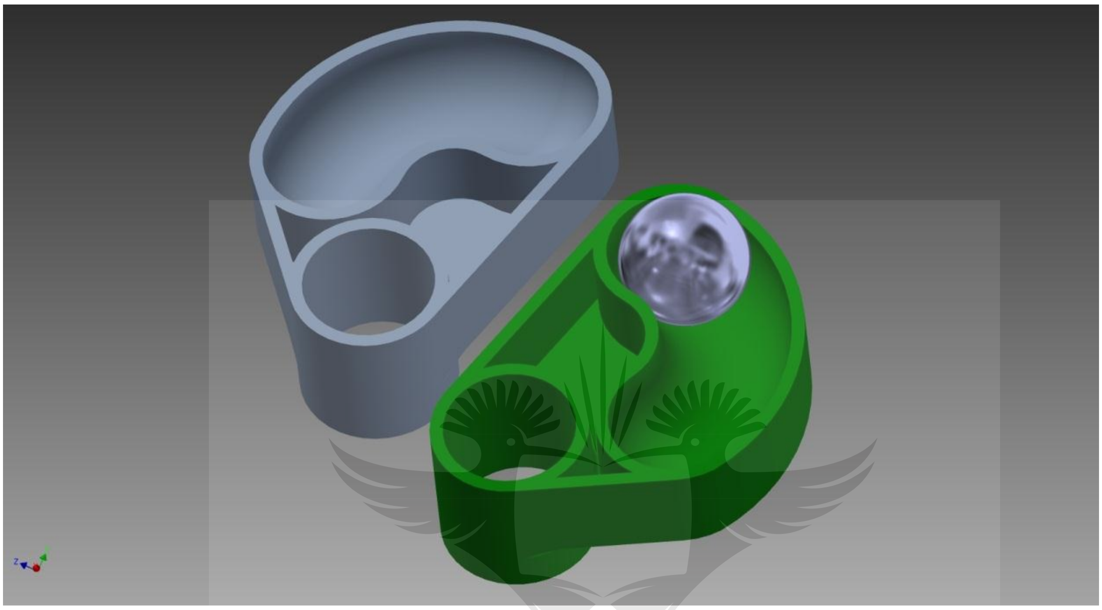
Anticlockwise rotation: the balls will roll to the other side of the eccentric weight as seen below in the figure, giving a smaller eccentric moment with different amplitude of operation, while maintaining the same angular speed or frequency as in the case of the clockwise rotation.



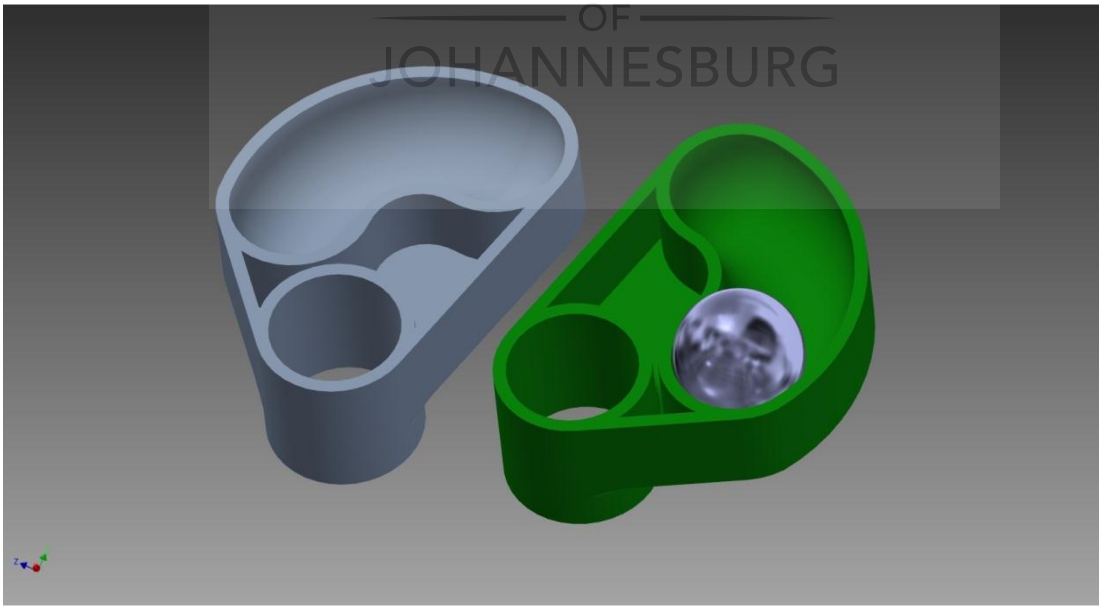
All other eccentric weight concepts work more or less on the same principle as the Caterpillar eccentric weight, but with different geometry and weight distribution. The next two concepts, Concept 1, Concept 2 and Concept 3, work on the same principle as the Caterpillar Eccentric Weight. The balls will move and alter the eccentric moment differently during clockwise or anticlockwise rotation. See the figures below.

Concept 1:

Clockwise rotation

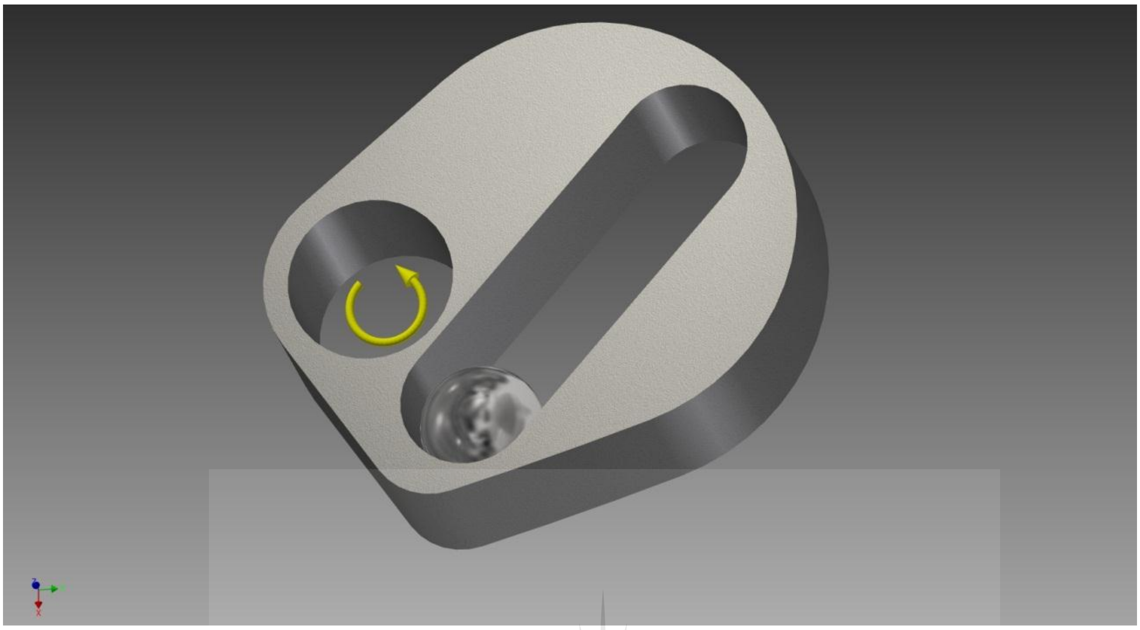


Anticlockwise rotation

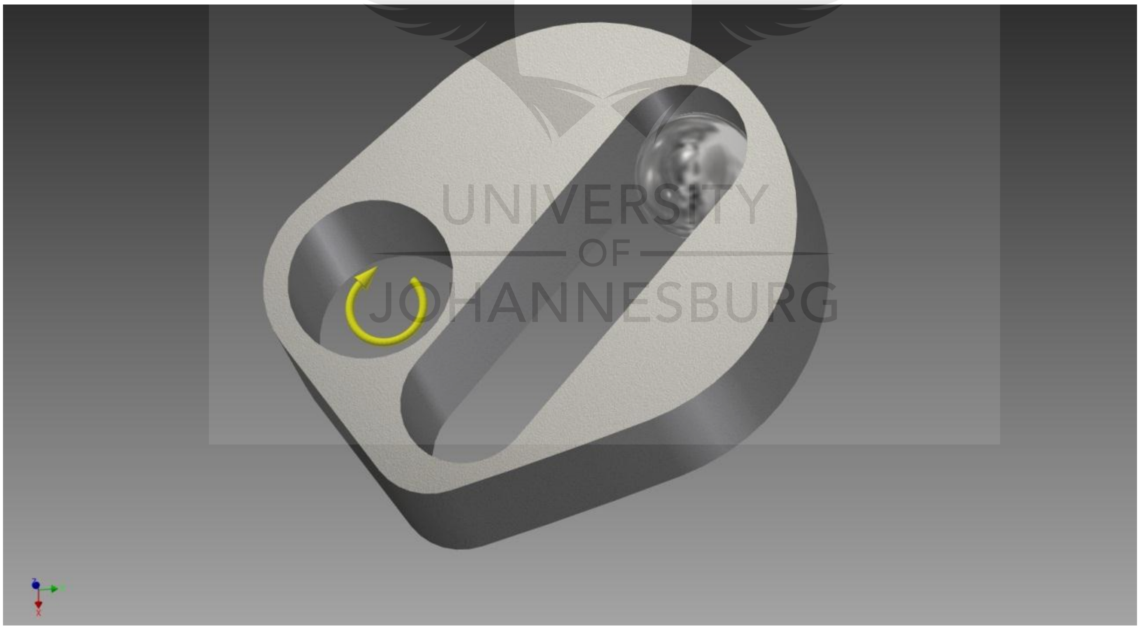


Concept 2:

Anticlockwise rotation



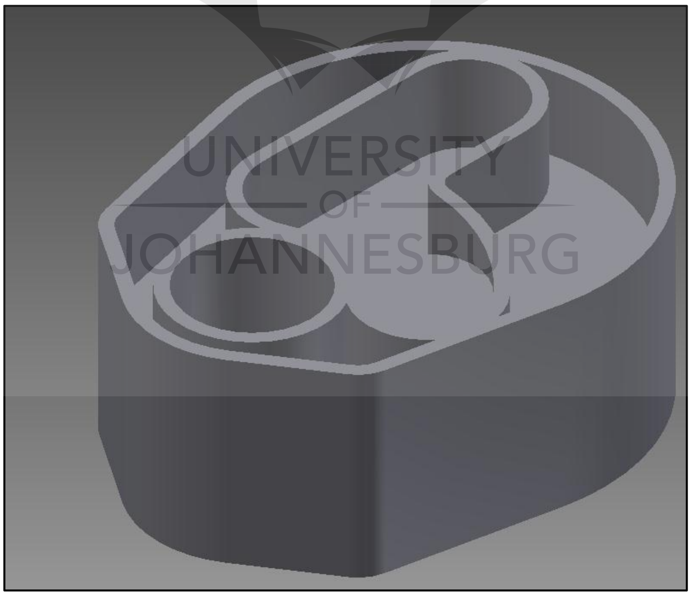
Clockwise Rotation



Concept 3:

Clockwise Rotation

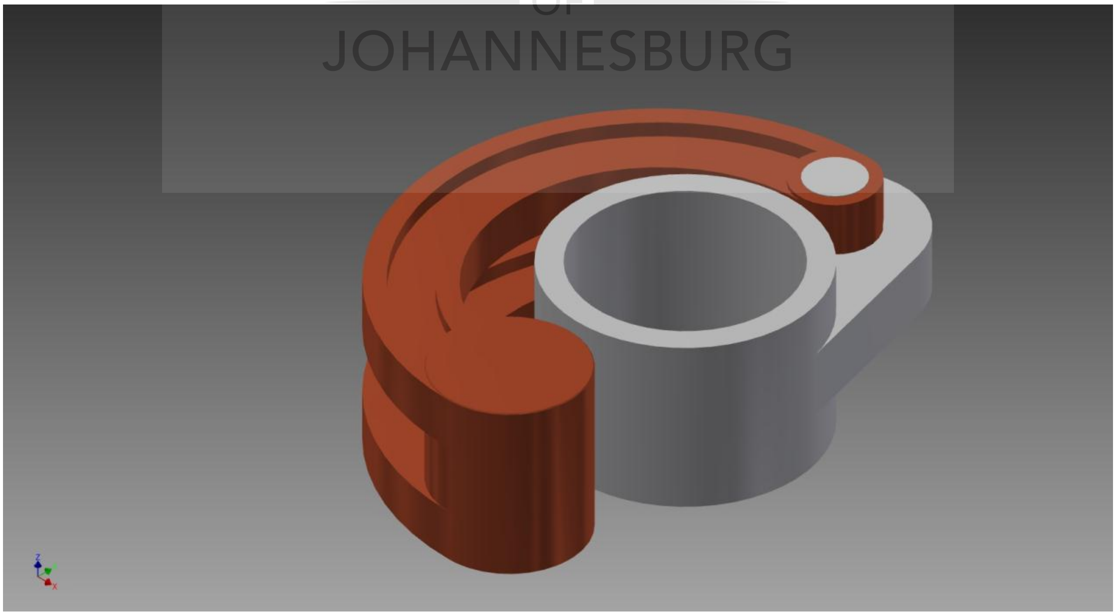
Anticlockwise rotation



Concept 4:

Anticlockwise rotation: This concept of an eccentric weight consists of an arch lever mechanism. When anticlockwise rotation is applied, the arch lever will flip open in a jack-knife type of action and in so doing, the eccentric moment is altered.

Clockwise rotation: During clockwise rotation, the arch lever will move in the opposite direction and yield a lesser eccentric moment and centrifugal force, as compared to the effect anticlockwise rotation with its greater centrifugal force and eccentric moment.



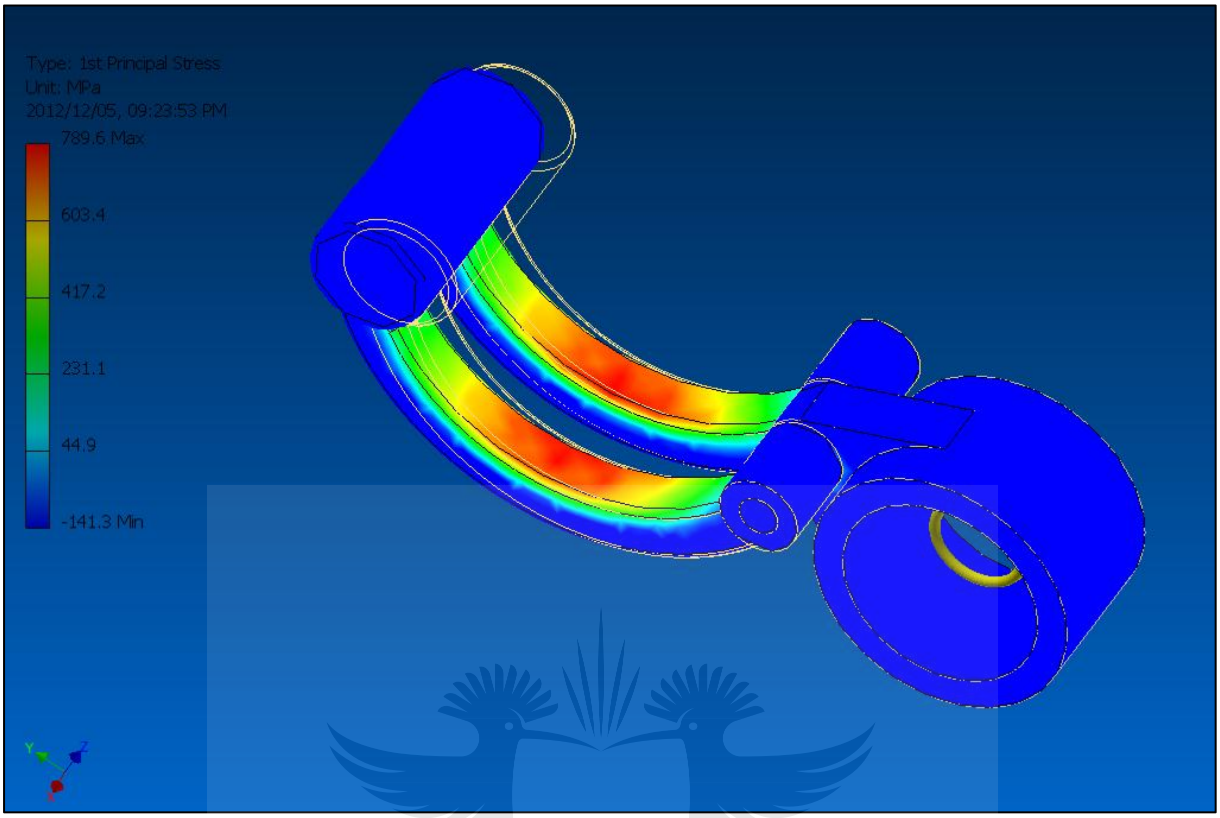
Appendix 25: Eccentric weight 1 design simulation.

The following is a simulation of a failed part that is an original idea of an eccentric weight system for the compactor being designed. The steps indicate how the problem was approached to make it work within the safe stress limits, but due to geometry and mass constraints, the design was a failure when it came to use mild steel, but stronger steel can be used such as the EN9-070M55 with heat treatment T.

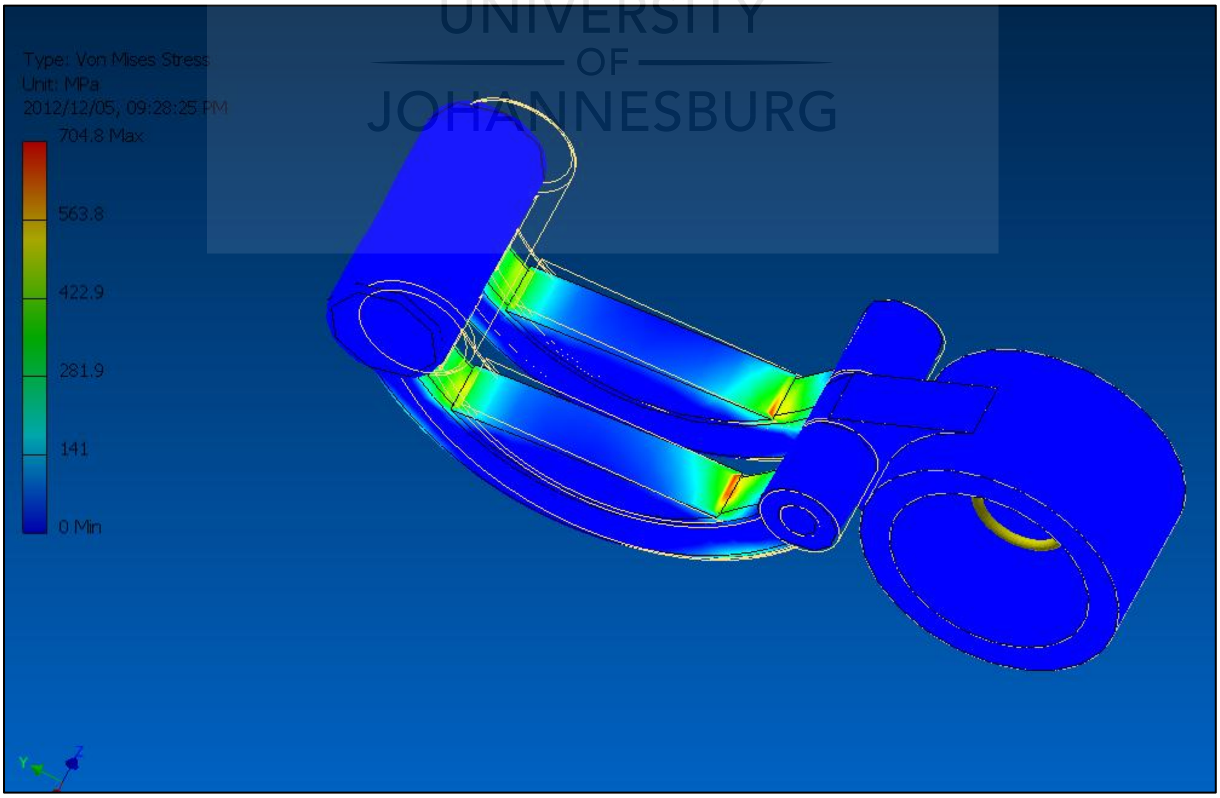
The material used in the simulation is mild steel 070M40 with yield strength of 215 MPa. A material with higher yield strength can be used for a safer design if needed. Two parts are presented in the FEA simulation. The first part is a hub that will be press fitted onto the rotating shaft which has spigots on both sides of the nose. The second is the pivoting eccentric weight with its arch lever system (ALS) connected onto the spigots. The system has the ability to achieve both high and low amplitude dependant on the direction of rotation and centrifugal force.

1. The eccentric weight is decreased to about half the mass of the previous one to alleviate the stresses experienced by the arch lever mechanism. The stresses are still too high at 789.6 MPa.

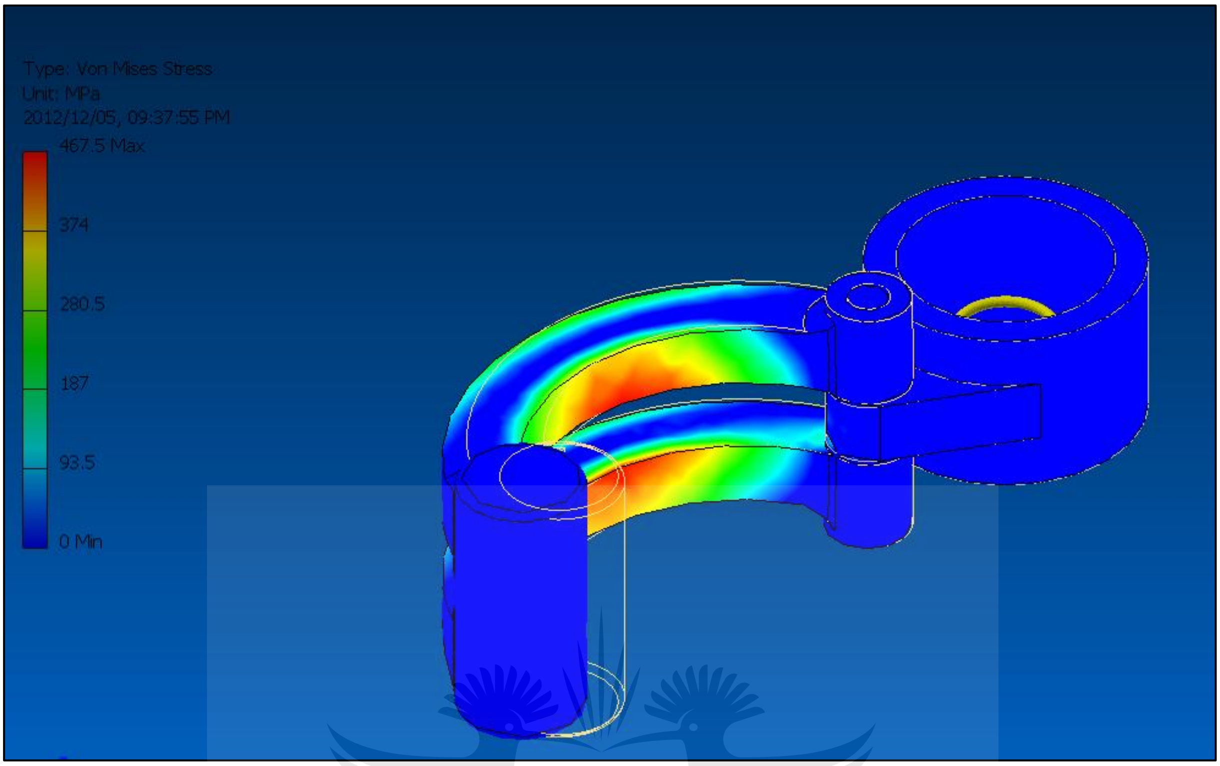
1. The cylindrical weight at the tip of the eccentric weight is too heavy for the arch lever mechanism, causing unusually high stresses in the center of the arch



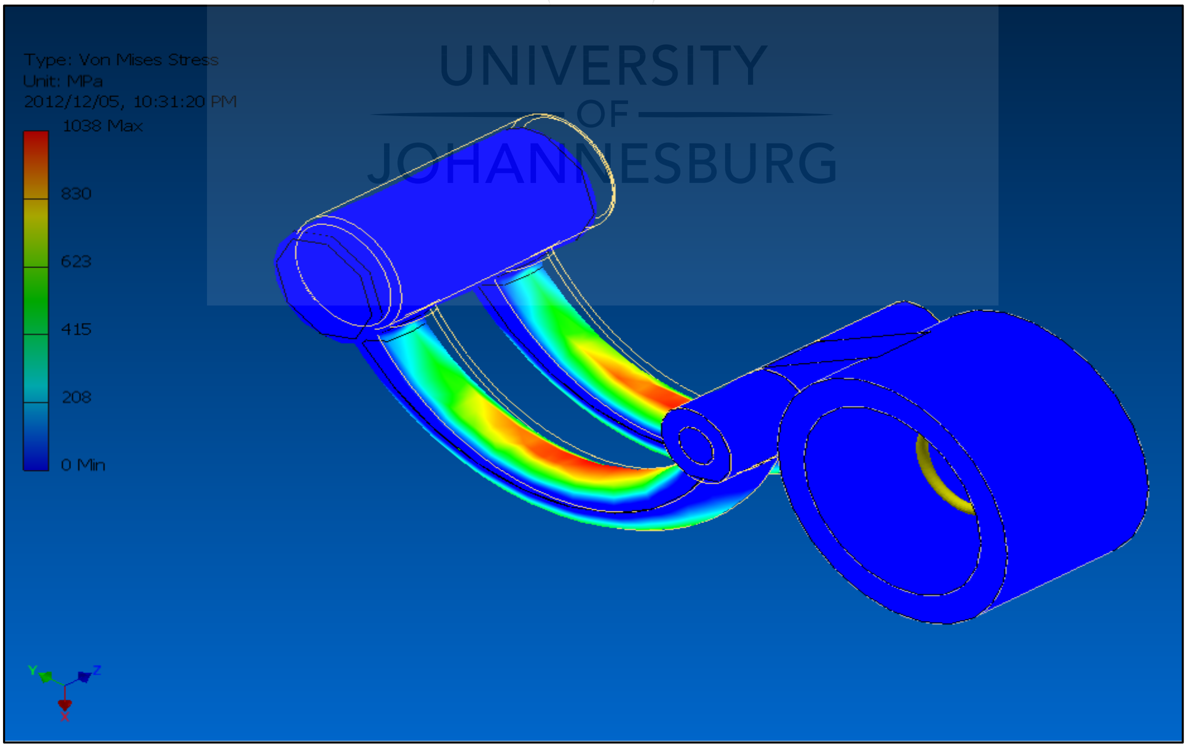
2. Two ribs are added to decrease stress in the arch. Very little stress change is observed, only 84.8 MPa less than the previous 789.6 MPa.



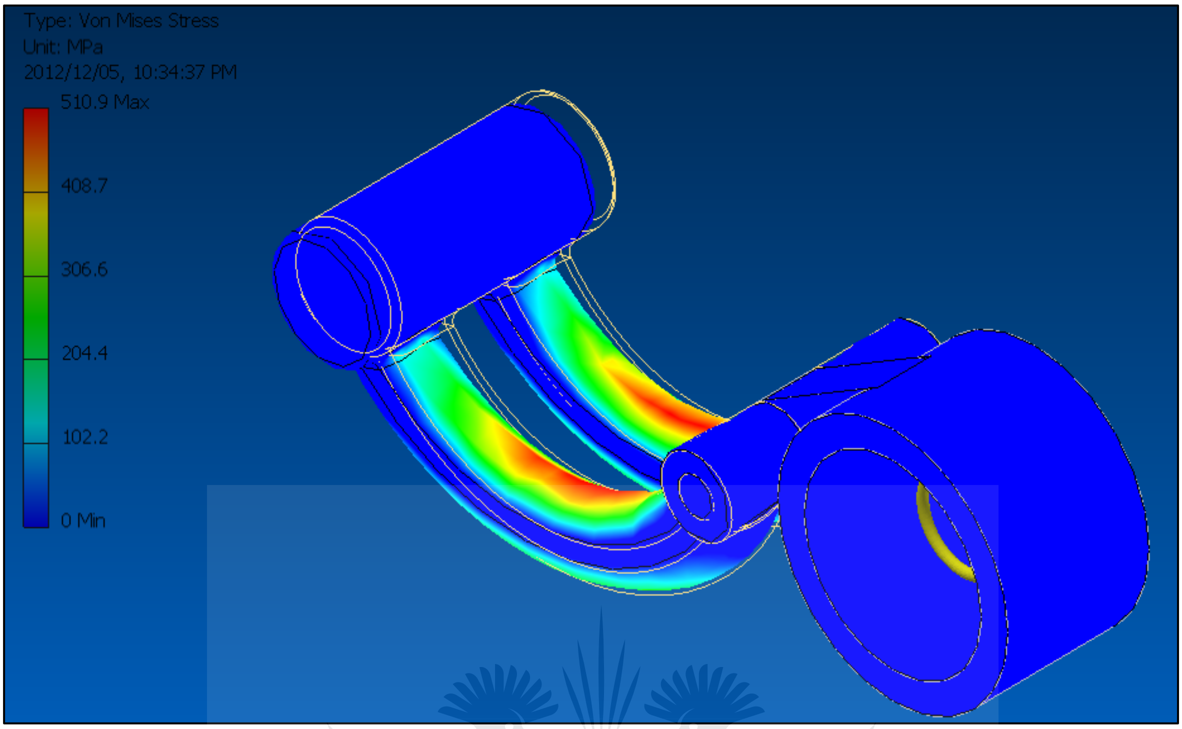
3. The eccentric weight arch geometry thickness is increased by 10 mm. The arch stress decreased to 467.5 MPa, but still no satisfactory result is achieved.



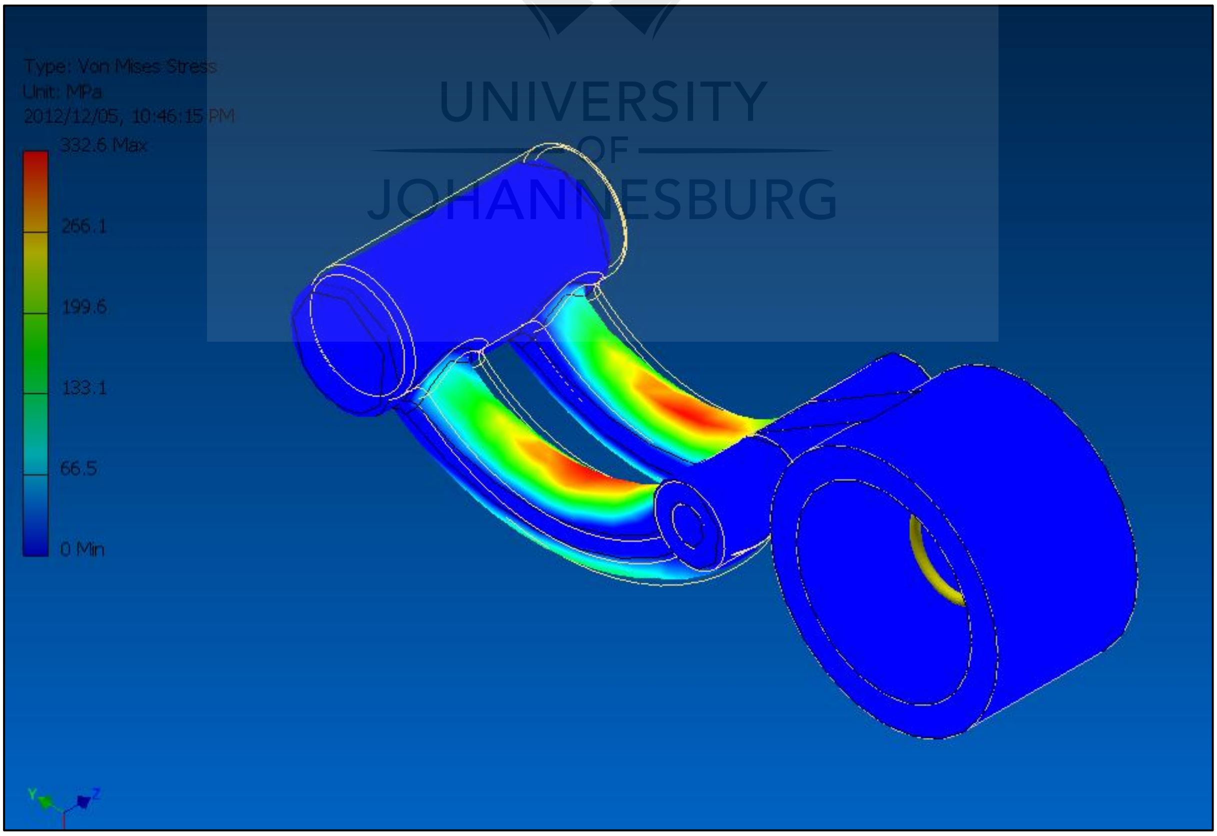
4. The square eccentric weight arch lever geometry is changed to circular and as a result, the stress increased from 467.5 MPa to 1038 MPa.



5. By increasing eccentric weight arch lever cross sectional diameter by 10 mm the maximum stress dropped to 510.9 MPa. This is half of the previous stress value.

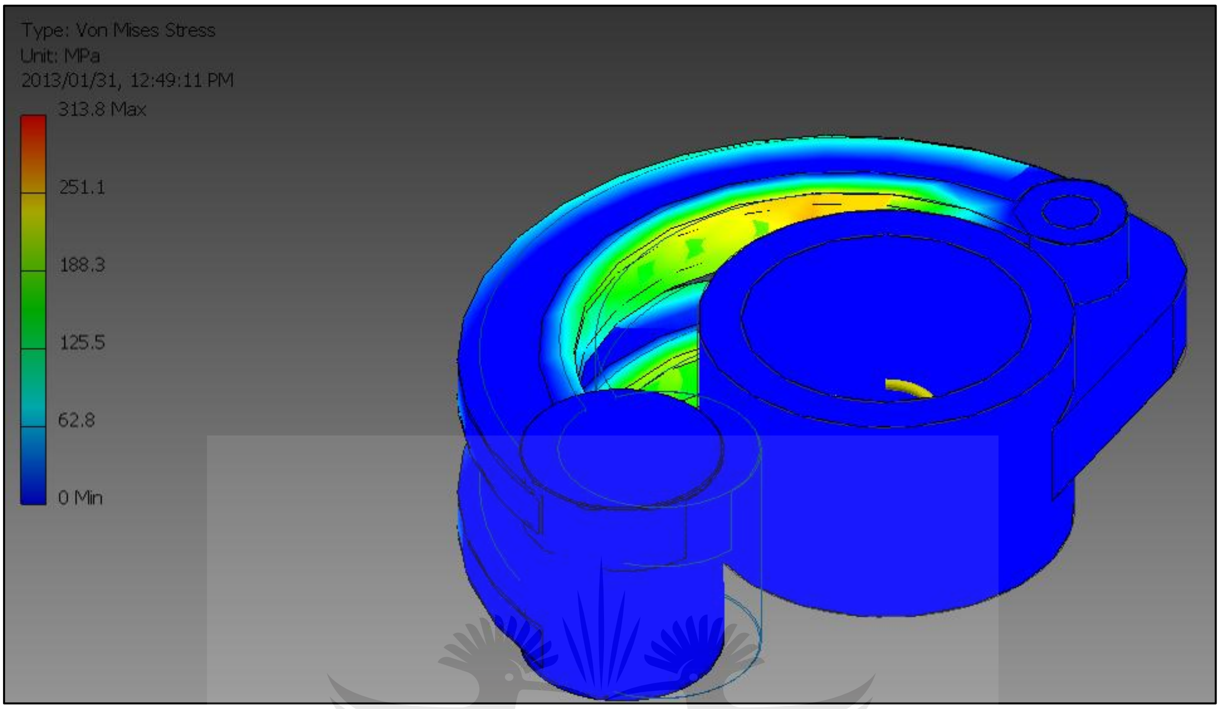


6. Decreasing the arch lever curvature radius caused the maximum stress to decrease by 36.86 % from510.9 MPa to 332.6 MPa.

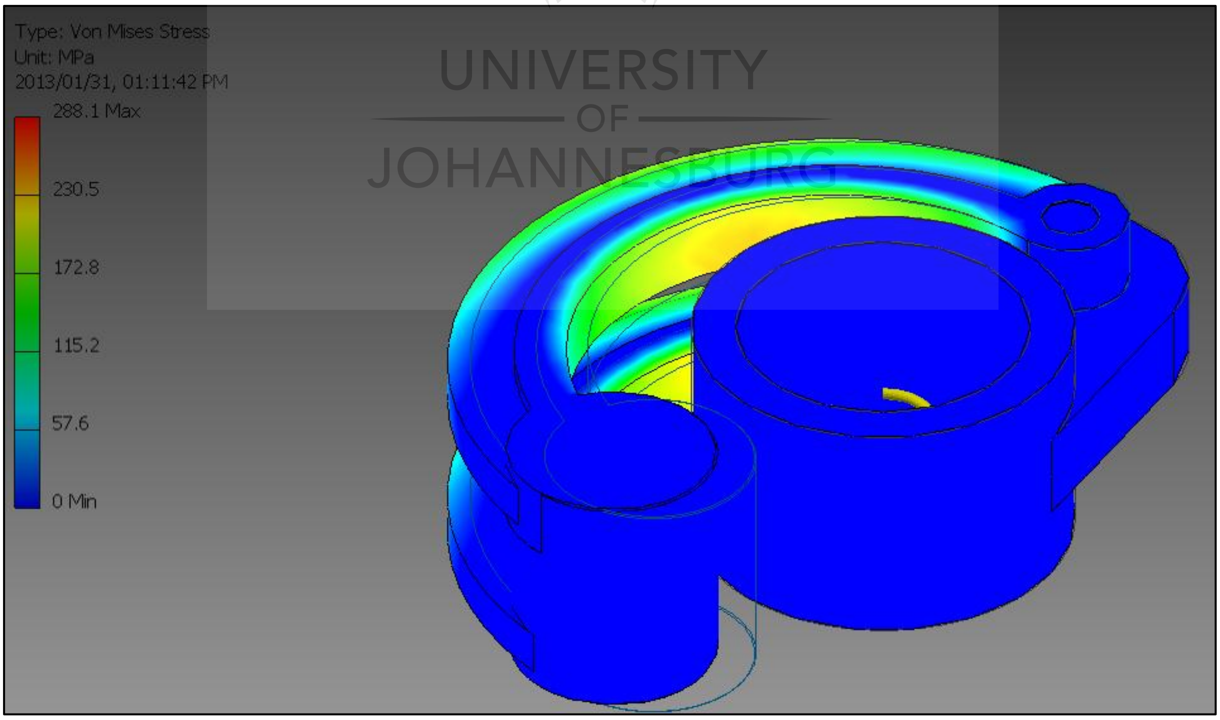


Considering a T-section Eccentric weigh arch lever EWAL

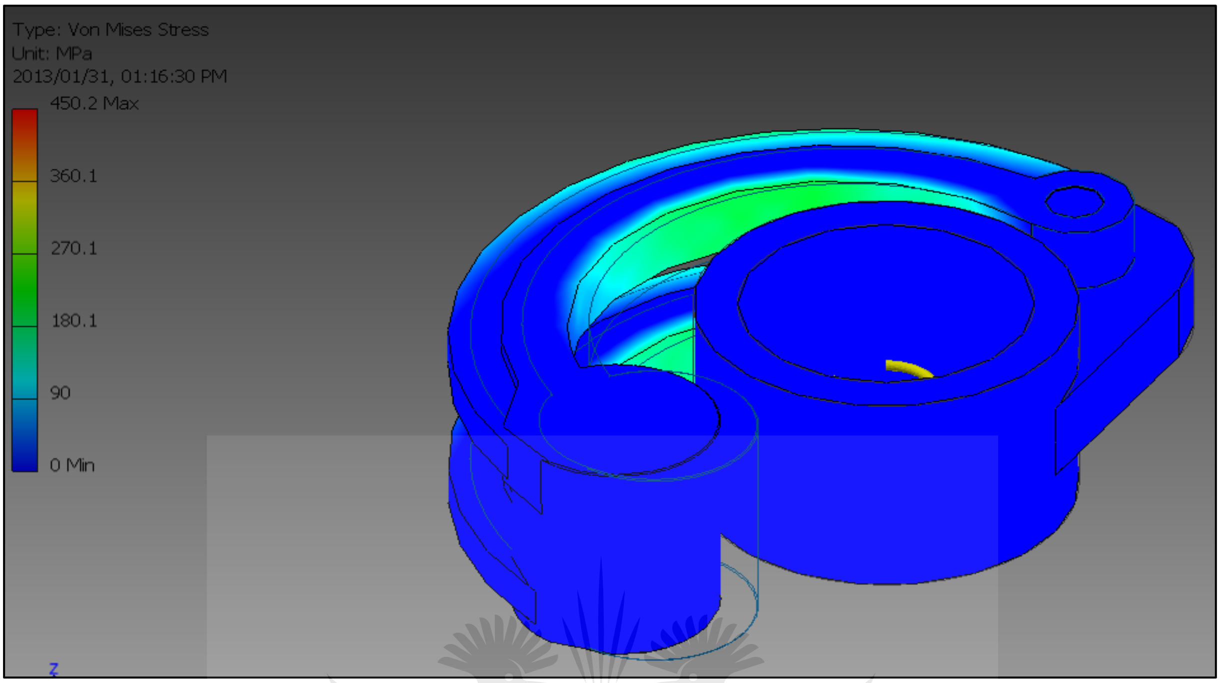
7. The maximum stress decreased from 332.6 MPa to 318.8 MPa.



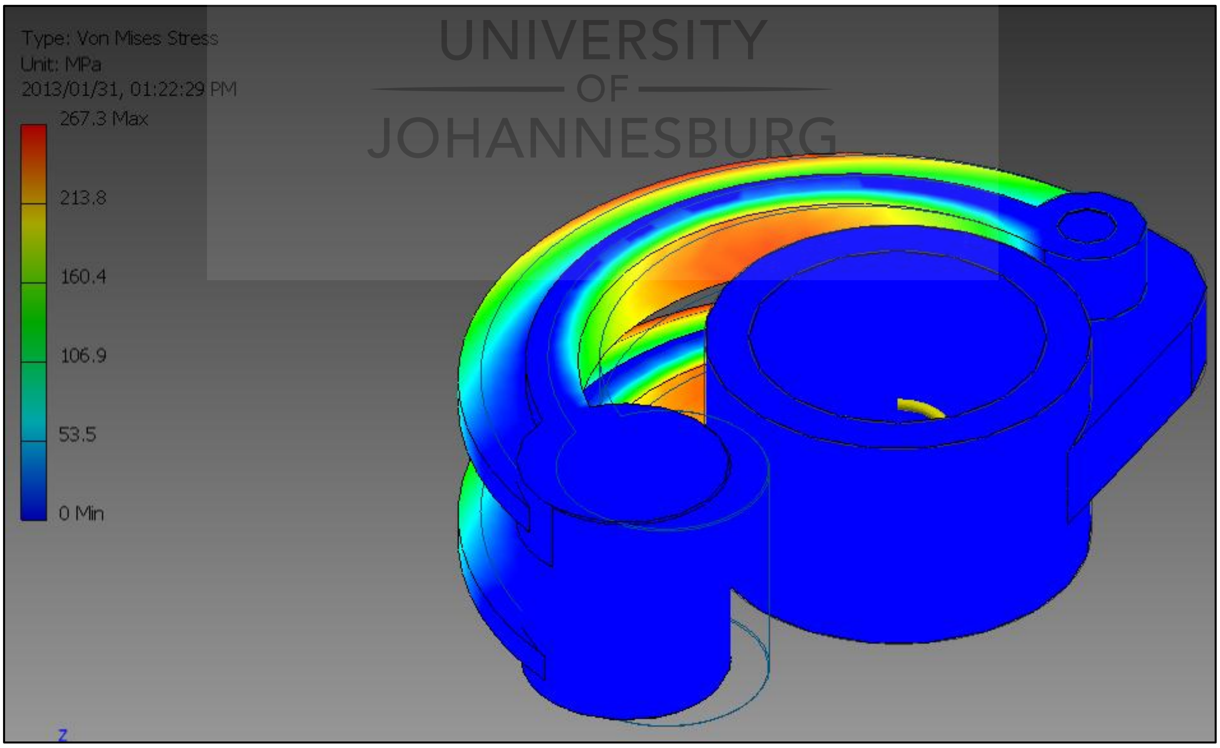
8. The T-section flange is increased from 12.5 mm thickness to 17.5 mm. This caused a stress decrease of 25.7 MPa and brought the stress down to 288.1 MPa in the EWAL vicinity.



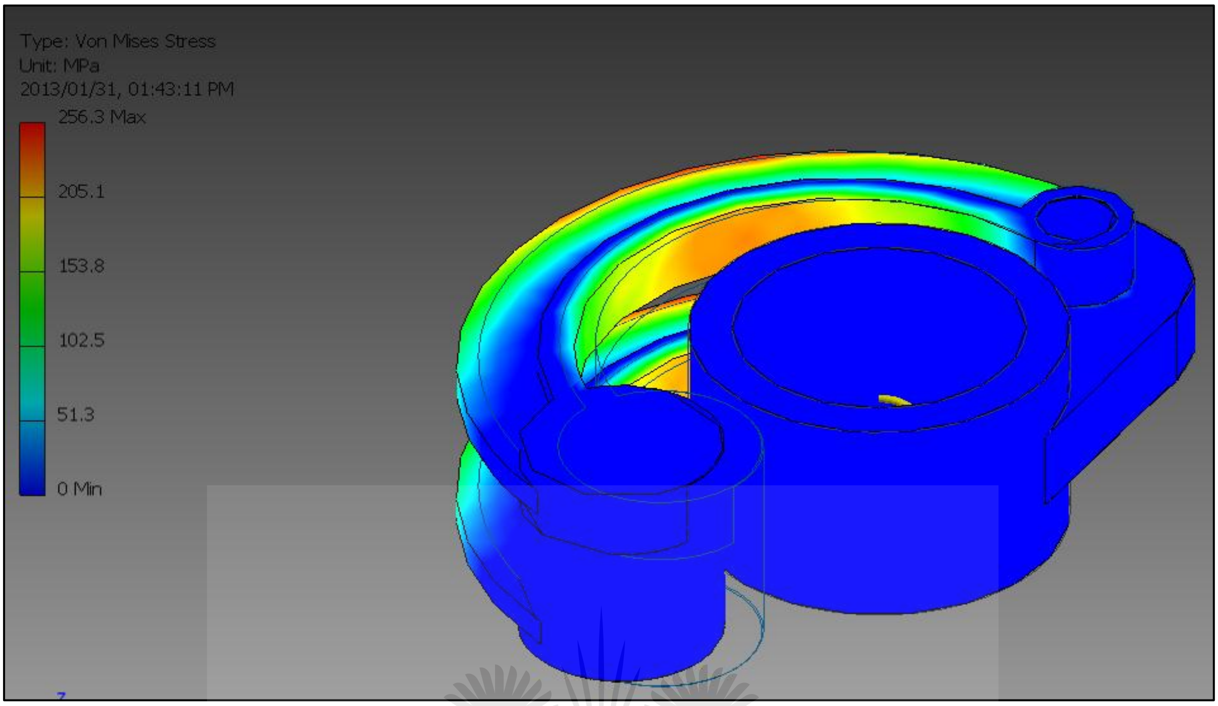
9. When the T-section web thickness was increased from 17.5 mm to 22.5 mm, it caused the stress to almost double from 288.1 MPa to 450.2 MPa. This indicates that increasing the web thickness any further, will cause the maximum stress to increase.



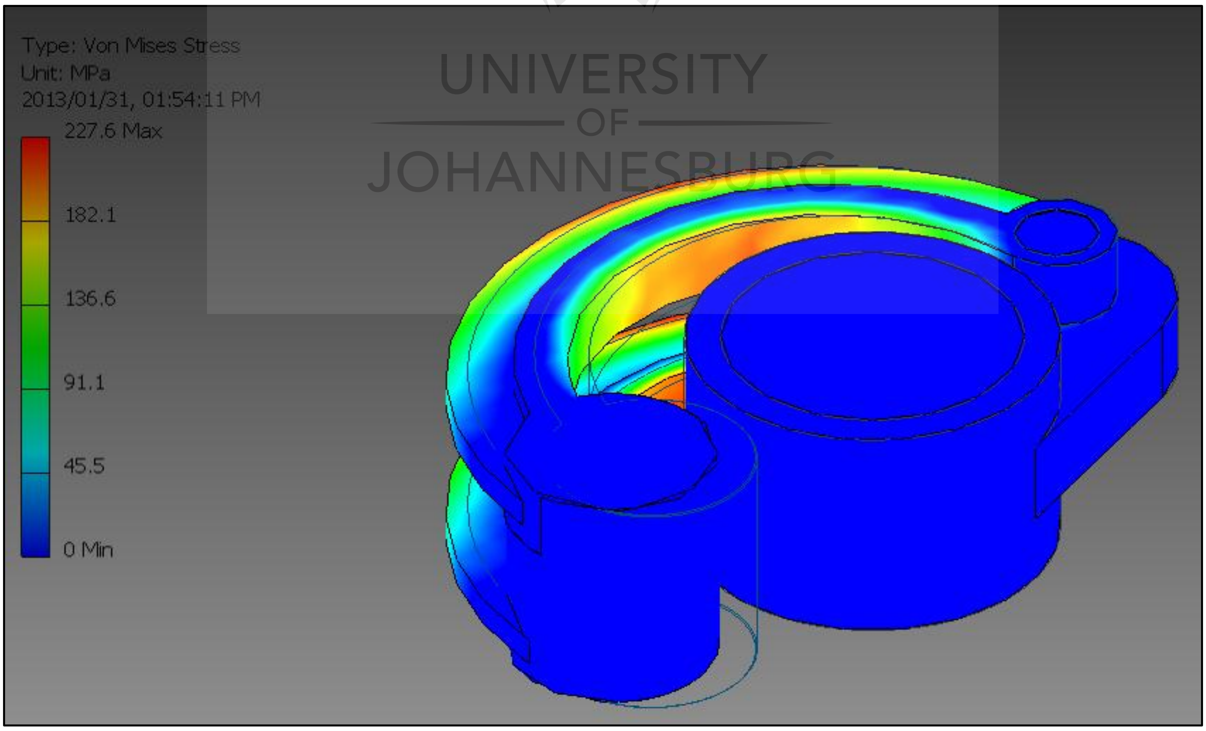
10. The web thickness is kept 17,5 mm and the flange reduced from 17.5 mm – 10 mm. A decrease of 267.3 MPa is noted. The maximum stress is even less than the previous 288.1 MPa in No 8.



11. The web thickness is decreased from 17.5 mm to 12, 5 mm and the flange thickness. The maximum stress dropped by a further 11 MPa and is currently at 256.3 MPa.



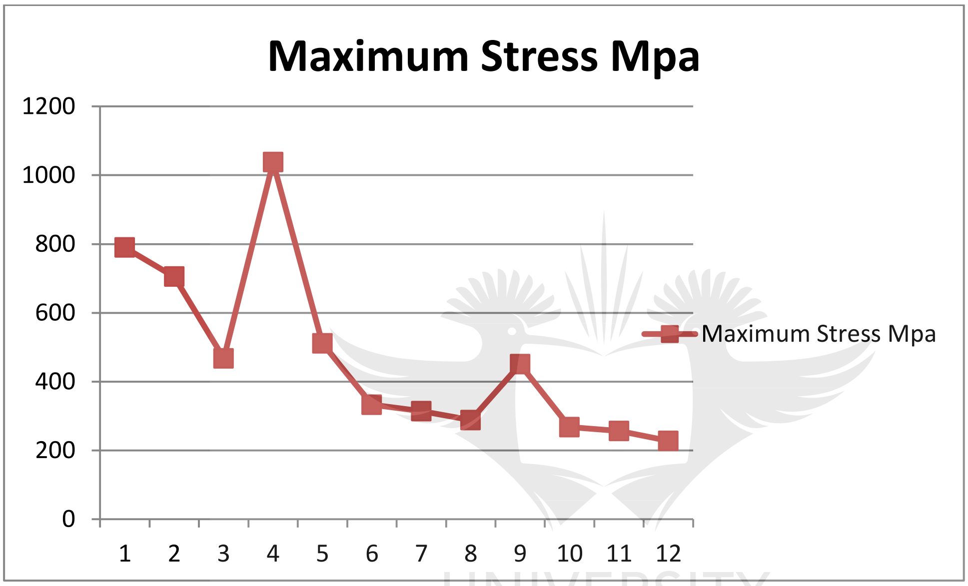
12. The arch radius was reduced from 200 mm to 189 mm thus further decreased the maximum stress by 28.7 MPa. This gives a final refined design with a maximum stress of 227.6 MPa in the EWAL.



Making the web and or the flange of the T-shape thicker will result in the eccentric weight mass to become too heavy, thus increasing the centrifugal force beyond the limit for this type of compactor. Greater force means greater machine mass due to bigger component geometries to resist the stresses concerned.

An I shaped lever was also introduced and caused the eccentric weight to become too heavy with not much extra strength gain.

A stress graph for all twelve simulations is shown below and it shows in detail how the stress varied during all twelve simulations.

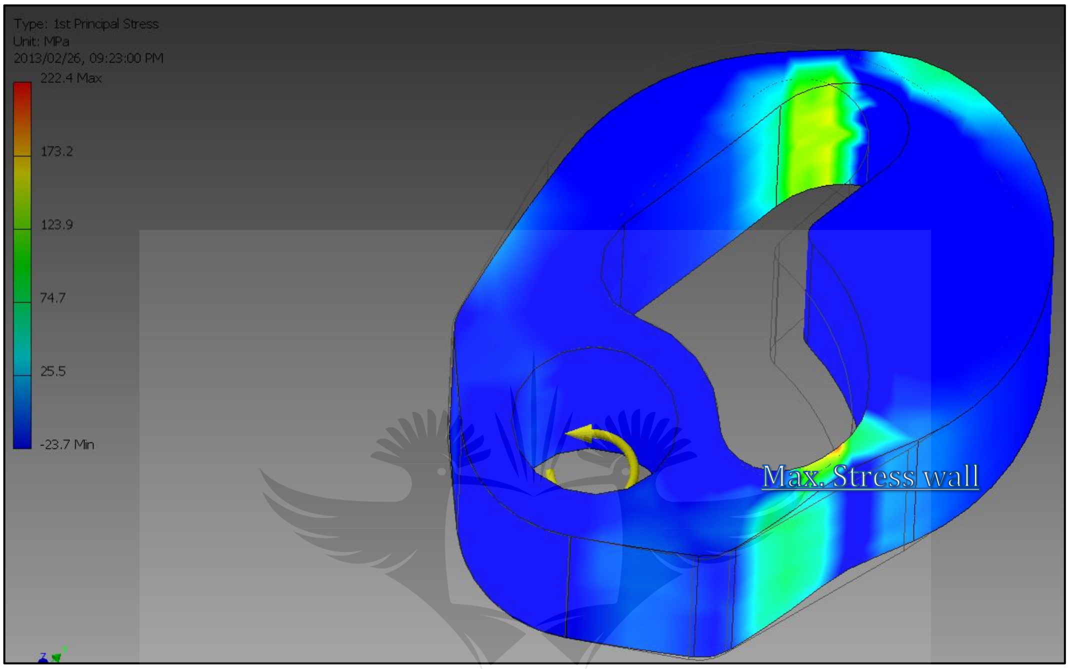


Stress Simulation Results			
Simulation Number	Maximum Stress MPa	Simulation Number	Maximum Stress MPa
1	789.6	7	313.8
2	704.8	8	288.1
3	467.5	9	450.2
4	1038	10	267.3
5	510.9	11	256.3
6	332.6	12	227.6

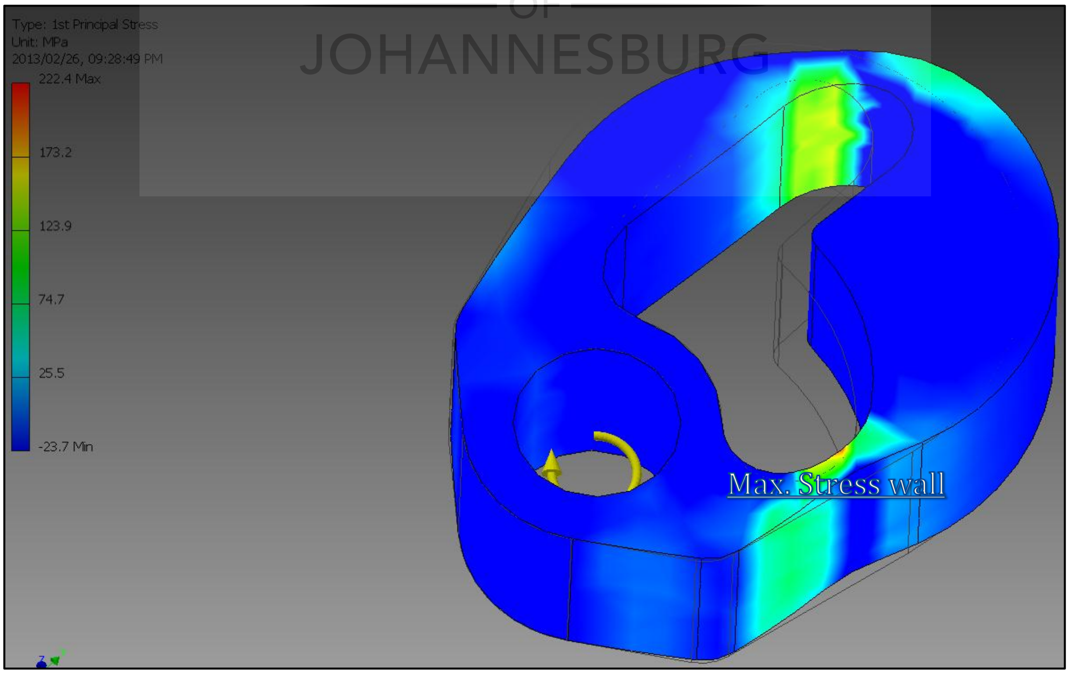
Appendix 26: Eccentric weight design simulation 2.

The two figures below illustrates that the direction of rotation of the centrifugal weight in the two figures below has no effect on the maximum stress.

1. Anticlockwise rotation as seen by the circular arrow yielded a maximum stress of 222.4 MPa at the wall.



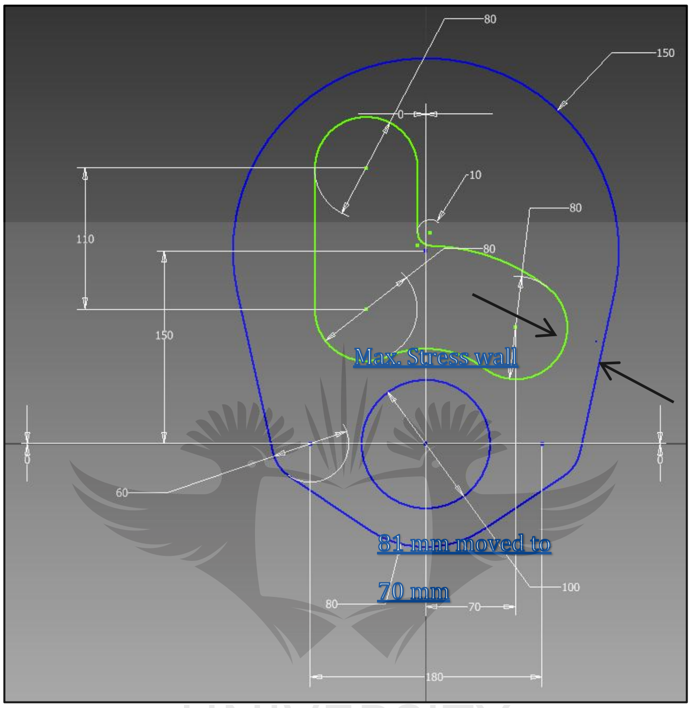
2. Clockwise rotation yielded a similar maximum stress result of 222.4 MPa at the wall.



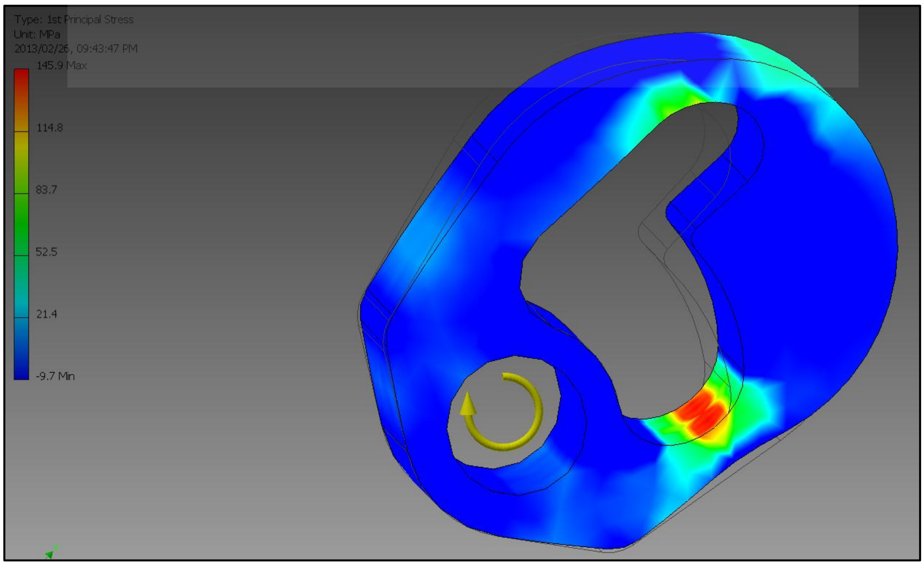
Increasing wall thickness

Since the maximum stress occurs at the wall, the wall needs to be thickened. The distance from the centre circle to the centre of the arc closest to the wall is decreased from 81 mm to 70 mm to thicken the wall where the maximum stress occurs. See the drawing below.

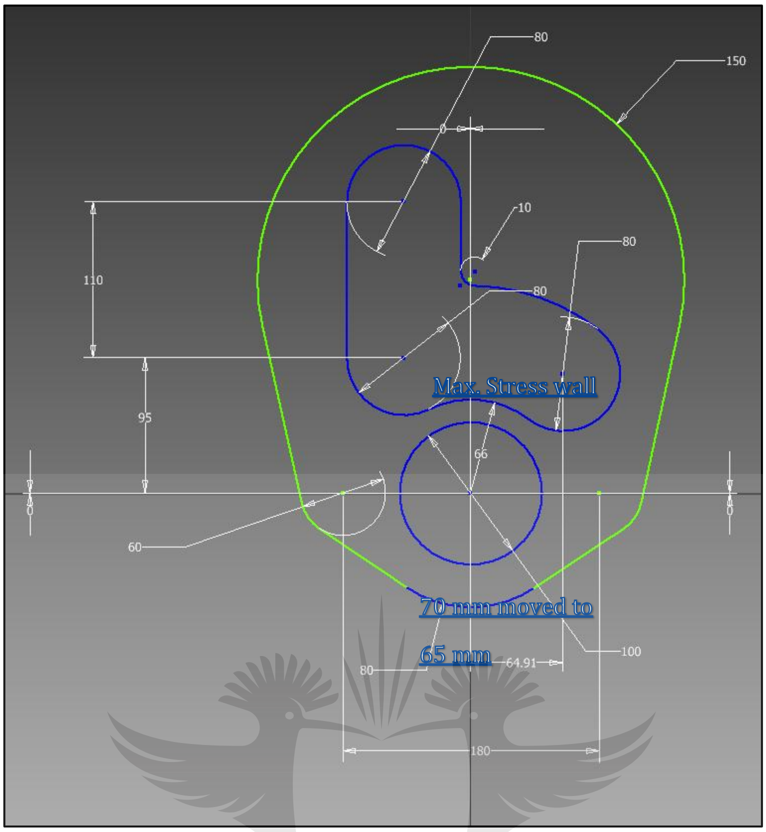
NB: This is not an actual engineering drawing drawn to standards, but a configurative drawing to test dimension sizes.



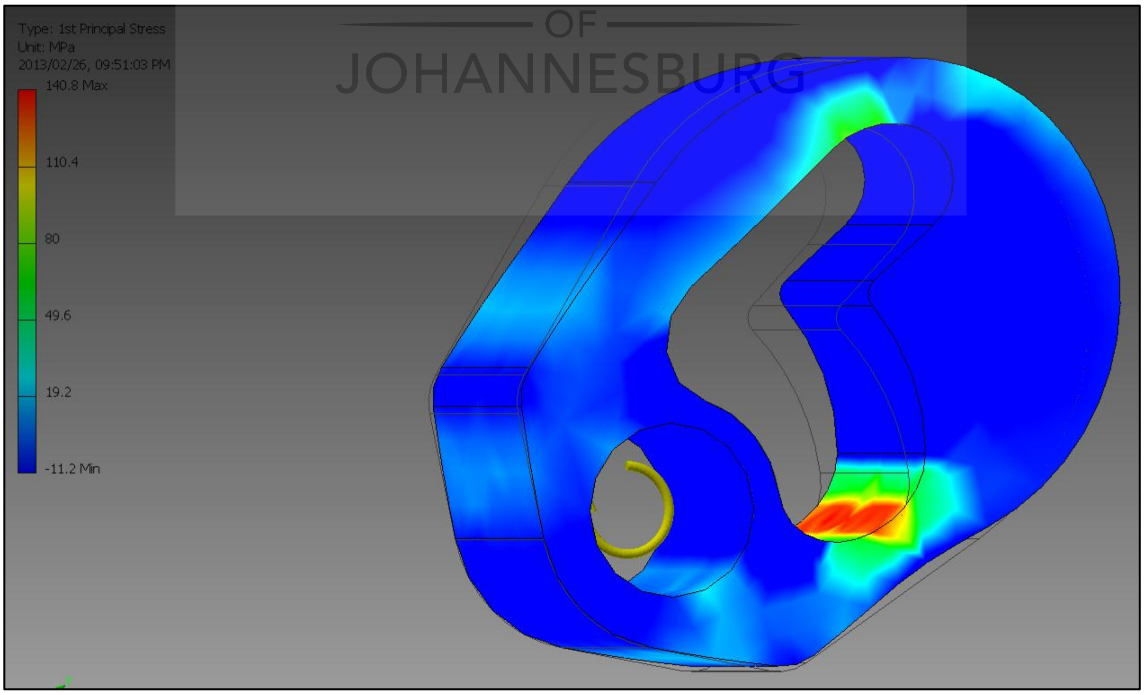
2. Second simulation, after wall thickening yielded a stress decrease from 222.4 MPa to 145.9 MPa.



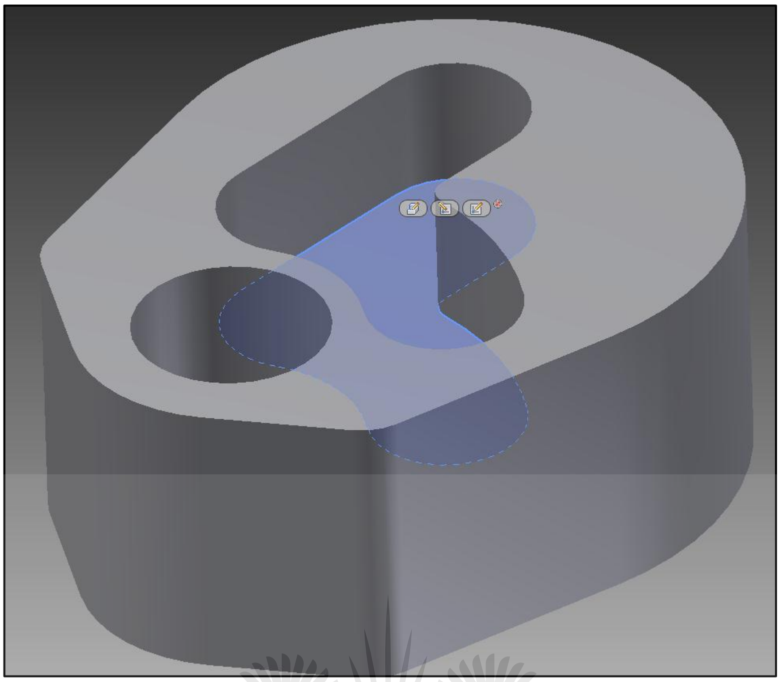
Further wall thickness increase is shown by changing the mentioned 70 mm to (64.91 mm) 65 mm about.



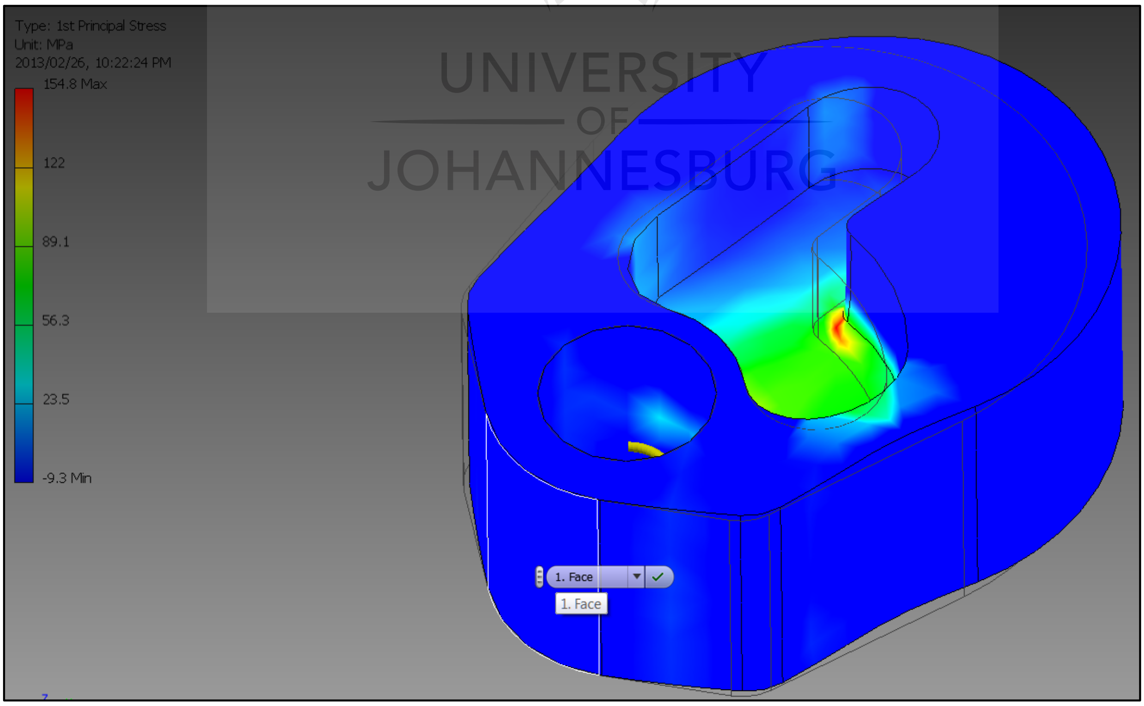
3. Not much difference, the principle stress is indicated at 140.8 MPa, only 5.1 MPa less than the previous stress.



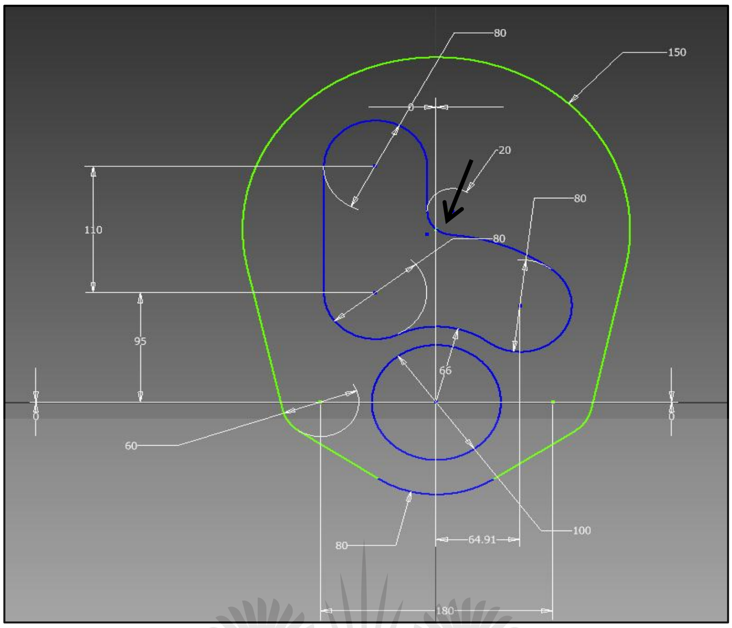
Drawing a 10 mm stabilizing rib in the centre of the L shaped hole to resist deformation at the walls.



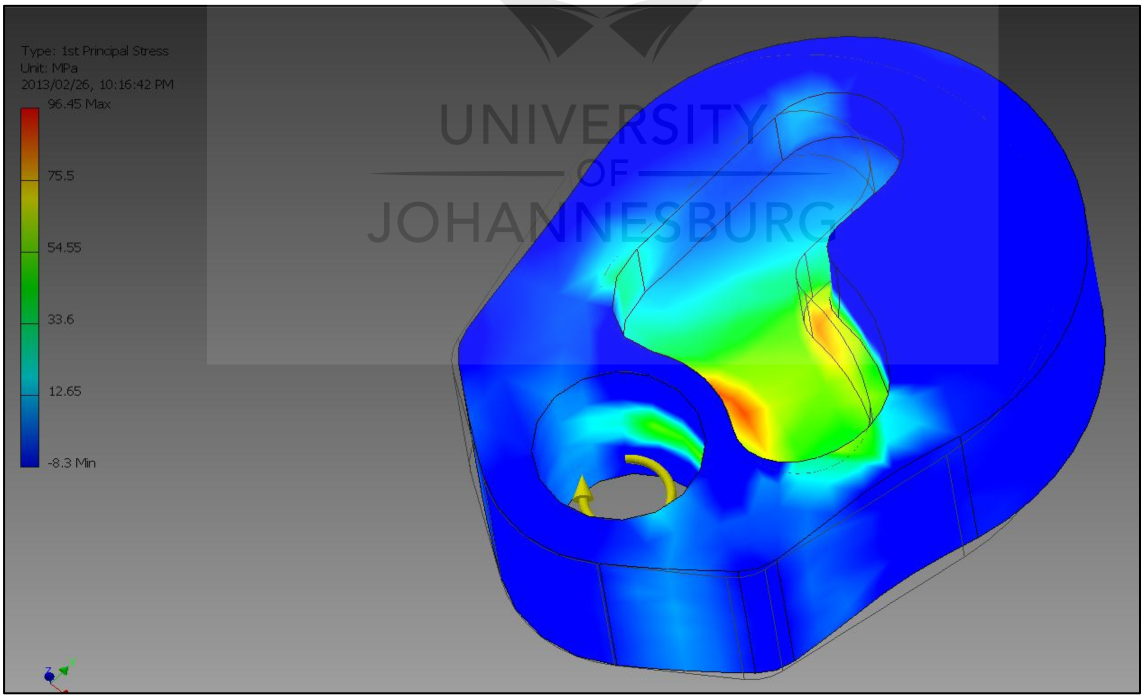
4. The maximum stress has increased at the wall from 140.8 MPa to 154.4 MPa. The higher maximum stress value has moved from the wall to the inside corner of the L shape due to the rib construction as indicated in the diagram below.



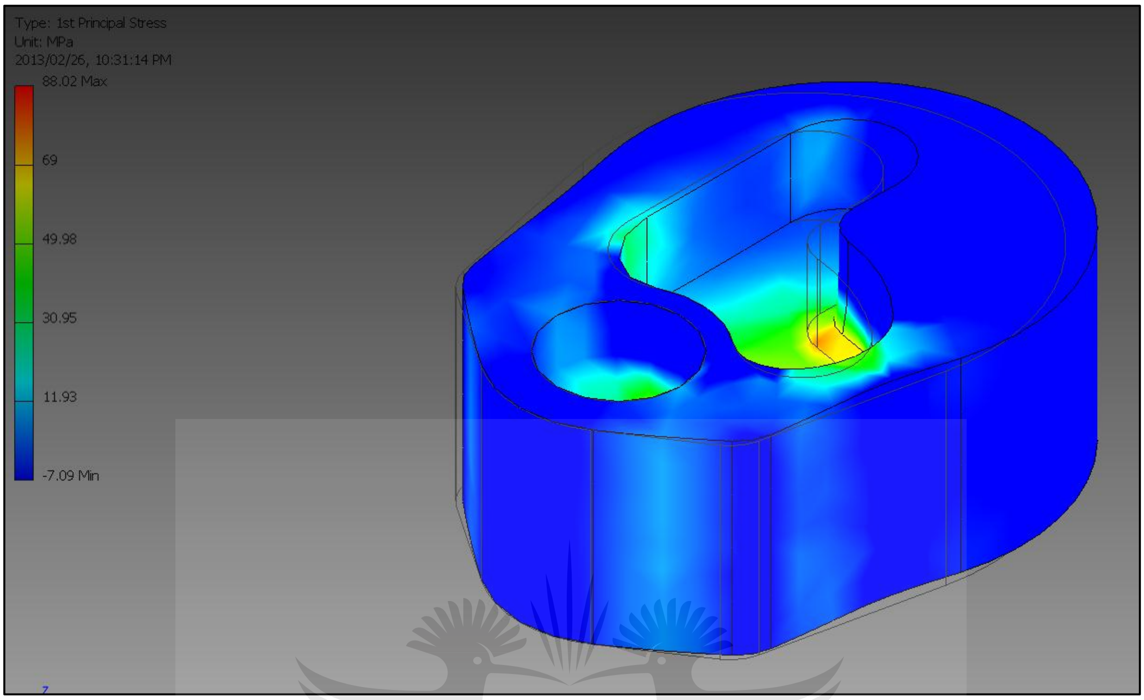
The inner corner radius of the L shape is increased from 10 mm to 20 mm to provide a much smoother stress flow. The L-shaped rib is also increased from 10 mm to 20 mm for extra stiffness.



5. The Maximum stress has reached a satisfactory safety level and the principle stress decreased considerably from 154.8 MPa to 96.45 MPa. This is very good progress.

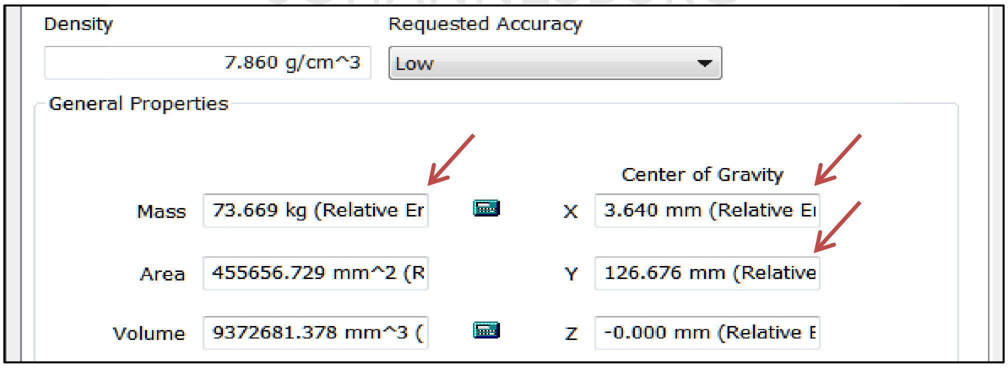


6. The inner corner radius of the L shape is increased again from 20 mm to 25 mm to provide a much smoother stress flow. The rib is also increased from 20 mm to 25 mm for extra stiffness. The final result yields a stress decrease from 96.45 MPa to 88.02 MPa.



Since the stress is now within satisfactory levels, the centrifugal force this weight can generate can be calculated.

See the figure below for the mass and the radius that is needed to calculate the centrifugal force. Pythagoras needs to be used to combine the X and Y coordinates.



Density: 7.860 g/cm^3

Requested Accuracy: Low

General Properties

Mass: 73.669 kg (Relative Er

Area: 455656.729 mm^2 (R

Volume: 9372681.378 mm^3 (

Center of Gravity

X: 3.640 mm (Relative Er

Y: 126.676 mm (Relative

Z: -0.000 mm (Relative E

$$\omega = \frac{2 \times \pi \times N}{60} = \frac{2 \times \pi \times 2200}{60} = 230,4\ rad/s$$

$$F_c = m \times r \times \omega^2 = 73.669 \times \sqrt{0.00364^2 + 0.126676^2} \times 230.4^2 = 495,491\ kN$$

The calculated centrifugal force is way too high when compared with the SDVRR's as seen in the Appendix15 - 20.

The lowest high amplitude force is 245 kN (Sakai catologue) and the highest is 412 kN (Bomag catologue). One of the contraints the customer Demco (Pty) Ltd wanted, was to stick as close as possible to the catipillar catologue where the high amplitude is 266 kN.

For further calculations, a centrifugal force of 250 kN on high amplitude was assumed. In order to achieve the asumed force, the mass will have to be reduced and the effective eccentric moment to the center of gravity has to be altered.

The following steps were taken to get closer to the assumed force of 250 kN (Focus need to be placed on the next three figures A, B and C);

1. the high amplitude and low amplitude radius center points (see the arrow) was taken further apart (121.2 mm to 160.5 mm) to yield a greater difference between the high and low centrifugal forces (Figure A),

2. the mass is decreased from 73,669 kg to 37.978 kg as seen in the Figure B,

3. and the maximum stress dropped even lower from 88.02 MPa to 68,69 MPa, because of the alteration. See Figure C.

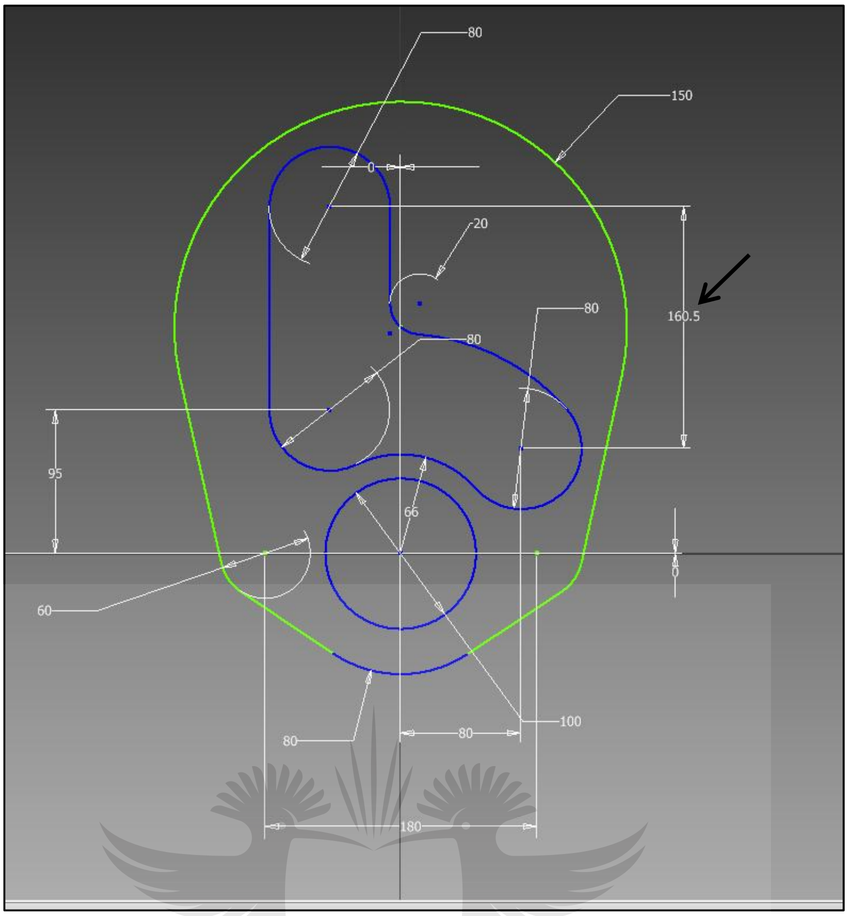
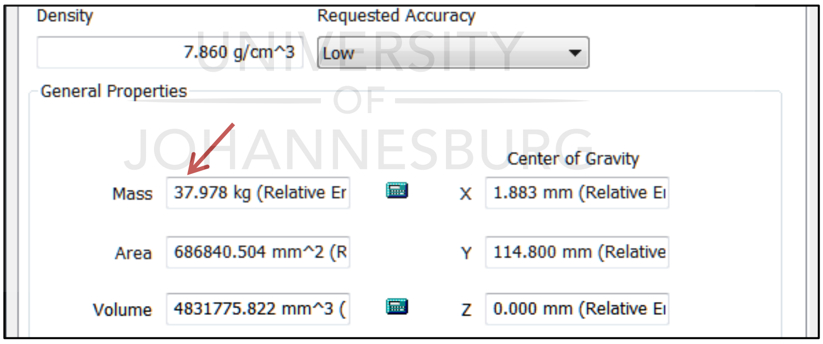


Figure A



Density

7.860 g/cm^3

Requested Accuracy

Low

General Properties

Center of Gravity

Mass	37.978 kg (Relative Er	X	1.883 mm (Relative Ei
Area	686840.504 mm^2 (R	Y	114.800 mm (Relative
Volume	4831775.822 mm^3 (	Z	0.000 mm (Relative Ei

Figure B

7. Max. stress decreased from 88.02 MPa to 68.68 MPa.

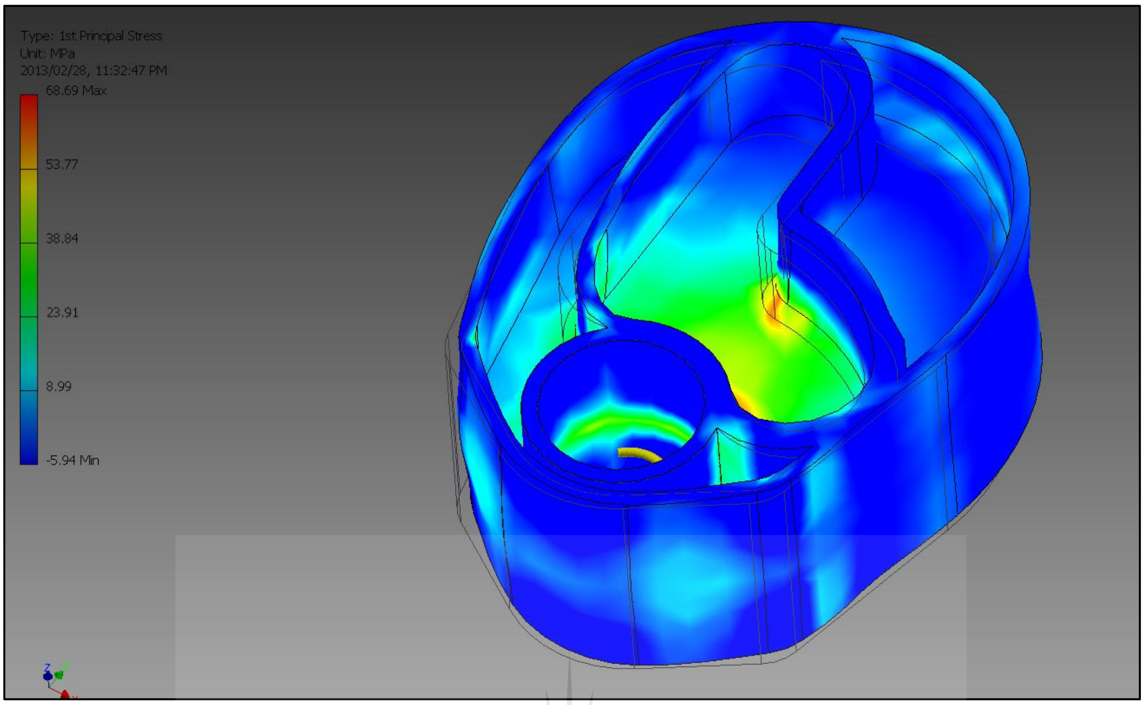
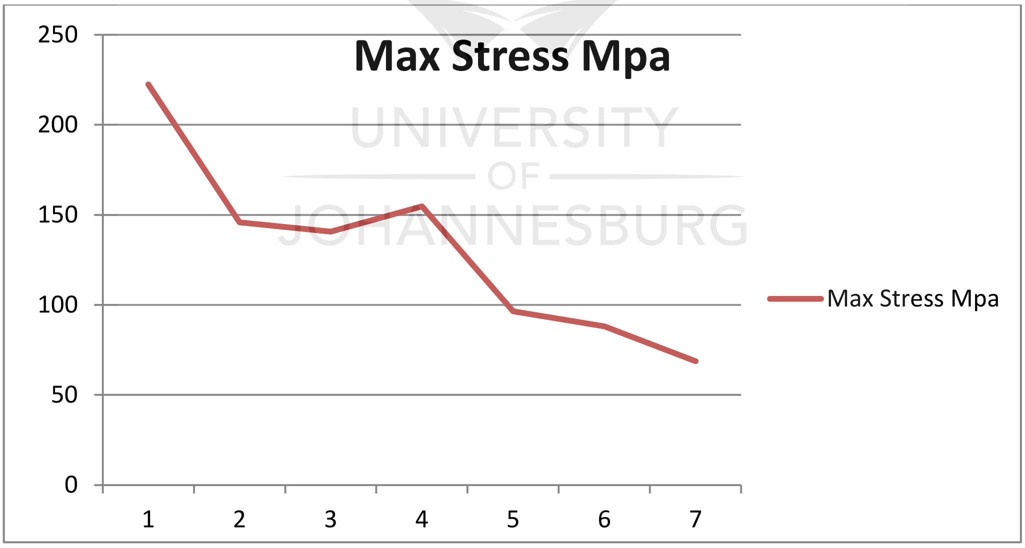


Figure C

Stress graph for all seven simulations.



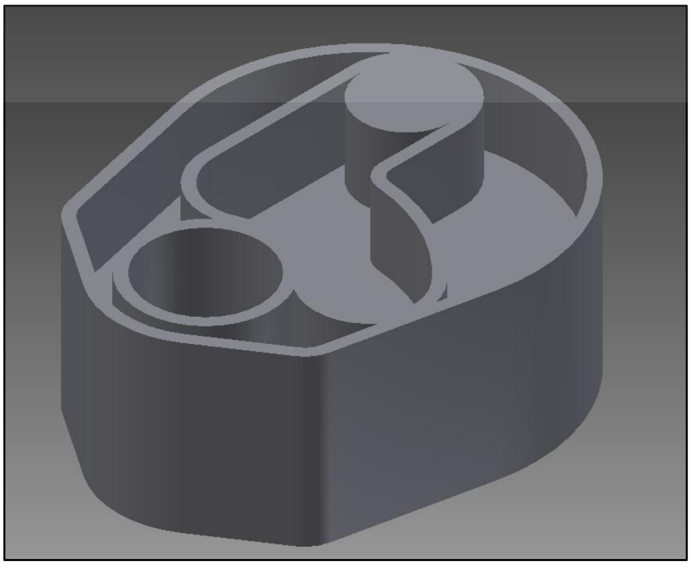
Tabular result for all simulations 1 to 7.

Stress Simulation Results	
Simulation Number	Maximum Stress MPa
1	222.4
2	145.9
3	140.8
4	154.8
5	96.45
6	88.02
7	68.69

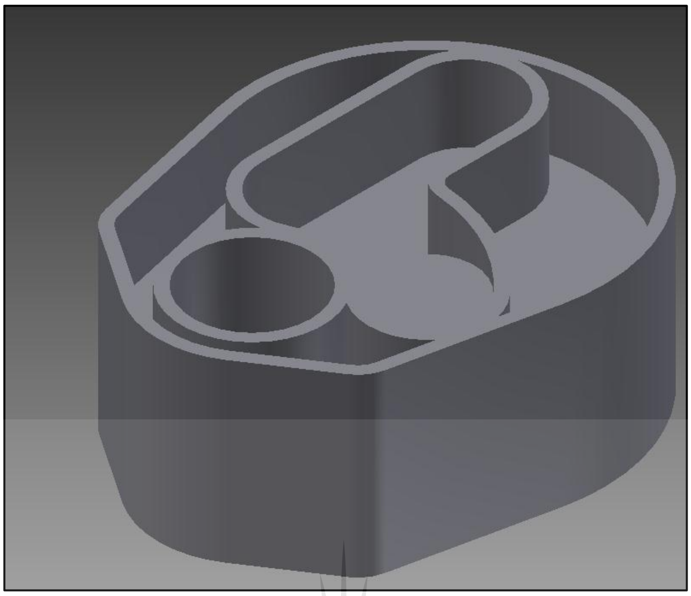
The graph indicates how geometry was minipulated step by step in order to achieve such a huge stress drop of 69.2 %.

Adding 5 kg of ball bearings, high amplitude and low amplitude compaction can be achieved. This is, because of the eccentric distance increases from the shaft centre to the ball bearings. For Anticlockwise rotation, low amplitude can be achieved due to all the ball bearings rolling into the lower end of the L-shape and visa versa for the high amplitude.

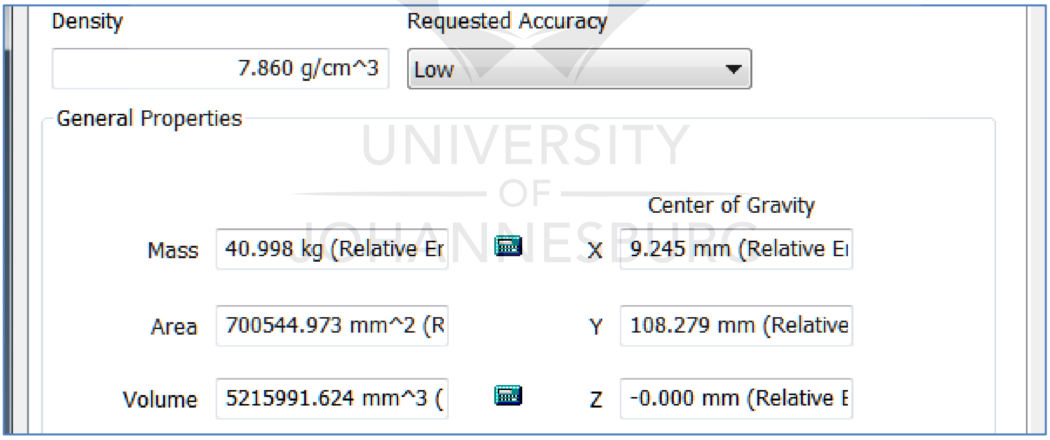
The figure below shows the high amplitude arrangement, where a cylinder represents the mass of the ball bearings altogether. The generated centrifugal force will have the maximum value.



The next figure shows the low amplitude arrangement, where the cylinder represents the mass of the ball bearings. The generated centrifugal force will in this case have a minimum value.



Calculation of the centrifugal force Fc.



Density
7.860 g/cm^3

Requested Accuracy
Low

General Properties

Center of Gravity

Mass 40.998 kg (Relative Er
X 9.245 mm (Relative E

Area 700544.973 mm^2 (R
Y 108.279 mm (Relative

Volume 5215991.624 mm^3 (
Z -0.000 mm (Relative E

$$\boldsymbol{F_c = m \times r \times \omega^2 = m \times \sqrt{X^2 + Y^2} \times \omega^2}$$

High frequency F_c = 63230.8 kN (5kg, 234.8mm, 230.4rad/s), this force is for the ball bearings only.

High amplitude = 279802.4 kN (128.5mm, 41kg, 230.4rad/s) (ball bearings + main eccentric mass). This force is for the eccentric weight itself.

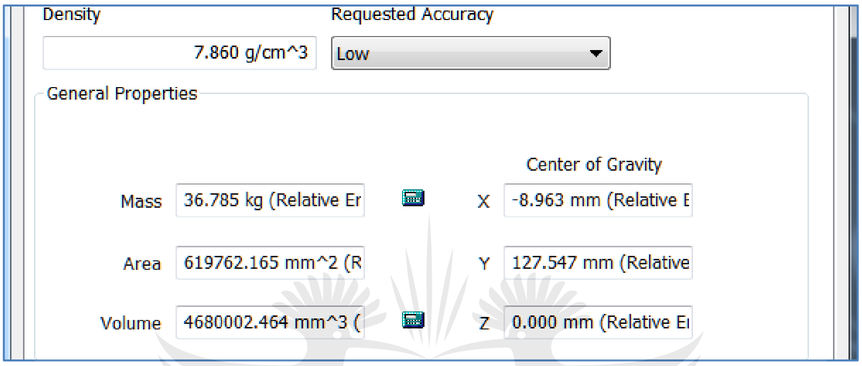
Low frequency F_c = 281315 kN (5kg, 106mm, 230.4rad/s), this force is for the ball bearings alone.

Low amplitude = 236521.418 kN (41kg, 108.673mm, 230.4rad/s) (ball bearings + main mass). This force is for the eccentric weight itself.

Since the assumed target is 250 kN for the maximum centrifugal force of the machine, the centrifugal weight geometry needs to be revisited and the maximum and minimum centrifugal forces need to be recalculated to achieve a lower force output.

Revisiting the eccentric weight geometry that will alter the mass and eccentric moment.

High amplitude



Density 7.860 g/cm^3

Requested Accuracy Low

General Properties

Center of Gravity

Mass 36.785 kg (Relative Er

X -8.963 mm (Relative E

Area 619762.165 mm^2 (R

Y 127.547 mm (Relative

Volume 4680002.464 mm^3 (

Z 0.000 mm (Relative Er

In the above table, the mass has been decreased from 40.98 kg to 36.785 kg and the eccentric moment has also been altered. The value for X, previously was 9.245 mm and now is -8.963 mm. The value for Y, previously was 108.279 mm and now 127.547 mm.

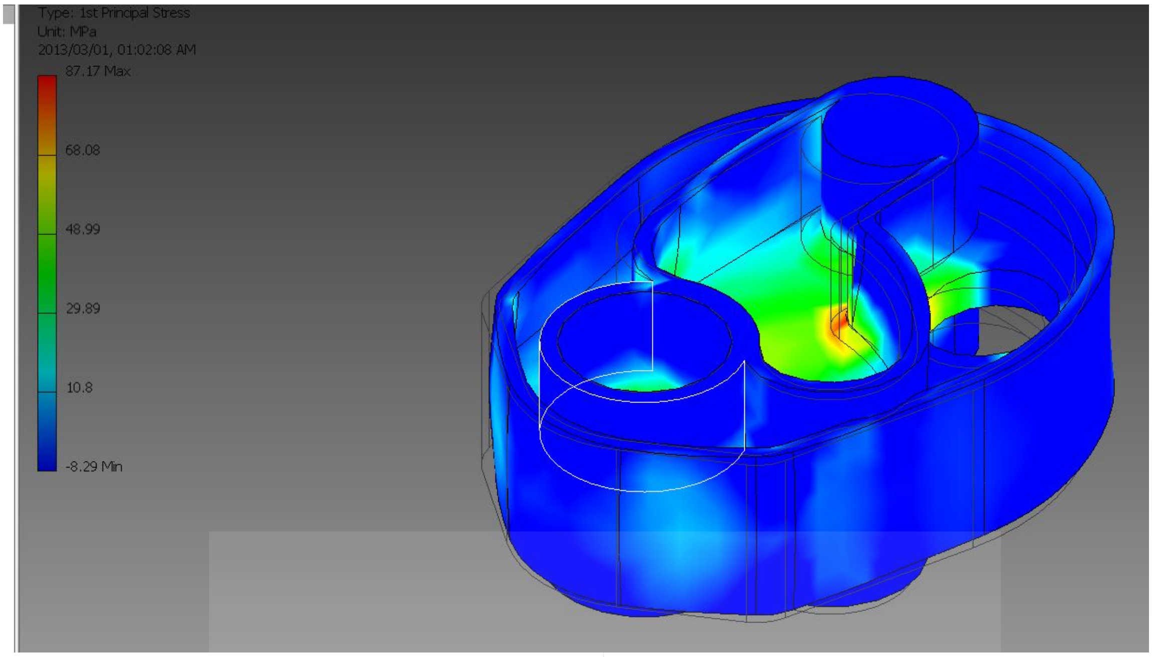
The centrifugal force for high amplitude will then be:

$$\boldsymbol{F_c = m \times \sqrt{X^2 + Y^2} \times \omega^2 = 36.785 \times \sqrt{-0.008963^2 + 0.127547^2} \times 230.4^2 = 249675.33\ N}$$

The 249.675 kN force is very close to the estimated 250 kN force and is therefore satisfactory.

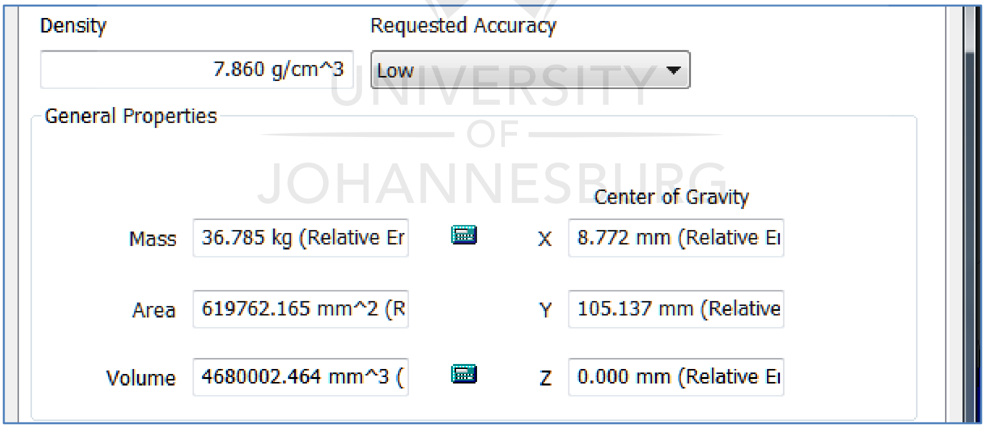
Now, the actual stress that the eccentric weight has to withstand can be simulated with the above calculated centrifugal force.

After re-simulation, the maximum stress is still low at 87.17 MPa.



Low Amplitude

With the cylinder in the lower position, the Y dimension for the eccentric moment changed from 127.547 mm to 105.137 mm. The X dimension remained the same.



Density: 7.860 g/cm^3

Requested Accuracy: Low

General Properties

Mass: 36.785 kg (Relative Er

Area: 619762.165 mm^2 (R

Volume: 4680002.464 mm^3 (

Center of Gravity

X: 8.772 mm (Relative Er

Y: 105.137 mm (Relative

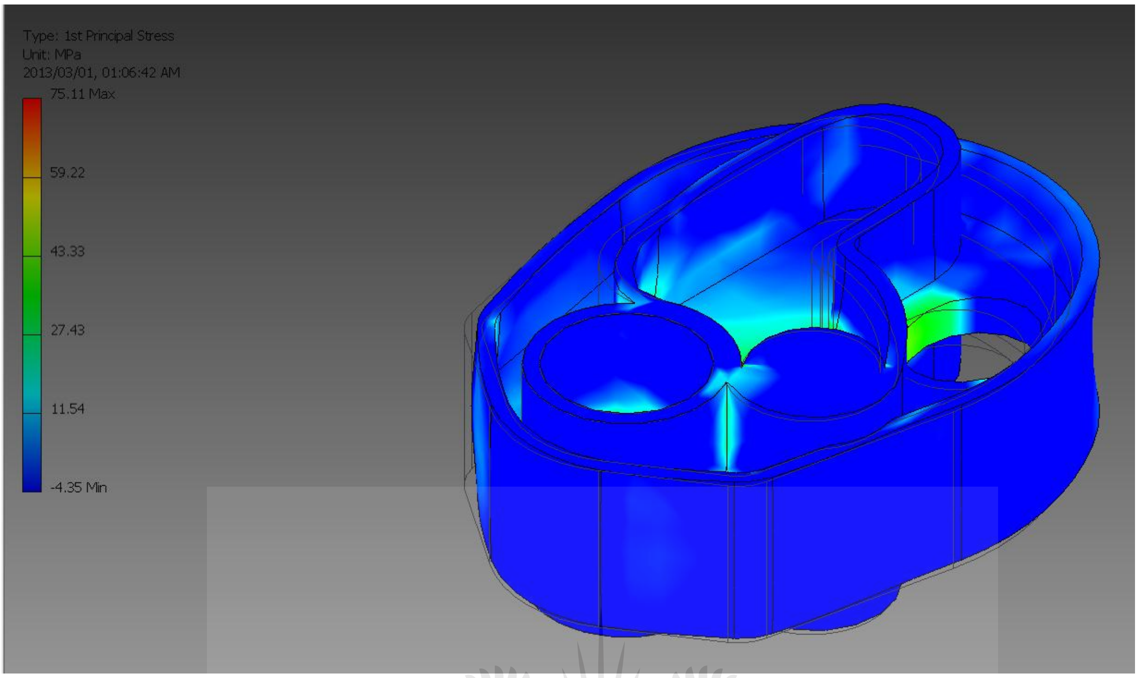
Z: 0.000 mm (Relative Er

Calculating the low amplitude centrifugal force:

$$\boldsymbol{F_c = m \times \sqrt{X^2 + Y^2} \times \omega^2 = 36.785 \times \sqrt{-0.008772^2 + 0.105137^2} \times 230.4^2 = 206\ kN}$$

The 206 kN low amplitude force is 44 kN less than its high amplitude counterpart and therefore satisfactory for low amplitude compaction.

Redoing a stress analysis in the low amplitude position yielded a maximum stress value of 75.11 MPa.



Appendix 27: Inventor MS design simulation results 1.

Material

Material		User material
Modulus of Elasticity	E	206000 MPa
Modulus of Rigidity	G	80000 MPa
Density	ρ	7860 kg/m^3

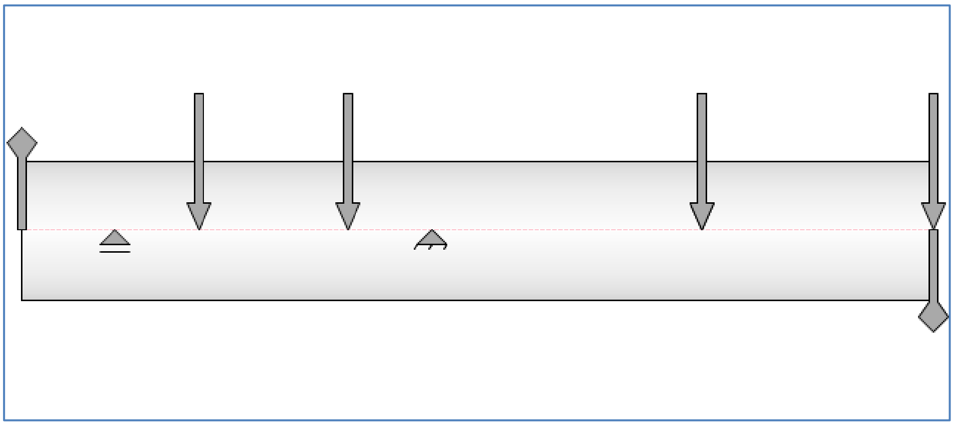
Calculation Properties

Include			
Yes	Density	ρ	7860 kg/m^3
Yes	Shear Displacement Ratio	β	1.188 ul
	Number of Divisions		1000 ul
	Mode of reduced stress		HMH

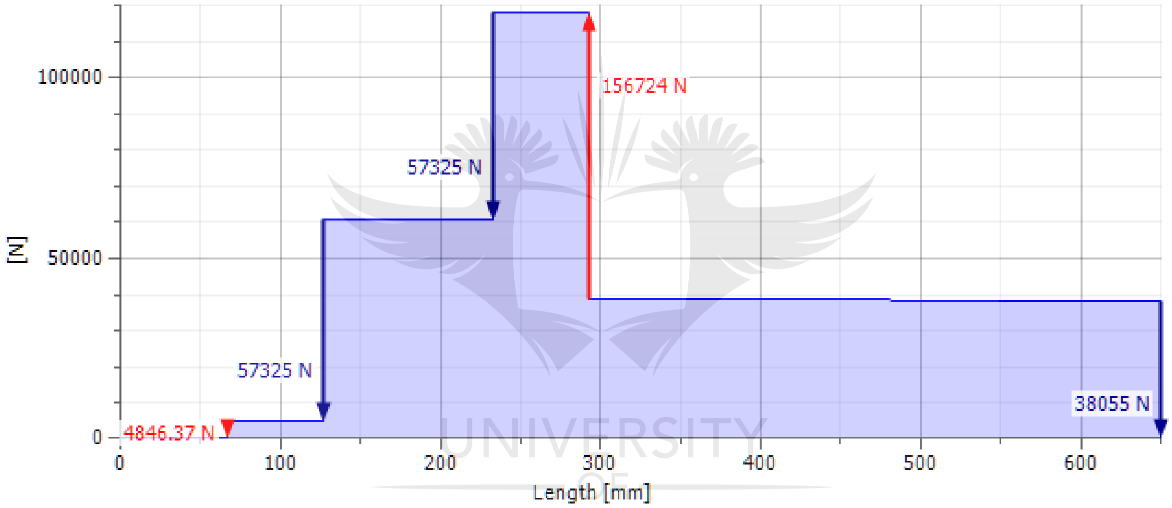
Results

Length	L	650.000 mm
Mass	Mass	40.126 kg
Maximal Bending Stress	σ_B	139.540 MPa
Maximal Shear Stress	τ_S	15.043 MPa
Maximal Torsional Stress	τ	1.006 MPa
Maximal Tension Stress	σ_T	0.000 MPa
Maximal Reduced Stress	σ_{red}	141.963 MPa
Maximal Deflection	f_{max}	906.556 microm
Angle of Twist	φ	0.01 deg

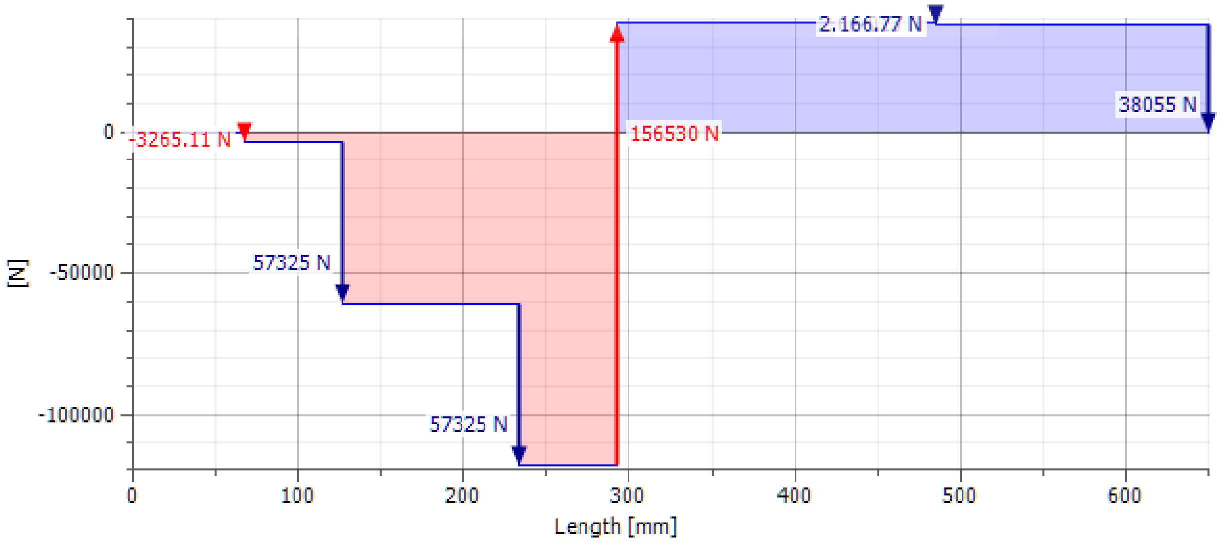
Preview



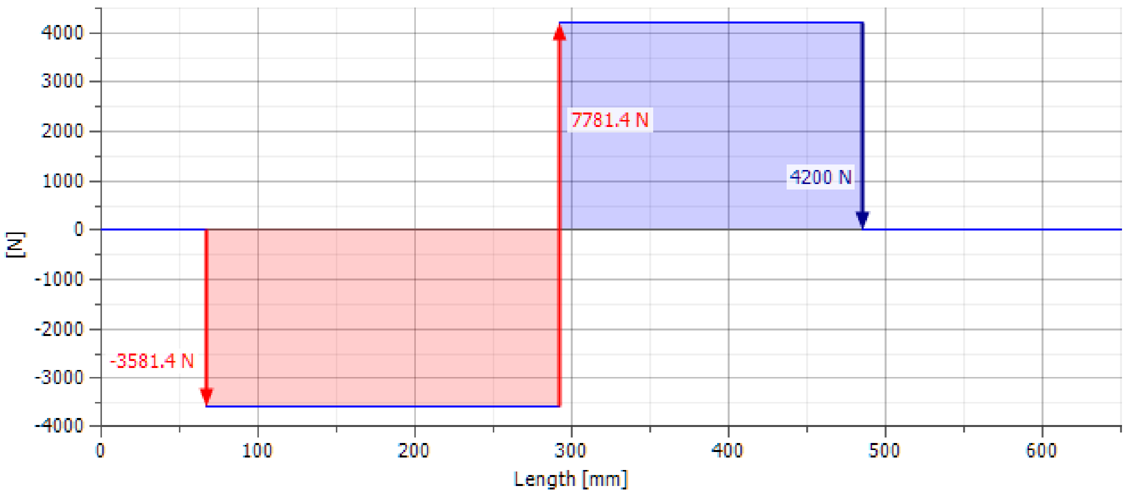
Shear Force



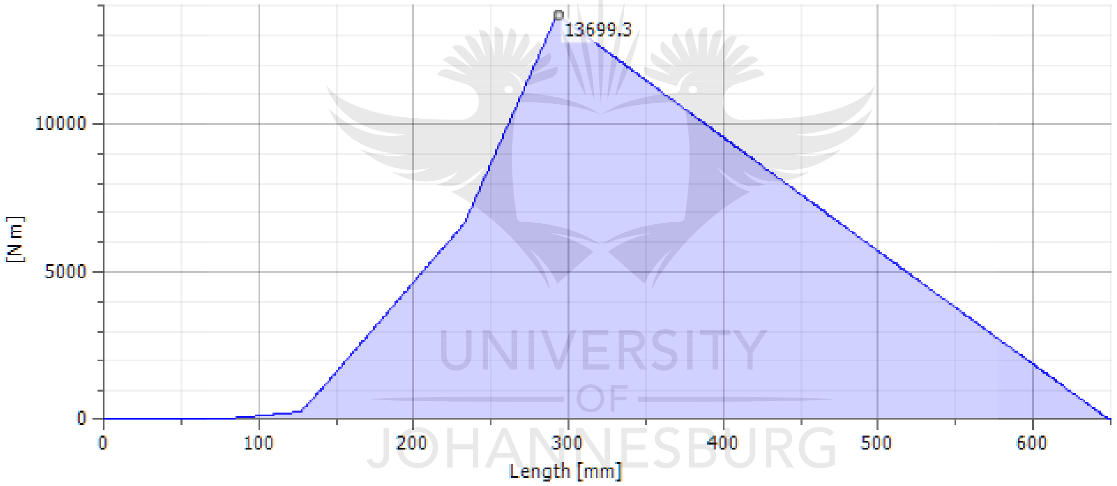
Shear Force, YZ Plane



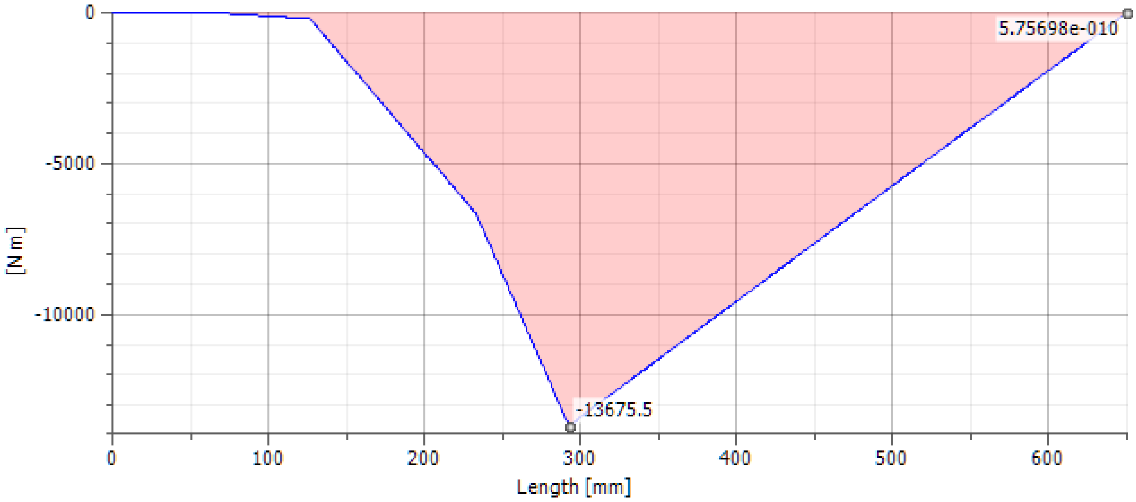
Shear Force, XZ Plane



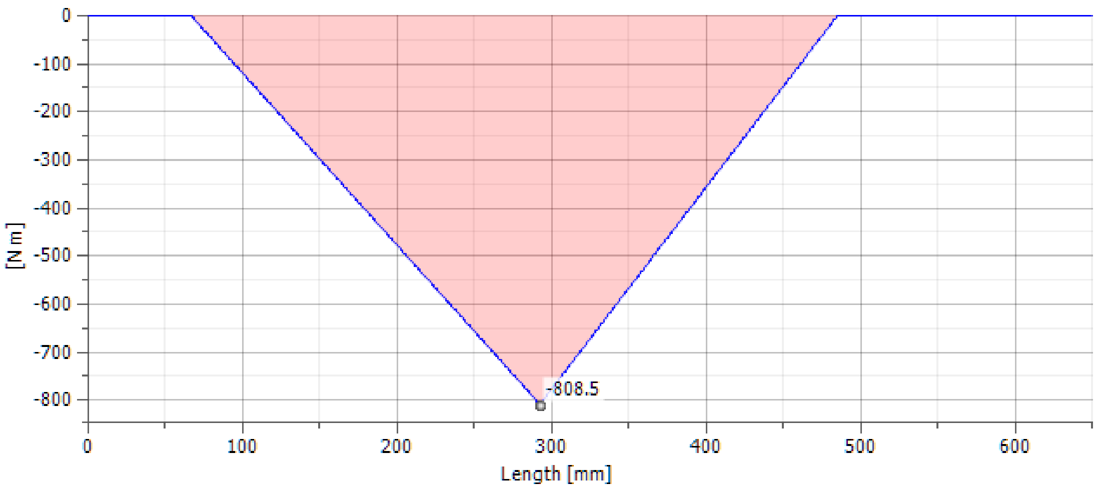
Bending Moment



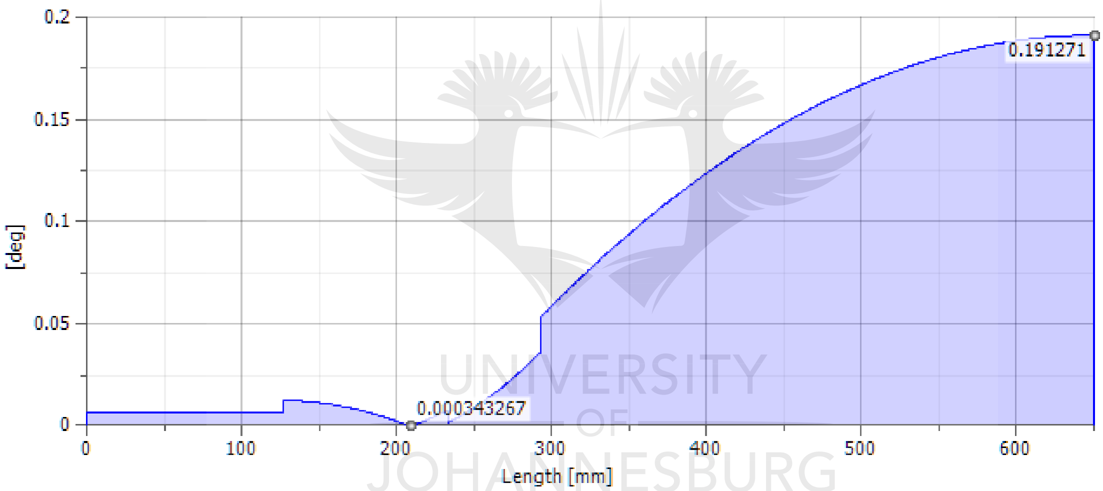
Bending Moment, YZ Plane



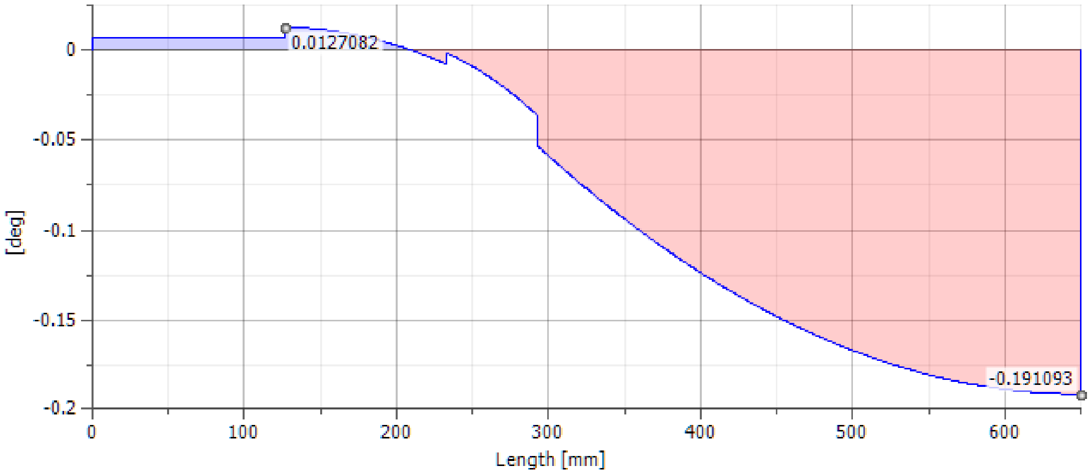
Bending Moment, XZ Plane



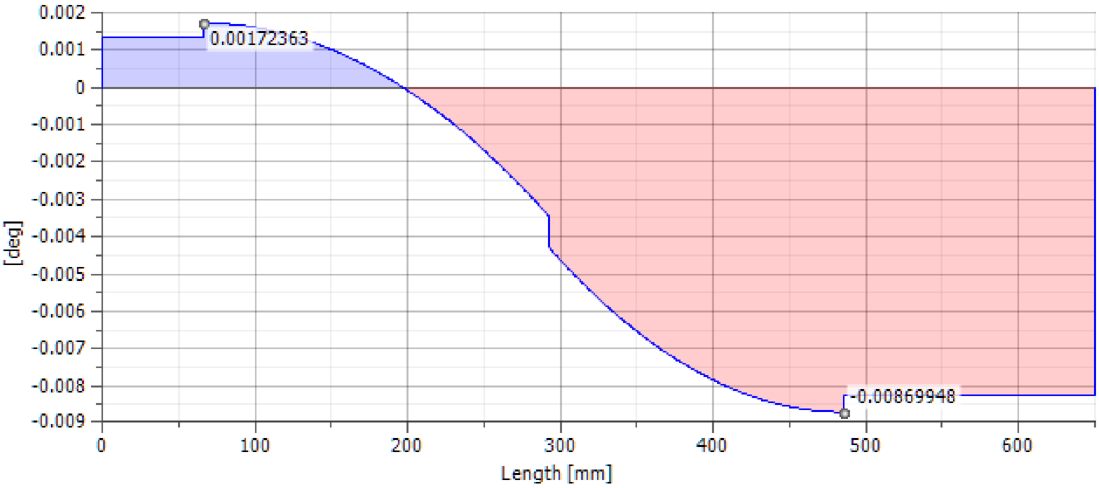
Deflection Angle



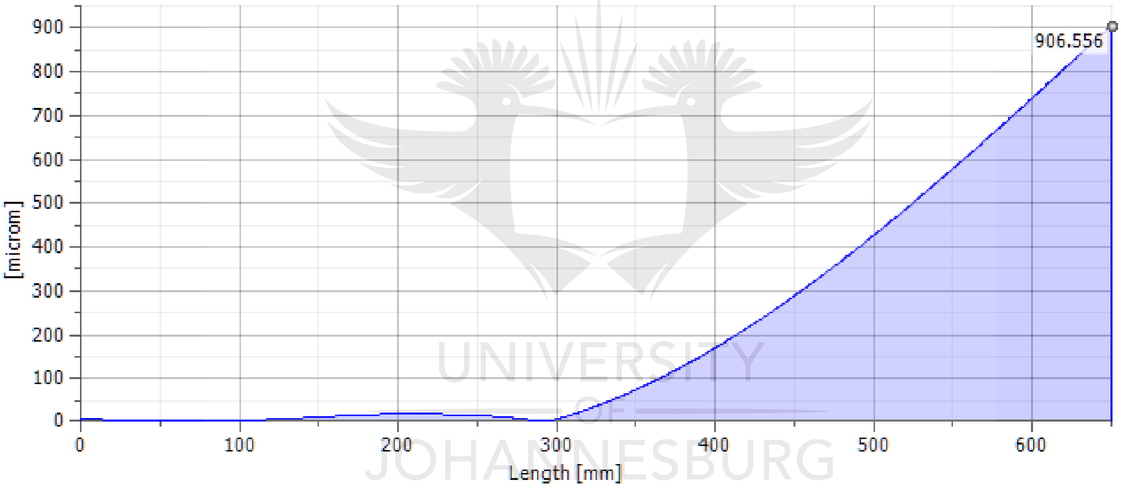
Deflection Angle, YZ Plane



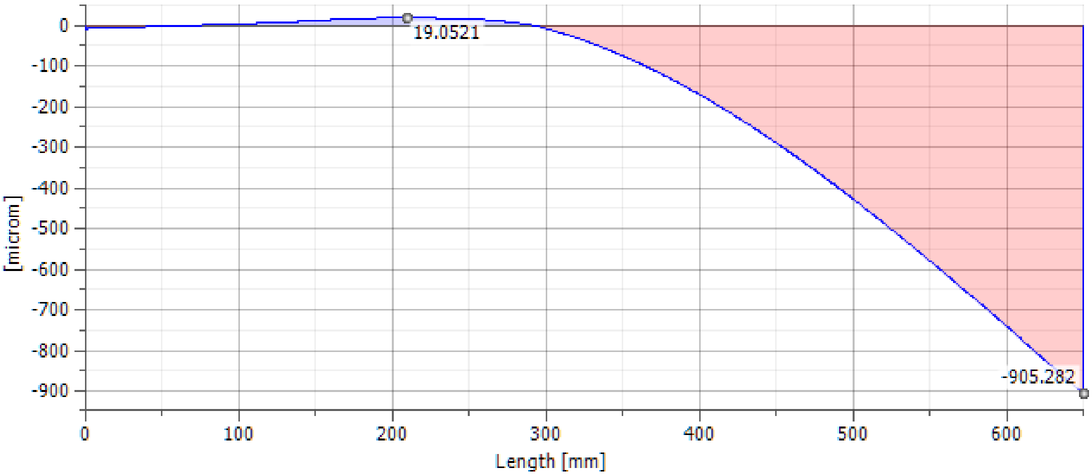
Deflection Angle, XZ Plane



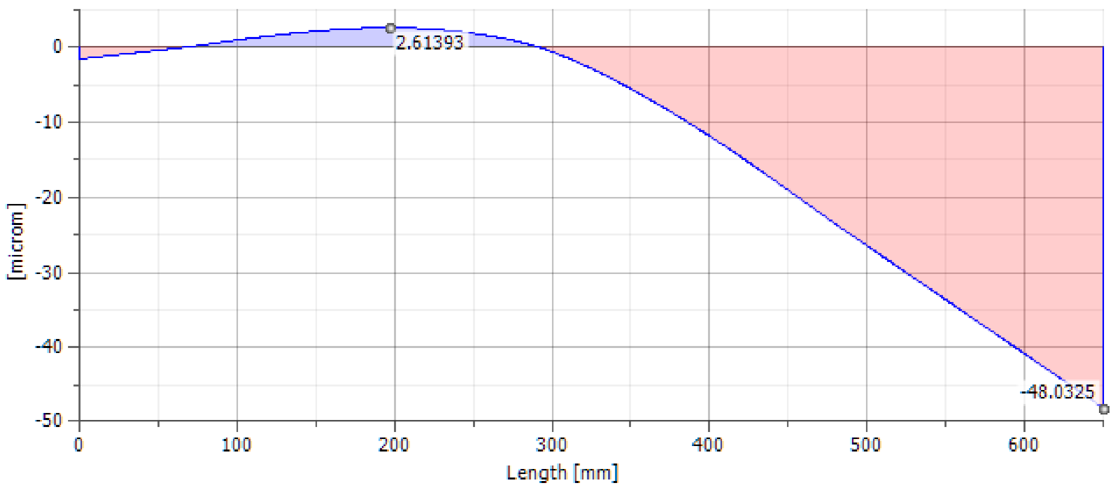
Deflection



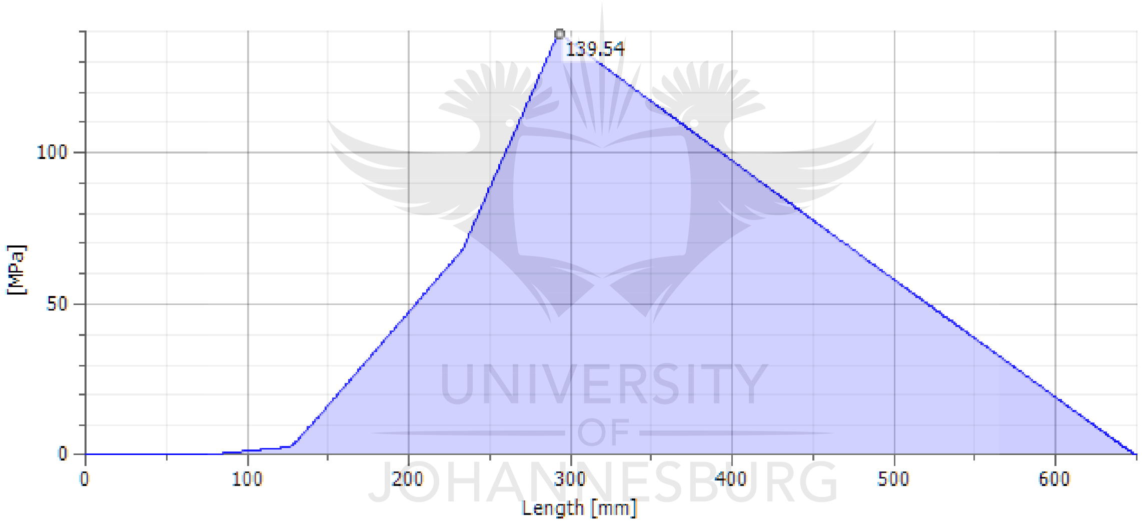
Deflection, YZ Plane



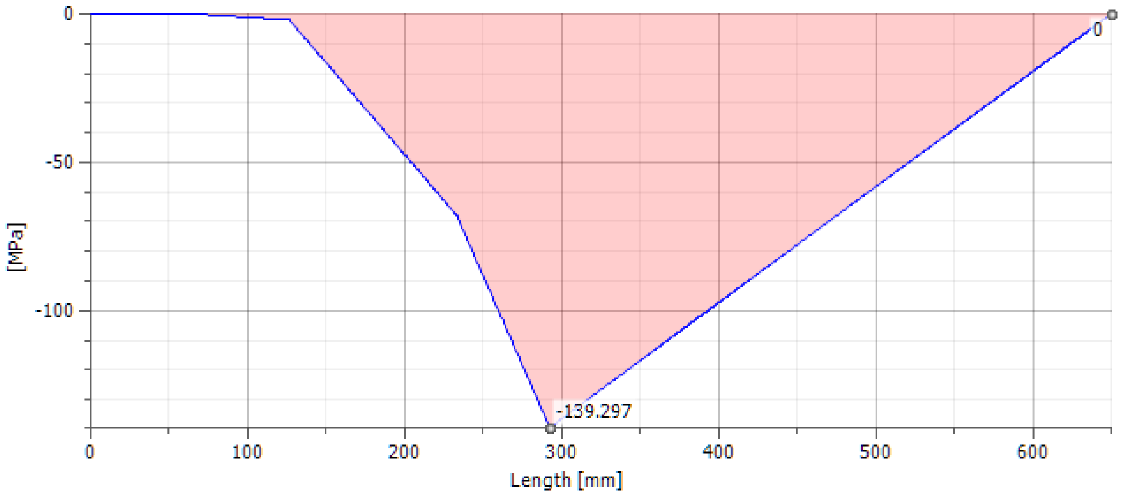
Deflection, XZ Plane



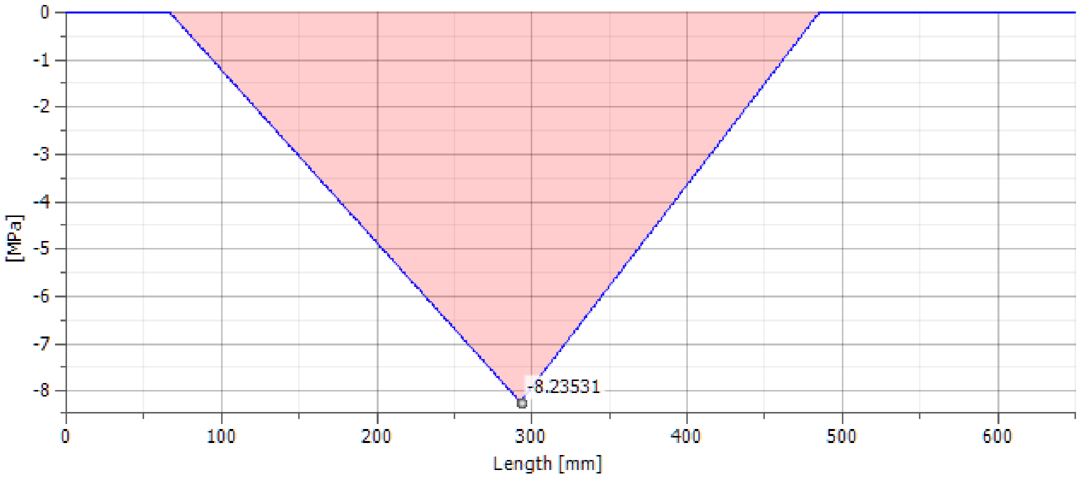
Bending Stress



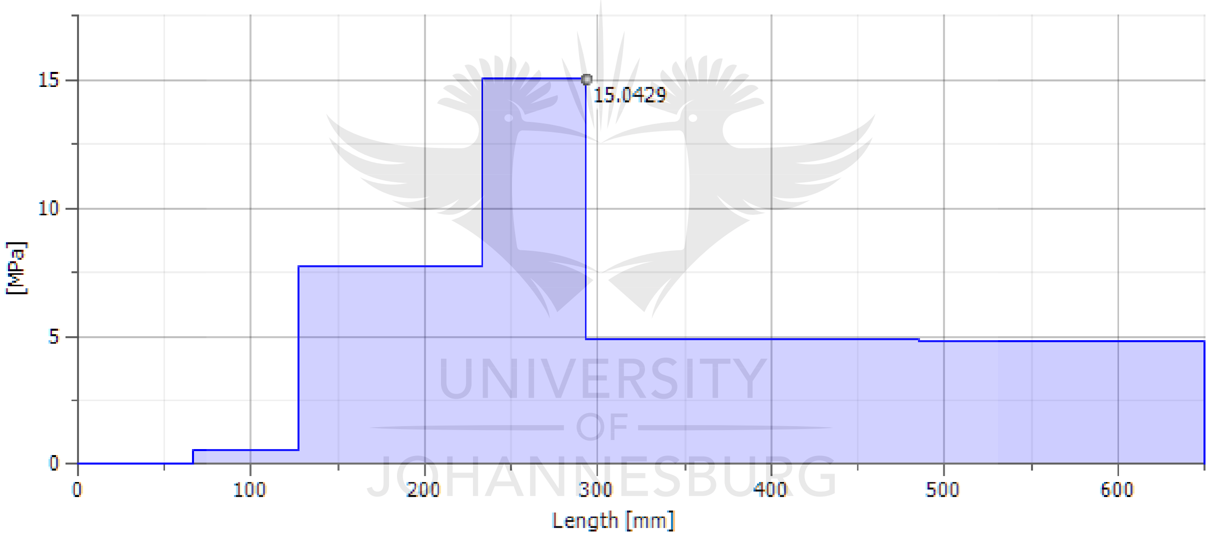
Bending Stress, YZ Plane



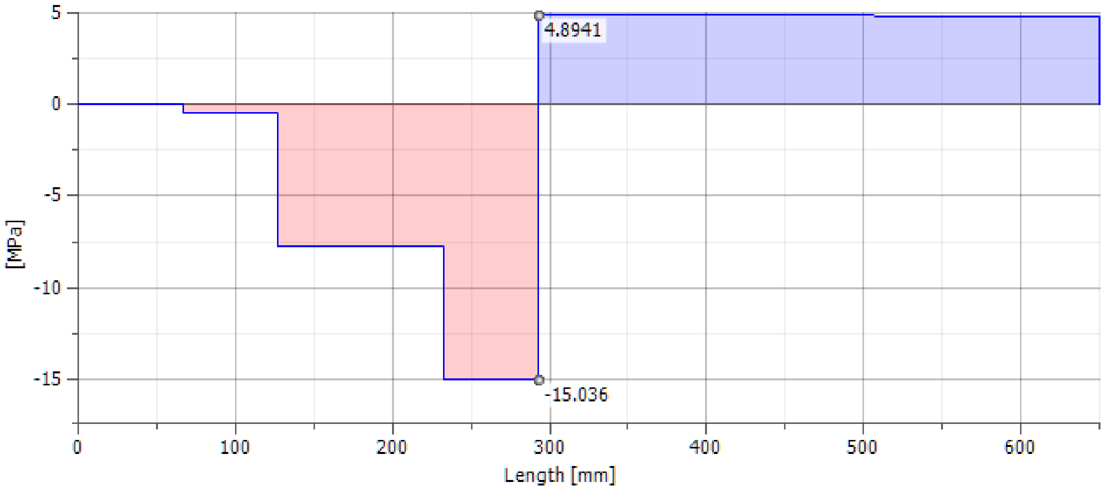
Bending Stress, XZ Plane



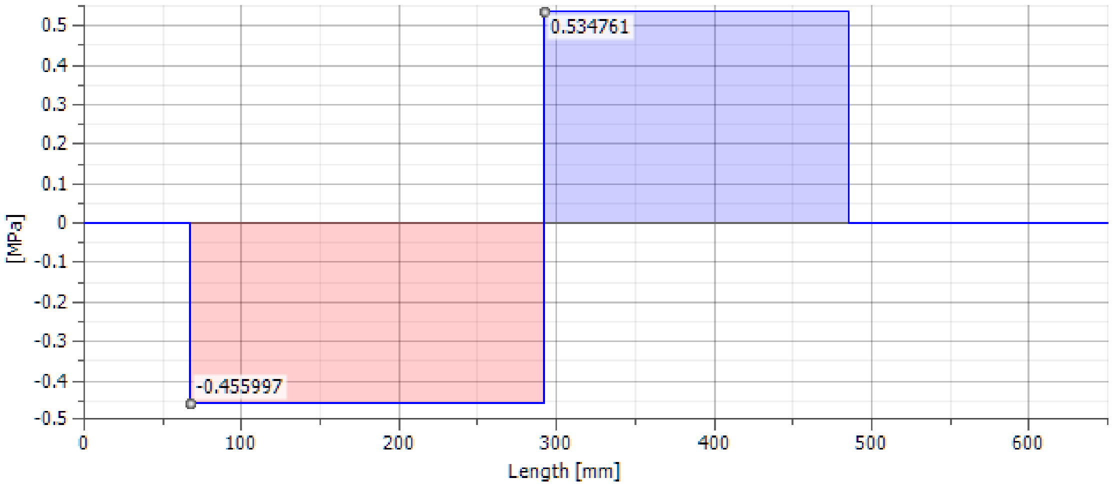
Shear Stress



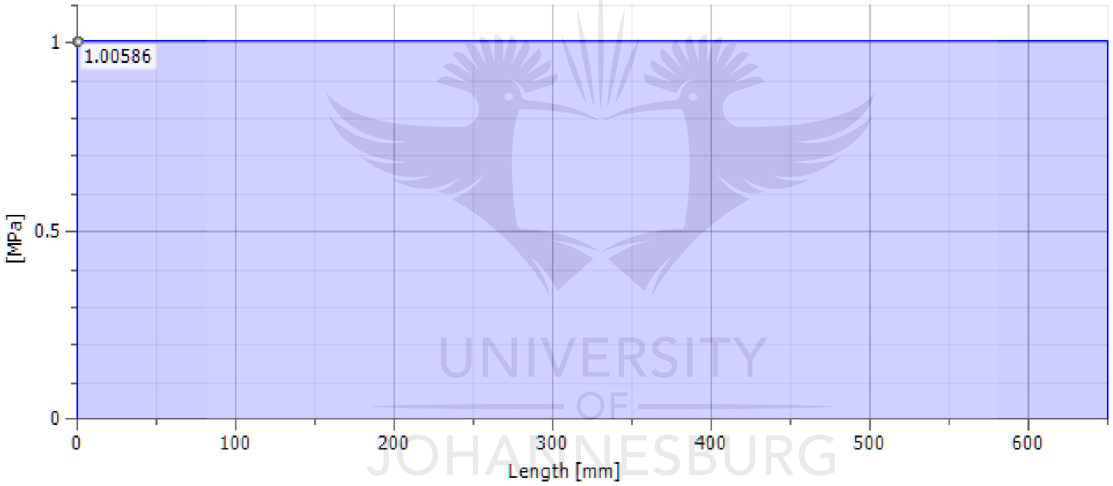
Shear Stress, YZ Plane



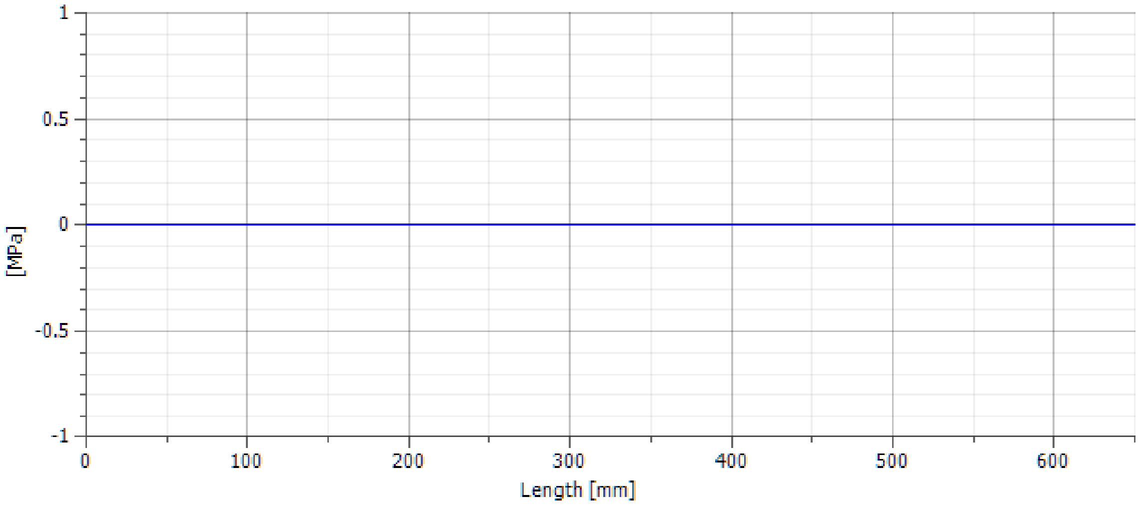
Shear Stress, XZ Plane



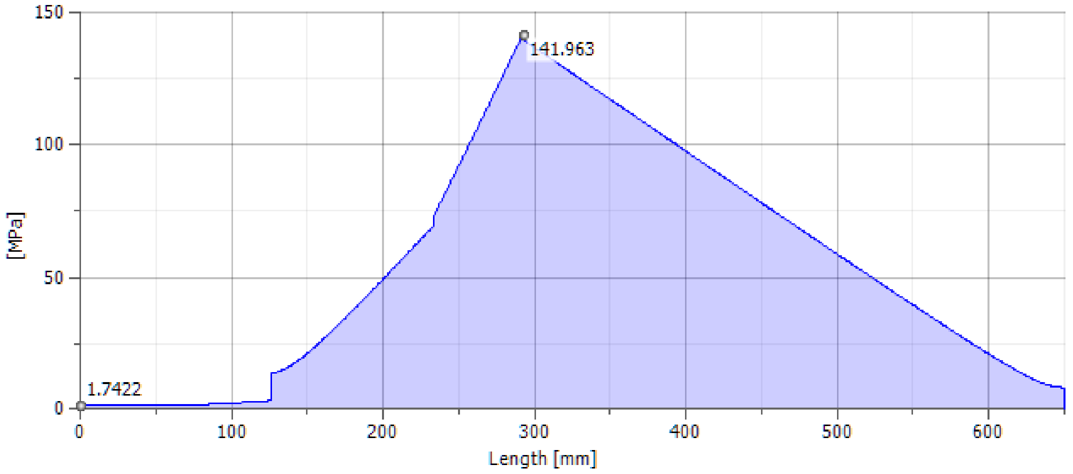
Torsional Stress



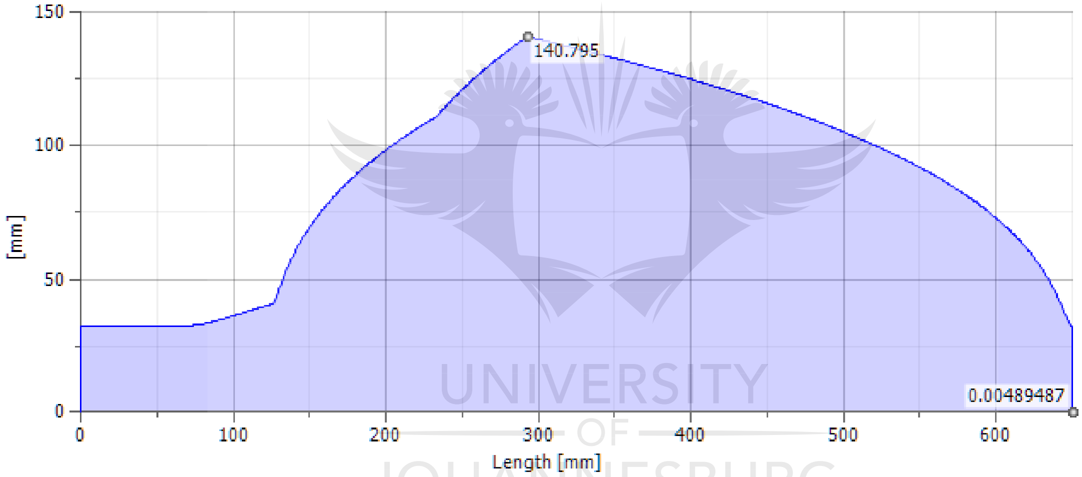
Tension Stress



Reduced Stress



Ideal Diameter



Appendix 28: Inventor MS refinement simulation design results.

Material

Material		User material
Modulus of Elasticity	E	206000 MPa
Modulus of Rigidity	G	80000 MPa
Density	ρ	7860 kg/m^3

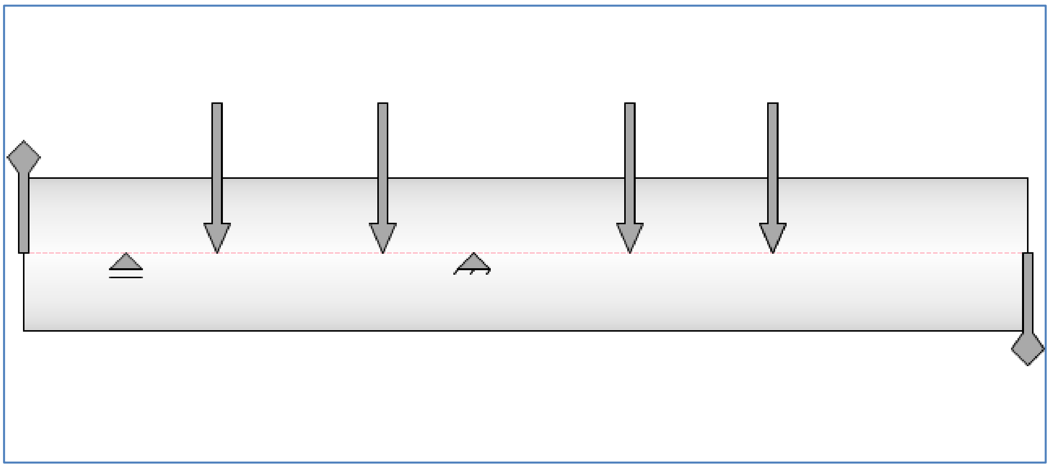
Calculation Properties

Include			
Yes	Density	ρ	7860 kg/m^3
Yes	Shear Displacement Ratio	β	1.188 ul
	Number of Divisions		1000 ul
	Mode of reduced stress		HMH

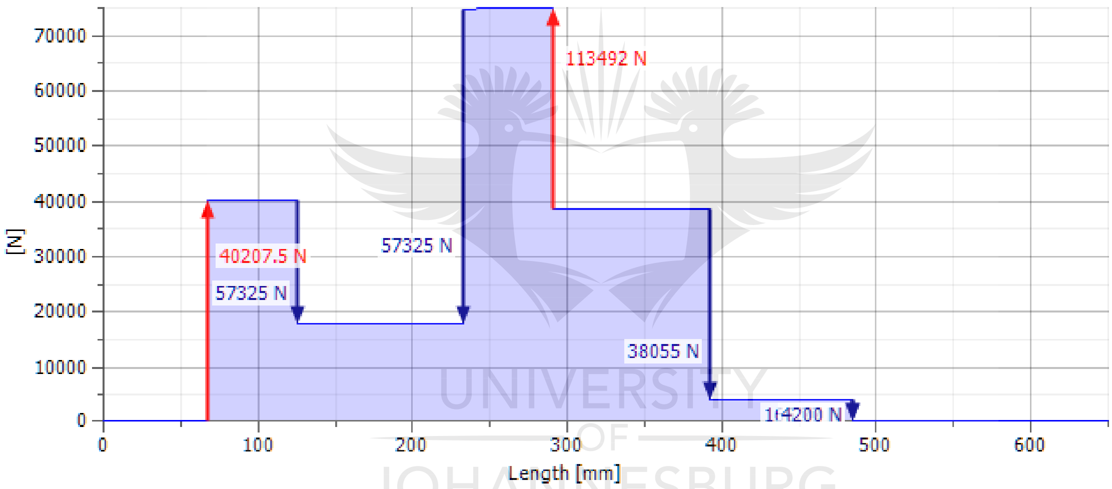
Results

Length	L	650.000 mm
Mass	Mass	40.126 kg
Maximal Bending Stress	σ_B	40.634 MPa
Maximal Shear Stress	τ_S	9.533 MPa
Maximal Torsional Stress	τ	1.006 MPa
Maximal Tension Stress	σ_T	0.000 MPa
Maximal Reduced Stress	σ_{red}	43.895 MPa
Maximal Deflection	f_{max}	100.988 microm
Angle of Twist	φ	0.01 deg

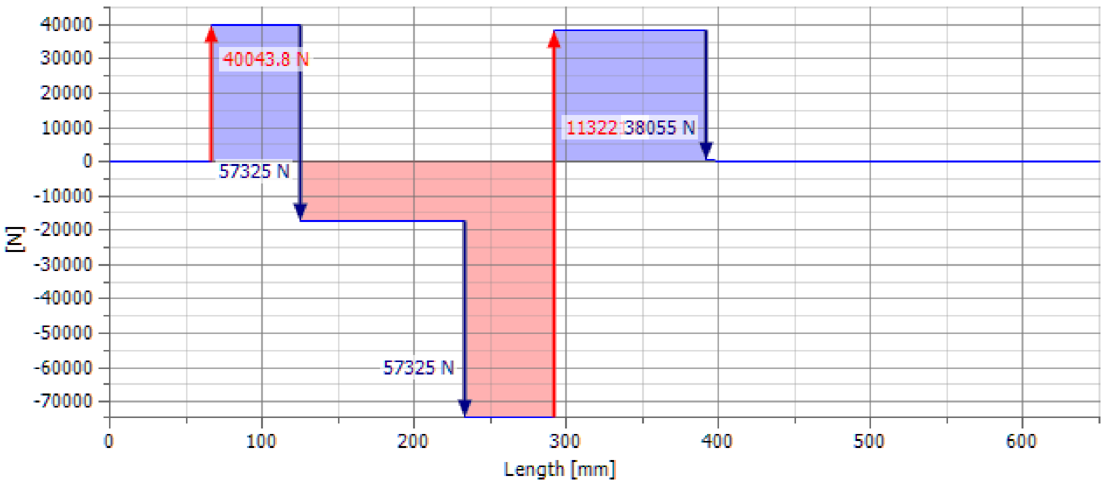
Preview



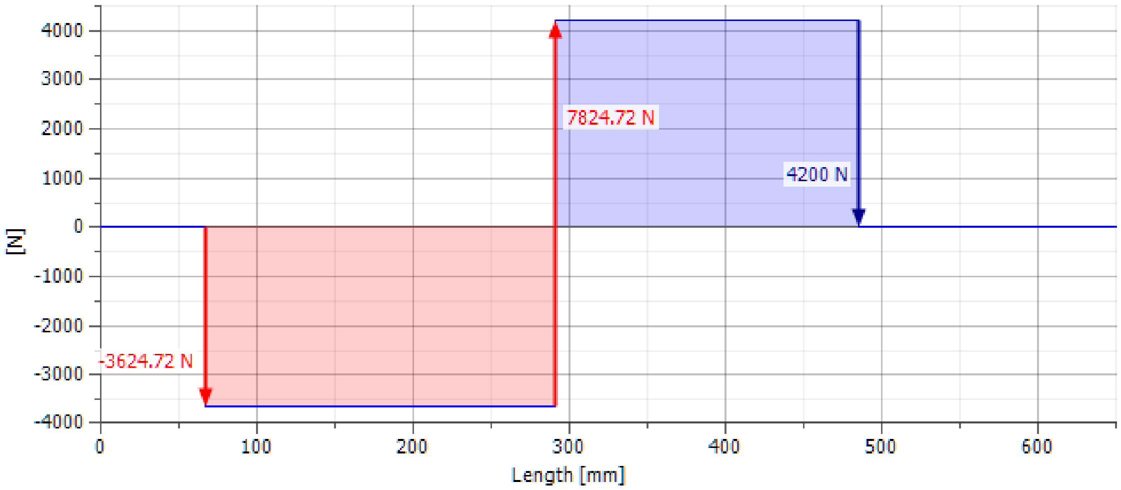
Shear Force



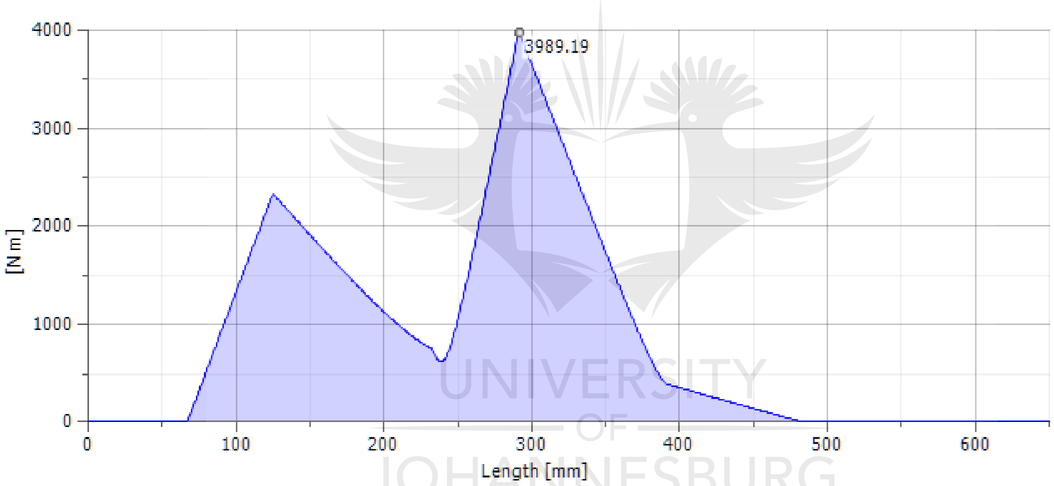
Shear Force, YZ Plane



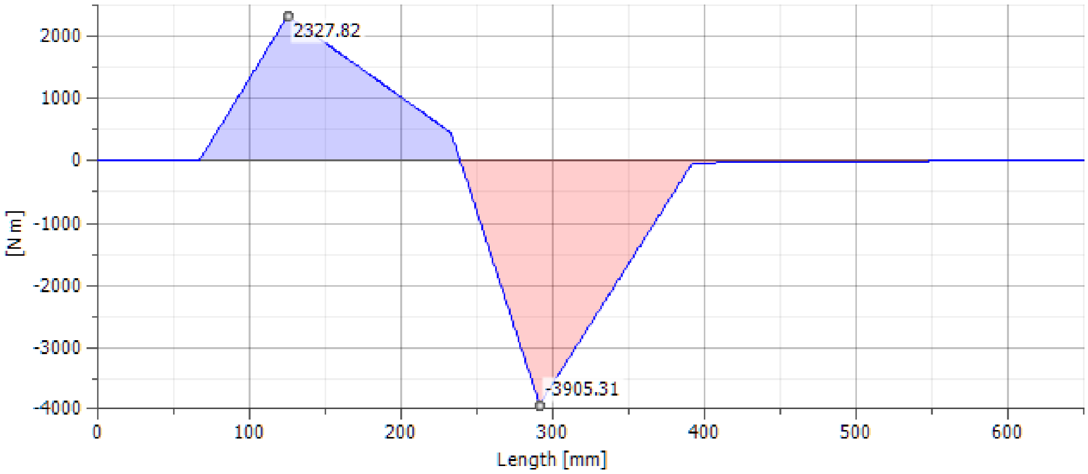
Shear Force, XZ Plane



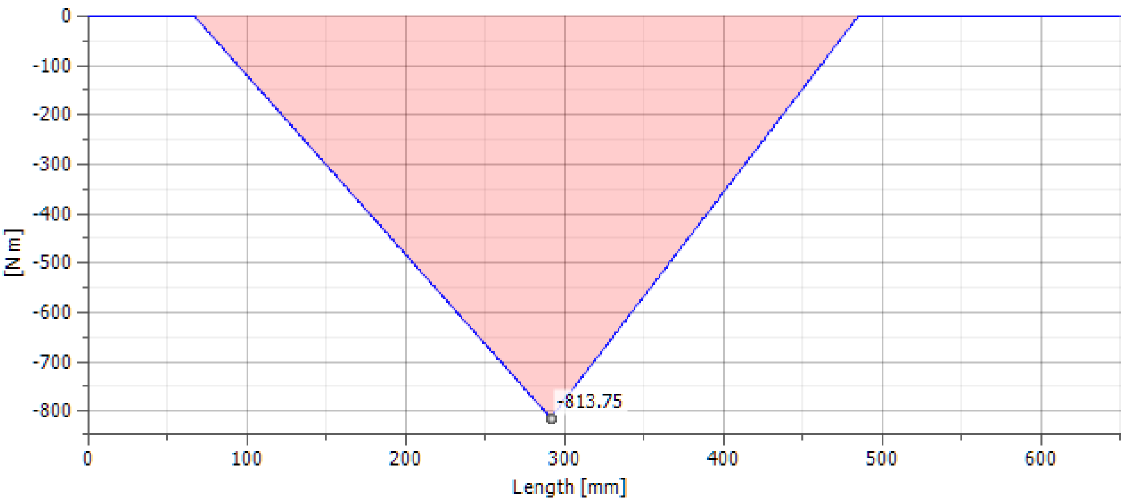
Bending Moment



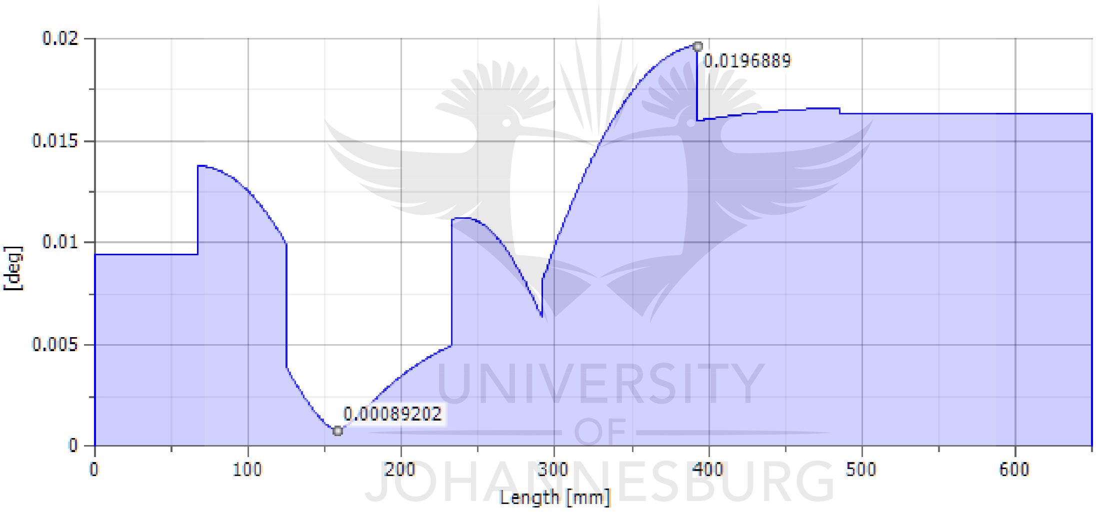
Bending Moment, YZ Plane



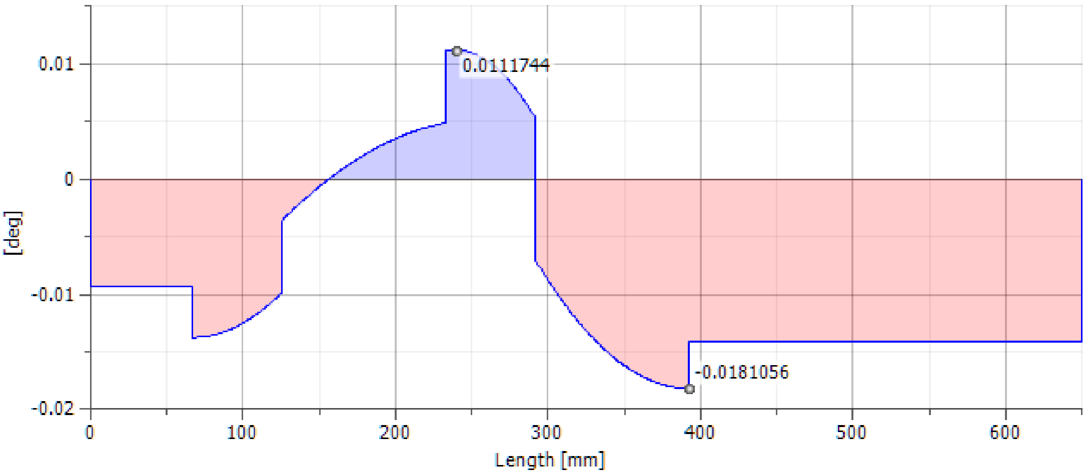
Bending Moment, XZ Plane



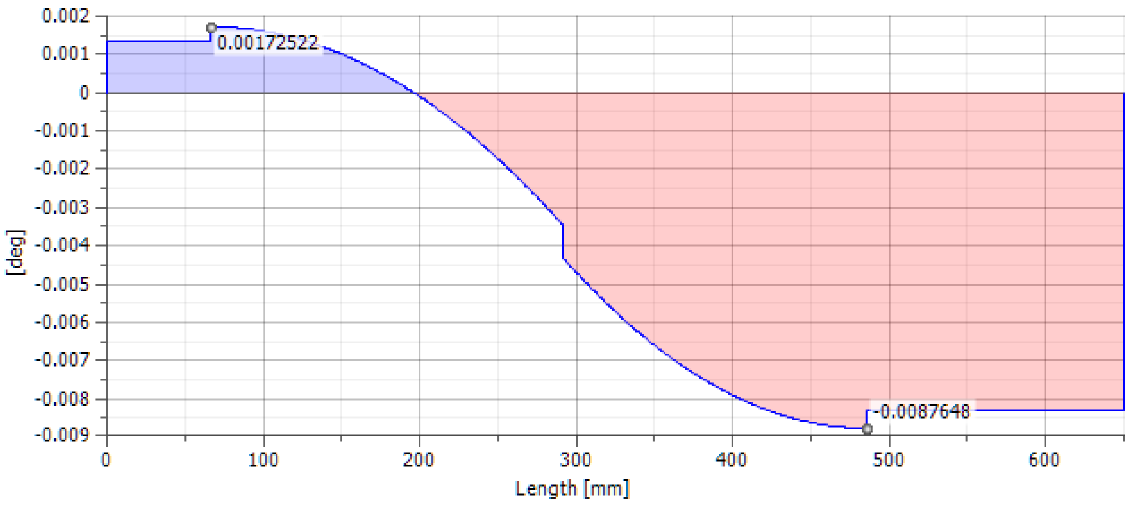
Deflection Angle



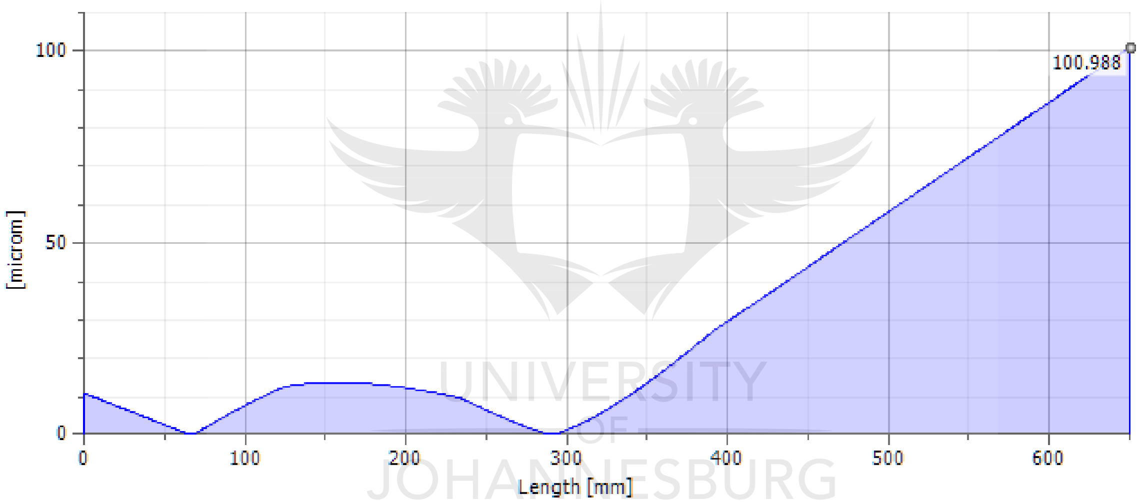
Deflection Angle, YZ Plane



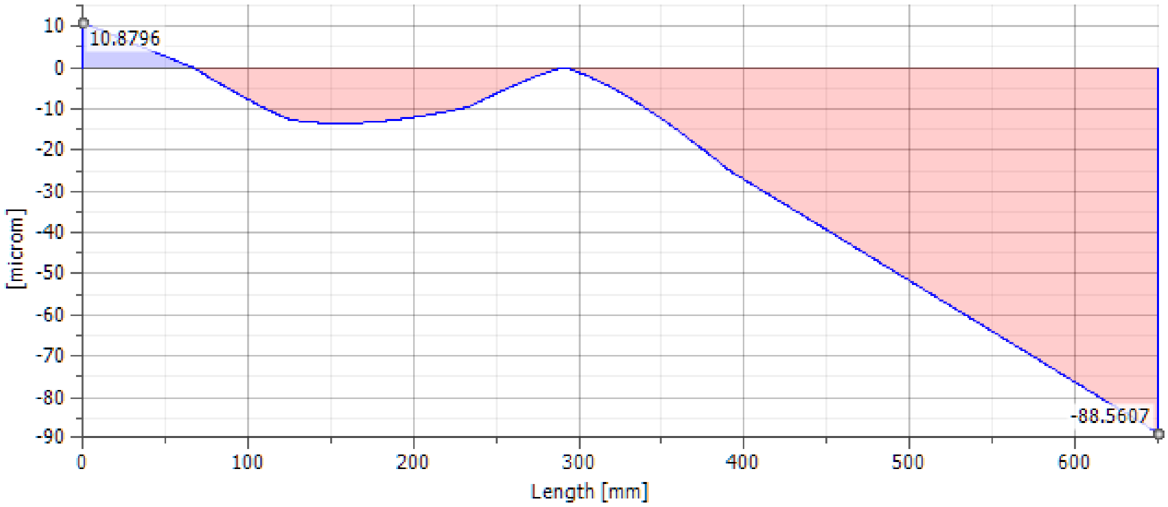
Deflection Angle, XZ Plane



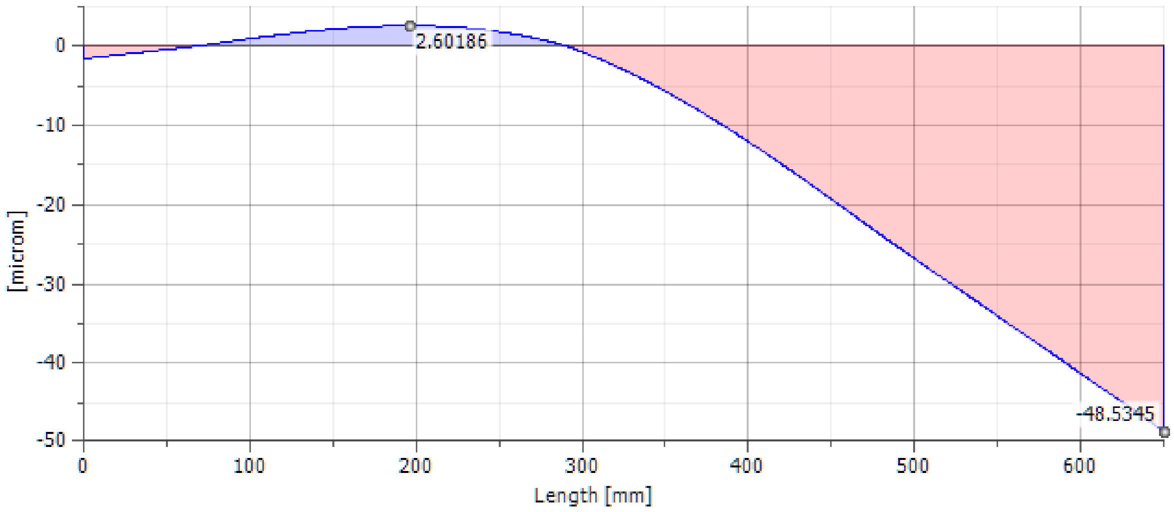
Deflection



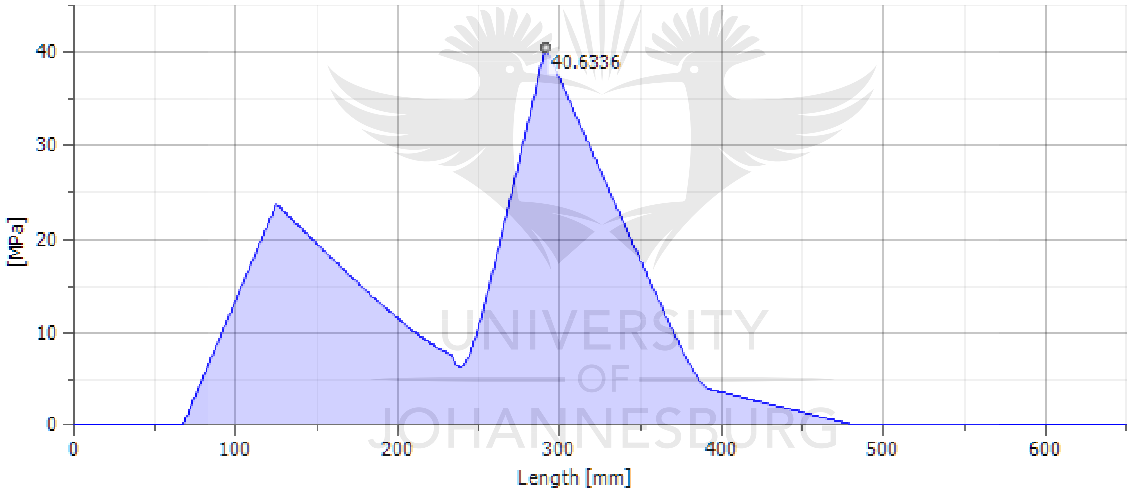
Deflection, YZ Plane



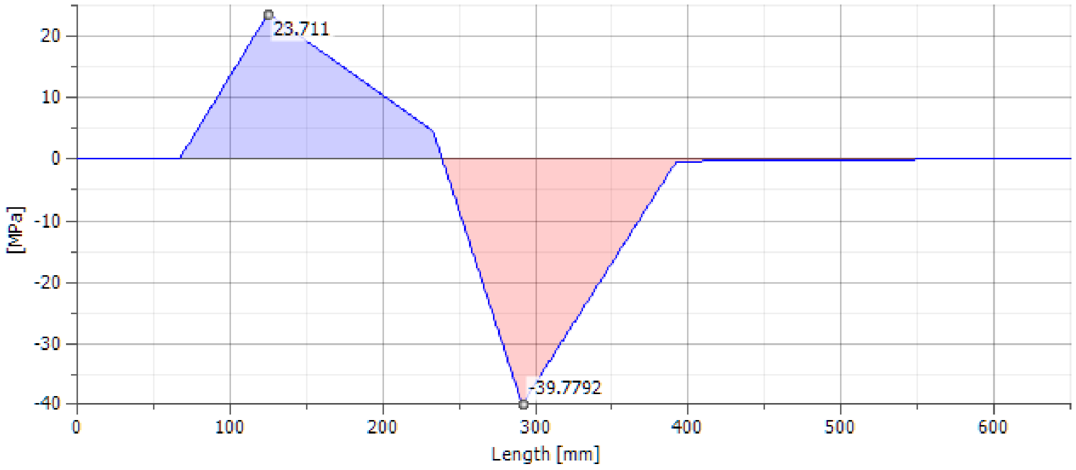
Deflection, XZ Plane



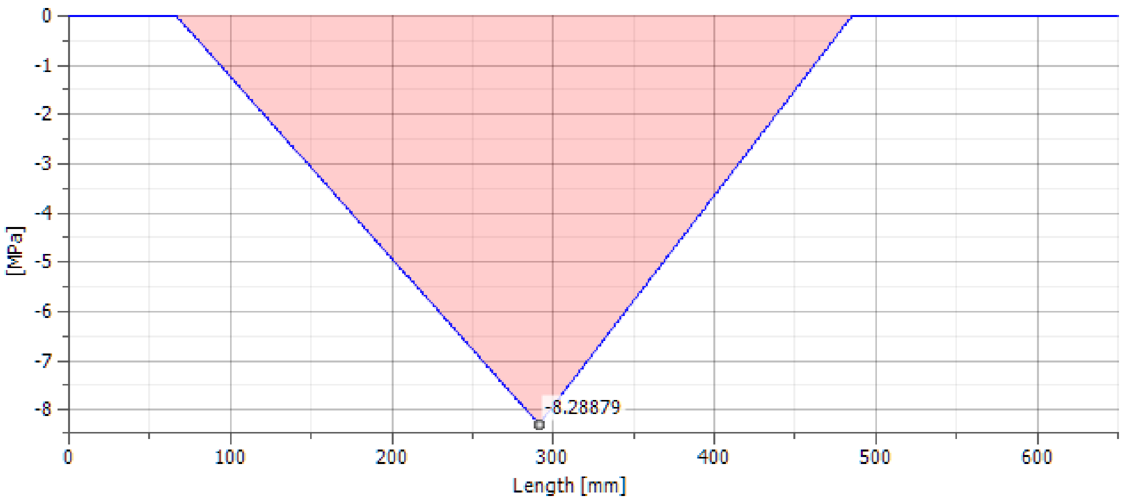
Bending Stress



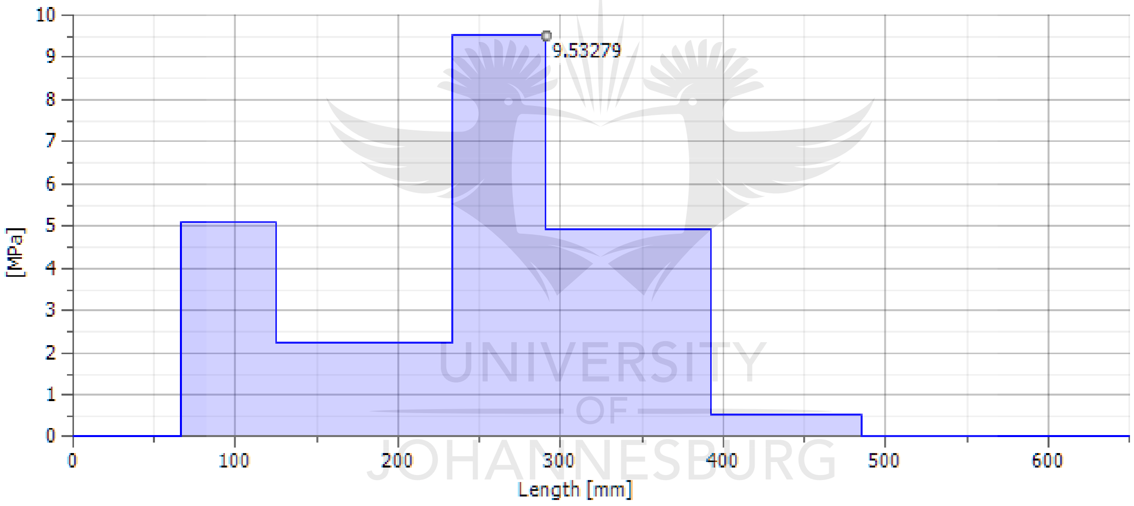
Bending Stress, YZ Plane



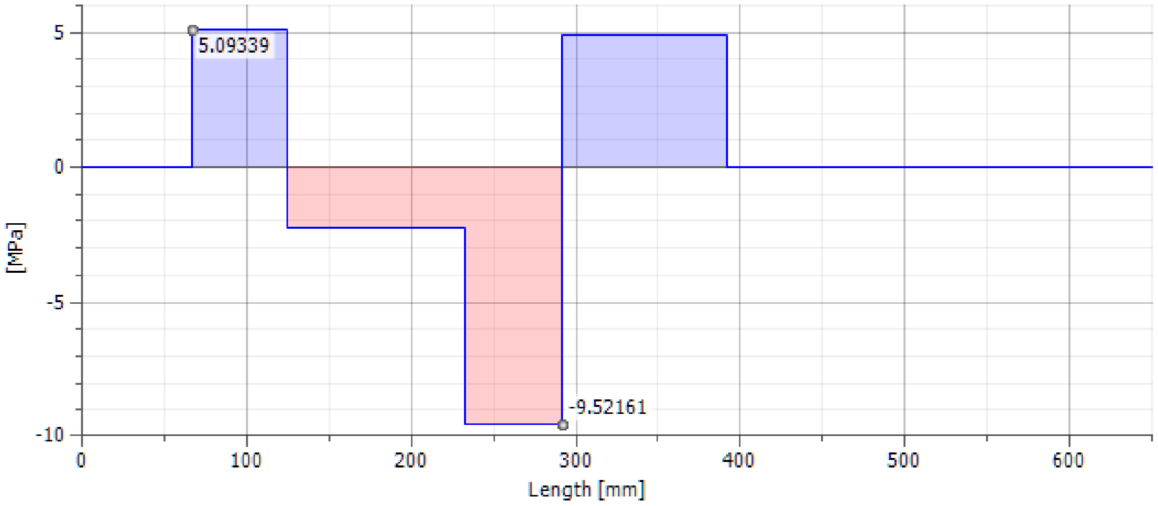
Bending Stress, XZ Plane



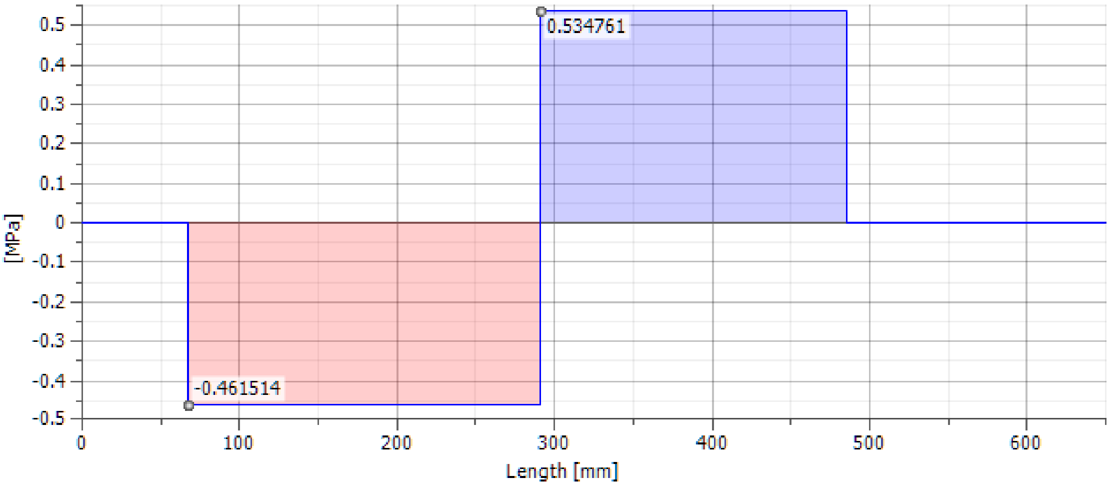
Shear Stress



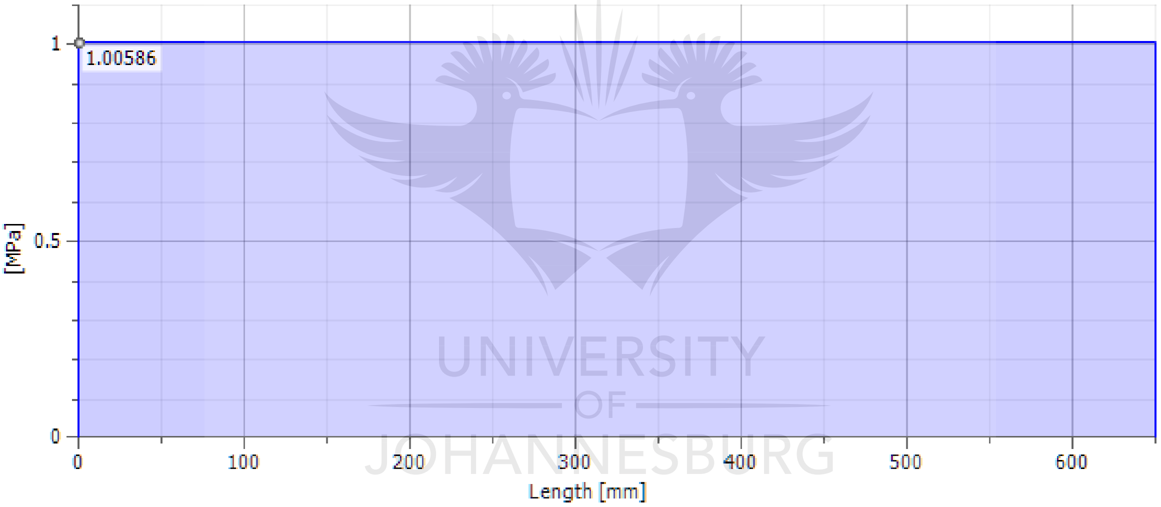
Shear Stress, YZ Plane



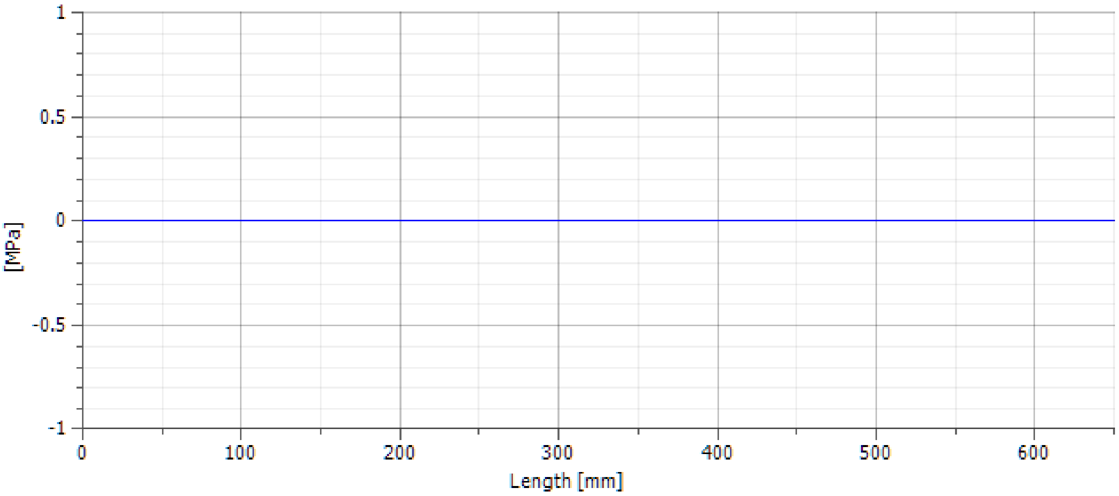
Shear Stress, XZ Plane



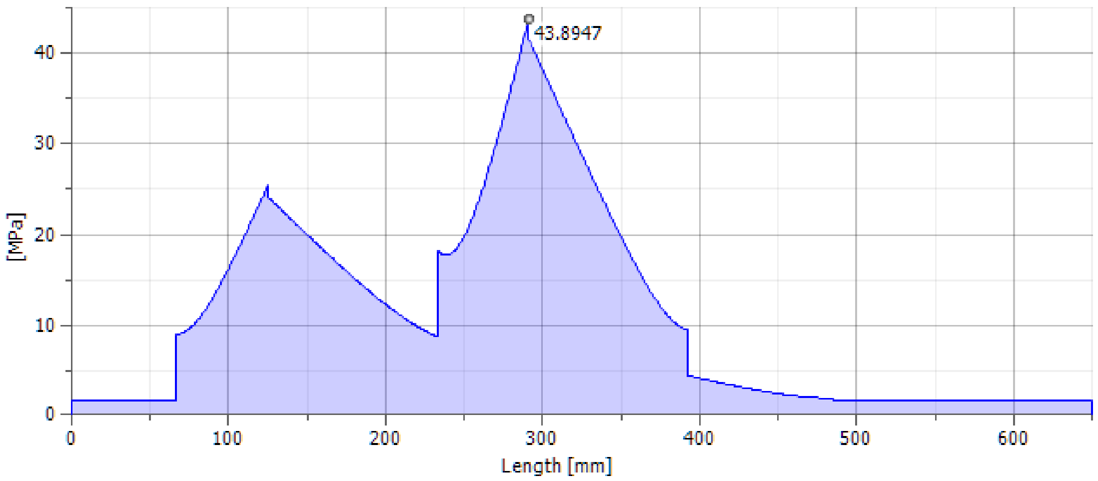
Torsional Stress



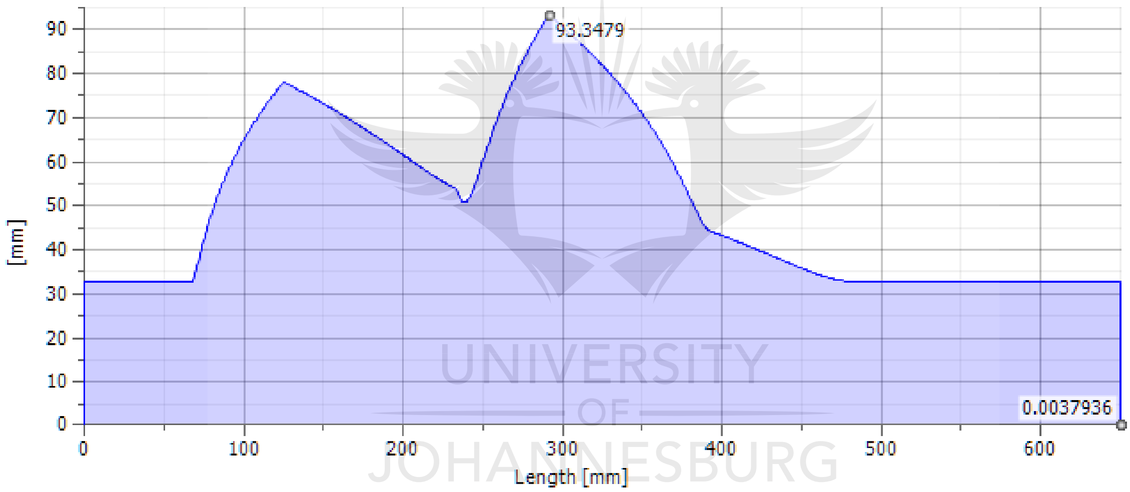
Tension Stress



Reduced Stress



Ideal Diameter



Appendix 29: IS design results.

Material

Material		080M40
Modulus of Elasticity	E	206000 MPa
Modulus of Rigidity	G	80 MPa
Density	ρ	7860 kg/m^3

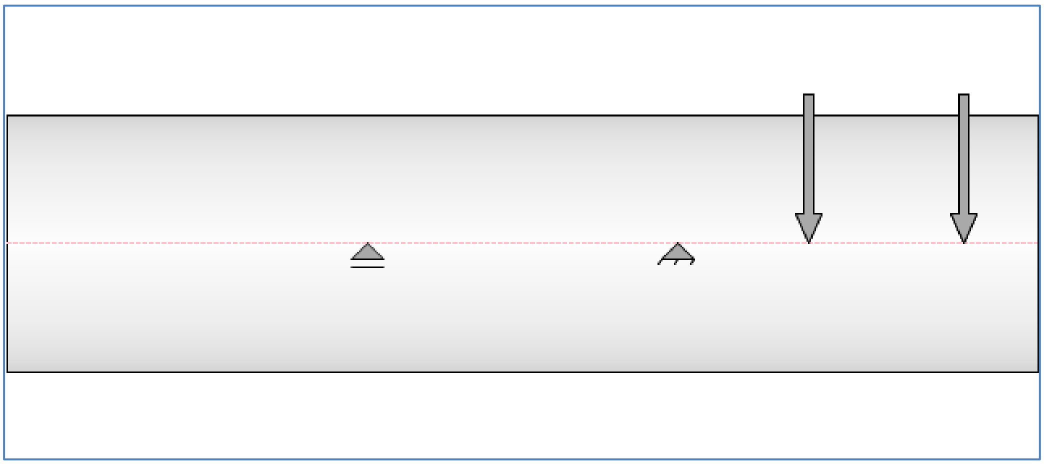
Calculation Properties

Include			
Yes	Density	ρ	7860 kg/m^3
Yes	Shear Displacement Ratio	β	1.188 ul
	Number of Divisions		1000 ul
	Mode of reduced stress		HMH

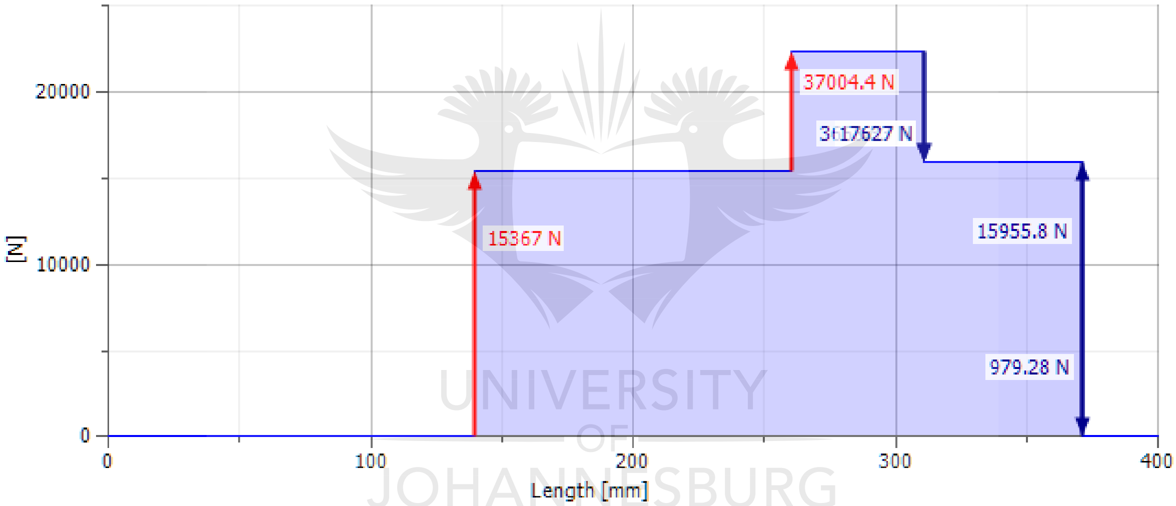
Results

Length	L	400.000 mm
Mass	Mass	24.693 kg
Maximal Bending Stress	σ_B	19.051 MPa
Maximal Shear Stress	τ_S	2.848 MPa
Maximal Torsional Stress	τ	0.000 MPa
Maximal Tension Stress	σ_T	0.000 MPa
Maximal Reduced Stress	σ_{red}	19.679 MPa
Maximal Deflection	f_{max}	7626.715 microm
Angle of Twist	φ	0.00 deg

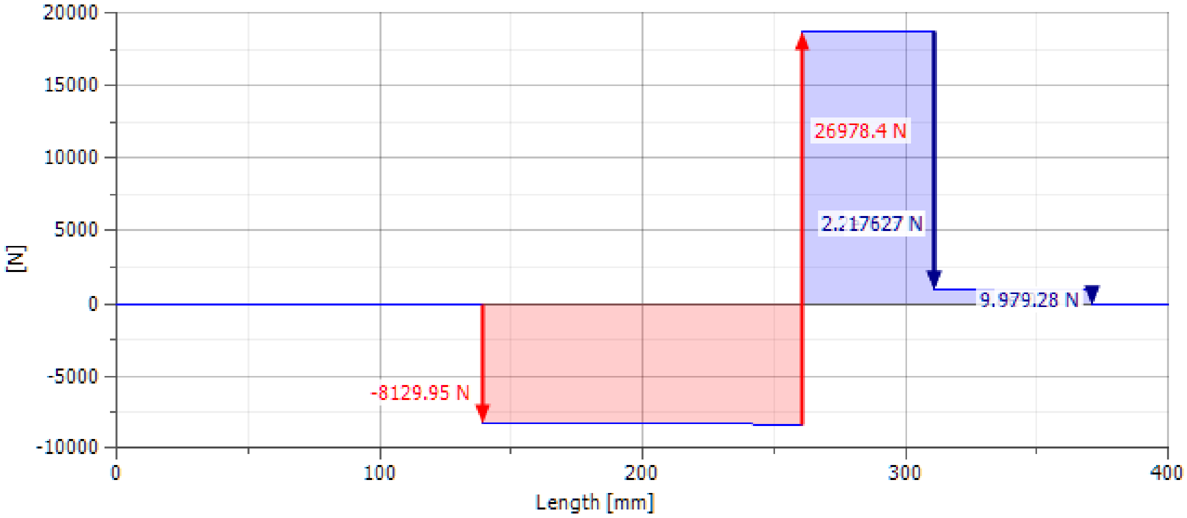
Preview



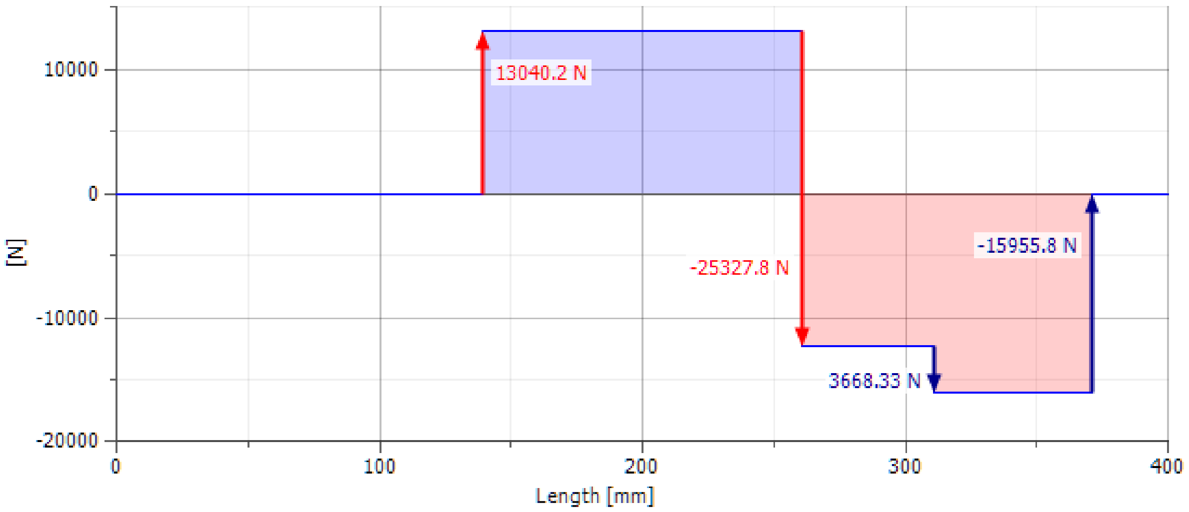
Shear Force



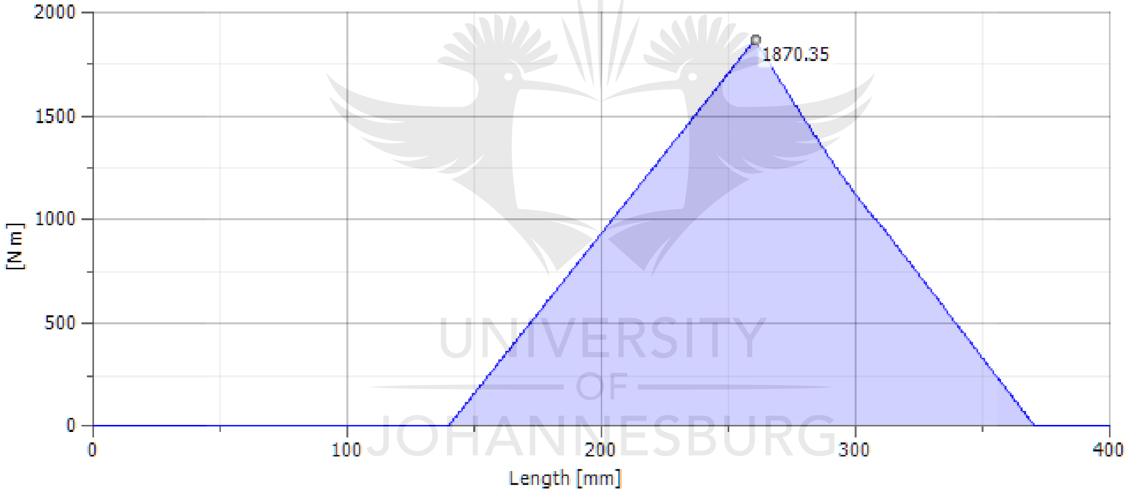
Shear Force, YZ Plane



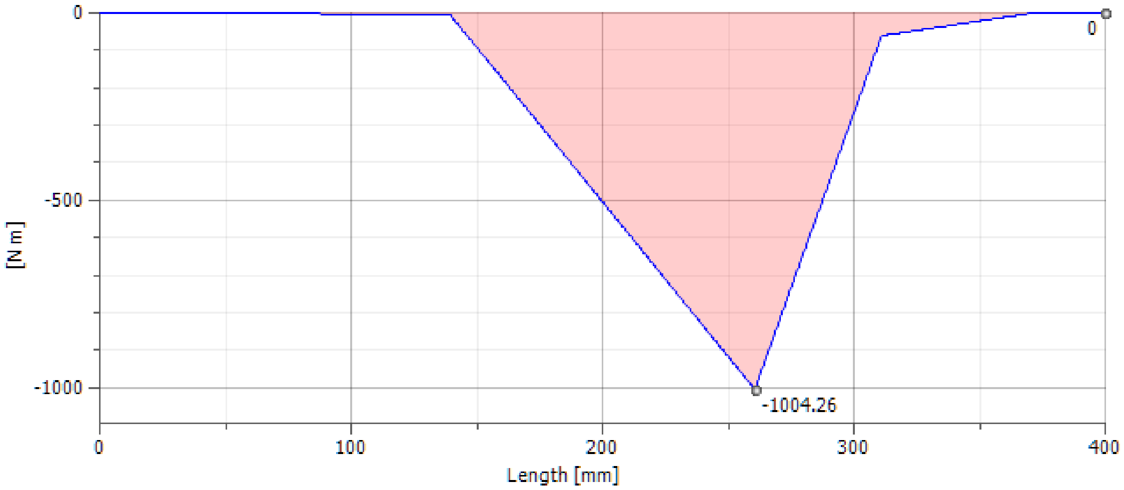
Shear Force, XZ Plane



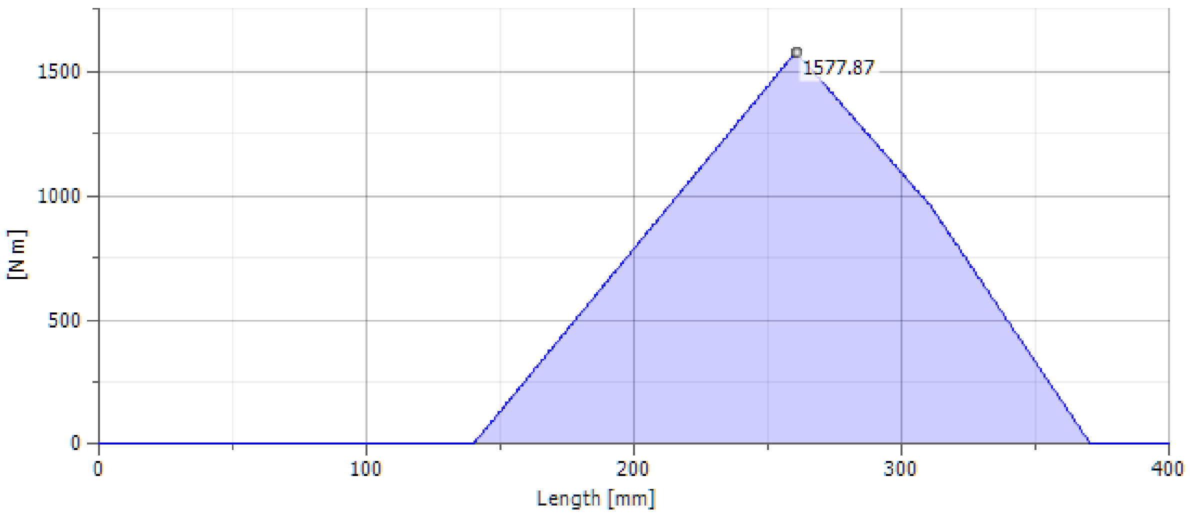
Bending Moment



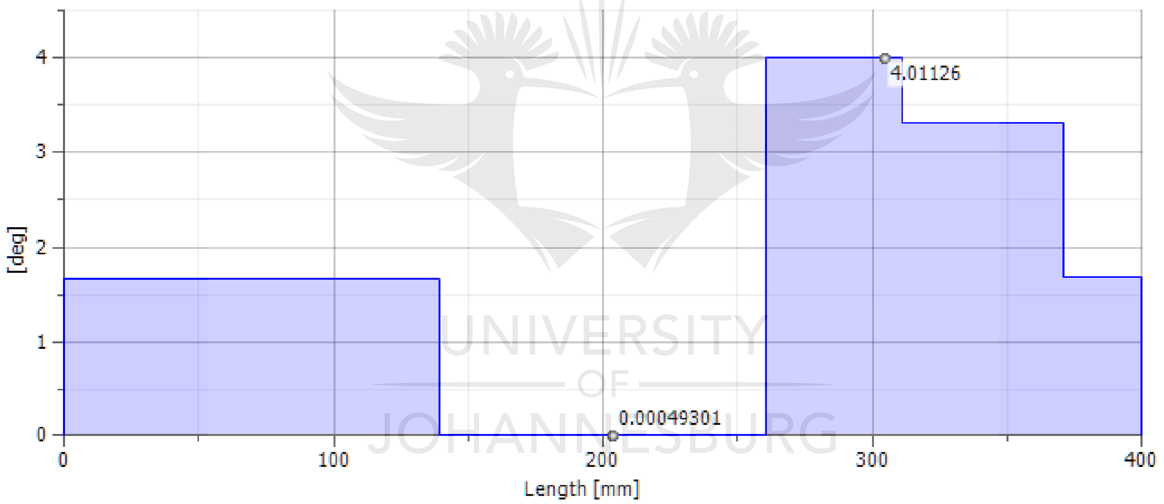
Bending Moment, YZ Plane



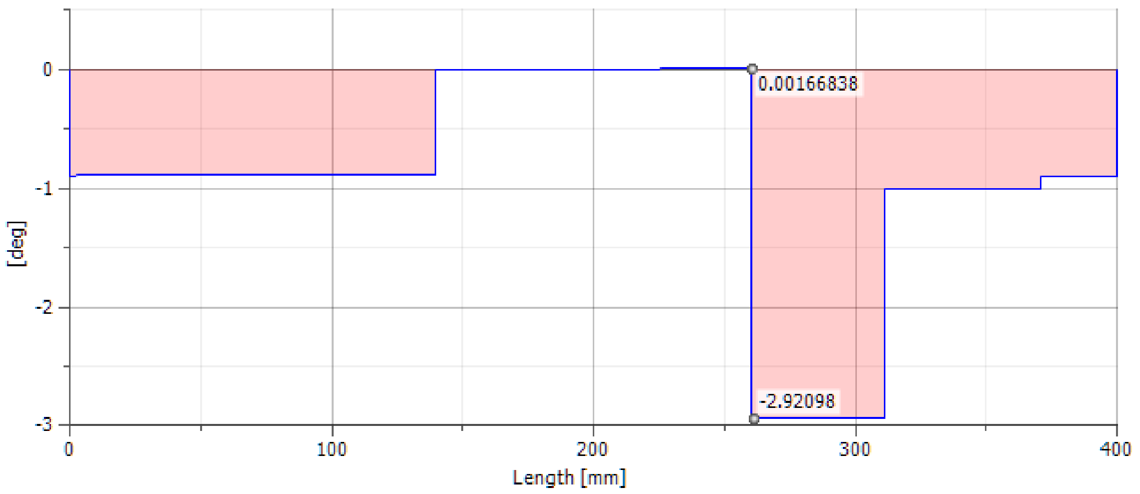
Bending Moment, XZ Plane



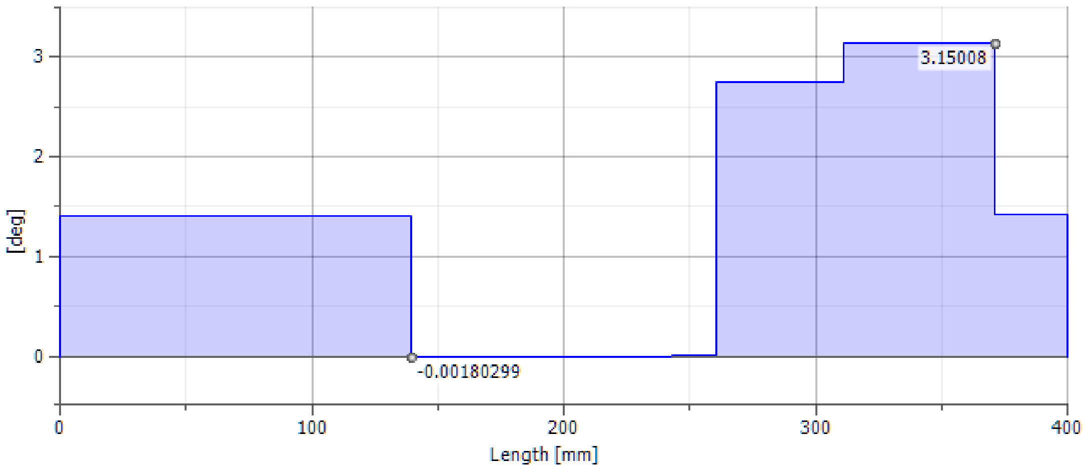
Deflection Angle



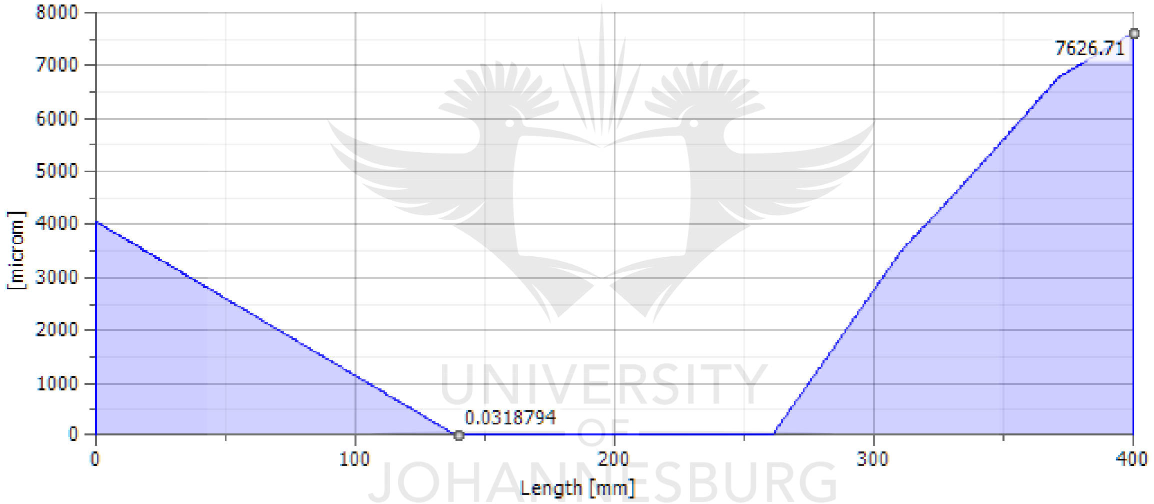
Deflection Angle, YZ Plane



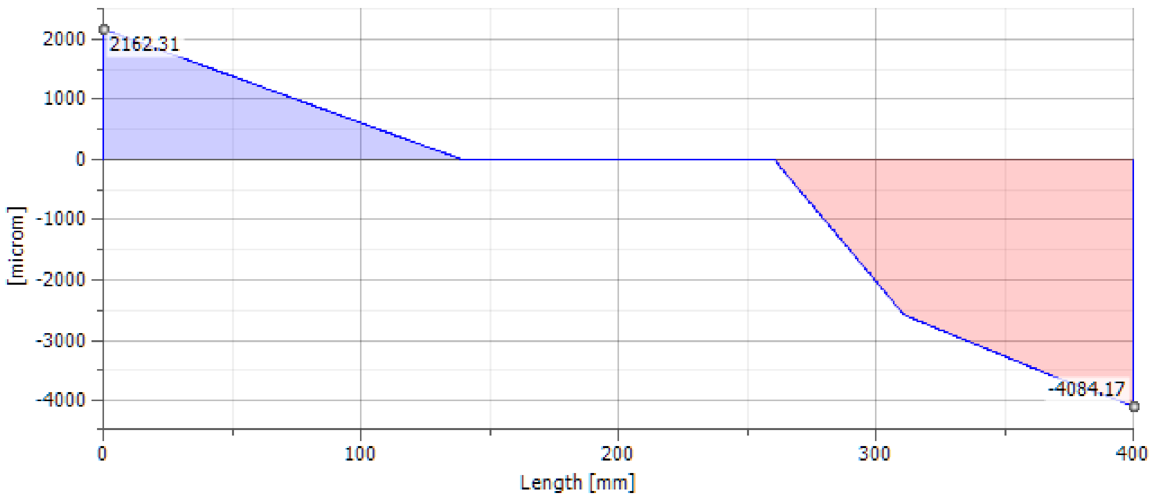
Deflection Angle, XZ Plane



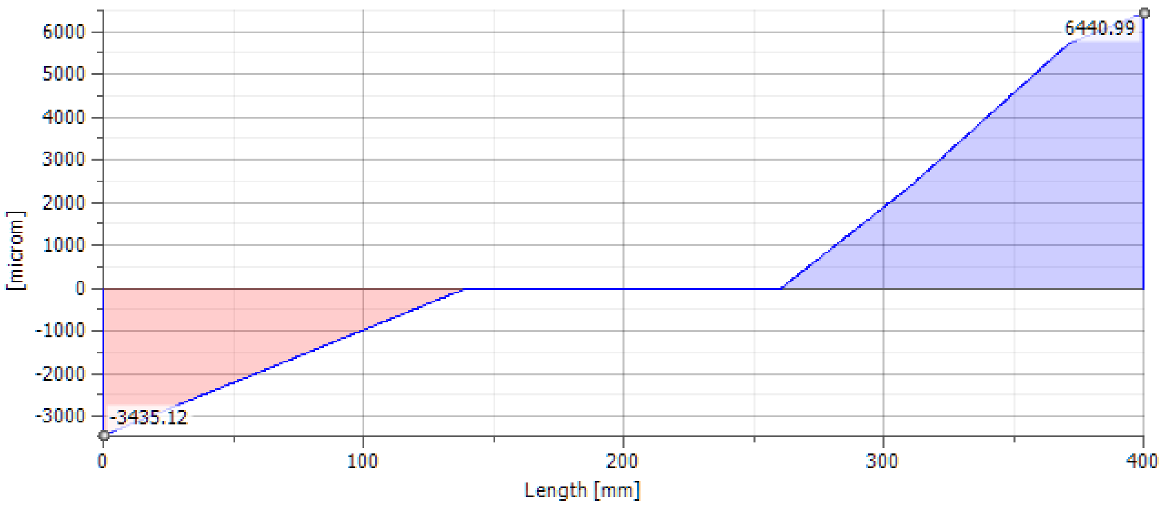
Deflection



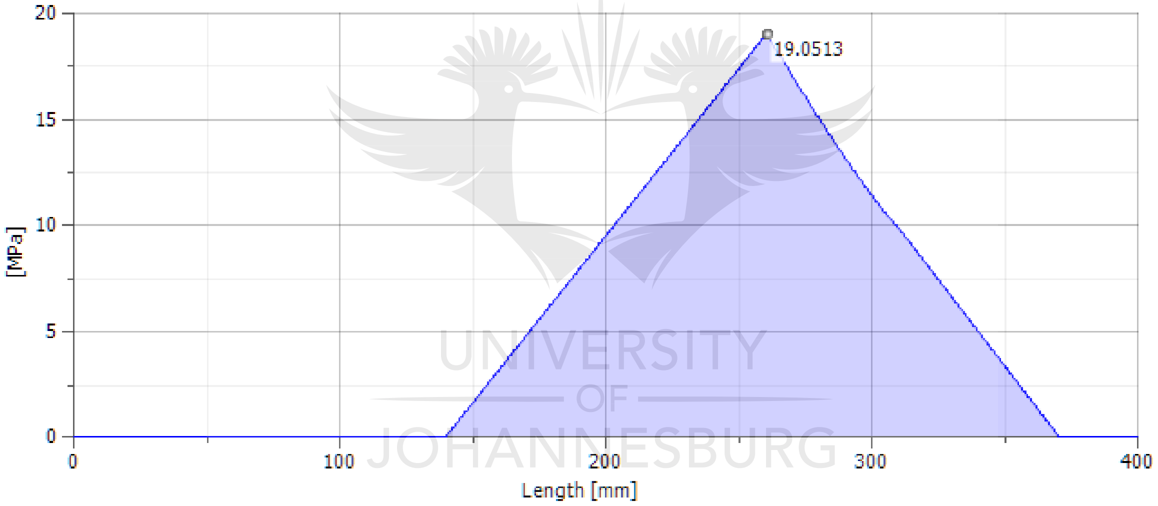
Deflection, YZ Plane



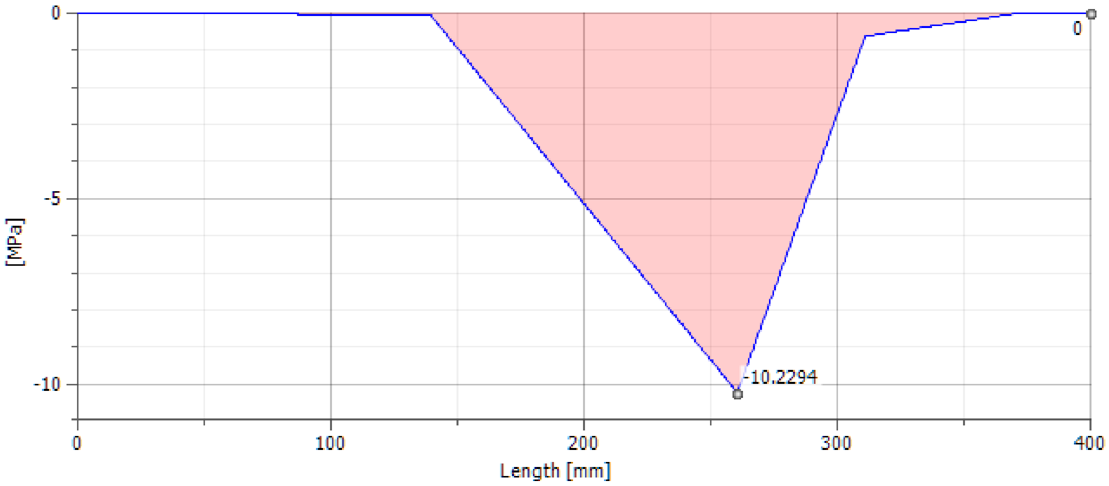
Deflection, XZ Plane



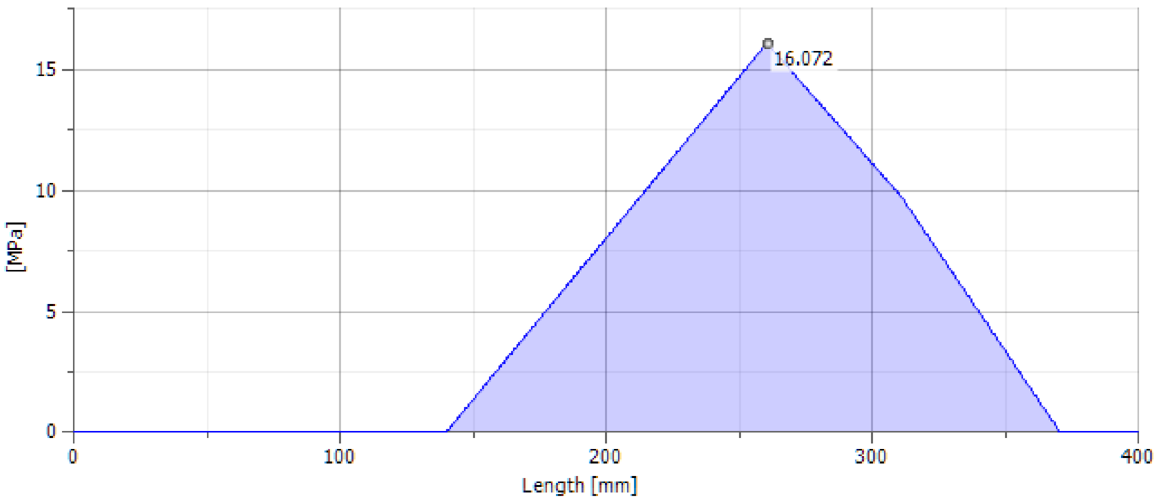
Bending Stress



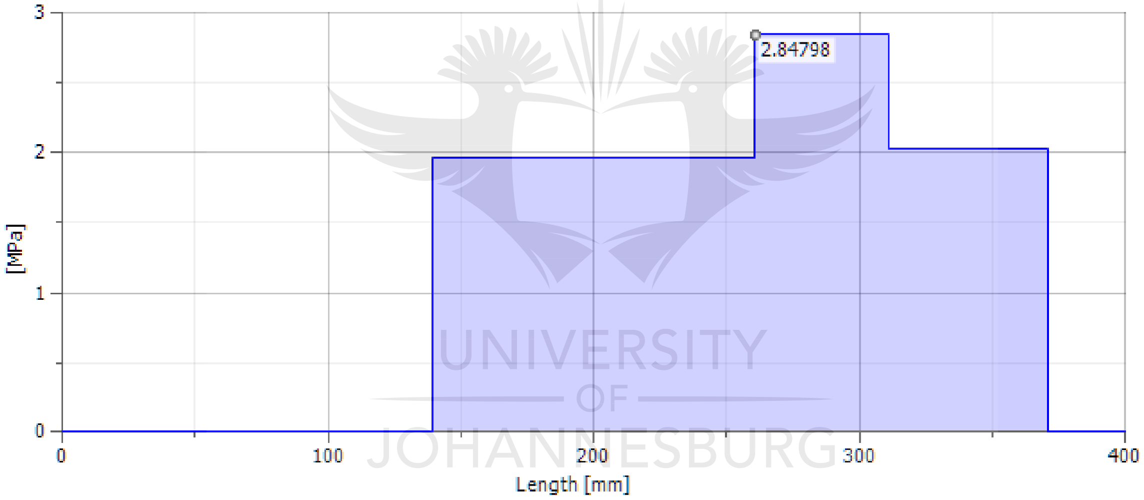
Bending Stress, YZ Plane



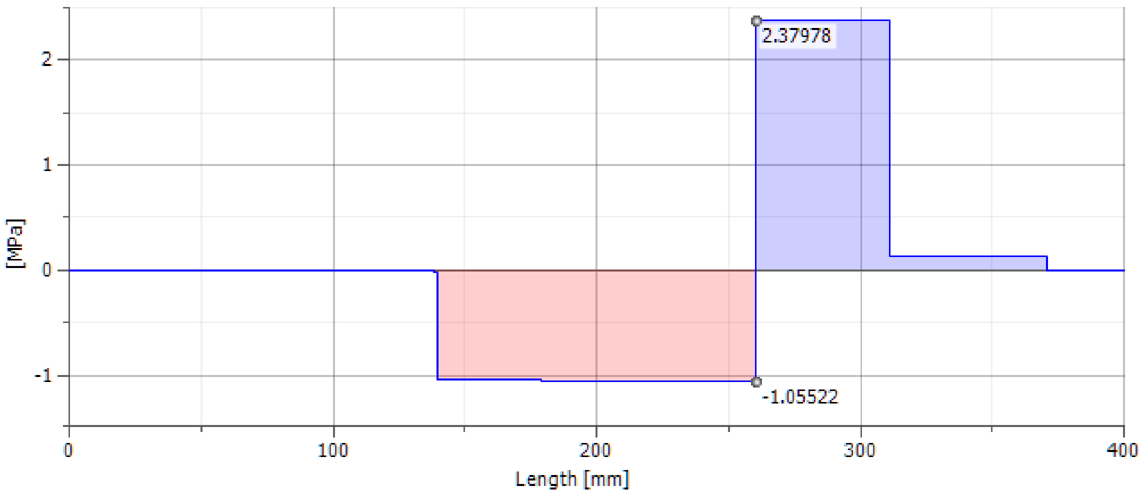
Bending Stress, XZ Plane



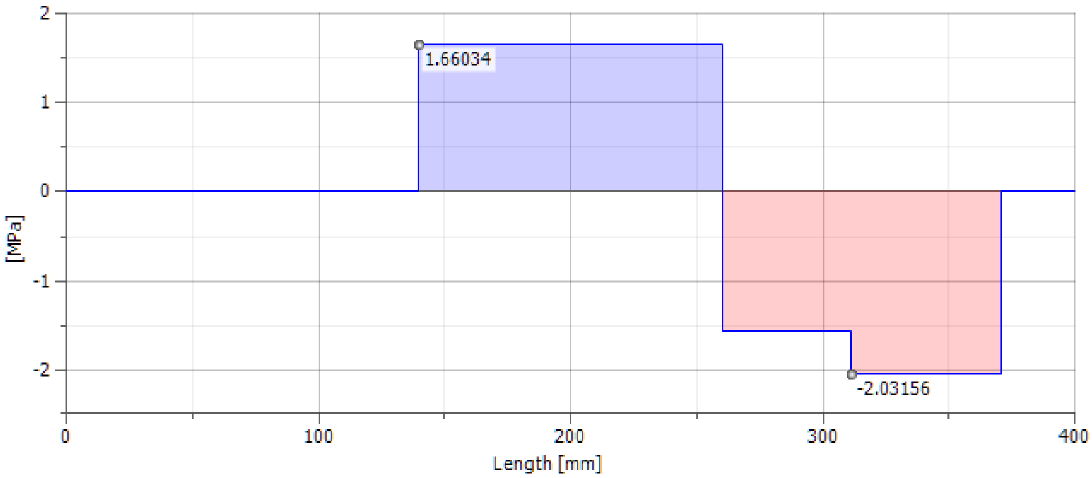
Shear Stress



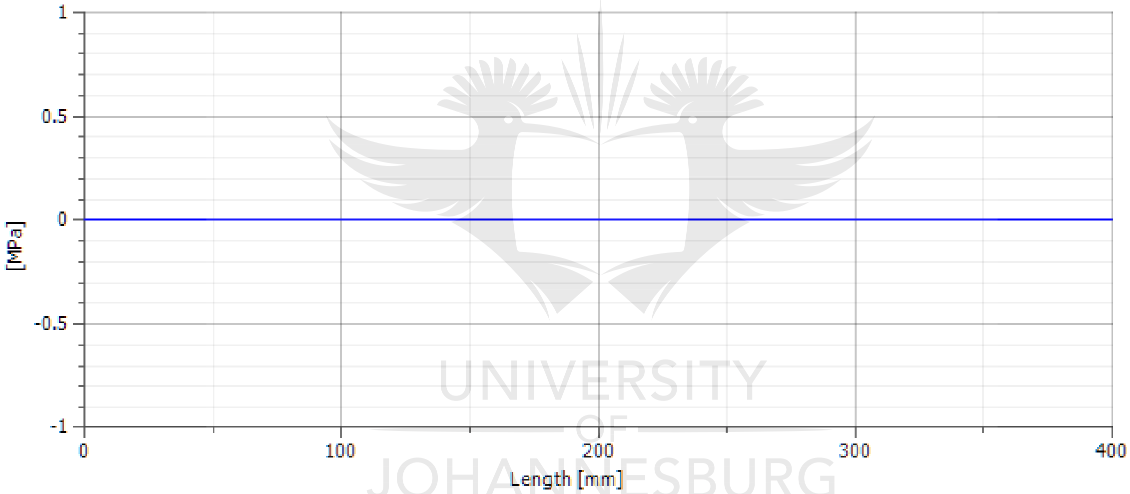
Shear Stress, YZ Plane



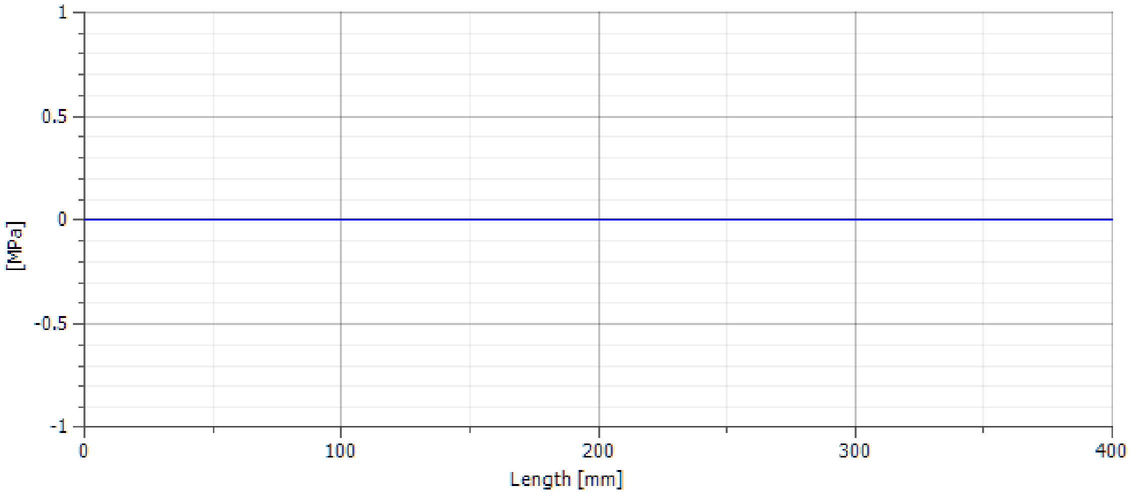
Shear Stress, XZ Plane



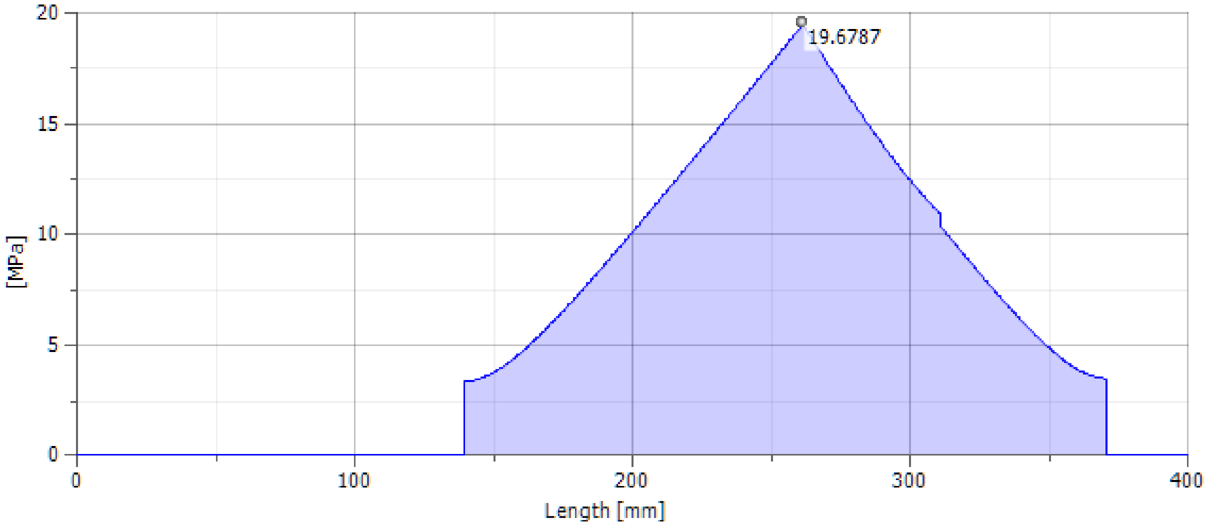
Torsional Stress



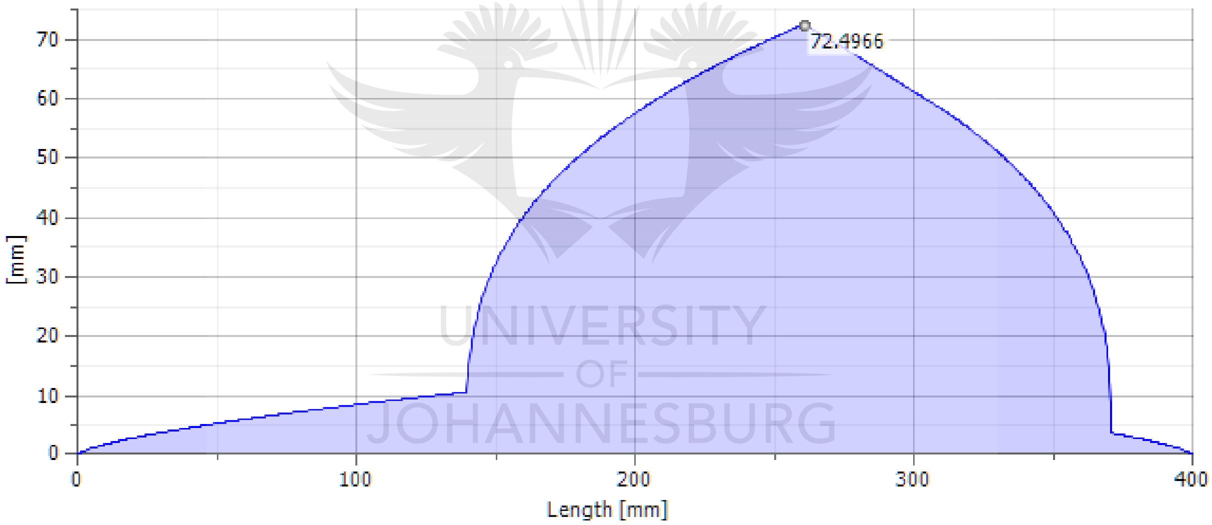
Tension Stress



Reduced Stress



Ideal Diameter



Appendix 30: FS Results.

Material

Material		080M40
Modulus of Elasticity	E	206000 MPa
Modulus of Rigidity	G	80 GPa
Density	ρ	7860 kg/m^3

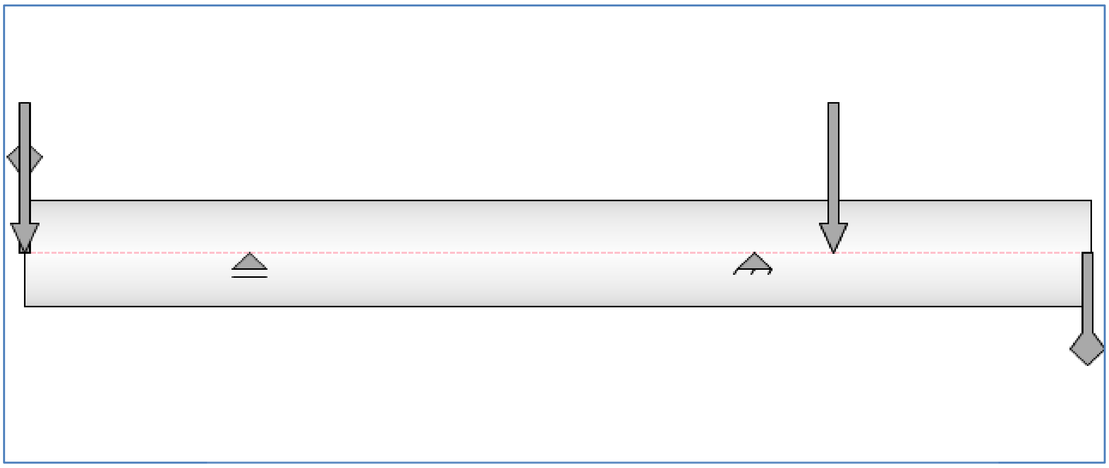
Calculation Properties

Include			
Yes	Density	ρ	7860 kg/m^3
Yes	Shear Displacement Ratio	β	1.188 ul
	Number of Divisions		1000 ul

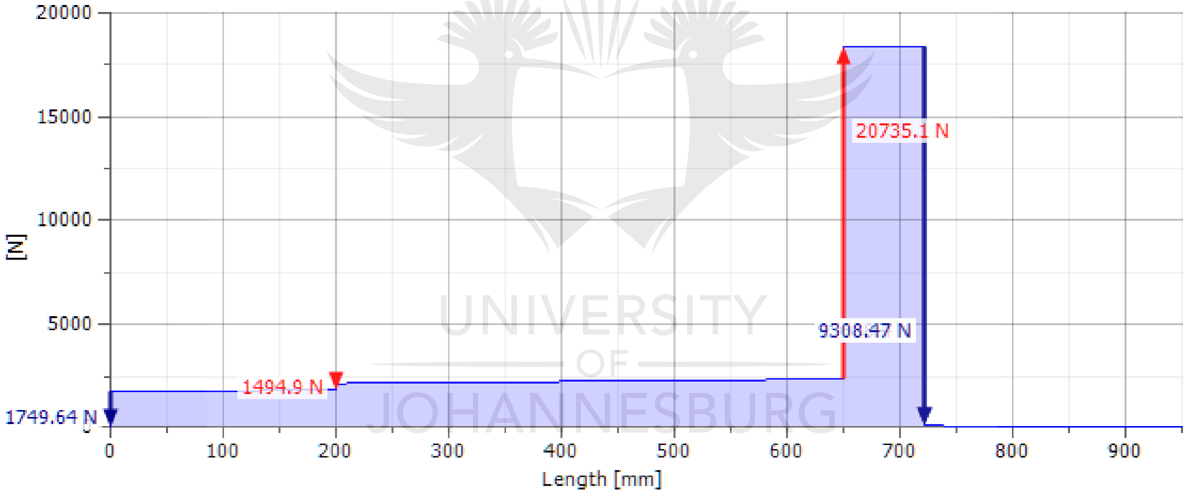
Results

Length	L	950.000 mm
Mass	Mass	58.646 kg
Maximal Bending Stress	σ_B	13.420 MPa
Maximal Shear Stress	τ_S	2.340 MPa
Maximal Torsional Stress	τ	17.350 MPa
Maximal Tension Stress	σ_T	0.000 MPa
Maximal Reduced Stress	σ_{red}	33.159 MPa
Maximal Deflection	f_{max}	3842.011 microm
Angle of Twist	φ	235.34 deg

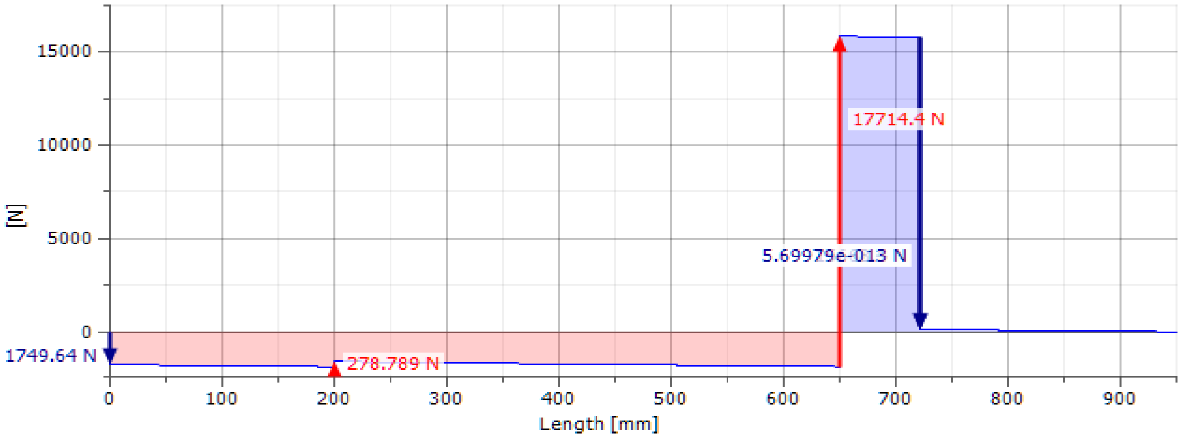
Preview



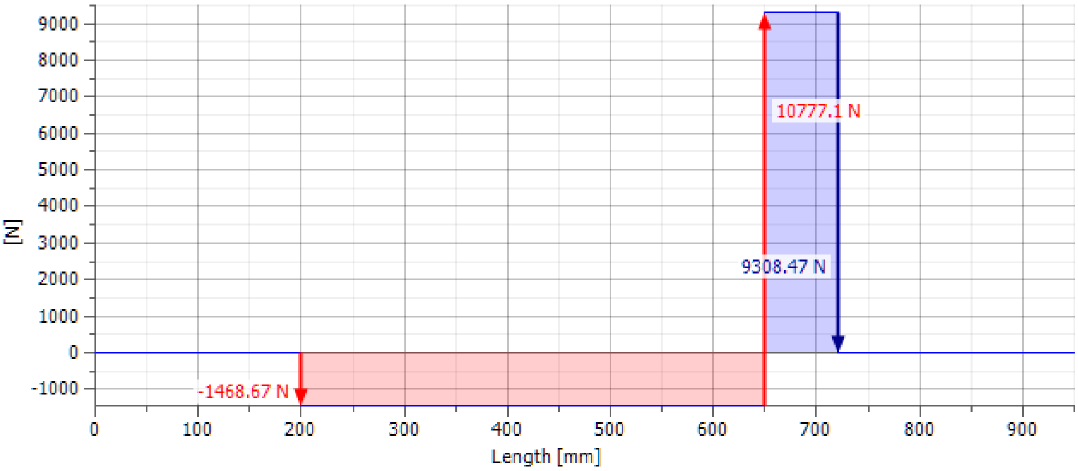
Shear Force



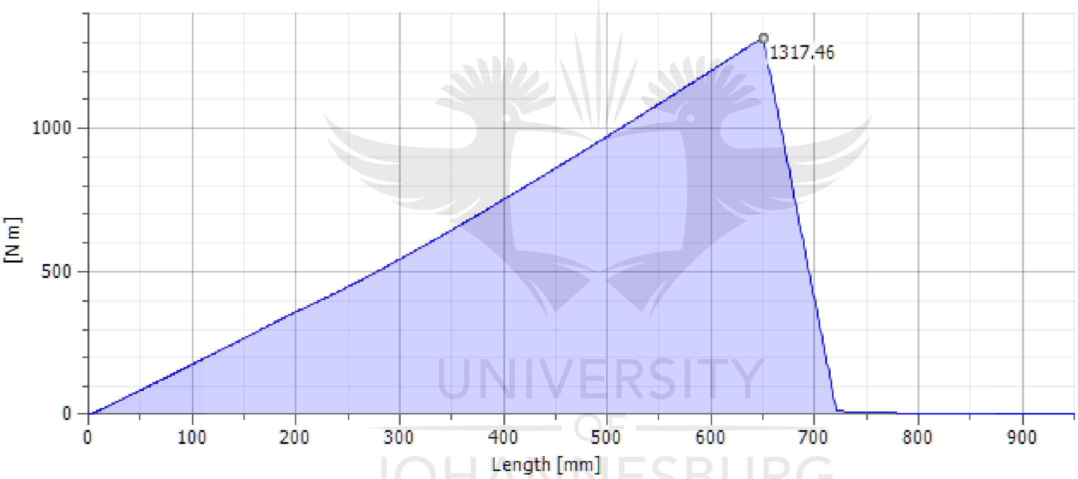
Shear Force, YZ Plane



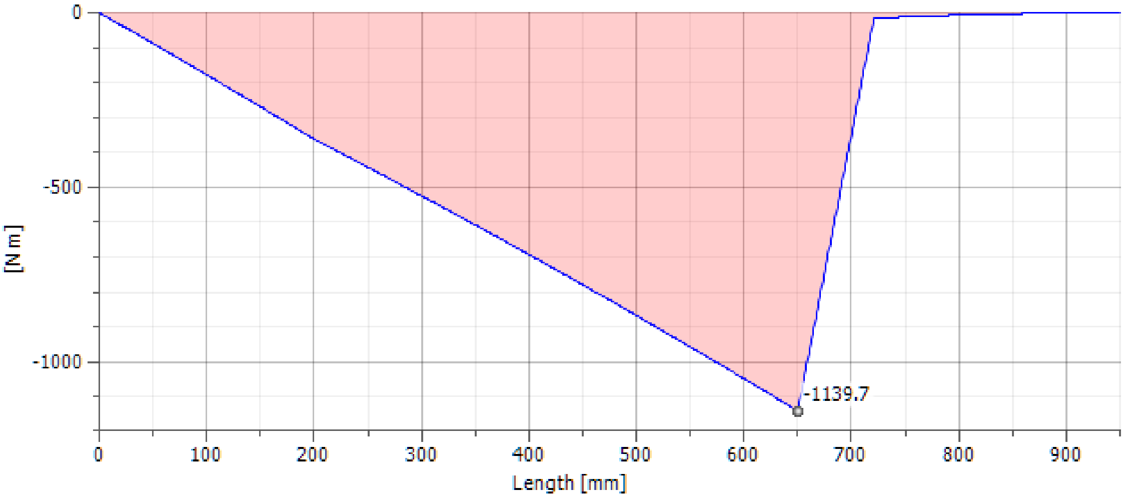
Shear Force, XZ Plane



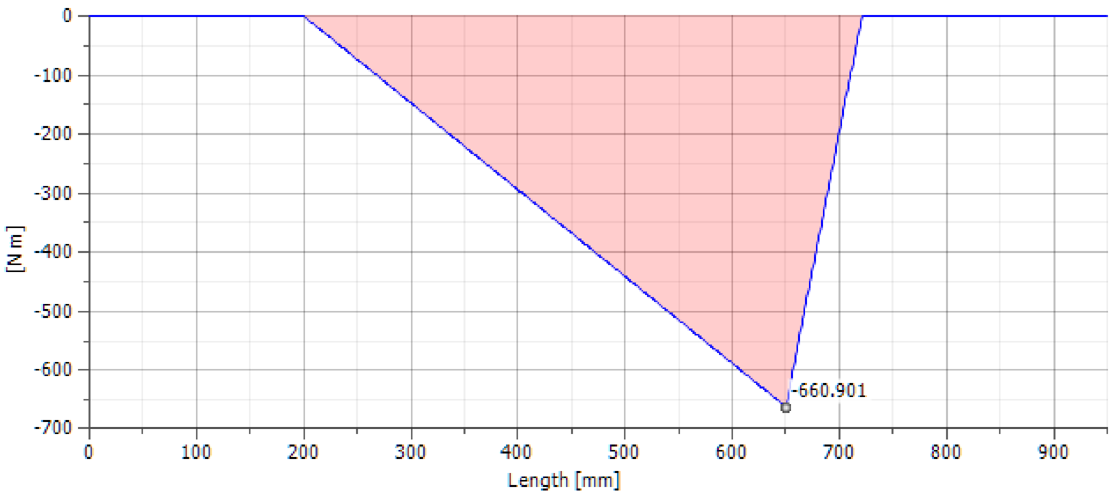
Bending Moment



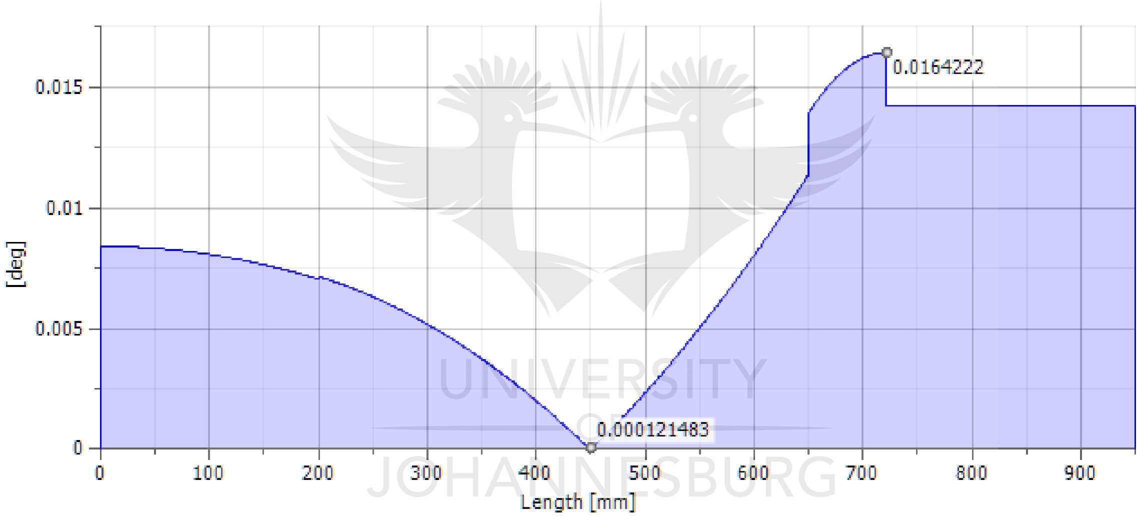
Bending Moment, YZ Plane



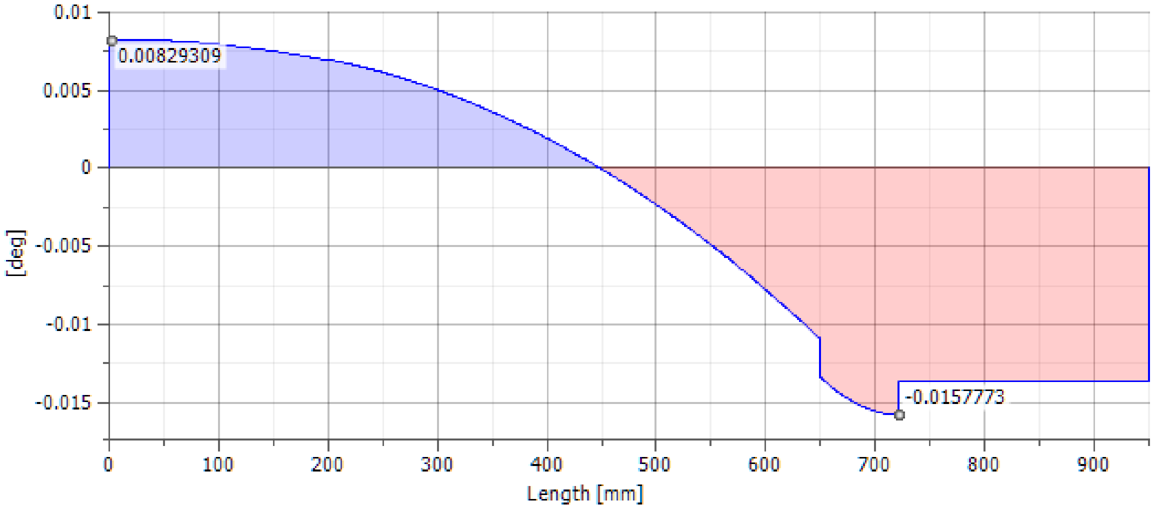
Bending Moment, XZ Plane



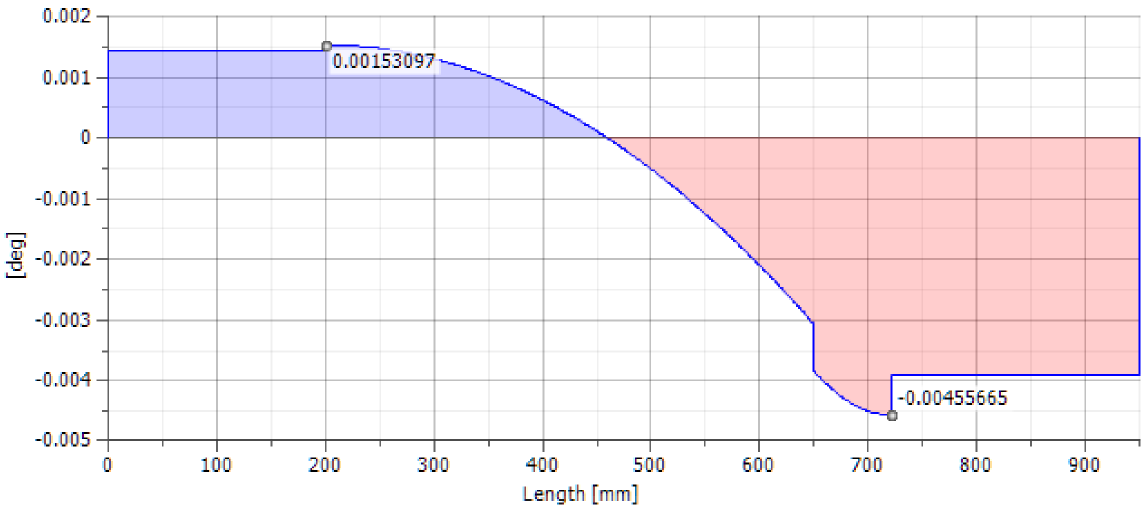
Deflection Angle



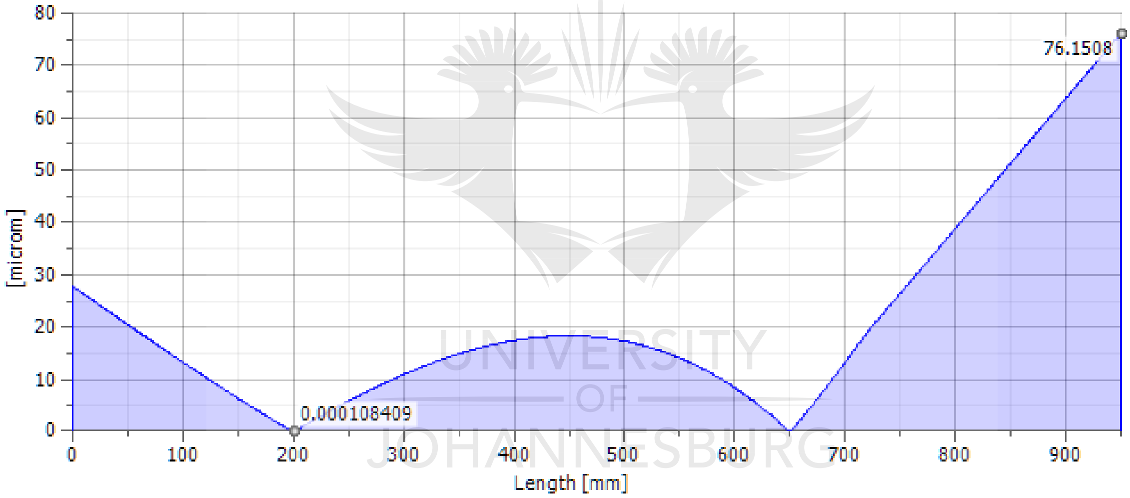
Deflection Angle, YZ Plane



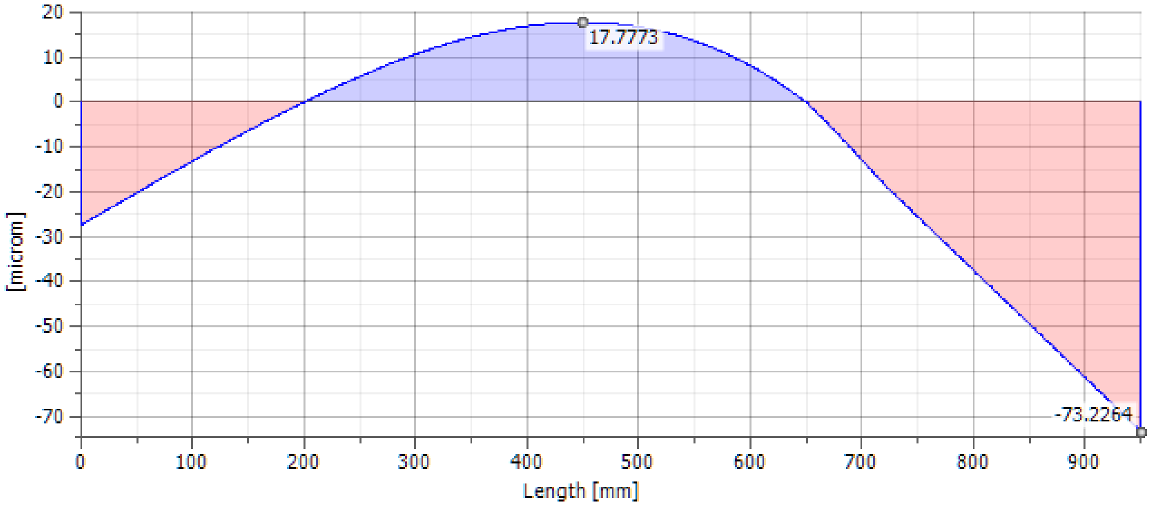
Deflection Angle, XZ Plane



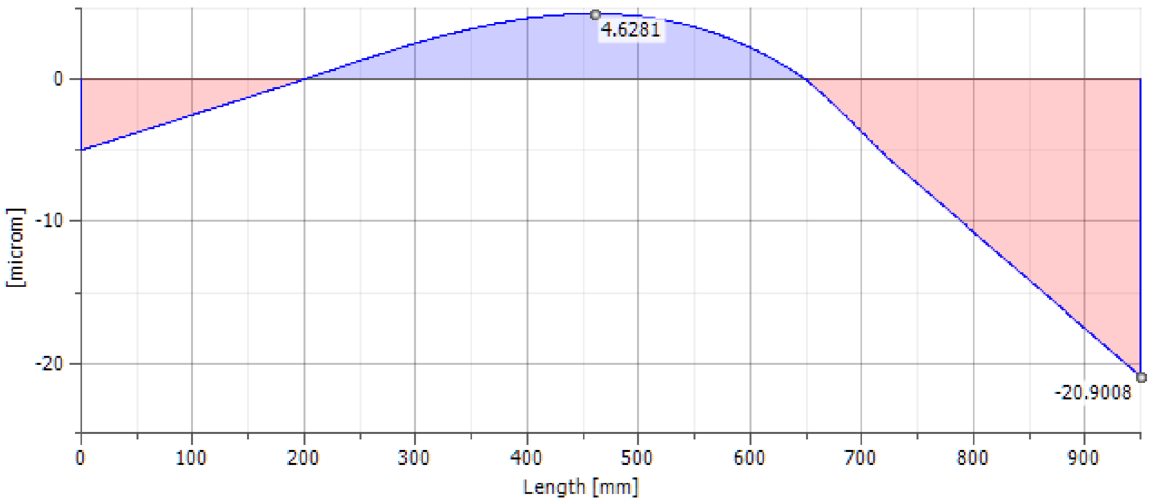
Deflection



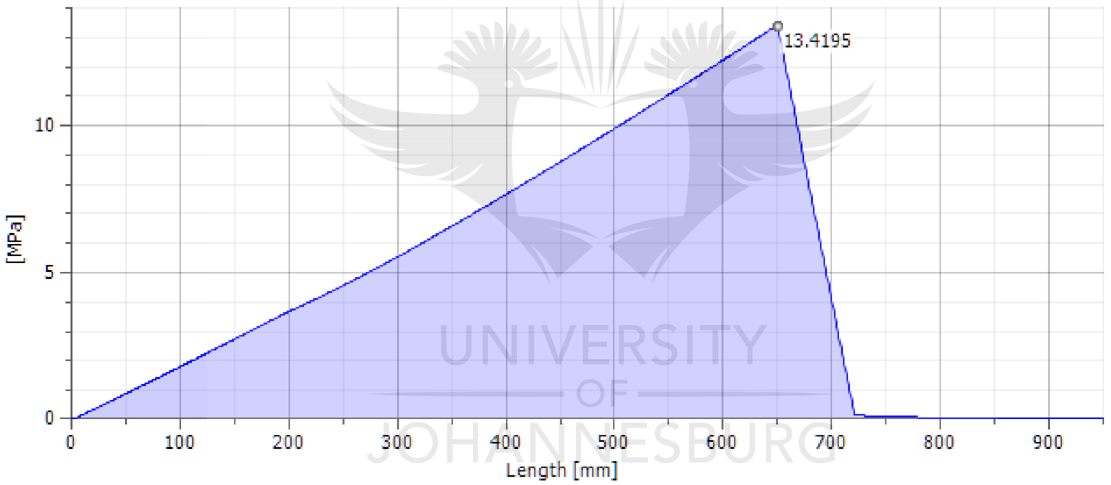
Deflection, YZ Plane



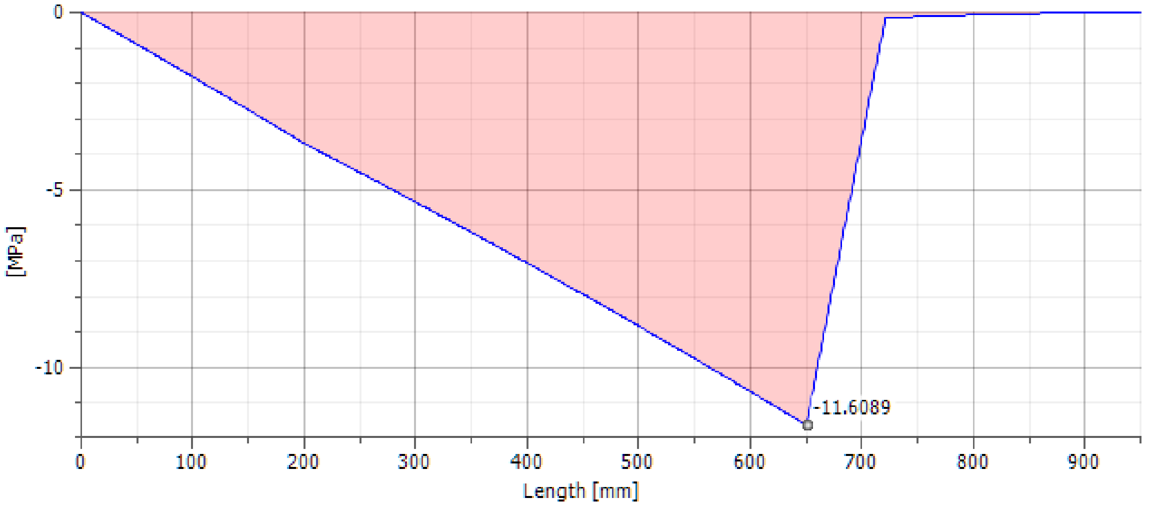
Deflection, XZ Plane



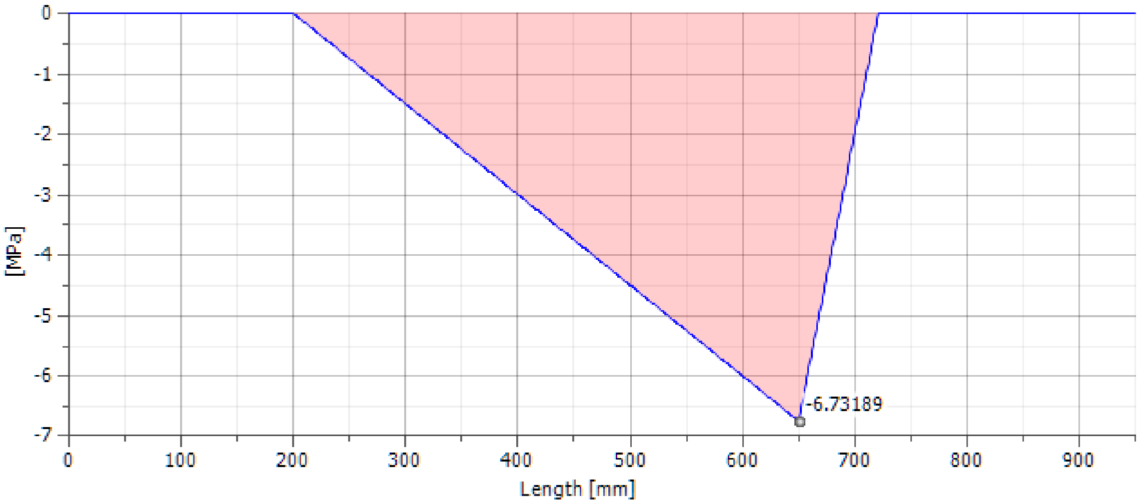
Bending Stress



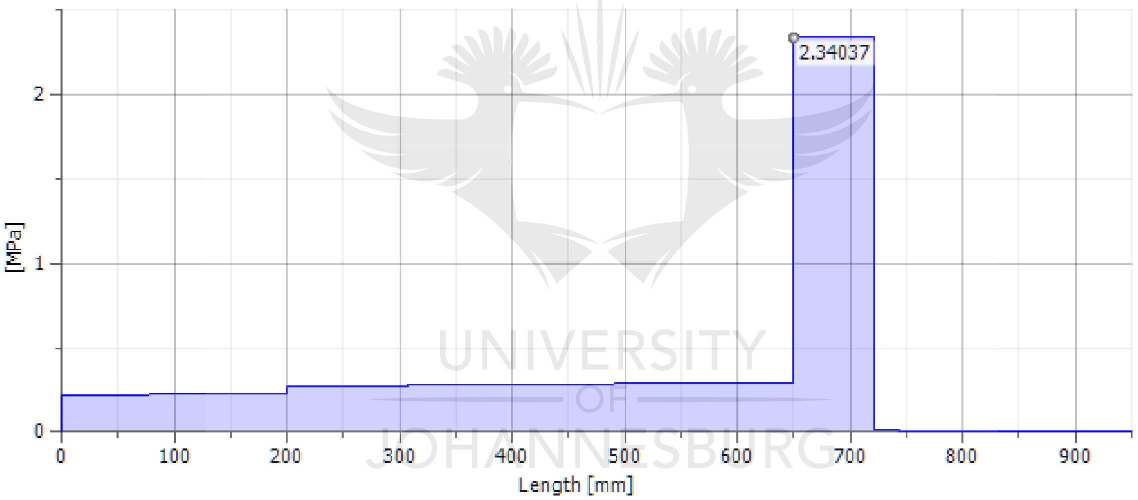
Bending Stress, YZ Plane



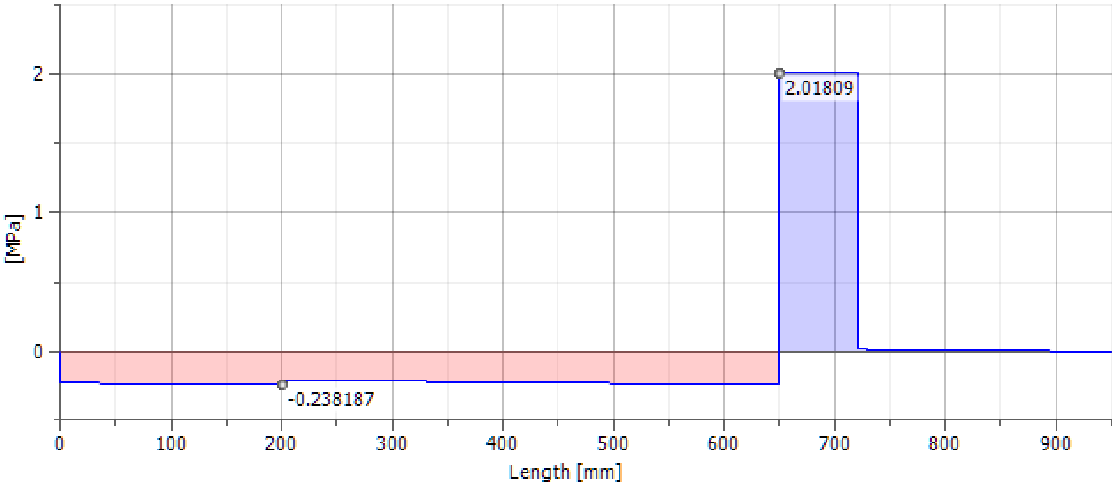
Bending Stress, XZ Plane



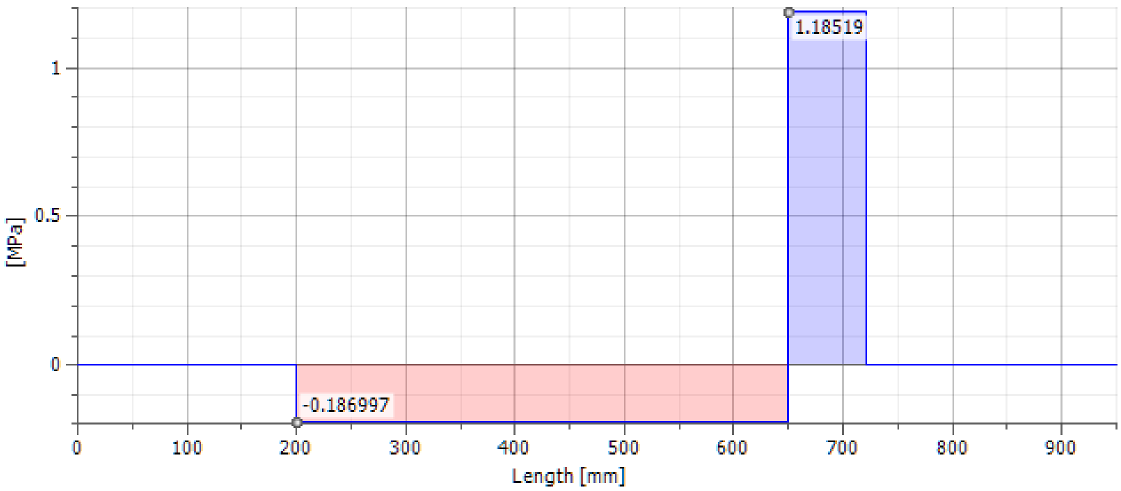
Shear Stress



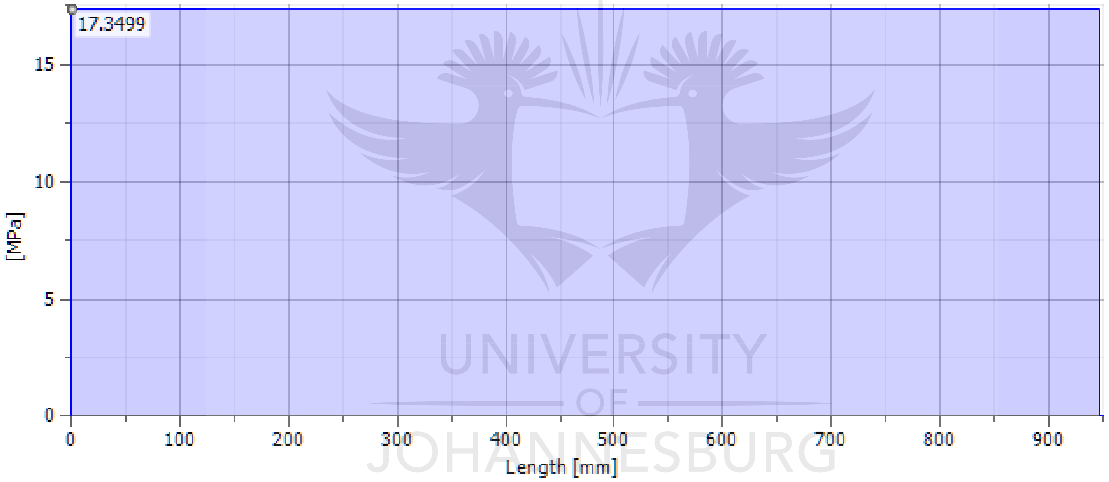
Shear Stress, YZ Plane



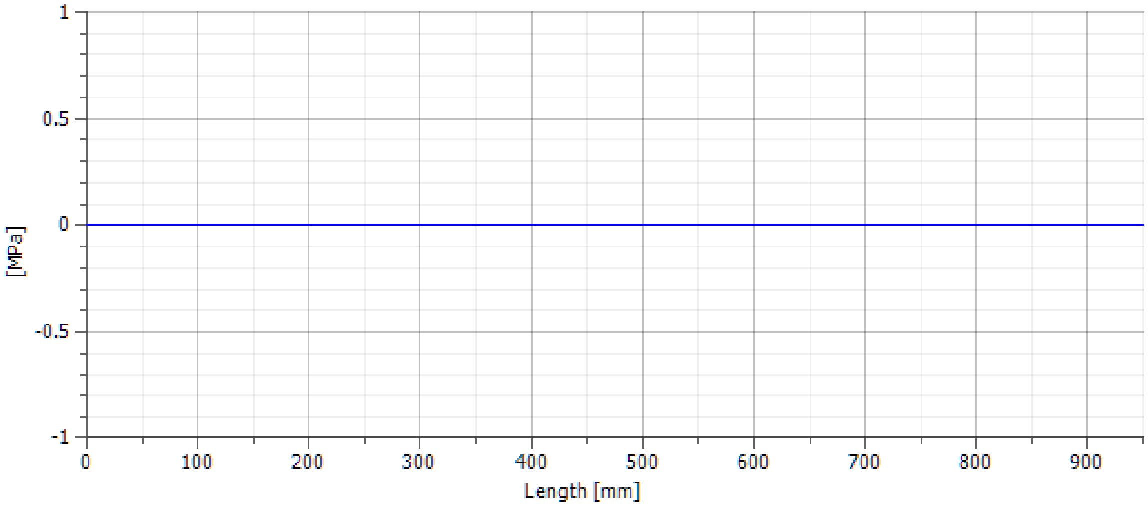
Shear Stress, XZ Plane



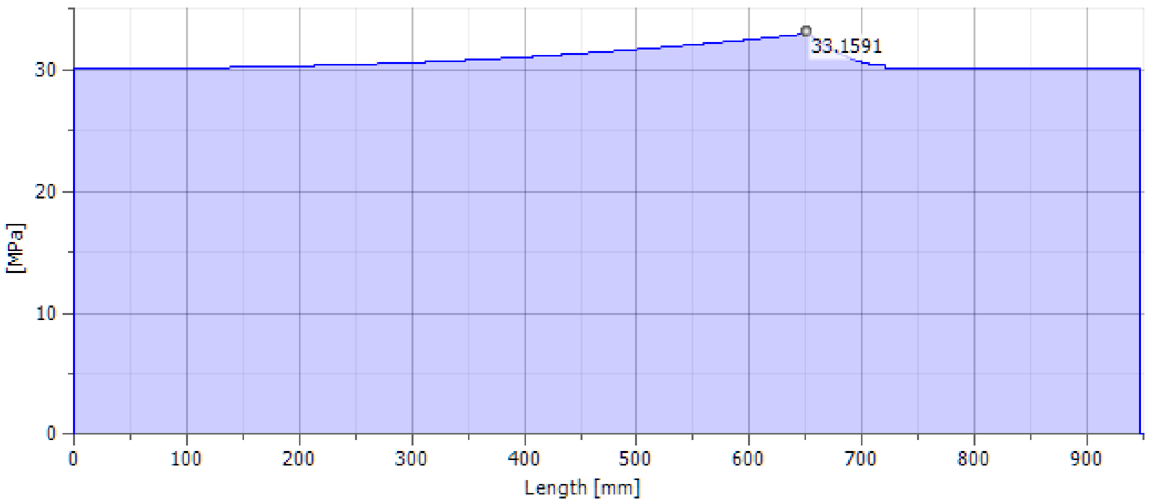
Torsional Stress



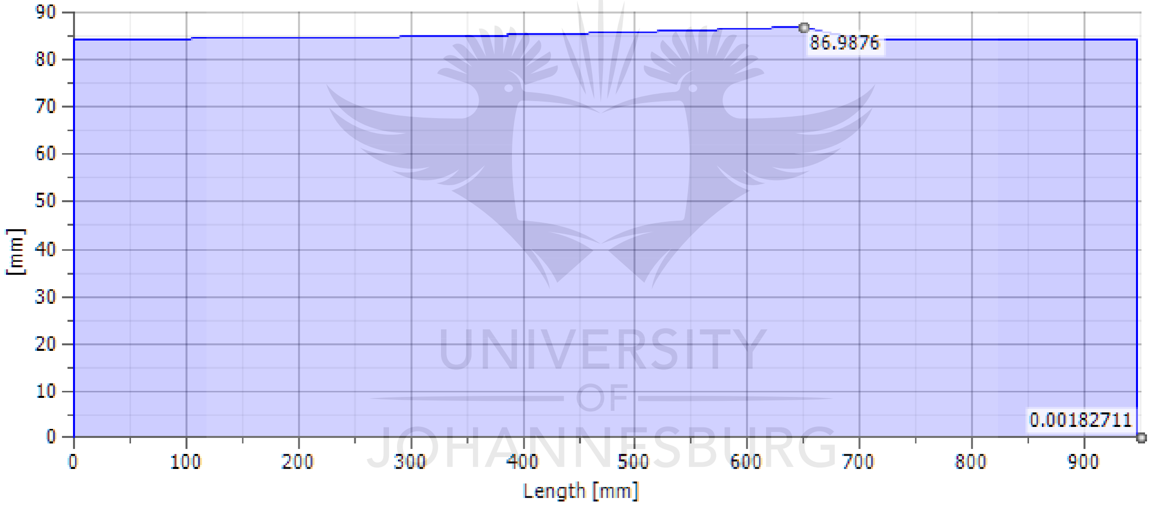
Tension Stress



Reduced Stress

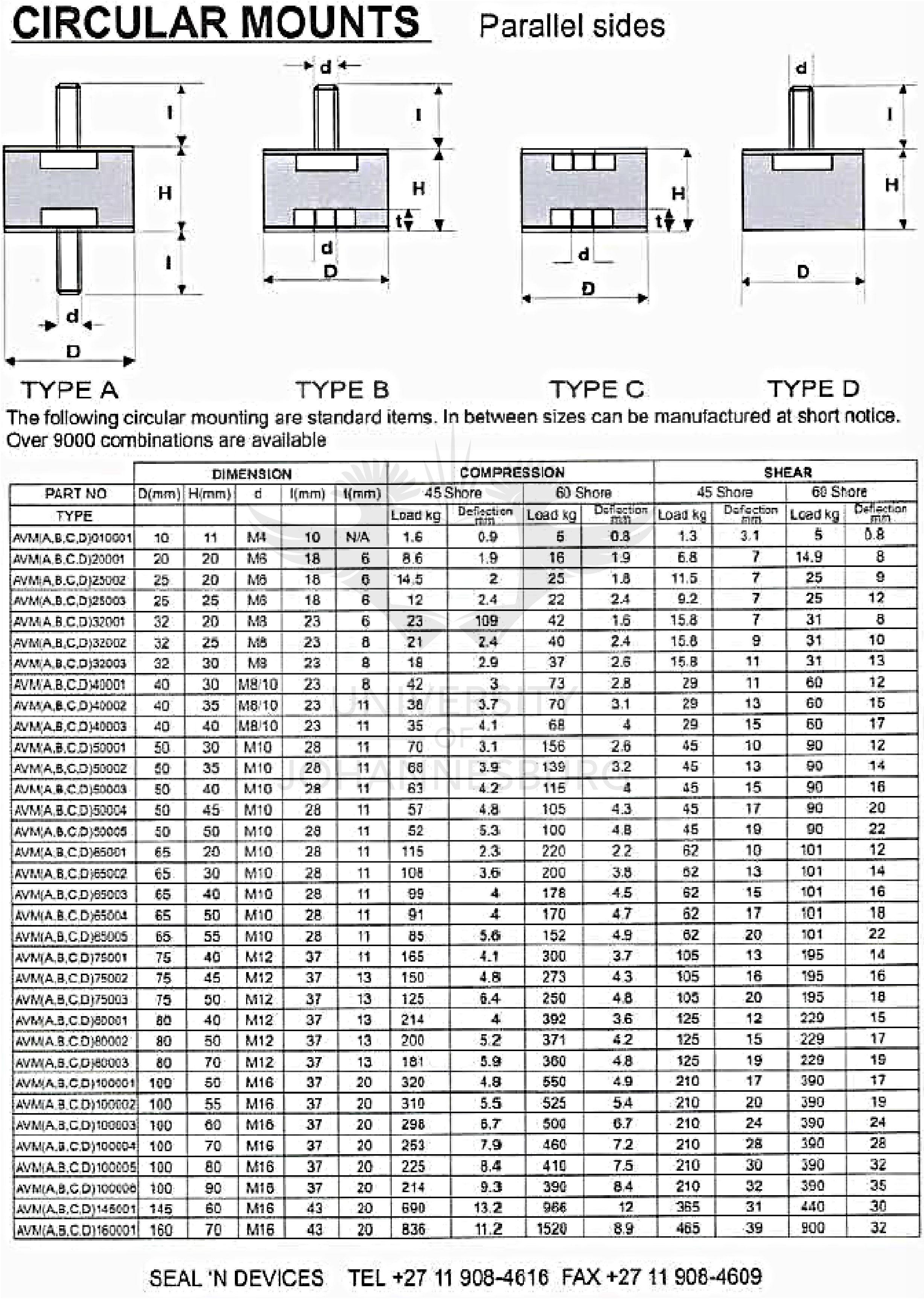


Ideal Diameter



Appendix 31: Anti vibration mounts.

Seal N' Devices



CIRCULAR MOUNTS

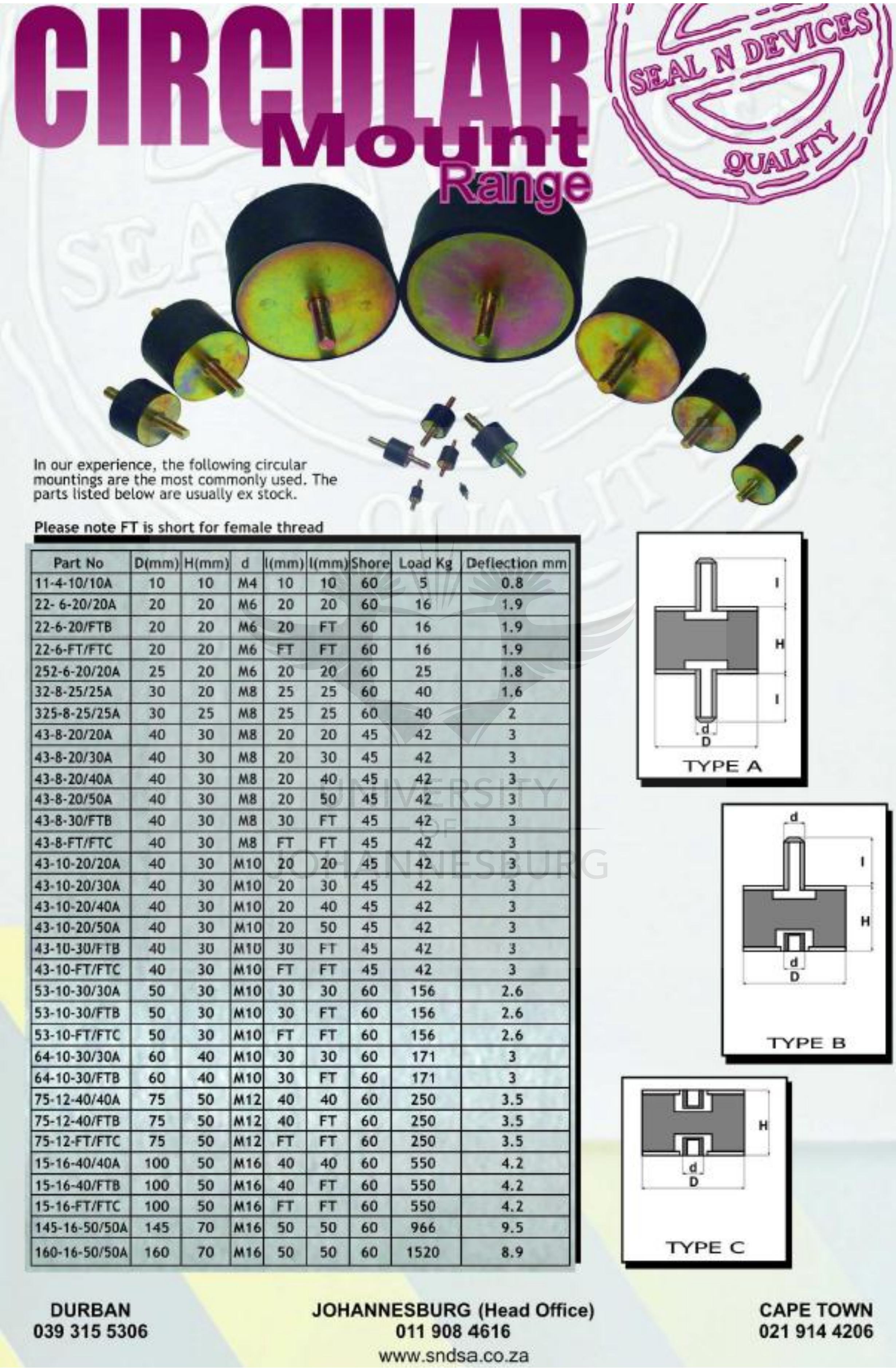
Parallel sides

TYPE A TYPE B TYPE C TYPE D

The following circular mounting are standard items. In between sizes can be manufactured at short notice. Over 9000 combinations are available

	DIMENSION					COMPRESSION				SHEAR			
PART NO	D(mm)	H(mm)	d	l(mm)	t(mm)	45 Shore		60 Shore		45 Shore		60 Shore	
TYPE						Load kg	Deflection mm	Load kg	Deflection mm	Load kg	Deflection mm	Load kg	Deflection mm
AVM(A,B,C,D)010001	10	11	M4	10	N/A	1.6	0.9	6	0.8	1.3	3.1	5	0.8
AVM(A,B,C,D)20001	20	20	M6	18	6	8.6	1.9	16	1.9	6.8	7	14.9	8
AVM(A,B,C,D)25002	25	20	M6	18	6	14.5	2	25	1.8	11.5	7	25	9
AVM(A,B,C,D)25003	25	25	M6	18	6	12	2.4	22	2.4	9.2	7	25	12
AVM(A,B,C,D)32001	32	20	M8	23	6	23	109	42	1.6	15.8	7	31	8
AVM(A,B,C,D)32002	32	25	M8	23	8	21	2.4	40	2.4	15.8	9	31	10
AVM(A,B,C,D)32003	32	30	M8	23	8	18	2.9	37	2.6	15.8	11	31	13
AVM(A,B,C,D)40001	40	30	M8/10	23	8	42	3	73	2.8	29	11	60	12
AVM(A,B,C,D)40002	40	35	M8/10	23	11	38	3.7	70	3.1	29	13	60	15
AVM(A,B,C,D)40003	40	40	M8/10	23	11	35	4.1	68	4	29	15	60	17
AVM(A,B,C,D)50001	50	30	M10	28	11	70	3.1	156	2.6	45	10	90	12
AVM(A,B,C,D)50002	50	35	M10	28	11	66	3.9	139	3.2	45	13	90	14
AVM(A,B,C,D)50003	50	40	M10	28	11	63	4.2	115	4	45	15	90	16
AVM(A,B,C,D)50004	50	45	M10	28	11	57	4.8	105	4.3	45	17	90	20
AVM(A,B,C,D)50005	50	50	M10	28	11	52	5.3	100	4.8	45	19	90	22
AVM(A,B,C,D)65001	65	20	M10	28	11	115	2.3	220	2.2	62	10	101	12
AVM(A,B,C,D)65002	65	30	M10	28	11	108	3.6	200	3.8	62	13	101	14
AVM(A,B,C,D)65003	65	40	M10	28	11	99	4	178	4.5	62	15	101	16
AVM(A,B,C,D)65004	65	50	M10	28	11	91	4	170	4.7	62	17	101	18
AVM(A,B,C,D)65005	65	55	M10	28	11	85	5.6	152	4.9	62	20	101	22
AVM(A,B,C,D)75001	75	40	M12	37	11	165	4.1	300	3.7	105	13	195	14
AVM(A,B,C,D)75002	75	45	M12	37	13	150	4.8	273	4.3	105	16	195	16
AVM(A,B,C,D)75003	75	50	M12	37	13	125	6.4	250	4.8	105	20	195	18
AVM(A,B,C,D)80001	80	40	M12	37	13	214	4	392	3.6	125	12	229	15
AVM(A,B,C,D)80002	80	50	M12	37	13	200	5.2	371	4.2	125	15	229	17
AVM(A,B,C,D)80003	80	70	M12	37	13	181	5.9	360	4.8	125	19	229	19
AVM(A,B,C,D)100001	100	50	M16	37	20	320	4.8	550	4.9	210	17	390	17
AVM(A,B,C,D)100002	100	55	M16	37	20	310	5.5	525	5.4	210	20	390	19
AVM(A,B,C,D)100003	100	60	M16	37	20	298	6.7	500	6.7	210	24	390	24
AVM(A,B,C,D)100004	100	70	M16	37	20	253	7.9	460	7.2	210	28	390	28
AVM(A,B,C,D)100005	100	80	M16	37	20	225	8.4	410	7.5	210	30	390	32
AVM(A,B,C,D)100006	100	90	M16	37	20	214	9.3	390	8.4	210	32	390	35
AVM(A,B,C,D)145001	145	60	M16	43	20	690	13.2	966	12	365	31	440	30
AVM(A,B,C,D)160001	160	70	M16	43	20	836	11.2	1520	8.9	465	39	900	32

SEAL 'N DEVICES TEL +27 11 908-4616 FAX +27 11 908-4609



CIRCULAR Mount Range

In our experience, the following circular mountings are the most commonly used. The parts listed below are usually ex stock.

Please note FT is short for female thread

Part No	D(mm)	H(mm)	d	l(mm)	l(mm)	Shore	Load Kg	Deflection mm
11-4-10/10A	10	10	M4	10	10	60	5	0.8
22- 6-20/20A	20	20	M6	20	20	60	16	1.9
22-6-20/FTB	20	20	M6	20	FT	60	16	1.9
22-6-FT/FTC	20	20	M6	FT	FT	60	16	1.9
252-6-20/20A	25	20	M6	20	20	60	25	1.8
32-8-25/25A	30	20	M8	25	25	60	40	1.6
325-8-25/25A	30	25	M8	25	25	60	40	2
43-8-20/20A	40	30	M8	20	20	45	42	3
43-8-20/30A	40	30	M8	20	30	45	42	3
43-8-20/40A	40	30	M8	20	40	45	42	3
43-8-20/50A	40	30	M8	20	50	45	42	3
43-8-30/FTB	40	30	M8	30	FT	45	42	3
43-8-FT/FTC	40	30	M8	FT	FT	45	42	3
43-10-20/20A	40	30	M10	20	20	45	42	3
43-10-20/30A	40	30	M10	20	30	45	42	3
43-10-20/40A	40	30	M10	20	40	45	42	3
43-10-20/50A	40	30	M10	20	50	45	42	3
43-10-30/FTB	40	30	M10	30	FT	45	42	3
43-10-FT/FTC	40	30	M10	FT	FT	45	42	3
53-10-30/30A	50	30	M10	30	30	60	156	2.6
53-10-30/FTB	50	30	M10	30	FT	60	156	2.6
53-10-FT/FTC	50	30	M10	FT	FT	60	156	2.6
64-10-30/30A	60	40	M10	30	30	60	171	3
64-10-30/FTB	60	40	M10	30	FT	60	171	3
75-12-40/40A	75	50	M12	40	40	60	250	3.5
75-12-40/FTB	75	50	M12	40	FT	60	250	3.5
75-12-FT/FTC	75	50	M12	FT	FT	60	250	3.5
15-16-40/40A	100	50	M16	40	40	60	550	4.2
15-16-40/FTB	100	50	M16	40	FT	60	550	4.2
15-16-FT/FTC	100	50	M16	FT	FT	60	550	4.2
145-16-50/50A	145	70	M16	50	50	60	966	9.5
160-16-50/50A	160	70	M16	50	50	60	1520	8.9

DURBAN
039 315 5306

JOHANNESBURG (Head Office)
011 908 4616
www.sndsa.co.za

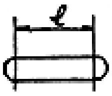
CAPE TOWN
021 914 4206

Appendix 32: Key way standards (BS 4235: Part1 : 1972).

Josco KEY WAY STANDARDS

METRIC

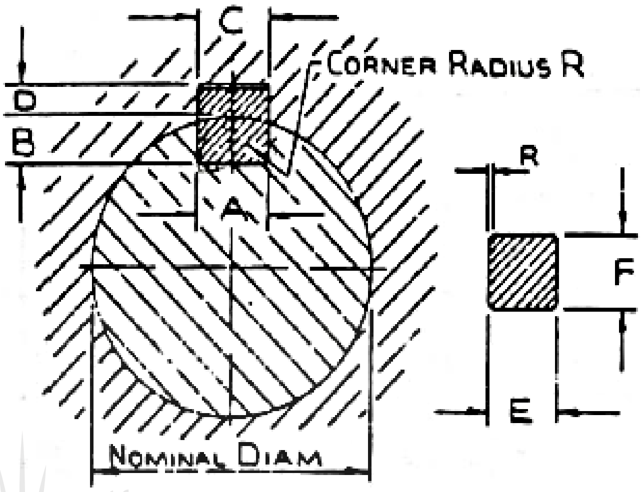
(BS 4235 : PART 1 : 1972)



Safe working loads on keys

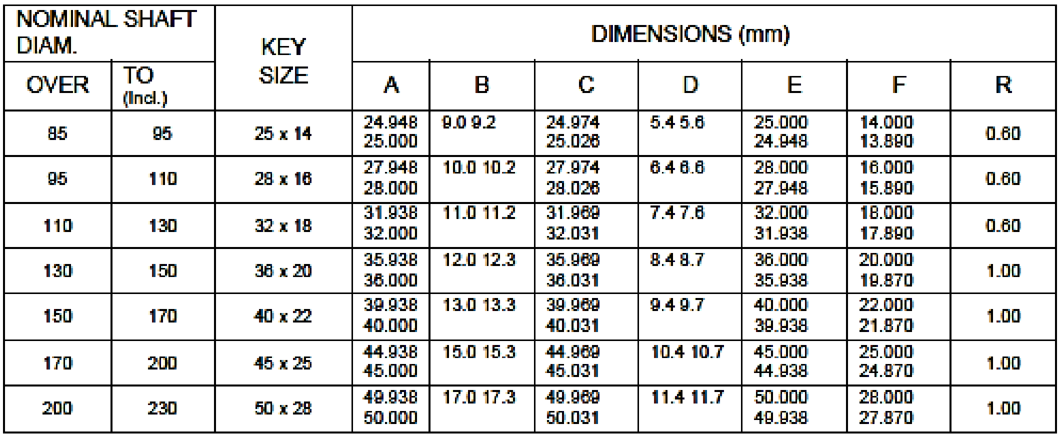
Torque Nm .065 x ℓ x F x shaft dia.(mm)

$$KW = \frac{\text{Torque (Nm) x RPM}}{9550}$$



NOMINAL SHAFT DIAM.		KEY SIZE	DIMENSIONS (mm)						
OVER	TO (Incl.)		A	B	C	D	E	F	R
10	12	4 x 4	3.970 4.000	2.5 2.6	3.985 4.015	1.8 1.9	4.030 3.970		0.16
12	17	5 x 5	4.970 5.000	3.0 3.1	4.985 5.015	2.3 2.4	5.030 4.970		0.25
17	22	6 x 6	5.970 6.000	3.5 3.6	5.985 6.015	2.8 2.9	6.030 5.970		0.25
		8 x 8	7.964 8.000	5.0 5.2	7.982 8.018	3.3 3.5	8.000 7.970		0.25
		10 x 10	9.964 10.000	6.0 6.2	9.982 10.018	4.3 4.5	10.000 9.964		0.40
		12 x 12	11.957 12.000	7.5 7.7	11.979 12.021	4.9 5.1	12.000 11.957		0.40
		16 x 16	15.957 16.000	10.0 10.2	15.979 16.021	6.4 6.6	16.000 15.957		0.40
		20 x 20	19.948 20.000	12.0 12.3	19.974 20.026	8.4 8.7	20.000 19.957		0.60
		22 x 22	21.948 22.000	13.0 13.3	21.974 22.026	9.4 9.7	22.000 21.948		0.60
22	30	8 x 7	7.964 8.000	4.0 4.2	7.982 8.018	3.3 3.5	8.000 7.964	7.000 6.910	0.25
30	38	10 x 8	9.964 10.000	5.0 5.2	9.982 10.018	3.3 3.5	10.000 9.964	8.000 7.910	0.40
38	44	12 x 8	11.957 12.000	5.0 5.2	11.979 12.021	3.3 3.5	12.000 11.957	8.000 7.910	0.40
		12 x 10	11.957 12.000	6.0 6.2	11.979 12.021	4.3 4.5	12.000 11.957	10.000 9.964	0.40
44	50	14 x 9	13.957 14.000	5.5 5.7	13.979 14.021	3.8 4.0	14.000 13.957	9.000 8.910	0.40
50	58	16 x 10	15.957 16.000	6.0 6.2	15.979 16.021	4.3 4.5	16.000 15.957	10.000 9.910	0.40
58	65	18 x 11	17.957 18.000	7.0 7.2	17.979 18.021	4.4 4.6	18.000 17.957	11.000 10.890	0.40
65	75	20 x 12	19.948 20.000	7.5 7.7	19.974 20.026	4.9 5.1	20.000 19.948	12.000 11.890	0.60
75	85	22 x 14	21.948 22.000	9.0 9.2	21.974 22.026	5.4 5.6	22.000 21.948	14.000 13.890	0.60

* :-Indicates Non Standard Kev.



NOMINAL SHAFT DIAM.		KEY SIZE	DIMENSIONS (mm)						
OVER	TO (Incl.)		A	B	C	D	E	F	R
85	95	25 x 14	24.948 25.000	9.0 9.2	24.974 25.026	5.4 5.6	25.000 24.948	14.000 13.890	0.60
95	110	28 x 16	27.948 28.000	10.0 10.2	27.974 28.026	6.4 6.6	28.000 27.948	16.000 15.890	0.60
110	130	32 x 18	31.938 32.000	11.0 11.2	31.969 32.031	7.4 7.6	32.000 31.938	18.000 17.890	0.60
130	150	36 x 20	35.938 36.000	12.0 12.3	35.969 36.031	8.4 8.7	36.000 35.938	20.000 19.870	1.00
150	170	40 x 22	39.938 40.000	13.0 13.3	39.969 40.031	9.4 9.7	40.000 39.938	22.000 21.870	1.00
170	200	45 x 25	44.938 45.000	15.0 15.3	44.969 45.031	10.4 10.7	45.000 44.938	25.000 24.870	1.00
200	230	50 x 28	49.938 50.000	17.0 17.3	49.969 50.031	11.4 11.7	50.000 49.938	28.000 27.870	1.00

Appendix 33: SKF machine type life specification.

Table 9

Guideline values of specification life for different machine types

Machine type	**Specification life** Operating hours
Household machines, agricultural machines, instruments, technical equipment for medical use	300 ... 3 000
Machines used for short periods or intermittently: electric hand tools, lifting tackle in workshops, construction equipment and machines	3 000 ... 8 000
Machines used for short periods or intermittently where high operational reliability is required: lifts (elevators), cranes for packaged goods or slings of drums etc.	8 000 ... 12 000
Machines for use 8 hours a day, but not always fully utilized: gear drives for general purposes, electric motors for industrial use, rotary crushers	10 000 ... 25 000
Machines for use 8 hours a day and fully utilized: machine tools, woodworking machines, machines for the engineering industry, cranes for bulk materials, ventilator fans, conveyor belts, printing equipment, separators and centrifuges	20 000 ... 30 000
Machines for continuous 24 hour use: rolling mill gear units, medium-size electrical machinery, compressors, mine hoists, pumps, textile machinery	40 000 ... 50 000
Wind energy machinery, this includes main shaft, yaw, pitching gearbox, generator bearings	30 000 ... 100 000
Water works machinery, rotary furnaces, cable stranding machines, propulsion machinery for ocean-going vessels	60 000 ... 100 000
Large electric machines, power generation plant, mine pumps, mine ventilator fans, tunnel shaft bearings for ocean-going vessels	> 100 000

Appendix 34: SKF Bearing installation, maintenance & technical data.

1 Lock nut tightening angle

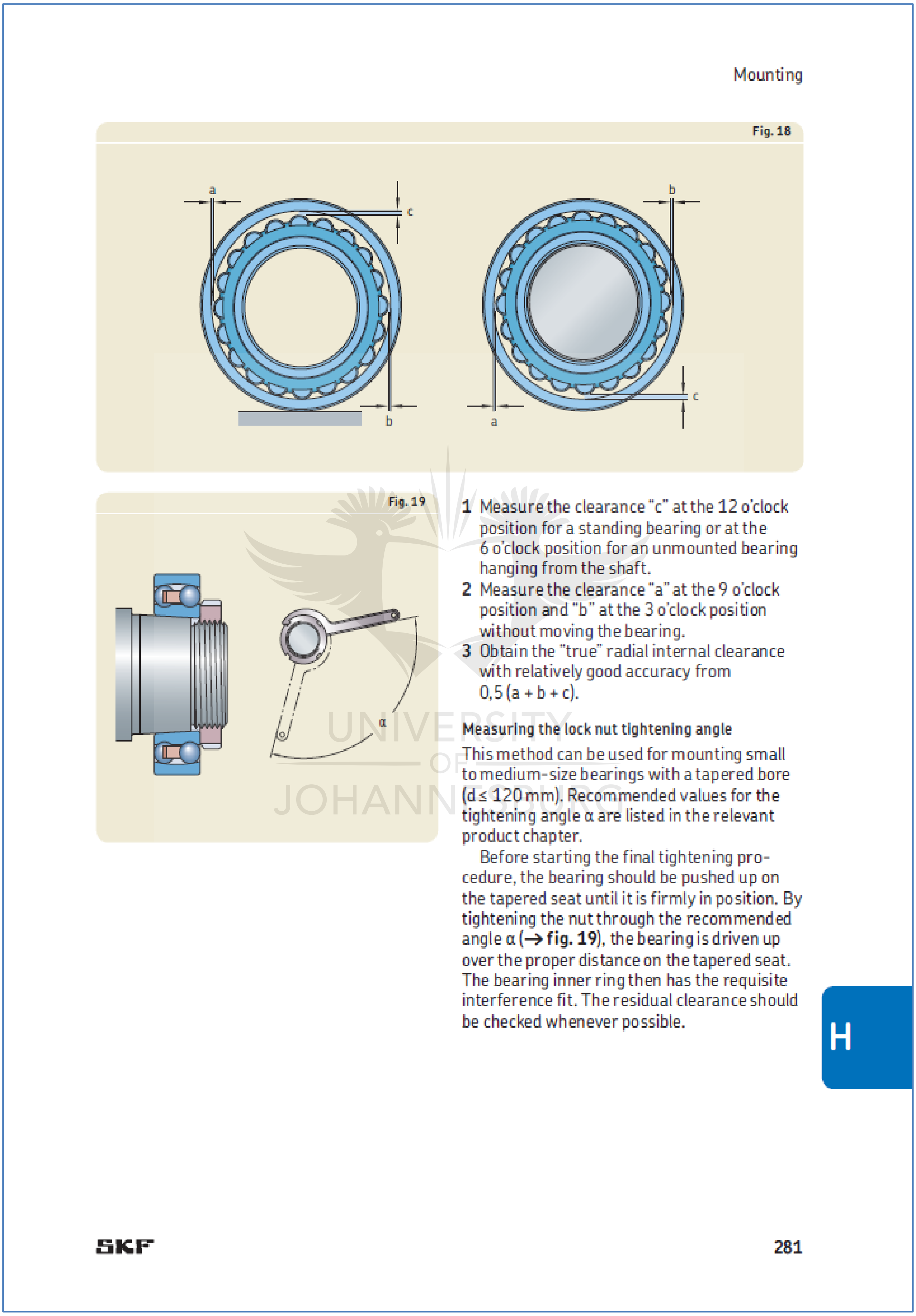


Fig. 18

Fig. 19

1 Measure the clearance "c" at the 12 o'clock position for a standing bearing or at the 6 o'clock position for an unmounted bearing hanging from the shaft.

2 Measure the clearance "a" at the 9 o'clock position and "b" at the 3 o'clock position without moving the bearing.

3 Obtain the "true" radial internal clearance with relatively good accuracy from 0,5 (a + b + c).

Measuring the lock nut tightening angle

This method can be used for mounting small to medium-size bearings with a tapered bore ($d \leq 120$ mm). Recommended values for the tightening angle α are listed in the relevant product chapter.

Before starting the final tightening procedure, the bearing should be pushed up on the tapered seat until it is firmly in position. By tightening the nut through the recommended angle α (→ **fig. 19**), the bearing is driven up over the proper distance on the tapered seat. The bearing inner ring then has the requisite interference fit. The residual clearance should be checked whenever possible.

Bearings for vibratory applications

For vibratory applications, SKF supplies spherical roller bearings with a cylindrical or tapered bore and surface-hardened stamped steel cages (series designations 223../VA405). These bearings have the same dimensions and performance characteristics as basic design bearings, but have C4 radial internal clearance as standard.

Bearings for vibratory applications are also available with a PTFE coated cylindrical bore (designation suffix VA406). These bearings are manufactured to VA405 specifications, with the exception of the bore. VA406 bearings are intended for the non-locating bearing position in vibratory applications with outer ring rotation. The PTFE coating prevents fretting corrosion between the shaft and the bearing bore. Therefore, shafts do not require special heat treatments or coatings.

All bearings are equipped with an annular groove and three lubrication holes in the outer ring.

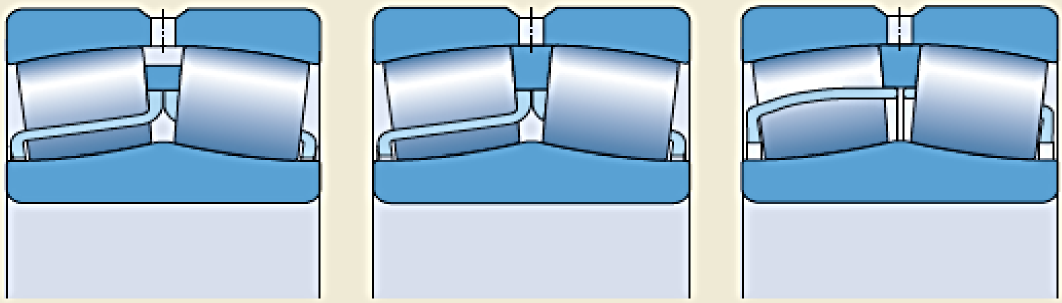
Depending on their size, SKF spherical roller bearings for vibratory applications are manufactured to one of following designs (→ **fig. 10**):

- E/VA405 spherical roller bearings have two surface-hardened stamped window-type steel cages, an inner ring without flanges and a floating guide ring centred on the inner ring or on the cages.
- EJA/VA405 and CCJA/W33VA405 spherical roller bearings have two surface-hardened stamped window-type steel cages, an inner ring without flanges and a floating guide ring centred on the outer ring raceway.

WARNING
PTFE coatings exposed to an open flame or temperatures above 300 °C (*570 °F*) are a health and environmental hazard! They remain dangerous even after they have cooled.

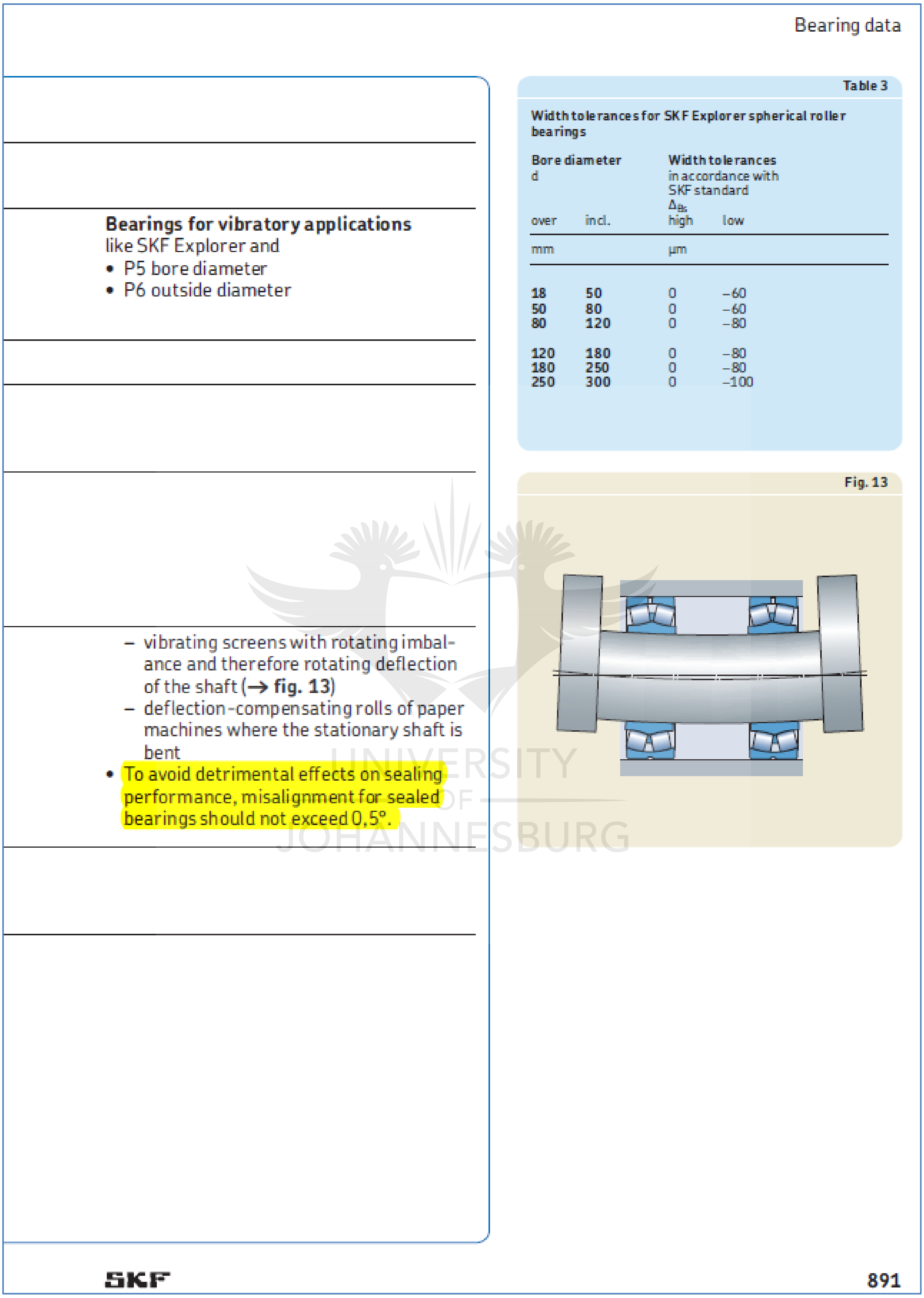
Read and follow the safety precautions under *Seal materials* (→ **page 155**).

Fig. 10



E/VA405 design EJA/VA405 design CCJA/W33VA405 design

3 Allowable degree of deflection



Bearings for vibratory applications
like SKF Explorer and

- P5 bore diameter
- P6 outside diameter

  - vibrating screens with rotating imbalance and therefore rotating deflection of the shaft (→ **fig. 13**)
  - deflection-compensating rolls of paper machines where the stationary shaft is bent
- To avoid detrimental effects on sealing performance, misalignment for sealed bearings should not exceed 0,5°.

Table 3

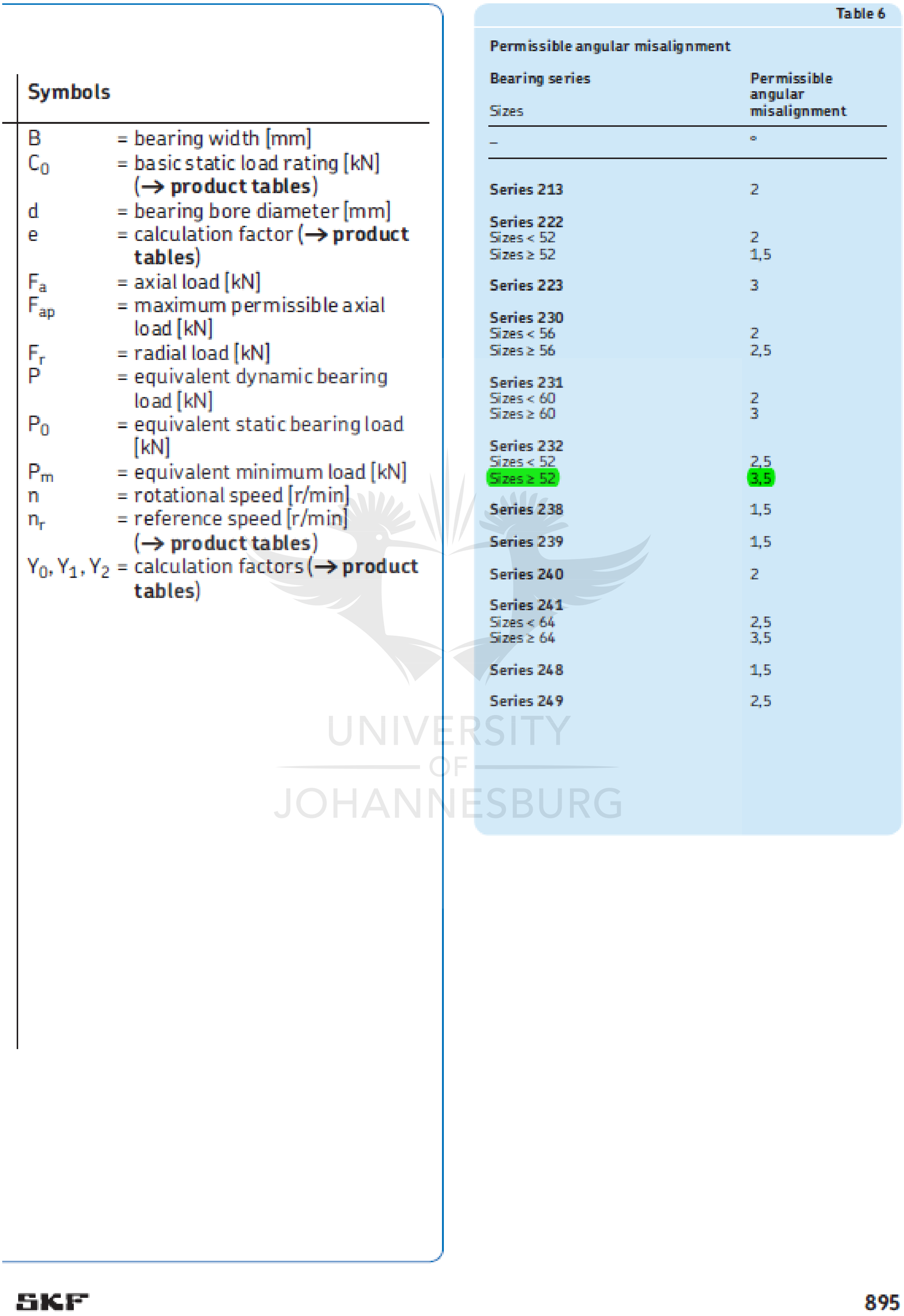
Width tolerances for SKF Explorer spherical roller bearings

Bore diameter d		**Width tolerances** in accordance with SKF standard Δ_{Bs}	
over	incl.	high	low
mm		µm	
18	**50**	0	–60
50	**80**	0	–60
80	**120**	0	–80
120	**180**	0	–80
180	**250**	0	–80
250	**300**	0	–100

Fig. 13

SKF

4 Maximum misalignments



Symbols

B = bearing width [mm]
C_0 = basic static load rating [kN] (→ **product tables**)
d = bearing bore diameter [mm]
e = calculation factor (→ **product tables**)
F_a = axial load [kN]
F_{ap} = maximum permissible axial load [kN]
F_r = radial load [kN]
P = equivalent dynamic bearing load [kN]
P_0 = equivalent static bearing load [kN]
P_m = equivalent minimum load [kN]
n = rotational speed [r/min]
n_r = reference speed [r/min] (→ **product tables**)
Y_0, Y_1, Y_2 = calculation factors (→ **product tables**)

Table 6

Permissible angular misalignment

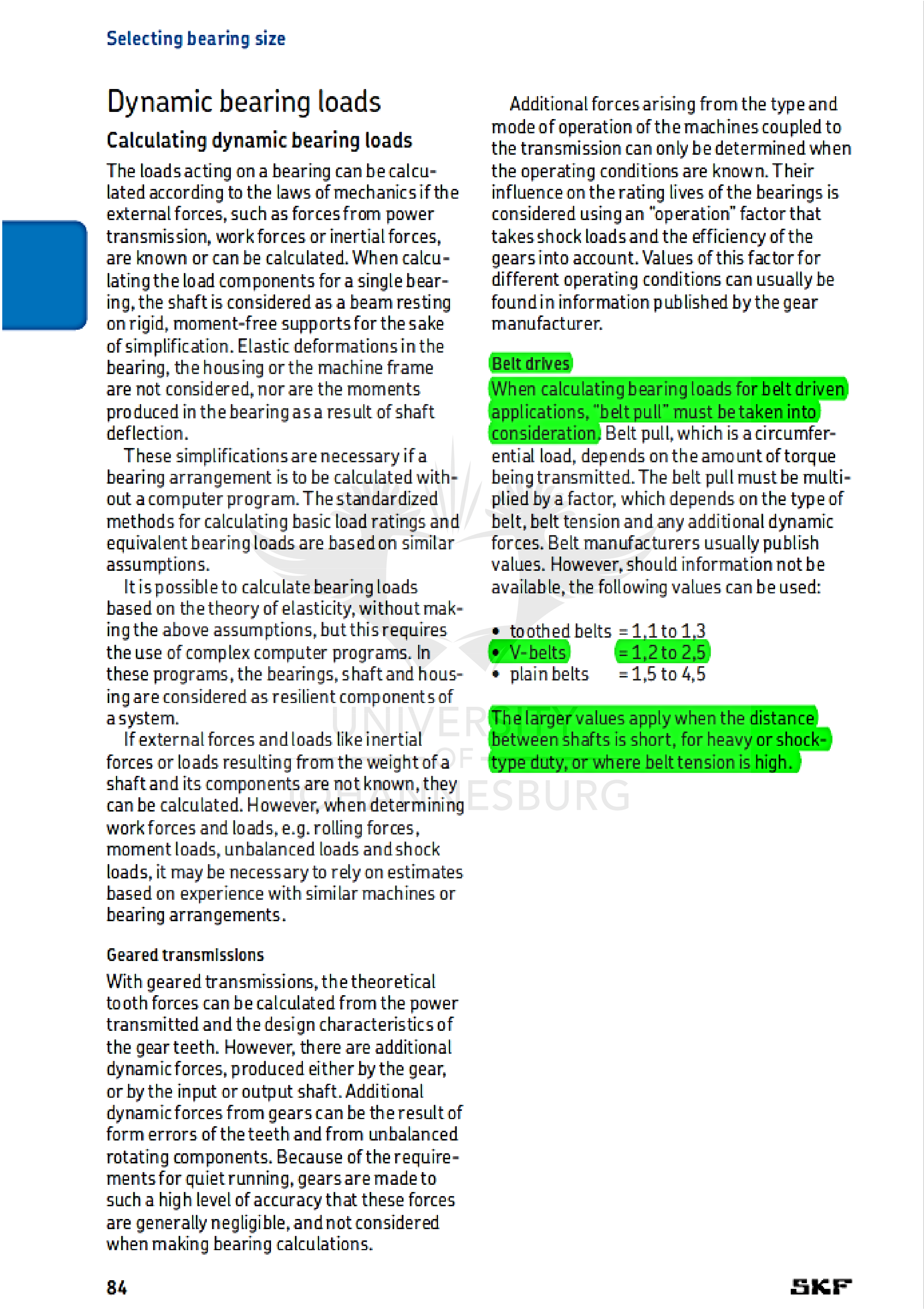
Bearing series Sizes	**Permissible angular misalignment**
–	°
Series 213	2
Series 222	
Sizes < 52	2
Sizes ≥ 52	1,5
Series 223	3
Series 230	
Sizes < 56	2
Sizes ≥ 56	2,5
Series 231	
Sizes < 60	2
Sizes ≥ 60	3
Series 232	
Sizes < 52	2,5
Sizes ≥ 52	3,5
Series 238	1,5
Series 239	1,5
Series 240	2
Series 241	
Sizes < 64	2,5
Sizes ≥ 64	3,5
Series 248	1,5
Series 249	2,5

SKF

895

Appendix 35: Bearing loading.

1 Dynamic bearing loads due to belt pull.



Dynamic bearing loads

Calculating dynamic bearing loads

The loads acting on a bearing can be calculated according to the laws of mechanics if the external forces, such as forces from power transmission, work forces or inertial forces, are known or can be calculated. When calculating the load components for a single bearing, the shaft is considered as a beam resting on rigid, moment-free supports for the sake of simplification. Elastic deformations in the bearing, the housing or the machine frame are not considered, nor are the moments produced in the bearing as a result of shaft deflection.

These simplifications are necessary if a bearing arrangement is to be calculated without a computer program. The standardized methods for calculating basic load ratings and equivalent bearing loads are based on similar assumptions.

It is possible to calculate bearing loads based on the theory of elasticity, without making the above assumptions, but this requires the use of complex computer programs. In these programs, the bearings, shaft and housing are considered as resilient components of a system.

If external forces and loads like inertial forces or loads resulting from the weight of a shaft and its components are not known, they can be calculated. However, when determining work forces and loads, e.g. rolling forces, moment loads, unbalanced loads and shock loads, it may be necessary to rely on estimates based on experience with similar machines or bearing arrangements.

Geared transmissions

With geared transmissions, the theoretical tooth forces can be calculated from the power transmitted and the design characteristics of the gear teeth. However, there are additional dynamic forces, produced either by the gear, or by the input or output shaft. Additional dynamic forces from gears can be the result of form errors of the teeth and from unbalanced rotating components. Because of the requirements for quiet running, gears are made to such a high level of accuracy that these forces are generally negligible, and not considered when making bearing calculations.

Additional forces arising from the type and mode of operation of the machines coupled to the transmission can only be determined when the operating conditions are known. Their influence on the rating lives of the bearings is considered using an "operation" factor that takes shock loads and the efficiency of the gears into account. Values of this factor for different operating conditions can usually be found in information published by the gear manufacturer.

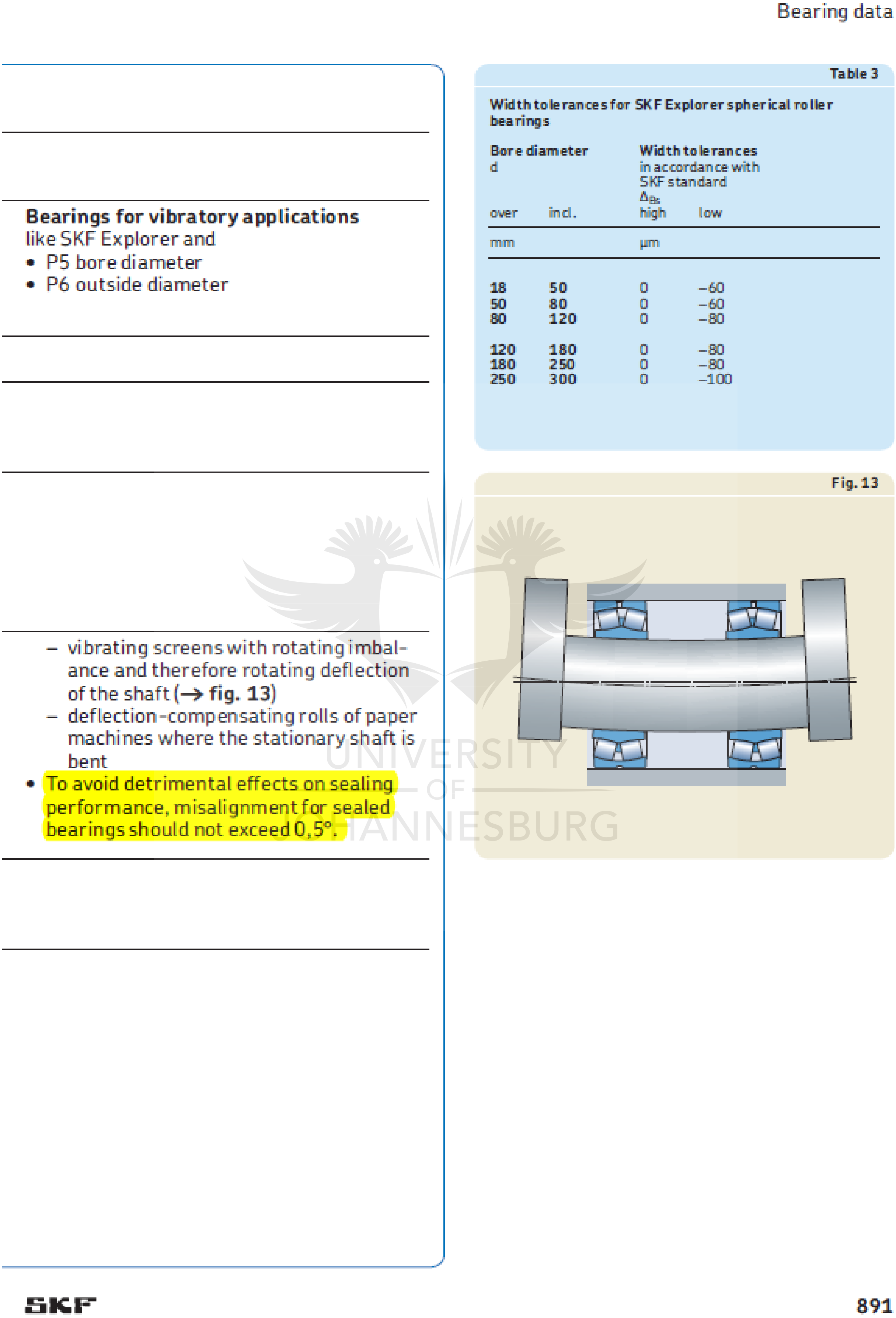
Belt drives

When calculating bearing loads for belt driven applications, "belt pull" must be taken into consideration. Belt pull, which is a circumferential load, depends on the amount of torque being transmitted. The belt pull must be multiplied by a factor, which depends on the type of belt, belt tension and any additional dynamic forces. Belt manufacturers usually publish values. However, should information not be available, the following values can be used:

- toothed belts = 1,1 to 1,3
- V-belts = 1,2 to 2,5
- plain belts = 1,5 to 4,5

The larger values apply when the distance between shafts is short, for heavy or shock-type duty, or where belt tension is high.

3 Maximum misalignment for sealed spherical roller bearings



Bearings for vibratory applications
like SKF Explorer and
- P5 bore diameter
- P6 outside diameter

- vibrating screens with rotating imbalance and therefore rotating deflection of the shaft (→ **fig. 13**)
- deflection-compensating rolls of paper machines where the stationary shaft is bent

- To avoid detrimental effects on sealing performance, misalignment for sealed bearings should not exceed 0,5°.

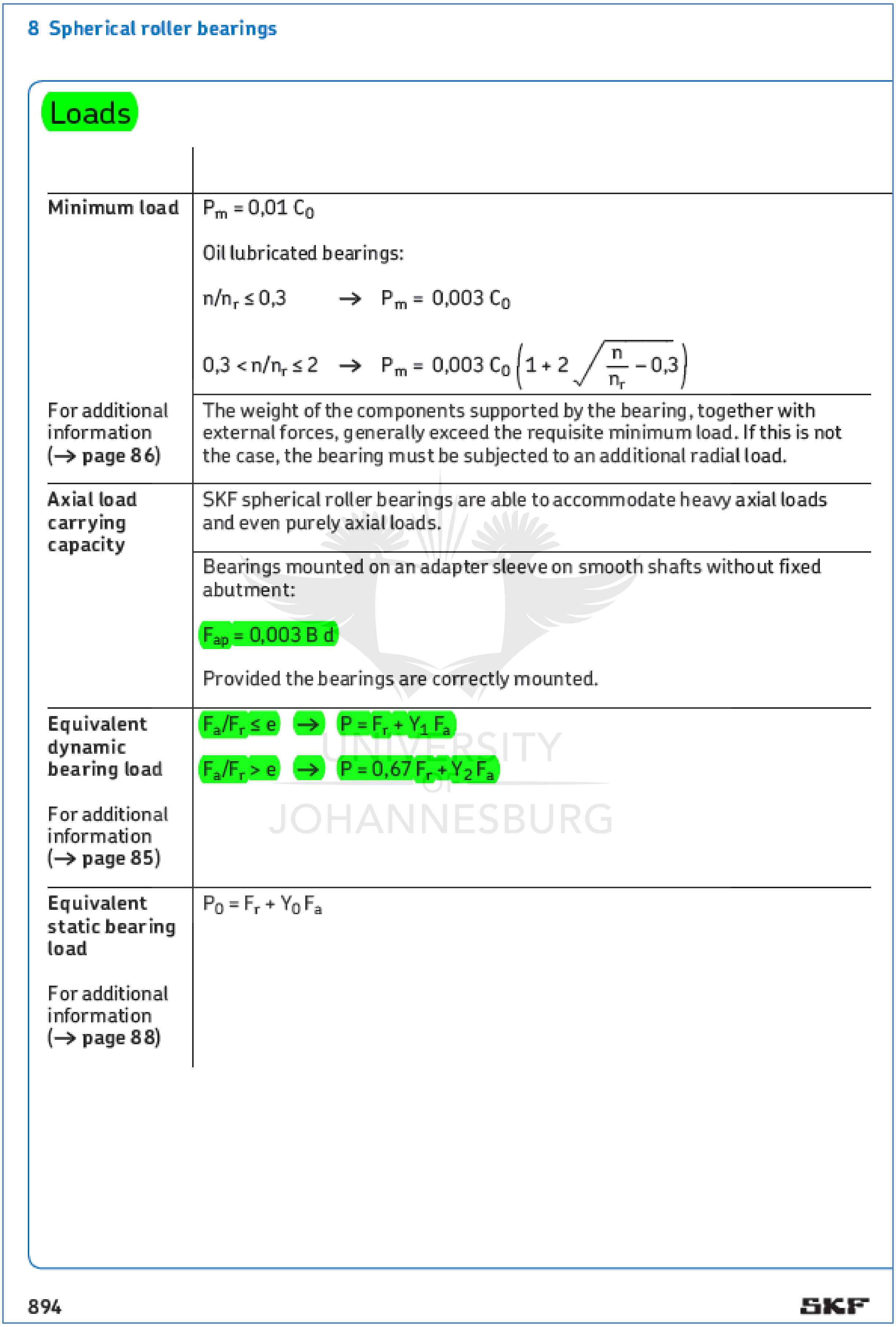
Table 3

Width tolerances for SKF Explorer spherical roller bearings

Bore diameter d		**Width tolerances** in accordance with SKF standard Δ_{Bs}	
over	incl.	high	low
mm		µm	
18	**50**	0	–60
50	**80**	0	–60
80	**120**	0	–80
120	**180**	0	–80
180	**250**	0	–80
250	**300**	0	–100

Fig. 13

3 Equivalent dynamic bearing load (P) calculation



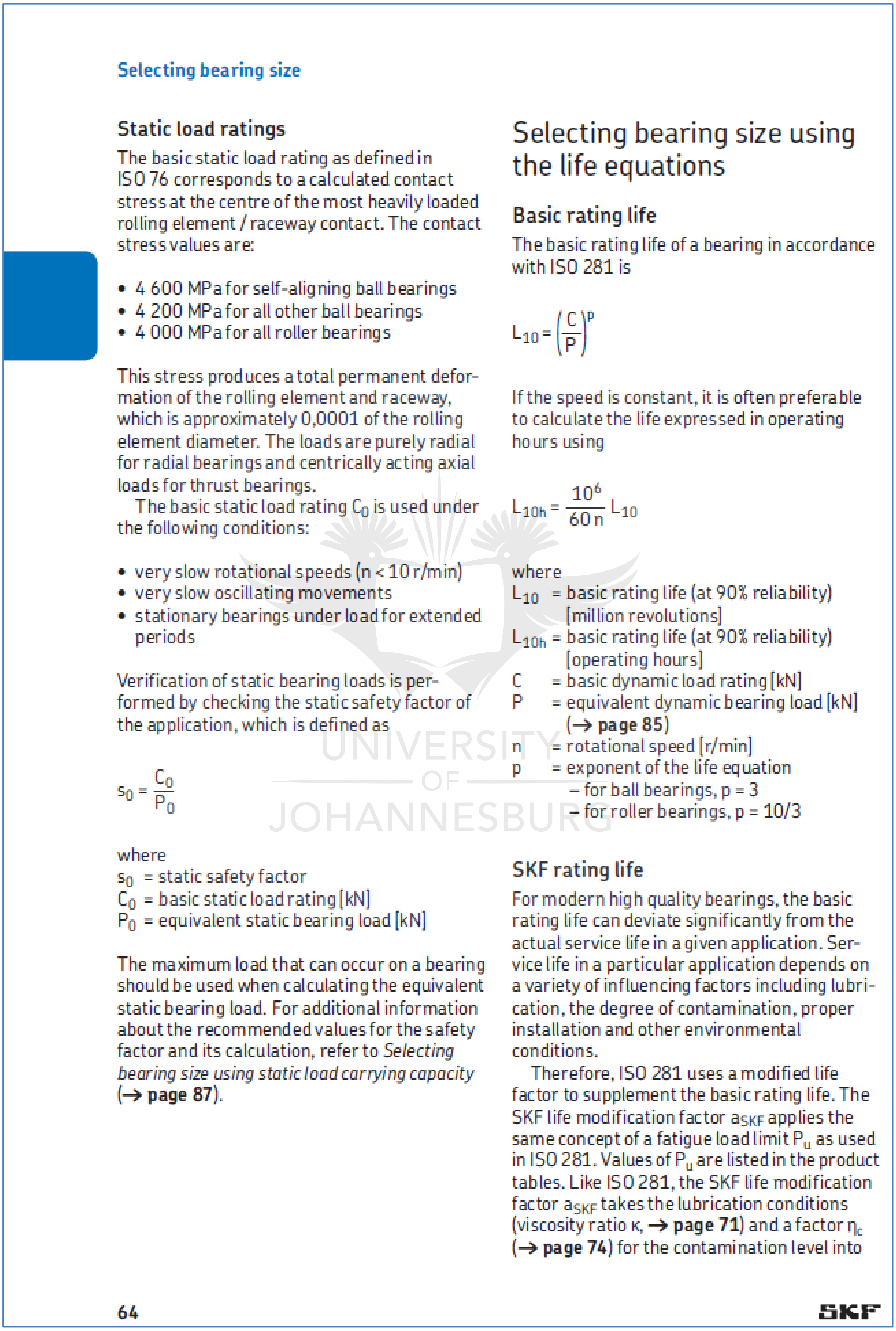
8 Spherical roller bearings

Loads

Minimum load	$P_m = 0{,}01\ C_0$ Oil lubricated bearings: $n/n_r \leq 0{,}3 \rightarrow P_m = 0{,}003\ C_0$ $0{,}3 < n/n_r \leq 2 \rightarrow P_m = 0{,}003\ C_0 \left(1 + 2\sqrt{\frac{n}{n_r} - 0{,}3}\right)$
For additional information (→ **page 86**)	The weight of the components supported by the bearing, together with external forces, generally exceed the requisite minimum load. If this is not the case, the bearing must be subjected to an additional radial load.
Axial load carrying capacity	SKF spherical roller bearings are able to accommodate heavy axial loads and even purely axial loads. Bearings mounted on an adapter sleeve on smooth shafts without fixed abutment: $F_{ap} = 0{,}003\ B\ d$ Provided the bearings are correctly mounted.
Equivalent dynamic bearing load	$F_a/F_r \leq e \rightarrow P = F_r + Y_1 F_a$ $F_a/F_r > e \rightarrow P = 0{,}67\ F_r + Y_2 F_a$
For additional information (→ **page 85**)	
Equivalent static bearing load	$P_0 = F_r + Y_0 F_a$
For additional information (→ **page 88**)	

894

SKF



Static load ratings

The basic static load rating as defined in ISO 76 corresponds to a calculated contact stress at the centre of the most heavily loaded rolling element / raceway contact. The contact stress values are:

- 4 600 MPa for self-aligning ball bearings
- 4 200 MPa for all other ball bearings
- 4 000 MPa for all roller bearings

This stress produces a total permanent deformation of the rolling element and raceway, which is approximately 0,0001 of the rolling element diameter. The loads are purely radial for radial bearings and centrically acting axial loads for thrust bearings.

The basic static load rating C_0 is used under the following conditions:

- very slow rotational speeds (n < 10 r/min)
- very slow oscillating movements
- stationary bearings under load for extended periods

Verification of static bearing loads is performed by checking the static safety factor of the application, which is defined as

$$s_0 = \frac{C_0}{P_0}$$

where

s_0 = static safety factor

C_0 = basic static load rating [kN]

P_0 = equivalent static bearing load [kN]

The maximum load that can occur on a bearing should be used when calculating the equivalent static bearing load. For additional information about the recommended values for the safety factor and its calculation, refer to *Selecting bearing size using static load carrying capacity* (→ **page 87**).

Selecting bearing size using the life equations

Basic rating life

The basic rating life of a bearing in accordance with ISO 281 is

$$L_{10} = \left(\frac{C}{P}\right)^p$$

If the speed is constant, it is often preferable to calculate the life expressed in operating hours using

$$L_{10h} = \frac{10^6}{60\,n}\,L_{10}$$

where

L_{10} = basic rating life (at 90% reliability) [million revolutions]

L_{10h} = basic rating life (at 90% reliability) [operating hours]

C = basic dynamic load rating [kN]

P = equivalent dynamic bearing load [kN] (→ **page 85**)

n = rotational speed [r/min]

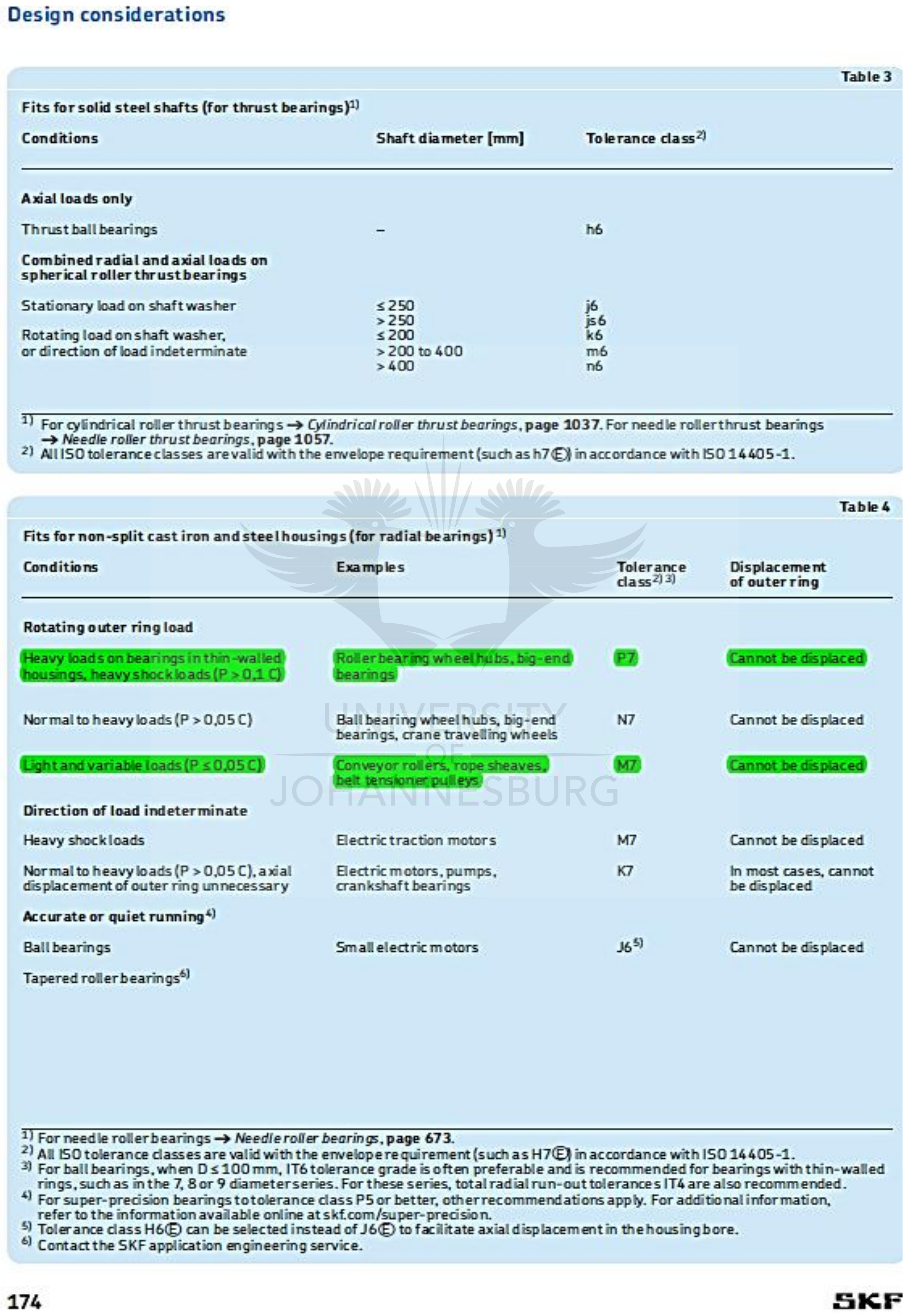
p = exponent of the life equation
  – for ball bearings, p = 3
  – for roller bearings, p = 10/3

SKF rating life

For modern high quality bearings, the basic rating life can deviate significantly from the actual service life in a given application. Service life in a particular application depends on a variety of influencing factors including lubrication, the degree of contamination, proper installation and other environmental conditions.

Therefore, ISO 281 uses a modified life factor to supplement the basic rating life. The SKF life modification factor a_{SKF} applies the same concept of a fatigue load limit P_u as used in ISO 281. Values of P_u are listed in the product tables. Like ISO 281, the SKF life modification factor a_{SKF} takes the lubrication conditions (viscosity ratio κ, → **page 71**) and a factor η_c (→ **page 74**) for the contamination level into

Appendix 36: Bearing tolerances.



Design considerations

Table 3

Fits for solid steel shafts (for thrust bearings)[1]

Conditions	Shaft diameter [mm]	Tolerance class[2]
Axial loads only		
Thrust ball bearings	–	h6
Combined radial and axial loads on spherical roller thrust bearings		
Stationary load on shaft washer	≤ 250 > 250	j6 js6
Rotating load on shaft washer, or direction of load indeterminate	≤ 200 > 200 to 400 > 400	k6 m6 n6

---

[1] For cylindrical roller thrust bearings → *Cylindrical roller thrust bearings*, **page 1037**. For needle roller thrust bearings → *Needle roller thrust bearings*, **page 1057**.

[2] All ISO tolerance classes are valid with the envelope requirement (such as h7Ⓔ) in accordance with ISO 14405-1.

Table 4

Fits for non-split cast iron and steel housings (for radial bearings)[1]

Conditions	Examples	Tolerance class[2) 3)]	Displacement of outer ring
Rotating outer ring load			
Heavy loads on bearings in thin-walled housings, heavy shock loads (P > 0,1 C)	Roller bearing wheel hubs, big-end bearings	P7	Cannot be displaced
Normal to heavy loads (P > 0,05 C)	Ball bearing wheel hubs, big-end bearings, crane travelling wheels	N7	Cannot be displaced
Light and variable loads (P ≤ 0,05 C)	Conveyor rollers, rope sheaves, belt tensioner pulleys	M7	Cannot be displaced
Direction of load indeterminate			
Heavy shock loads	Electric traction motors	M7	Cannot be displaced
Normal to heavy loads (P > 0,05 C), axial displacement of outer ring unnecessary	Electric motors, pumps, crankshaft bearings	K7	In most cases, cannot be displaced
Accurate or quiet running[4]			
Ball bearings	Small electric motors	J6[5]	Cannot be displaced
Tapered roller bearings[6]			

---

[1] For needle roller bearings → *Needle roller bearings*, **page 673**.

[2] All ISO tolerance classes are valid with the envelope requirement (such as H7Ⓔ) in accordance with ISO 14405-1.

[3] For ball bearings, when D ≤ 100 mm, IT6 tolerance grade is often preferable and is recommended for bearings with thin-walled rings, such as in the 7, 8 or 9 diameter series. For these series, total radial run-out tolerances IT4 are also recommended.

[4] For super-precision bearings to tolerance class P5 or better, other recommendations apply. For additional information, refer to the information available online at skf.com/super-precision.

[5] Tolerance class H6Ⓔ can be selected instead of J6Ⓔ to facilitate axial displacement in the housing bore.

[6] Contact the SKF application engineering service.

174

SKF

Table 8e

Housing tolerances and resultant fits

Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Housing bore diameter deviations, resultant fits Tolerance classes							
				N6Ⓔ		N7Ⓔ		P6Ⓔ		P7Ⓔ	
				Deviations (housing bore diameter) Theoretical interference (–)/clearance (+) Probable interference (–)/clearance (+)							
over	incl.	high	low								
mm		µm		µm							
6	**10**	0	–8	–16	–7	–19	–4	–21	–12	–24	–9
				–16	+1	–19	+4	–21	–4	–24	–1
				–14	–1	–16	+1	–19	–6	–21	–4
10	**18**	0	–8	–20	–9	–23	–5	–26	–15	–29	–11
				–20	–1	–23	+3	–26	–7	–29	–3
				–18	–3	–20	0	–24	–9	–26	–6
18	**30**	0	–9	–24	–11	–28	–7	–31	–18	–35	–14
				–24	–2	–28	+2	–31	–9	–35	–5
				–21	–5	–25	–1	–28	–12	–32	–8
30	**50**	0	–11	–28	–12	–33	–8	–37	–21	–42	–17
				–28	–1	–33	+3	–37	–10	–42	–6
				–25	–4	–29	–1	–34	–13	–38	–10
50	**80**	0	–13	–33	–14	–39	–9	–45	–26	–51	–21
				–33	–1	–39	+4	–45	–13	–51	–8
				–29	–5	–34	–1	–41	–17	–46	–13
80	**120**	0	–15	–38	–16	–45	–10	–52	–30	–59	–24
				–38	–1	–45	+5	–52	–15	–59	–9
				–33	–6	–40	0	–47	–20	–54	–14
120	**150**	0	–18	–45	–20	–52	–12	–61	–36	–68	–28
				–45	–2	–52	+6	–61	–18	–68	–10
				–39	–8	–45	–1	–55	–24	–61	–17
150	**180**	0	–25	–45	–20	–52	–12	–61	–36	–68	–28
				–45	+5	–52	+13	–61	–11	–68	–3
				–38	–2	–44	+5	–54	–18	–60	–11
180	**250**	0	–30	–51	–22	–60	–14	–70	–41	–79	–33
				–51	+8	–60	+16	–70	–11	–79	–3
				–43	0	–50	+6	–62	–19	–69	–13
250	**315**	0	–35	–57	–25	–66	–14	–79	–47	–88	–36
				–57	+10	–66	+21	–79	–12	–88	–1
				–48	+1	–54	+9	–70	–21	–76	–13
315	**400**	0	–40	–62	–26	–73	–16	–87	–51	–98	–41
				–62	+14	–73	+24	–87	–11	–98	–1
				–51	+3	–60	+11	–76	–22	–85	–14
400	**500**	0	–45	–67	–27	–80	–17	–95	–55	–108	–45
				–67	+18	–80	+28	–95	–10	–108	0
				–55	+6	–65	+13	–83	–22	–93	–15
500	**630**	0	–50	–88	–44	–114	–44	–122	–78	–148	–78
				–88	+6	–114	+6	–122	–28	–148	–28
				–75	–7	–98	–10	–109	–41	–132	–44

Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Shaft and housing tolerances and fits* (→ **page 171**).

Table 8d

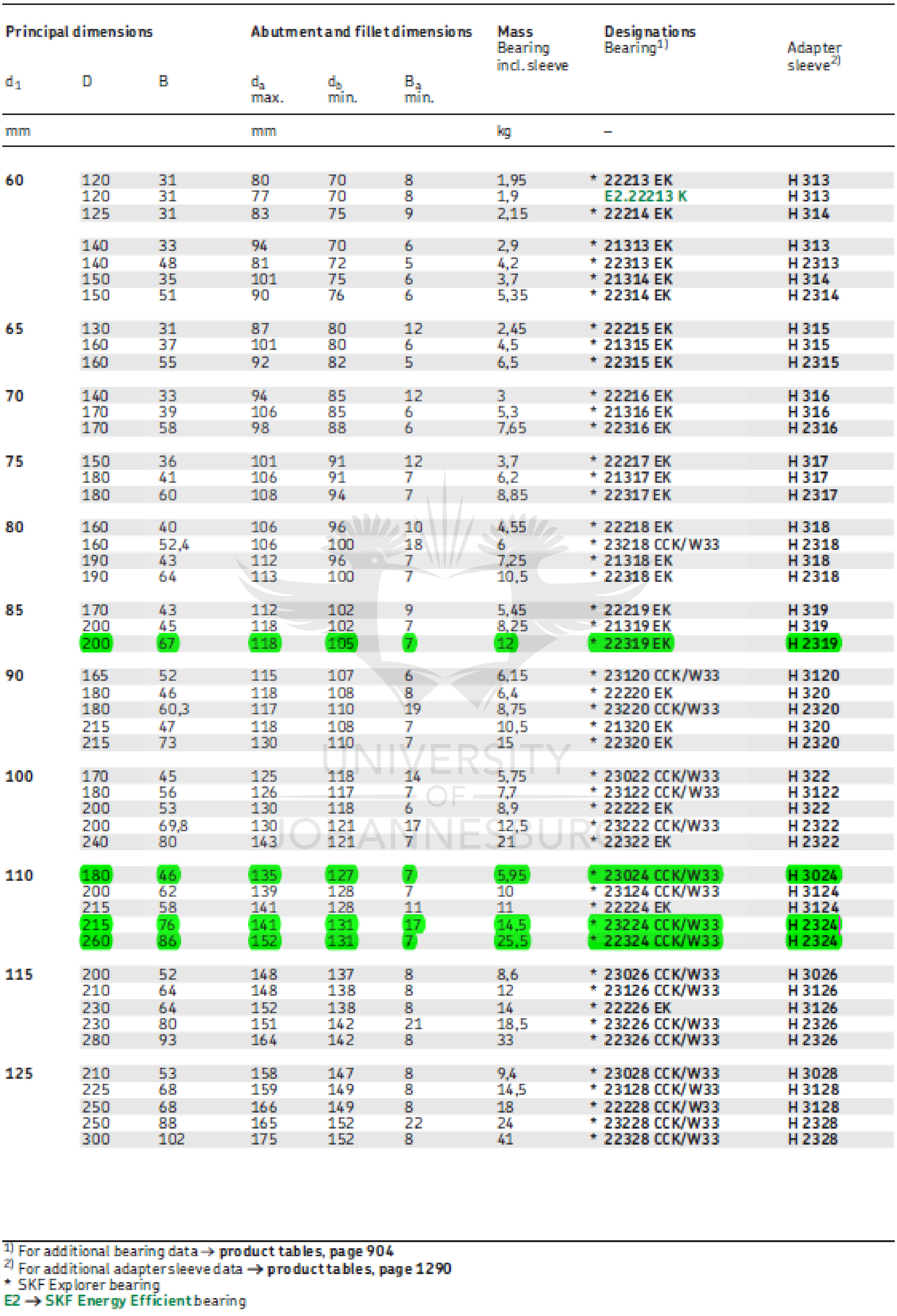
Housing tolerances and resultant fits

Deviations (housing bore diameter)
Theoretical interference (–)/clearance (+)
Probable interference (–)/clearance (+)

Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance Δ_{Dmp}		**Housing bore diameter deviations, resultant fits** Tolerance classes									
				K6Ⓔ		K7Ⓔ		M5Ⓔ		M6Ⓔ		M7Ⓔ	
over	incl.	high	low										
mm		µm		µm									
6	**10**	0	–8	–7	+2	–10	+5	–10	–4	–12	–3	–15	0
				–7	+10	–10	+13	–10	+4	–12	+5	–15	+8
				–5	+8	–7	+10	–8	+2	–10	+3	–12	+5
10	**18**	0	–8	–9	+2	–12	+6	–12	–4	–15	–4	–18	0
				–9	+10	–12	+14	–12	+4	–15	+4	–18	+8
				–7	+8	–9	+11	–10	+2	–13	+2	–15	+5
18	**30**	0	–9	–11	+2	–15	+6	–14	–4	–17	–4	–21	0
				–11	+11	–15	+15	–14	+4	–17	+5	–21	+9
				–8	+8	–12	+12	–12	+2	–14	+2	–18	+6
30	**50**	0	–11	–13	+3	–18	+7	–16	–5	–20	–4	–25	0
				–13	+14	–18	+18	–16	+6	–20	+7	–25	+11
				–10	+11	–14	+14	–13	+3	–17	+4	–21	+7
50	**80**	0	–13	–15	+4	–21	+9	–19	–6	–24	–5	–30	0
				–15	+17	–21	+22	–19	+7	–24	+8	–30	+13
				–11	+13	–16	+17	–16	+4	–20	+4	–25	+8
80	**120**	0	–15	–18	+4	–25	+10	–23	–8	–28	–6	–35	0
				–18	+19	–25	+25	–23	+7	–28	+9	–35	+15
				–13	+14	–20	+20	–19	+3	–23	+4	–30	+10
120	**150**	0	–18	–21	+4	–28	+12	–27	–9	–33	–8	–40	0
				–21	+22	–28	+30	–27	+9	–33	+10	–40	+18
				–15	+16	–21	+23	–22	+4	–27	+4	–33	+11
150	**180**	0	–25	–21	+4	–28	+12	–27	–9	–33	–8	–40	0
				–21	+29	–28	+37	–27	+16	–33	+17	–40	+25
				–14	+22	–20	+29	–21	+10	–26	+10	–32	+17
180	**250**	0	–30	–24	+5	–33	+13	–31	–11	–37	–8	–46	0
				–24	+35	–33	+43	–31	+19	–37	+22	–46	+30
				–16	+27	–23	+33	–25	+13	–29	+14	–36	+20
250	**315**	0	–35	–27	+5	–36	+16	–36	–13	–41	–9	–52	0
				–27	+40	–36	+51	–36	+22	–41	+26	–52	+35
				–18	+31	–24	+39	–28	+14	–32	+17	–40	+23
315	**400**	0	–40	–29	+7	–40	+17	–39	–14	–46	–10	–57	0
				–29	+47	–40	+57	–39	+26	–46	+30	–57	+40
				–18	+36	–27	+44	–31	+18	–35	+19	–44	+27
400	**500**	0	–45	–32	+8	–45	+18	–43	–16	–50	–10	–63	0
				–32	+53	–45	+63	–43	+29	–50	+35	–63	+45
				–20	+41	–30	+48	–34	+20	–38	+23	–48	+30
500	**630**	0	–50	–44	0	–70	0	–	–	–70	–26	–96	–26
				–44	+50	–70	+50	–	–	–70	+24	–96	+24
				–31	+37	–54	+34	–	–	–57	+11	–80	+8

Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Shaft and housing tolerances and fits* (→ **page 171**).

Appendix 37: Bearing calculation and selection tables.



Principal dimensions			Abutment and fillet dimensions			Mass	Designations	
dd_1	D	B	d_a max.	d_b min.	B_a min.	Bearing incl. sleeve	Bearing[1]	Adapter sleeve[2]
mm			mm			kg	–	
60	120	31	80	70	8	1,95	* 22213 EK	H 313
	120	31	77	70	8	1,9	E2.22213 K	H 313
	125	31	83	75	9	2,15	* 22214 EK	H 314
	140	33	94	70	6	2,9	* 21313 EK	H 313
	140	48	81	72	5	4,2	* 22313 EK	H 2313
	150	35	101	75	6	3,7	* 21314 EK	H 314
	150	51	90	76	6	5,35	* 22314 EK	H 2314
65	130	31	87	80	12	2,45	* 22215 EK	H 315
	160	37	101	80	6	4,5	* 21315 EK	H 315
	160	55	92	82	5	6,5	* 22315 EK	H 2315
70	140	33	94	85	12	3	* 22216 EK	H 316
	170	39	106	85	6	5,3	* 21316 EK	H 316
	170	58	98	88	6	7,65	* 22316 EK	H 2316
75	150	36	101	91	12	3,7	* 22217 EK	H 317
	180	41	106	91	7	6,2	* 21317 EK	H 317
	180	60	108	94	7	8,85	* 22317 EK	H 2317
80	160	40	106	96	10	4,55	* 22218 EK	H 318
	160	52,4	106	100	18	6	* 23218 CCK/W33	H 2318
	190	43	112	96	7	7,25	* 21318 EK	H 318
	190	64	113	100	7	10,5	* 22318 EK	H 2318
85	170	43	112	102	9	5,45	* 22219 EK	H 319
	200	45	118	102	7	8,25	* 21319 EK	H 319
	200	67	118	105	7	12	* 22319 EK	H 2319
90	165	52	115	107	6	6,15	* 23120 CCK/W33	H 3120
	180	46	118	108	8	6,4	* 22220 EK	H 320
	180	60,3	117	110	19	8,75	* 23220 CCK/W33	H 2320
	215	47	118	108	7	10,5	* 21320 EK	H 320
	215	73	130	110	7	15	* 22320 EK	H 2320
100	170	45	125	118	14	5,75	* 23022 CCK/W33	H 322
	180	56	126	117	7	7,7	* 23122 CCK/W33	H 3122
	200	53	130	118	6	8,9	* 22222 EK	H 322
	200	69,8	130	121	17	12,5	* 23222 CCK/W33	H 2322
	240	80	143	121	7	21	* 22322 EK	H 2322
110	180	46	135	127	7	5,95	* 23024 CCK/W33	H 3024
	200	62	139	128	7	10	* 23124 CCK/W33	H 3124
	215	58	141	128	11	11	* 22224 EK	H 3124
	215	76	141	131	17	14,5	* 23224 CCK/W33	H 2324
	260	86	152	131	7	25,5	* 22324 CCK/W33	H 2324
115	200	52	148	137	8	8,6	* 23026 CCK/W33	H 3026
	210	64	148	138	8	12	* 23126 CCK/W33	H 3126
	230	64	152	138	8	14	* 22226 EK	H 3126
	230	80	151	142	21	18,5	* 23226 CCK/W33	H 2326
	280	93	164	142	8	33	* 22326 CCK/W33	H 2326
125	210	53	158	147	8	9,4	* 23028 CCK/W33	H 3028
	225	68	159	149	8	14,5	* 23128 CCK/W33	H 3128
	250	68	166	149	8	18	* 22228 CCK/W33	H 3128
	250	88	165	152	22	24	* 23228 CCK/W33	H 2328
	300	102	175	152	8	41	* 22328 CCK/W33	H 2328

---

[1] For additional bearing data → **product tables, page 904**

[2] For additional adapter sleeve data → **product tables, page 1290**

* SKF Explorer bearing

E2 → SKF Energy Efficient bearing

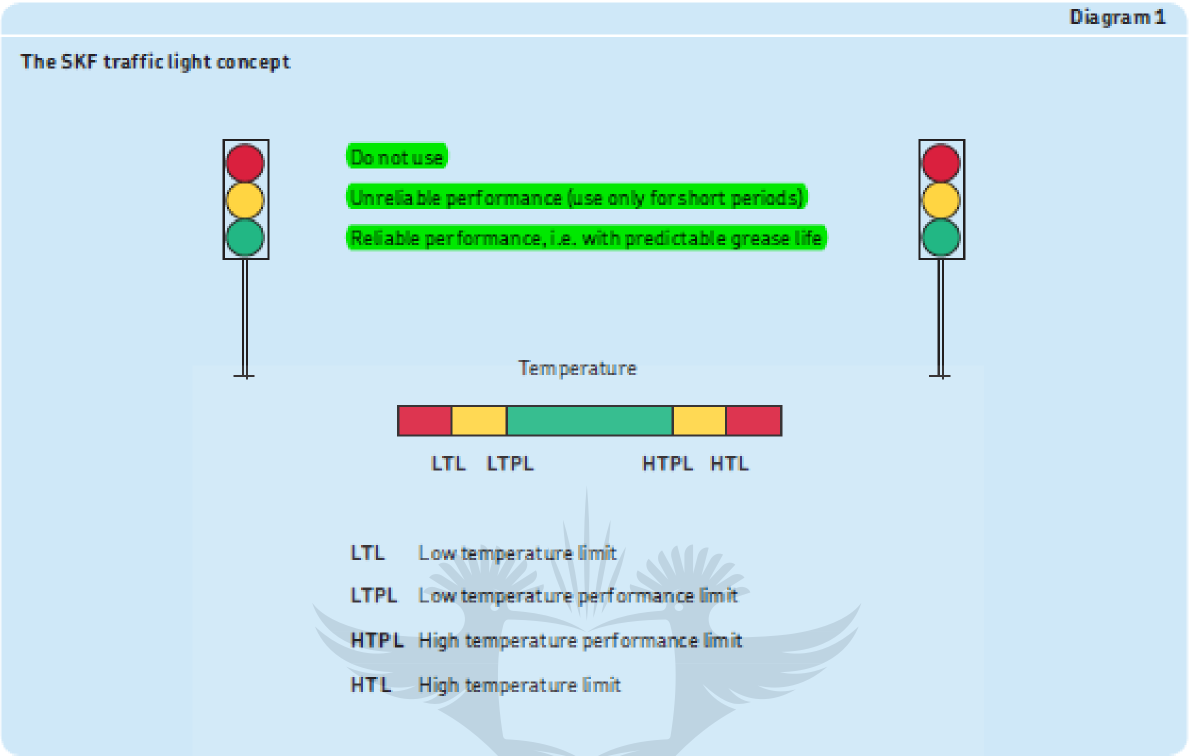
Appendix 38: Bearing grease selection

1 Bearing grease traffic light identification

Lubricating greases

Diagram 1

The SKF traffic light concept



An amber zone also exists for low temperatures. With decreasing temperature, the consistency (stiffness) of grease increases and the tendency of grease to bleed decreases. This ultimately leads to an insufficient supply of lubricant reaching the contact surfaces of the rolling elements and raceways. In **diagram 1**, this temperature limit is indicated by the low temperature performance limit (LTPL). Values for the low temperature performance limit are different for roller bearings than for ball bearings. Since ball bearings are easier to lubricate than roller bearings, the low temperature performance limit is less important for ball bearings. For roller bearings, however, serious damage can result when the bearings are operated continuously below this limit. Short periods in this zone, such as during a cold start, are not harmful because the heat caused by frictional heat brings the bearing temperature into the green zone.

Greases for sealed bearings

Sealed spherical roller bearings are filled as standard with SKF LGEP 2 grease. Bearings filled with SKF LGHB 2 grease can be supplied on request. Technical specification of both greases are listed in **table 1**.

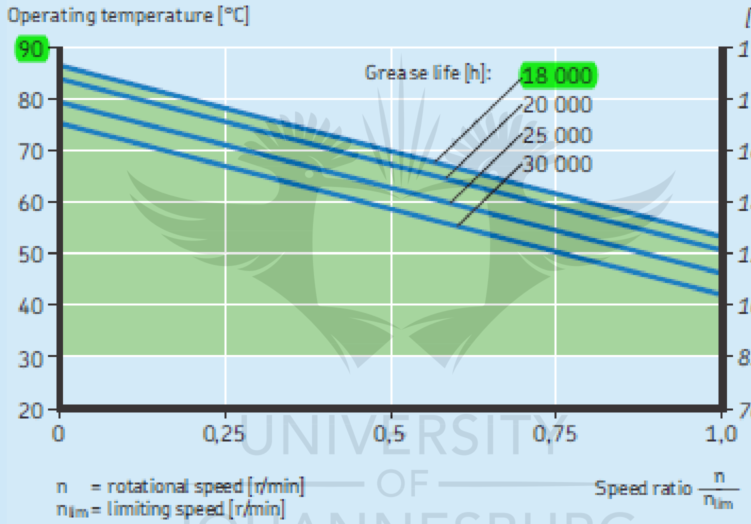
For additional information about greases, refer to *Lubrication* (→ **page 239**).

Relubrication of sealed bearings

Sealed spherical roller bearings are designed to operate relubrication-free. For bearings lubricated with the standard grease LGEP 2 (designation suffix VT143), the relubrication-free operating conditions can be identified using **diagram 1**. The diagram is valid for bearings in light to normal load applications ($P \leq 0,1\ C$) on a horizontal shaft. For other operating conditions, the grease life can be estimated by multiplying the relubrication

Diagram 1

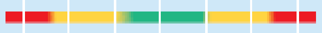
Relubrication-free operating conditions for sealed spherical roller bearings with standard SKF LGEP 2 grease (designation suffix VT143)



n = rotational speed [r/min]
n_{lim} = limiting speed [r/min]

Table 1

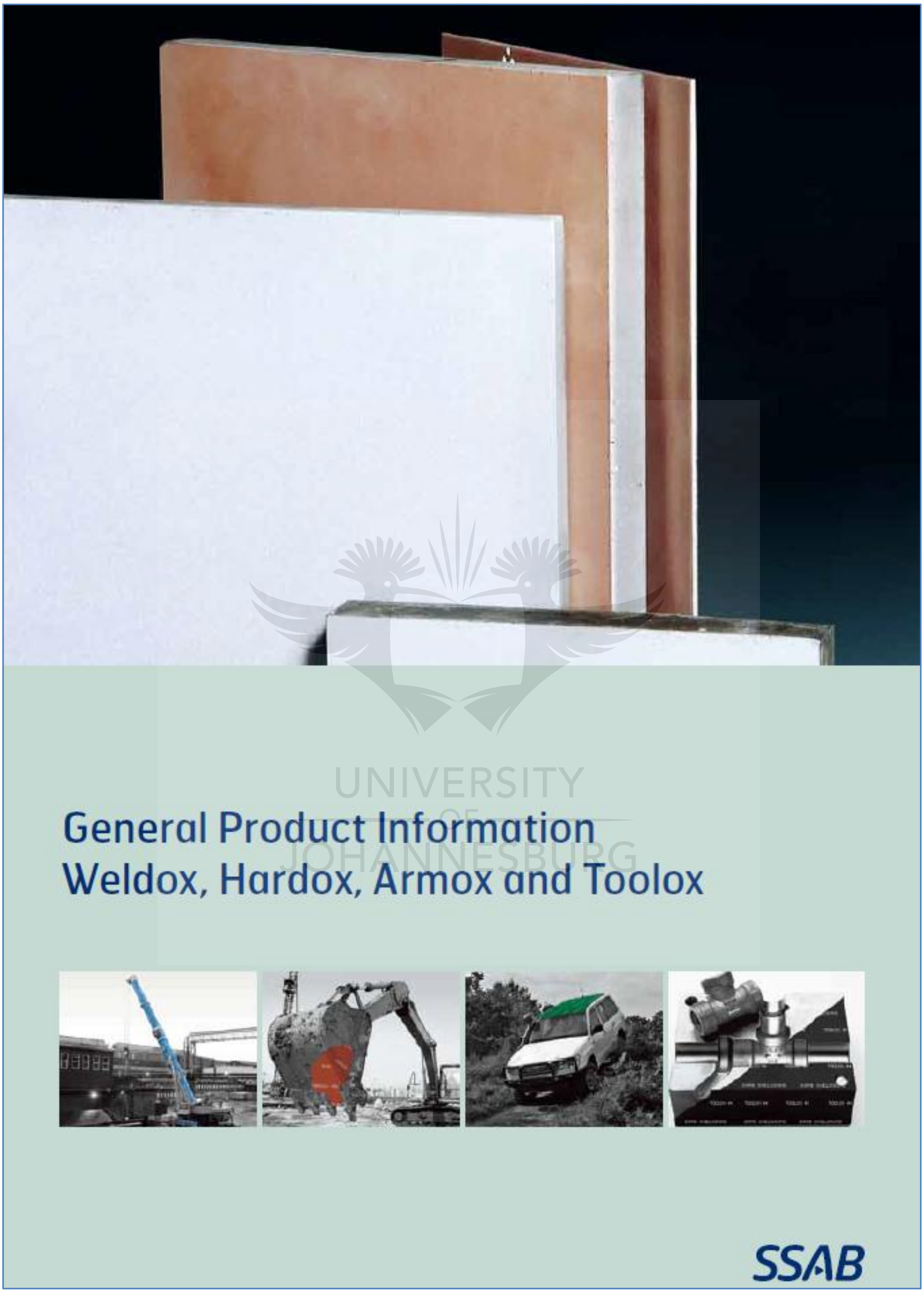
Technical specifications of SKF greases for sealed spherical roller bearings

Grease	**Designation suffix**	**Temperature range**[1] –50 0 50 100 150 200 250 °C –60 30 120 210 300 390 480 °F	**Thickener**	**Base oil type**	**NLGI consistency class**	**Base oil viscosity** [mm²/s] at 40 °C *(105 °F)*	at 100 °C *(210 °F)*
LGEP 2	VT143		Lithium soap	Mineral	2	200	16
LGHB 2	GEM9		Calcium complex sulphonate	Mineral	2	400	26,5

---

[1] Refer to the SKF traffic light concept → **page 244**

Appendix 39: Weldox, Hardox, Armox and Toolox uses.



General Product Information
Weldox, Hardox, Armox and Toolox

SSAB

Weldox high strength steel

SSAB produces high strength steels that conform to most international and national standards. Our high strength steels are marketed under the Weldox brand name.

Weldox has been developed to provide excellent weldability, combined with high strength and toughness. The ore-based metallurgy and advanced processing in the steel shop ensures very low contents of residual elements in the steel. Weldox high strength steels have excellent bendability and machinability properties. Due to the high strength of the steel, the end products can be strong but lightweight, which allows for substantial reductions in the cost of materials, welding and transport. Good flatness and fine surface quality are also distinguishing features of Weldox plate.

Weldox high strength steel is produced in thicknesses ranging from 4 to 160 mm, and with guaranteed yield strengths between 700 MPa and up to 1300 MPa. The flexible production system enables us to deliver plate with tailored properties to suit the customer's requirements. We can supply plate in thermomechanically rolled or quenched and tempered condition. In addition, most Weldox steels can be supplied with guaranteed impact toughness at temperatures down to –60°C.

Weldox high strength steels conform to EN 10025-6. However, Weldox 1100 and Weldox 1300 do not have any standardized equivalents.

Further information concerning the properties of the plate and the options in the standard that are employed are given in the relevant data sheets.

Z-plate

All high strength steels with yield strengths of up to 960 MPa can be supplied with guaranteed properties in the through thickness direction, often called Z-plate. Restrictions may apply.

WELDOX® HIGH STRENGTH STEEL

HARDOX®
WEAR PLATE

HARDOX ON SITE
Roadbuilding

A Part of Your Success

SSAB

Wear: The inside story

Your choice of wear plate has consequences for your business. Hardox® maximizes the wear performance of your equipment and machines, reducing workshop lead times and increasing the overall productivity of your operations.

Thanks to its consistent properties, Hardox's performance remains invariable across its lifetime. That also makes its service life very predictable, allowing you to rationalize your repair schedule.

With its combination of high hardness, high strength and good toughness, Hardox can be used in a variety of applications, from forest harvesting and ground preparation to asphalting work.

What's the secret of Hardox's top performance? The production processes include the state-of-the-art metallurgical cleaning of steel and a unique hardening process, resulting in wear plates with outstanding hardness, toughness and workshop friendliness.

Expertise at your service

In addition to plate, SSAB Plate provides you with expertise. We share our knowledge with you through our Technical Managers, Conceptual Design Group™ and Wear Technology Group™.

The Conceptual Design Group consists of experts that can help optimize your product from a design perspective.

The Wear Technology Group is committed to developing the technical knowledge of wear. We offer you access to Ph.D's and experts with decades of experience in solving wear challenges. You can get applied support and information on wear-critical components.

Information about wear

Wear comes in different forms and each has a different impact on the service life of your application.

The most common wear types are sliding wear and impact wear. Abrasive particles trapped in a narrow gap between two rigid surfaces causing squeezing wear is also a common wear type.

Each variety of rock is composed of a unique set of minerals and these also contribute to the specific type of abrasive wear damage.

WearCalc software, available from our Technical Managers, describes and calculates the relative differences between materials. It allows you to predict relative wear life and compare different wear solutions.

Whatever your application and wear situation, Hardox is your ticket to outstanding wear performance.

SLIDING

In sliding wear, abrasive bodies such as aggregate rocks are free to slide and roll. By selecting a harder Hardox grade, service life can be improved considerably.

IMPACT

In impact wear, the aggregate rocks hit the surface of the wear component at various angles. A harder grade of Hardox will deliver a longer service life here as well.

SQUEEZING

With squeezing wear, the improvement in service life of wear components is more difficult to quantify. However, an increased Hardox plate hardness often improves the service life significantly.

Hardox – Complete product program

You'll always find a Hardox plate to fit your wear challenge. With a wide range of hardness grades, thicknesses and widths to choose from, you'll always be able to maximize your application's performance.

Hardox 400 and 450 are versatile wear plates with high toughness, good bendability and excellent weldability.

Hardox 500 is a tough, bendable and weldable abrasion resistant plate used in applications requiring high resistance.

Hardox 550, with a hardness of 550 Brinell and a toughness equal to Hardox 500, is designed to increase wear life but not at the expense of reduced resistance to cracking.

Hardox 600 has a hardness of 600 Brinell but can still be cut and welded—an excellent plate for high performance applications.

Hardox HiTuf is a wear resistant plate with extra high toughness intended for heavy section wear parts requiring extraordinary wear and crack resistance.

Hardox Extreme is intended for applications requiring extremely high abrasion resistance. It can replace costly wear products like hard-faced overlay plates and high chrome white iron. Despite its hardness, it can be welded, cut, milled and drilled using standard workshop practices.

Breaking the ground for roadbuilding makes tough demands on machines and materials...

1. Hydralic hammer
2. Excavator bucket
3. Bucket
4. Excavator bucket
5. Crusher
6. Bucket
7. Bucket
8. Asphalt Plant
9. Tipper (asphalt)
10. Tipper (gravel)
11. Bulldozer
12. Motor Grader
13. Asphalt Roller
14. Asphalt Paver
15. Tipper (asphalt)
16. Asphalt milling machine

HYDRAULIC HAMMER
Hardox 400/450 in the housing of the hydraulic hammer exposed to wear from rocks.

EXCAVATOR BUCKET
Hardox 450/500/550 in the excavator bucket. Hardox may be used in the entire bucket or just in the cutting edge and sides depending on the wear situation.

BUCKET
Hardox 450/500/550 in the entire bucket or cutting edge and sides depending on the wear situation.

CHUTE
In the chute that leads into the crusher Hardox 500/550/600 is the obvious choice in the tough environment.

CRUSHER
Hardox 500/550 used as liner plates in the interior in areas with high wear.

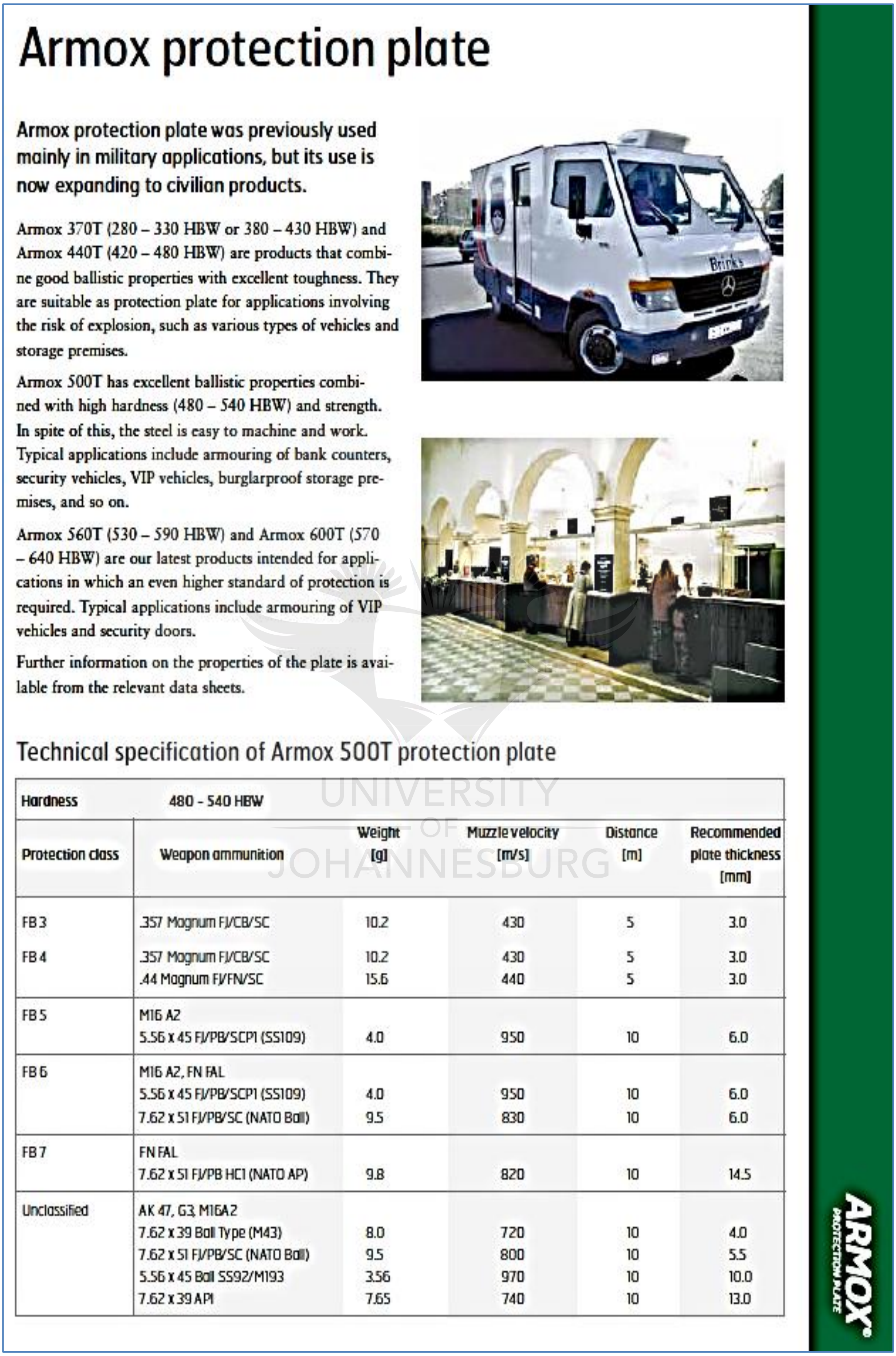
ASPHALT PLANT
Hardox 450/500/550 as liner plates and in gravel chutes. Hardox 450/500 in the wings of the drum. Hardox 450/500 in the transport container from the drum to the truck delivery shaft.

TIPPER BODY
Hardox 450/500 in the body of the tippers and dumpers. Hardox is an outstanding wear fighter in these applications. Hardox also has great impact resistance, making it an excellent design solution.

BULLDOZER
Hardox 450/500/550 in the blade of the bulldozer.

MOTOR GRADER
Hardox 450/500/550 in the blades of the motor grader.

ASPHALT PAVER
Hardox 450/500/550 in the screed plates.



Armox protection plate

Armox protection plate was previously used mainly in military applications, but its use is now expanding to civilian products.

Armox 370T (280 – 330 HBW or 380 – 430 HBW) and Armox 440T (420 – 480 HBW) are products that combine good ballistic properties with excellent toughness. They are suitable as protection plate for applications involving the risk of explosion, such as various types of vehicles and storage premises.

Armox 500T has excellent ballistic properties combined with high hardness (480 – 540 HBW) and strength. In spite of this, the steel is easy to machine and work. Typical applications include armouring of bank counters, security vehicles, VIP vehicles, burglarproof storage premises, and so on.

Armox 560T (530 – 590 HBW) and Armox 600T (570 – 640 HBW) are our latest products intended for applications in which an even higher standard of protection is required. Typical applications include armouring of VIP vehicles and security doors.

Further information on the properties of the plate is available from the relevant data sheets.

Technical specification of Armox 500T protection plate

Hardness	480 - 540 HBW				
Protection class	**Weapon ammunition**	**Weight [g]**	**Muzzle velocity [m/s]**	**Distance [m]**	**Recommended plate thickness [mm]**
FB 3	.357 Magnum FJ/CB/SC	10.2	430	5	3.0
FB 4	.357 Magnum FJ/CB/SC	10.2	430	5	3.0
	.44 Magnum FJ/FN/SC	15.6	440	5	3.0
FB 5	M16 A2				
	5.56 x 45 FJ/PB/SCP1 (SS109)	4.0	950	10	6.0
FB 6	M16 A2, FN FAL				
	5.56 x 45 FJ/PB/SCP1 (SS109)	4.0	950	10	6.0
	7.62 x 51 FJ/PB/SC (NATO Ball)	9.5	830	10	6.0
FB 7	FN FAL				
	7.62 x 51 FJ/PB HC1 (NATO AP)	9.8	820	10	14.5
Unclassified	AK 47, G3, M16A2				
	7.62 x 39 Ball Type (M43)	8.0	720	10	4.0
	7.62 x 51 FJ/PB/SC (NATO Ball)	9.5	800	10	5.5
	5.56 x 45 Ball SS92/M193	3.56	970	10	10.0
	7.62 x 39 API	7.65	740	10	13.0

Toolox prehardened tool steel

Toolox prehardened tool steels represent a unique and special concept for producing forming tools and machine components. Toolox is a modern prehardened tool steel based on many years of experience at SSAB in the development and production of Hardox wear plates and Weldox high strength steels.

The fundamental idea behind Toolox is to deliver a steel that is hardened and ready for use, with tested and guaranteed physical properties. Due to the high metallurgical purity, the freedom from slag achieved corresponds to ESR remelted material. Every plate is uniquely produced, and each individual plate is tested for hardness, toughness and homogeneity.

Toolox is directly machinable and requires no further hardening and remachining. The fact that the material is fully prehardened means that it has low residual stresses and guaranteed stable properties. The hardness of the steel creates unique conditions for precision and surface finish.

Toolox allows for a new and modern toolmaking process. The greatest benefits are shorter production times, and more uniform and stable material properties. In addition, the steel enables a number of risky processes to be eliminated, such as hardening, with the associated risk of damage. Due to its pure metallurgy and hardening, Toolox has unique toughness and fatigue properties, which greatly increases the lifespan of the tool or machine component.

Excellent properties for processes such as etching, polishing and coating are also a distinctive feature of Toolox. As a result, Toolox offers very flexible opportunities in applications for which it is used. Typical applications include plastic moulds, edge-pressing tools, wear strips, plate pressing tools, etc.

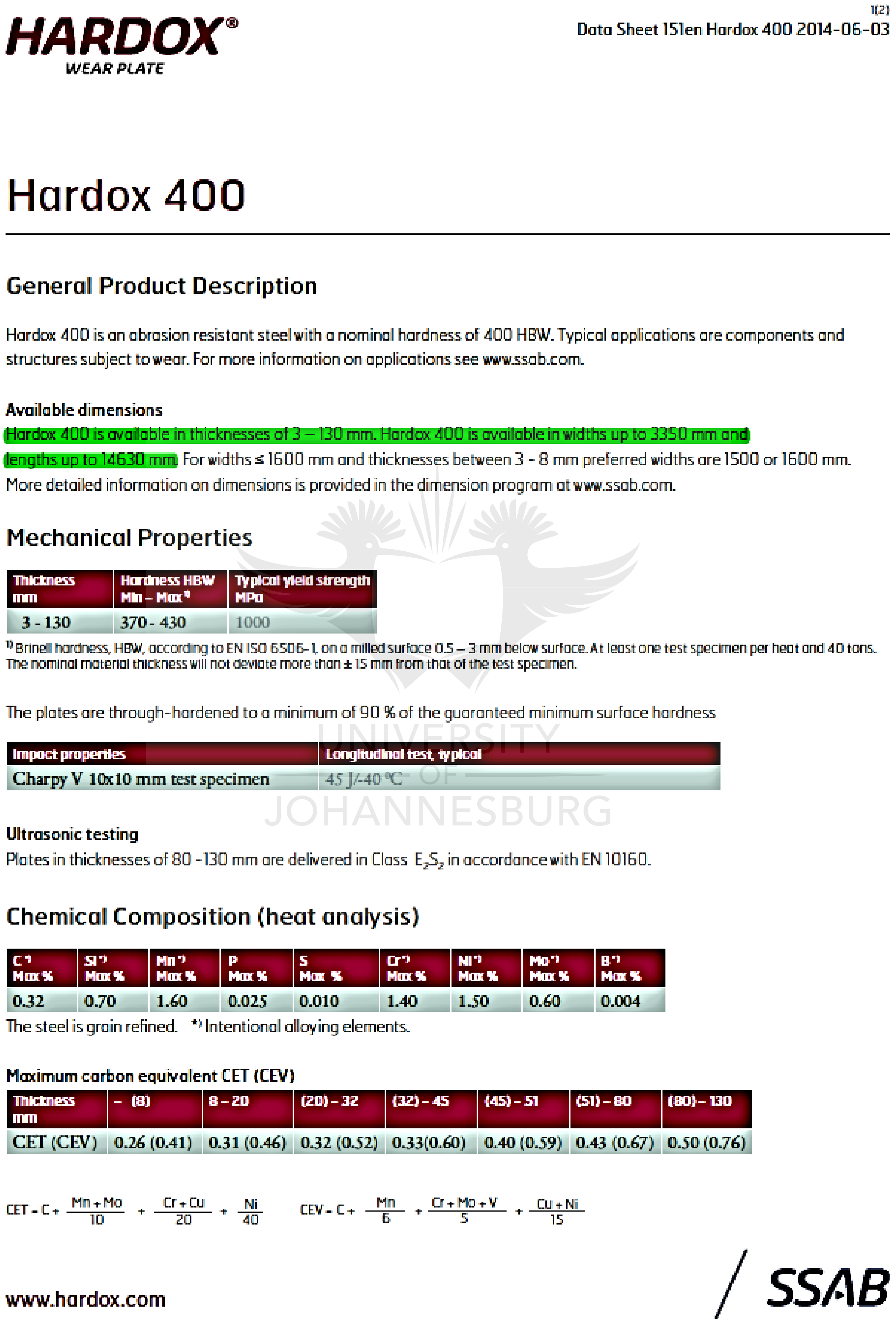
Toolox is available in two hardnesses: Toolox 33 with a hardness of 300 HBW and Toolox 44 with a hardness of 45 HRC – the world's hardest fully prehardened tool steel with the same hardness all the way through.

As an option, Toolox can also be nitrided and surfaced with an even harder coating for surface hardnesses between 60 and 65 HRC. This increases further the intervals between tool service, which improves the overall economy.

For further information on the range of sizes, tolerances, surface finishes, testing and other properties unique to Toolox, visit www.toolox.com

TOOLOX®
PREHARDENED TOOL STEEL

Appendix 40: Hardox specifications.



HARDOX®
WEAR PLATE

1(2)
Data Sheet 151en Hardox 400 2014-06-03

Hardox 400

General Product Description

Hardox 400 is an abrasion resistant steel with a nominal hardness of 400 HBW. Typical applications are components and structures subject to wear. For more information on applications see www.ssab.com.

Available dimensions
Hardox 400 is available in thicknesses of 3 – 130 mm. Hardox 400 is available in widths up to 3350 mm and lengths up to 14630 mm. For widths ≤ 1600 mm and thicknesses between 3 - 8 mm preferred widths are 1500 or 1600 mm. More detailed information on dimensions is provided in the dimension program at www.ssab.com.

Mechanical Properties

Thickness mm	Hardness HBW Min – Max [1]	Typical yield strength MPa
3 - 130	370 - 430	1000

[1] Brinell hardness, HBW, according to EN ISO 6506-1, on a milled surface 0.5 – 3 mm below surface. At least one test specimen per heat and 40 tons. The nominal material thickness will not deviate more than ± 15 mm from that of the test specimen.

The plates are through-hardened to a minimum of 90 % of the guaranteed minimum surface hardness

Impact properties	Longitudinal test, typical
Charpy V 10x10 mm test specimen	45 J/-40 °C

Ultrasonic testing
Plates in thicknesses of 80 -130 mm are delivered in Class E_2S_2 in accordance with EN 10160.

Chemical Composition (heat analysis)

C *) Max %	Si *) Max %	Mn *) Max %	P Max %	S Max %	Cr *) Max %	Ni *) Max %	Mo *) Max %	B *) Max %
0.32	0.70	1.60	0.025	0.010	1.40	1.50	0.60	0.004

The steel is grain refined. *) Intentional alloying elements.

Maximum carbon equivalent CET (CEV)

Thickness mm	- (8)	8 – 20	(20) – 32	(32) – 45	(45) – 51	(51) – 80	(80) – 130
CET (CEV)	0.26 (0.41)	0.31 (0.46)	0.32 (0.52)	0.33(0.60)	0.40 (0.59)	0.43 (0.67)	0.50 (0.76)

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40} \qquad CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

www.hardox.com

SSAB

HARDOX®
WEAR PLATE

Hardox 500

General Product Description

Hardox 500 is an abrasion resistant steel with a nominal hardness of 500 HBW. Typical applications are components and structures subject to wear. For more information on applications see www.ssab.com

Available dimensions

Hardox 500 is available in thicknesses of 4.0 – 80 mm. Hardox 500 Tuf is available in thicknesses of 4 – 65 mm. Both grades are available in widths up to 3350 mm and lengths up to 14630 mm. More detailed information on dimensions is provided in the dimension program at www.ssab.com.

Mechanical Properties

Thickness mm	Hardness HBW min – max[1]	Typical yield strength MPa, not guaranteed
4 - 32	470 – 530	1250
(32) - 80	450 - 540	1250

[1] Brinell hardness, HBW, according to EN ISO 6506-1, on a milled surface 0.5 – 3 mm below surface. At least one test specimen per heat and 40 tons. The nominal material thickness will not deviate more than ±15 mm from that of the test specimen.

The plates are through-hardened to a minimum of 90 % of the guaranteed minimum surface hardness.

Impact properties	Hardox 500	Hardox 500 Tuf Transverse test, guaranteed	Longitudinal test, typical
Impact energy (J) for transverse tests Charpy V 10x10 mm test specimen [2]	–	27 J/0 °C	37 J/-40 °C

[2] For thicknesses between 6 - 11.9 mm, subsize Charpy V-specimens are used. The specified minimum value is then proportional to the cross-sectional area of the test specimen, compared to a full-size specimen (10 x 10 mm). Impact testing according to ISO EN 148 per heat and thickness group. Average of three tests. Single value minimum 70% of specified average. Impact test is performed from 6 mm.

Ultrasonic testing

Plates in thickness of 80 mm are delivered in Class E_2S_2 in accordance with EN 10 160, other thicknesses are delivered in Class E_1S_1.

Chemical Composition (heat analysis)

C *) Max %	Si *) Max %	Mn *) Max %	P Max %	S Max %	Cr *) Max %	Ni *) Max %	Mo *) Max %	B *) Max %
0.30	0.70	1.60	0.020	0.010	1.50	1.5	0.60	0.005

The steel is grain refined. *) Intentional alloying elements.

Maximum carbon equivalent CET (CEV)

Thickness mm	– (5)	5 - (10)	10 - (20)	20 - (40)	40 - 80
CET (CEV)	0.34 (0.49)	0.36 (0.52)	0.43 (0.64)	0.45 (0.66)	0.47 (0.75)

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40} \qquad CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

HARDOX®
WEAR PLATE

Hardox 600

General Product Description

Hardox 600 is an abrasion resistant steel with a nominal hardness of 600 HBW. Typical applications are components with abrasions resistance . For more information on applications see www.ssab.com

Available dimensions
Hardox 600 is available in thicknesses of 8 – 51 mm. Hardox 600 is available in widths up to 2000 mm and lengths up to 14630 mm. Preferred dimensions are 2000 x 4000 mm, other dimensons on request. More detailed information on dimensions is provided in the dimension program at www.ssab.com.

Mechanical Properties

Thickness mm	Hardness HBW min – max[1]
8 - 51	570 - 640

[1] Brinell hardness, HBW, according to EN ISO 6506-1, on a milled surface 0.5 – 3 mm below surface. At least one test specimen per heat and 40 tons. The nominal material thickness will not deviate more than ±15 mm from that of the test specimen.

The plates are through-hardened to a minimum of 90 % of the guaranteed minimum surface hardness

Chemical Composition (heat analysis)

C *) Max %	Si *) Max %	Mn *) Max %	P Max %	S Max %	Cr *) Max %	Ni *) Max %	Mo *) Max %	B *) Max %
0.47	0.70	1.00	0.015	0.010	1.20	2.50	0.70	0.005

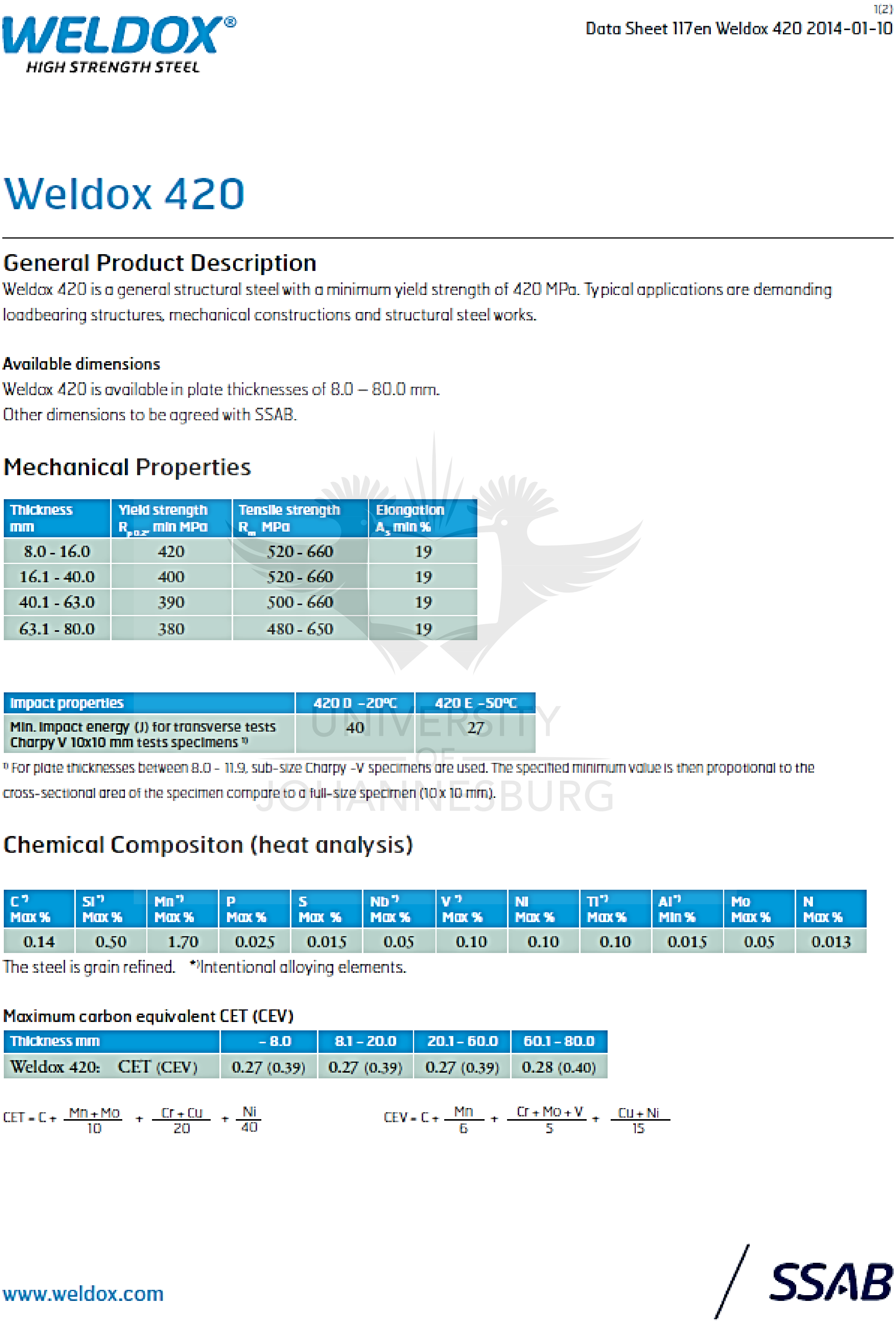
The steel is grain refined. *) Intentional alloying elements.

Maximum carbon equivalent CET (CEV)

Thickness mm	8 – (25)	25 – 51
CET (CEV)	0.58 (0.76)	0.61 (0.87)

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40} \qquad CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

Appendix 41: Weldox Specifications.



WELDOX®
HIGH STRENGTH STEEL

Data Sheet 117en Weldox 420 2014-01-10

Weldox 420

General Product Description

Weldox 420 is a general structural steel with a minimum yield strength of 420 MPa. Typical applications are demanding loadbearing structures, mechanical constructions and structural steel works.

Available dimensions

Weldox 420 is available in plate thicknesses of 8.0 – 80.0 mm.
Other dimensions to be agreed with SSAB.

Mechanical Properties

Thickness mm	Yield strength $R_{p0.2}$ min MPa	Tensile strength R_m MPa	Elongation A_5 min %
8.0 - 16.0	420	520 - 660	19
16.1 - 40.0	400	520 - 660	19
40.1 - 63.0	390	500 - 660	19
63.1 - 80.0	380	480 - 650	19

Impact properties	420 D −20°C	420 E −50°C
Min. impact energy (J) for transverse tests Charpy V 10x10 mm tests specimens [1]	40	27

[1] For plate thicknesses between 8.0 - 11.9, sub-size Charpy -V specimens are used. The specified minimum value is then propotional to the cross-sectional area of the specimen compare to a full-size specimen (10 x 10 mm).

Chemical Compositon (heat analysis)

C *) Max %	Si *) Max %	Mn *) Max %	P Max %	S Max %	Nb *) Max %	V *) Max %	Ni Max %	Ti *) Max %	Al *) Min %	Mo Max %	N Max %
0.14	0.50	1.70	0.025	0.015	0.05	0.10	0.10	0.10	0.015	0.05	0.013

The steel is grain refined. *)Intentional alloying elements.

Maximum carbon equivalent CET (CEV)

Thickness mm	- 8.0	8.1 – 20.0	20.1 – 60.0	60.1 – 80.0
Weldox 420: CET (CEV)	0.27 (0.39)	0.27 (0.39)	0.27 (0.39)	0.28 (0.40)

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40} \qquad CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

www.weldox.com

SSAB

WELDOX®
HIGH STRENGTH STEEL

Weldox 500

General Product Description

Weldox 500 is a general structural steel with a minimum yield strength of 500 MPa. Typical applications are demanding loadbearing structures, mechanical constructions and strutural steelworks.

Available dimensions

Weldox 500 is available in plate thicknesses of 8.0 – 80.0 mm.
Other dimensions to be agreed with SSAB.

Mechanical Properties

Thickness mm	Yield strength $R_{p0.2}$, min MPa	Tensile strength R_m MPa	Elongation A_5 min %
8.0 - 16.0	500	570 - 720	16
16.1 - 40.0	480	570 - 720	16
40.1 - 80.0	460	550 - 720	16

Impact properties	500 D -20°C	500 E -40°C
Min. impact energy (J) for transverse tests Charpy V 10x10 mm tests specimens 1)	40	40

1) For plate thicknesses between 8.0 - 11.9 sub-size Charpy - V specimens are used. The specified minimum value is then propotional to the cross-sectional area of the specimen compare to a full-size specimen (10 x 10 mm).

Chemical Compositon (heat analysis)

C *) Max %	Si *) Max %	Mn *) Max %	P Max %	S Max %	Nb *) Max %	V *) Max %	Ni Max %	Ti *) Max %	Al *) Min %	Mo Max %	N Max %
0.17	0.55	1.70	0.020	0.015	0.050	0.12	0.10	0.10	0.015	0.05	0.013

The steel is grain refined. *)Intentional alloying elements.

Maximum carbon equivalent CET (CEV)

Thickness mm	- 8.0	8.1 - 20.0	20.1 - 60.0	60.1 - 80.0
Weldox 500: CET (CEV)	0.27 (0.39)	0.27 (0.39)	0.30 (0.42)	0.32 (0.44)

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40} \qquad CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

Weldox 700

General Product Description

Weldox 700 is a general structural steel with a minimum yield strength of 650 - 700 MPa depending on thickness. Weldox 700 meets the requirements of EN 10025 for the corresponding grades. Typical applications are demanding loadbearing structures.

Available dimensions

Weldox 700 E is available in plate thicknesses of 4 – 160 mm and Weldox 700 F is available in plate thicknesses of 4 - 130 mm. Both grades are available in widths up to 3350 mm and lengths up to 14630 mm. For thicknesses over 100 mm preferred width is 1650 mm with untrimmed edge.

More detailed information on dimensions is provided in the dimension program at www.ssab.com.

Mechanical Properties

Thickness mm	Yield strength [1] $R_{p0.2}$ min MPa	Tensile strength [1] R_m MPa	Elongation A_5 min %	Typical hardness HBW
4 - 53	700	780 - 930	14	260 - 310
(53) - 100	650	780 - 930	14	260 - 310
(100) - 160	650	710 - 900	14	240 - 290

[1] For transverse test pieces according to EN 10025.

Impact properties	E –40°C	F –60°C
Min. impact energy (J) for transverse tests Charpy V 10x10 mm tests specimens [2]	69 J	27 J
Meet the requirements for	S 690 QL	S 690 QL1

[2] Unless otherwise agreed, transverse impact testing according to EN 10025-6 option 30 will apply. For thicknesses between 6 - 11.9 mm, subsize Charpy V-specimens are used. The specified minimum value is then propotional to the cross-sectional area of the specimen compare to a fullsize specimen (10 x 10 mm).

Chemical Compositon (heat analysis)

C [*] Max %	Si [*] Max %	Mn [*] Max %	P Max %	S Max %	Cr [*] Max %	Cu [*] Max %	Ni [*] Max %	Mo [*] Max %	B [*] Max %
0.20	0.60	1.60	0.020	0.010	0.70	0.30	2.0	0.70	0.005

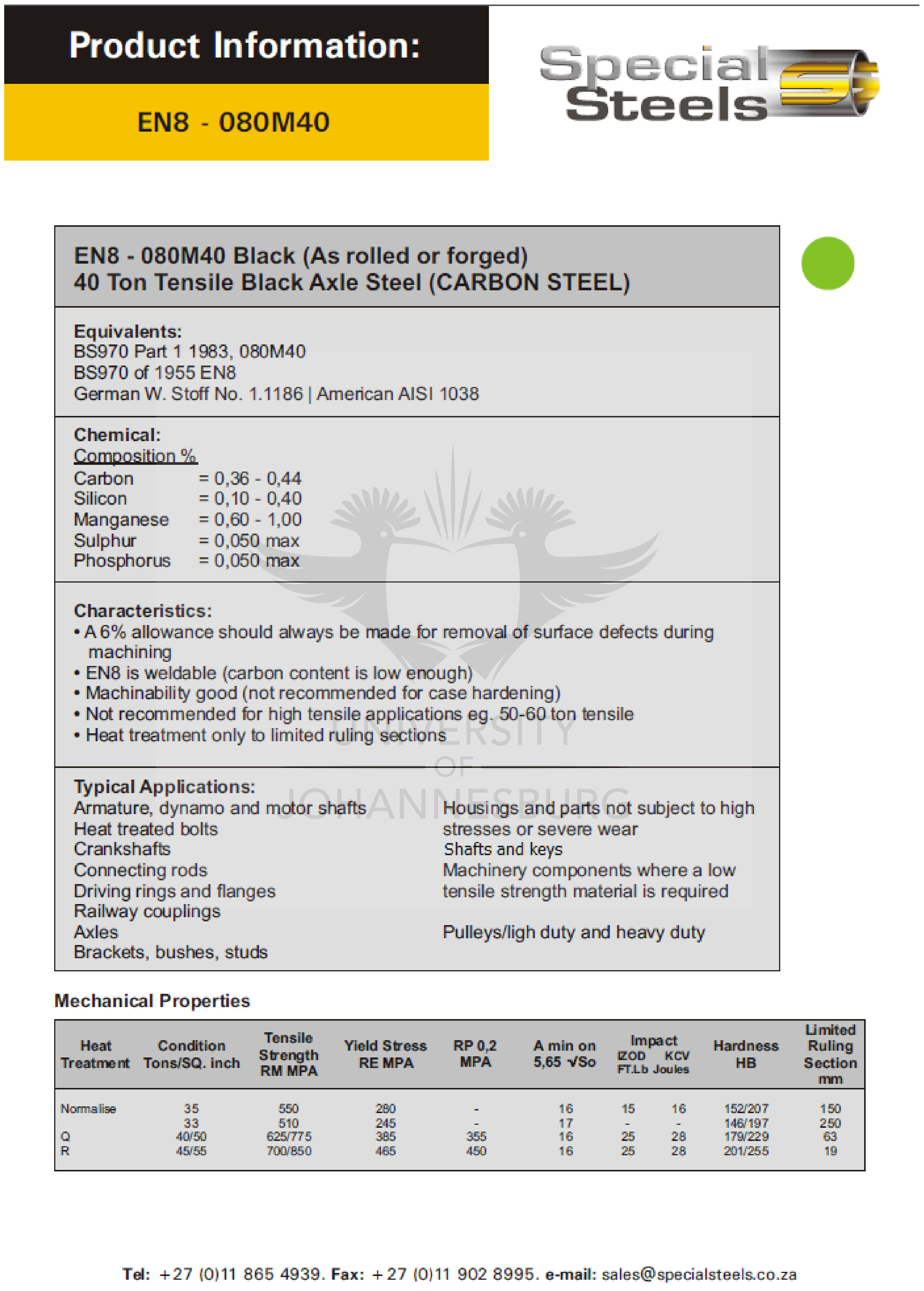
The steel is grain refined. [*] Intentional alloying elements.

Maximum carbon equivalent CET (CEV)

Thickness mm	– 5	(5)- (10)	10 – (20)	20 – (40)	40 – (80)	80 – (100)	100 – 160
Weldox 700E: CET (CEV)	0.34 (0.48)	0.31 (0.48)	0.31 (0.48)	0.36 (0.52)	0.39 (0.58)	0.39 (0.58)	0.41 (0.67)
Weldox 700F: CET (CEV)	0.38 (0.57)	0.38 (0.57)	0.38 (0.57)	0.38 (0.57)	0.39 (0.58)	0.39 (0.58)	0.41 (0.67)

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40} \qquad CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

Appendix 42: Carbon Steels.



Product Information:

EN8 - 080M40

Special Steels

EN8 - 080M40 Black (As rolled or forged)
40 Ton Tensile Black Axle Steel (CARBON STEEL)

Equivalents:
BS970 Part 1 1983, 080M40
BS970 of 1955 EN8
German W. Stoff No. 1.1186 | American AISI 1038

Chemical:
Composition %
Carbon = 0,36 - 0,44
Silicon = 0,10 - 0,40
Manganese = 0,60 - 1,00
Sulphur = 0,050 max
Phosphorus = 0,050 max

Characteristics:
- A 6% allowance should always be made for removal of surface defects during machining
- EN8 is weldable (carbon content is low enough)
- Machinability good (not recommended for case hardening)
- Not recommended for high tensile applications eg. 50-60 ton tensile
- Heat treatment only to limited ruling sections

Typical Applications:
Armature, dynamo and motor shafts
Heat treated bolts
Crankshafts
Connecting rods
Driving rings and flanges
Railway couplings
Axles
Brackets, bushes, studs
Housings and parts not subject to high stresses or severe wear
Shafts and keys
Machinery components where a low tensile strength material is required
Pulleys/ligh duty and heavy duty

Mechanical Properties

Heat Treatment	Condition Tons/SQ. inch	Tensile Strength RM MPA	Yield Stress RE MPA	RP 0,2 MPA	A min on 5,65 √So	Impact IZOD FT.Lb	Impact KCV Joules	Hardness HB	Limited Ruling Section mm
Normalise	35	550	280	-	16	15	16	152/207	150
	33	510	245	-	17	-	-	146/197	250
Q	40/50	625/775	385	355	16	25	28	179/229	63
R	45/55	700/850	465	450	16	25	28	201/255	19

Tel: +27 (0)11 865 4939. **Fax:** +27 (0)11 902 8995. **e-mail:** sales@specialsteels.co.za

Product Information:

EN9 - 070M55

EN9 - 070M55 Black (As rolled or forged) (CARBON STEEL)

Equivalents:
BS970 Part 1 1983, 070M55
BS970 of 1955 EN9
German W. Stoff No. 1.1203 | American AISI 1055

Chemical:
Composition %

Carbon = 0,50 - 0,60
Silicon = 0,10 - 0,40
Manganese = 0,50 - 0,90
Sulphur = 0,050 max
Phosphorus = 0,050 max

Characteristics:

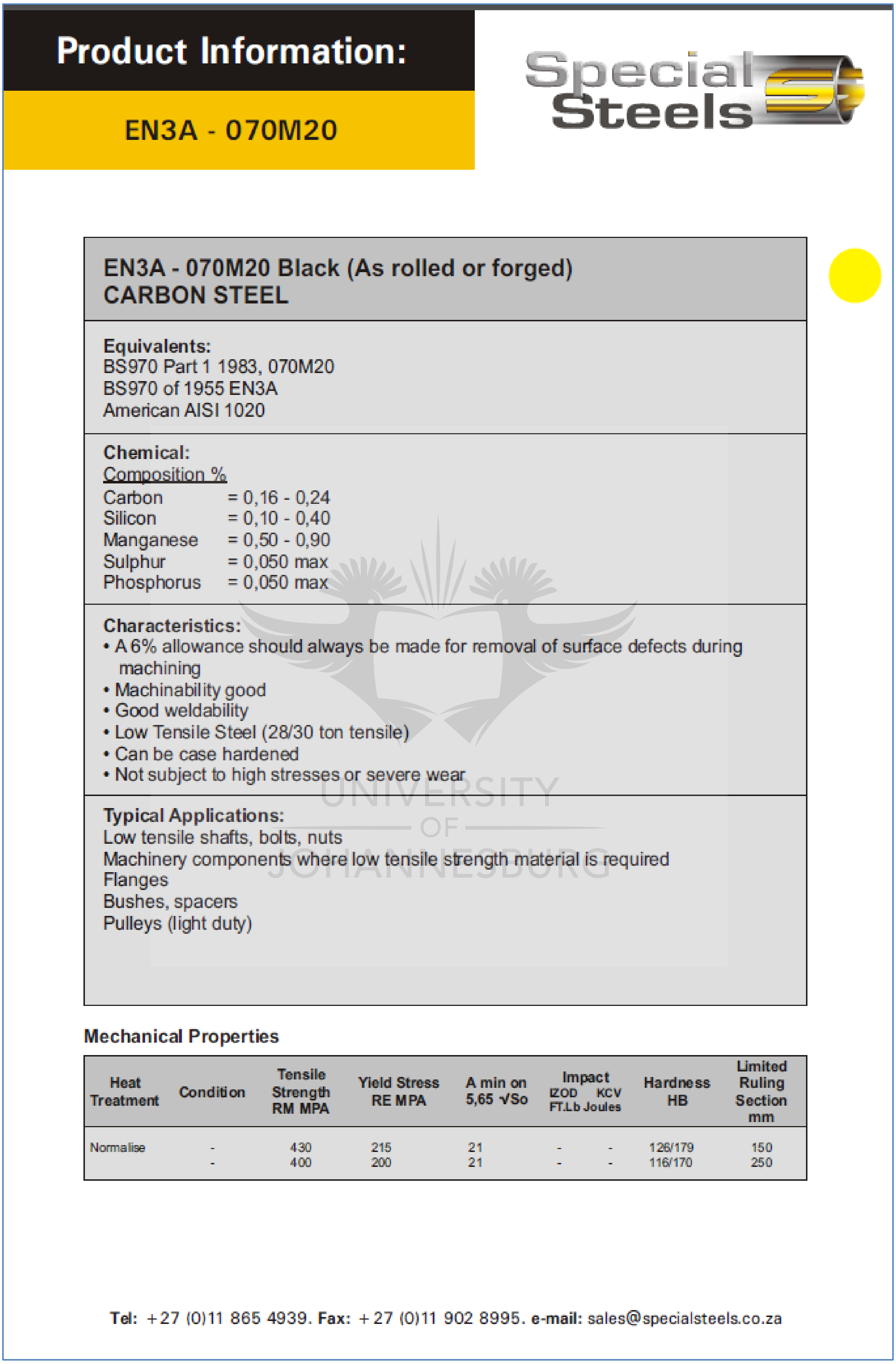
- A 6% allowance should always be made for removal of surface defects during machining
- EN9 is weldable (but not recommended)
- Machinability good
- Ideal for 45 ton tensile applications
- Not recommended for carburising
- Heat treatment only to limited ruling sections

Typical Applications:

Sprockets
Cylinders
Cams
Crankshafts
Keys
Small Gears
Machine Tools
Grinding Balls for Ball Mills
Ball Race Rings
Bolts, Nuts
Pulleys

Mechanical Properties

Heat Treatment	Condition Tons/SQ. inch	Tensile Strength RM MPA	Yield Stress RE MPA	RP 0,2 MPA	A min on 5,65 √So	Impact		Hardness HB	Limited Ruling Section mm
						IZOD FT.Lb	KCV Joules		
Normalise	45	700	355	-	12	-	-	201/255	63
	39	600	310	-	13	-	-	170/223	250
R	45/55	700/850	415	385	14	-	-	201/255	100
S	50/60	775/925	480	450	14	-	-	223/277	63
T	55/65	850/1000	570	555	12	-	-	248/302	19



Product Information:

EN3A - 070M20

EN3A - 070M20 Black (As rolled or forged)
CARBON STEEL

Equivalents:
BS970 Part 1 1983, 070M20
BS970 of 1955 EN3A
American AISI 1020

Chemical:
Composition %
Carbon = 0,16 - 0,24
Silicon = 0,10 - 0,40
Manganese = 0,50 - 0,90
Sulphur = 0,050 max
Phosphorus = 0,050 max

Characteristics:
- A 6% allowance should always be made for removal of surface defects during machining
- Machinability good
- Good weldability
- Low Tensile Steel (28/30 ton tensile)
- Can be case hardened
- Not subject to high stresses or severe wear

Typical Applications:
Low tensile shafts, bolts, nuts
Machinery components where low tensile strength material is required
Flanges
Bushes, spacers
Pulleys (light duty)

Mechanical Properties

Heat Treatment	Condition	Tensile Strength RM MPA	Yield Stress RE MPA	A min on 5,65 √So	Impact		Hardness HB	Limited Ruling Section mm
					IZOD FT.Lb	KCV Joules		
Normalise	-	430	215	21	-	-	126/179	150
	-	400	200	21	-	-	116/170	250

Tel: +27 (0)11 865 4939. **Fax:** +27 (0)11 902 8995. **e-mail:** sales@specialsteels.co.za

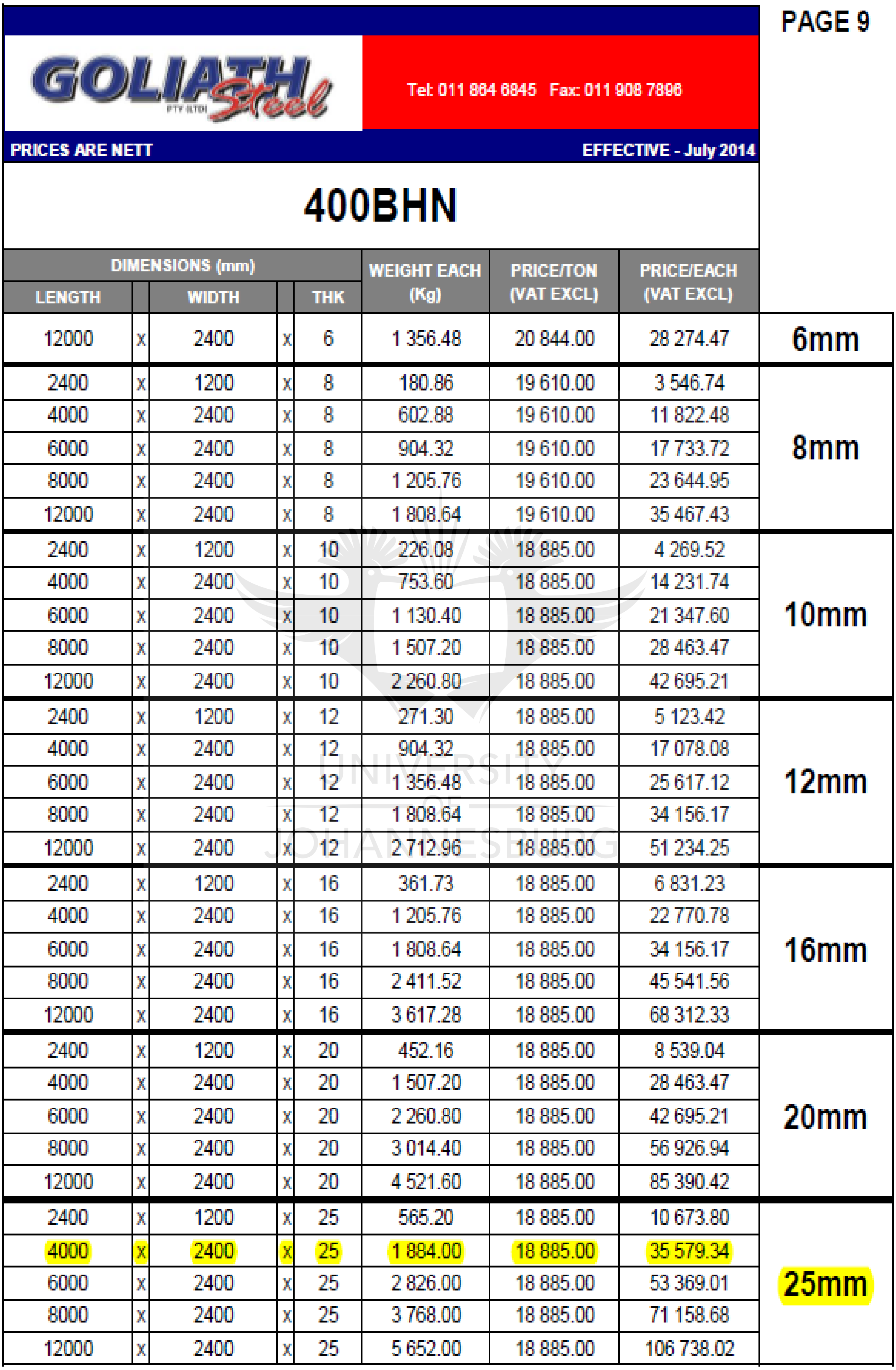
Appendix 43: John Deere Tractor Mass to Power ratio.

Specifications

Engine Performance	6130	6230	6330	6430	6534	6630	6830	6930
Stage III certified acc. 97/68/EC								
Rated power (97/68 EC), kW (hp)	63 (85)	70 (95)	77 (105)	88 (120)	92 (125)	99 (135)	107(145)	114 (155)
Maximum Power (97/68/EC), kW (hp)	66 (89)	74 (101)	81 (110)	93 (126)	97 (131)	104 (142)	112(152)	120 (163)
Rated power (ECE-R24), kW (hp)	60 (82)	67 (91)	74 (100)	85 (115)	88 (120)	95 (130)	103 (140)	110 (150)
Maximum Power (ECE-R24), kW (hp)	64 (86)	70 (96)	78 (106)	89 (121)	93 (126)	100 (136)	108 (147)	116 (158)
Cylinders/Turbo/Charge Air Cooler/Displacement, ccm	4/T/X/4530	4/T/X/4530	4/T/X/4530	4/T/X/4530	4/T/X/4530	6/T/X/6780	6/T/X/6780	6/T/X/6780
Rated Speed, rpm	2300	2300	2300	2300	2300	2300	2100	2100
Max Power at engine speed, rpm	2100	2100	2100	2100	2100	2100	2100	2100
Engine Speed at max. Torque, rpm	1500	1500	1500	1500	1500	1500	1300	1300
Constant power range, rpm	500	500	500	500	500	500	450	450
Torque Reserve, %	33	33	33	33	33	33	33	33
Fuel injection system	Full authority electronic engine management							
Type of injection	2 Valve High Pressure CommonRail							
Cooling	Dual Temperature Cooling							
Fan	Temperature controlled viscous fan							
Air cleaner	PowerCore precleaning							
Clutch	Oil cooled, PermaClutch 2, 225 mm diameter discs							
Transmission								
SyncroPlus								
12/4; 2.5 – 30 km//h	·	·	·	·	·	·	–	–
PowerReverser	Electrical Left Hand PowerReverser with neutral position; F/R Ratio 1:1							
16/16; 1.9 – 30 km/h or 2.5 – 40 km/h	·	·	·	·	·	·	–	–
PowrQuad Plus	4 powershiftable gears; Electrical Left Hand PowerReverser with neutral position; F/R Ratio 1:1							
16/16; 2.5 – 30 km/h	·	·	·	·	·	·	–	–
16/16; 3.1 – 40 km/h	·	·	·	·	–	–	–	–
20/20; 2.5 – 40 km/h	–	–	–	–	–	–	·	·
24/24; 1.6 – 40 km/h	·	·	·	·	·	·	–	–
Additional 12/12 Creeper reduction 1:10	Option for PowerReverser and PowrQuad Transmission							
PTO – Rear								
PTO Clutch	Electro-hydraulically operated oil cooled multi-disc clutch							
540/540E/1000 Rear PTO at rated PTO speeds	2143/1684/2208	2143/1684/2208	2143/1684/2208	2143/1684/2208	2143/1684/2208	2143/1684/2208	1995/1743/1995	1995/1743/1995
Brakes								
Foot brake	Oil cooled discs, self equalizing, self adjusting							
Parking lock	Transmission parking position.							
4WD braking	Automatic 4WD engagement							
Axles								
2WD version	·	·	·	·	·	–	–	–

Engine Performance	6130	6230	6330	6430	6534	6630	6830	6930
Engagement front differential lock	Automatic self locking under full load							
Engagement rear differential lock	Electro-hydraulically operated, oil-cooled multi-disc							
Caster angle/turning radius w/o brakes 4WD in mm	12°/4440	12°/4440	12°/4500	12°/4500	12°/5100	12°/5100	12°/5200	12°/5200
Hydraulic System and 3-point-hitch								
Type (base)	Load sensing with constant flow pump (PC = pressure compensated)							
Type (option)	–	–	–	–	–	–	Pressure and flow compensated (PFC)	
Pump capacity (l/min) / maximum pressure with PC	65 or 80 with 200 bar (PC)						80 with 200 bar (PC)	
Pump capacity (l/min) / maximum pressure with PFC	–	–	–	–	–	–	110 with 200 bar (PFC)	
Category	II opt. II/IIIN	II opt. II/IIIN	II opt. II/IIIN		II opt. II/IIIN	II opt. II/IIIN	II/IIIN	II/IIIN
Sensing type	Electronic lower link							
Sensing modes	Load & depth control, infinite mix, float							
Control modes	Height limiter, working depth, rate of drop, quick raise, quick pull-in; Hitch dampening							
Max. lift capacity base/optional, kN	37.4/49.8	43.4/49.8	43.4/49.8	49.8/57.2	49.8/57.2	49.8/64.6	66.6/75.2	66.6/75.2
Capacity								
Fuel Tank base/optional, l	165/185	165/185	165/185	165/185	207/250	207/250	250/325	250/325
Alternator/Battery, A/Ah	90/110 opt. 120/154	90/110 opt. 120/154	90/110 opt. 120/154	90/110 opt. 120/154	90/154 opt. 120/174	90/154 opt. 120/174	90/154 opt. 120/174	90/154 opt. 120/174
Hydraulic & Transmission oil/change intervall in l/h	50/1500	50/1500	50/1500	50/1500	50/1500	50/1500	50/1500	50/1500
Engine Oil/change intervall in l/h	16/500	16/500	16/500	16/500	16/500	19.5/500	19.5/500	19.5/500
Engine Coolant/change intervall in l/h	24/3000	24/3000	24/3000	24/3000	24/3000	28/3000	28/3000	28/3000
Cab								
Specification	Tiltable cab; 310° all-round vision; climate control, telescoping and tiltable steering column/wheel; opt. FieldOffice							
Sound Reduction/Level, dB(A)	71	71	71	71	70	70	72	72
Display	Tilt with steering wheel	Tilt with steering wheel	Tilt with steering wheel	Tilt with steering wheel	Tilt with steering wheel	Tilt with steering wheel	Tilt with steering wheel	Tilt with steering wheel
Dimensions and Weights								
Wheelbase, mm	2400	2400	2400	2400	2650	2650	2650	2650
Width x Height x Length, mm	2275 x 2718 x 4289	2275 x 2718 x 4289	2316 x 2743 x 4289	2316 x 2743 x 4289	2316 x 2833 x 4728	2316 x 2833 x 4728	2382 x 2923 x 4758	2382 x 2923 x 4758
With tyre size	14.9R24 and 16.9R38	14.9R24 and 16.9R38	16.9R24 and 18.4R38	16.9R24 and 18.4R38	420/70R28 and 18.4R38	420/70R28 and 18.4R38	16.9R28 and 20.8R38	16.9R28 and 20.8R38
Minimum shipping weight, kg	4640	4640	4745	4745	5165	5165	5635	5635
Maximum Permissible Gross Weight at 40 km/h, kg	7600	7600	8200	8200	8500	9500	10000	10500

Appendix 44: Steel Prices.



PAGE 9

GOLIATH Steel PTY (LTD)

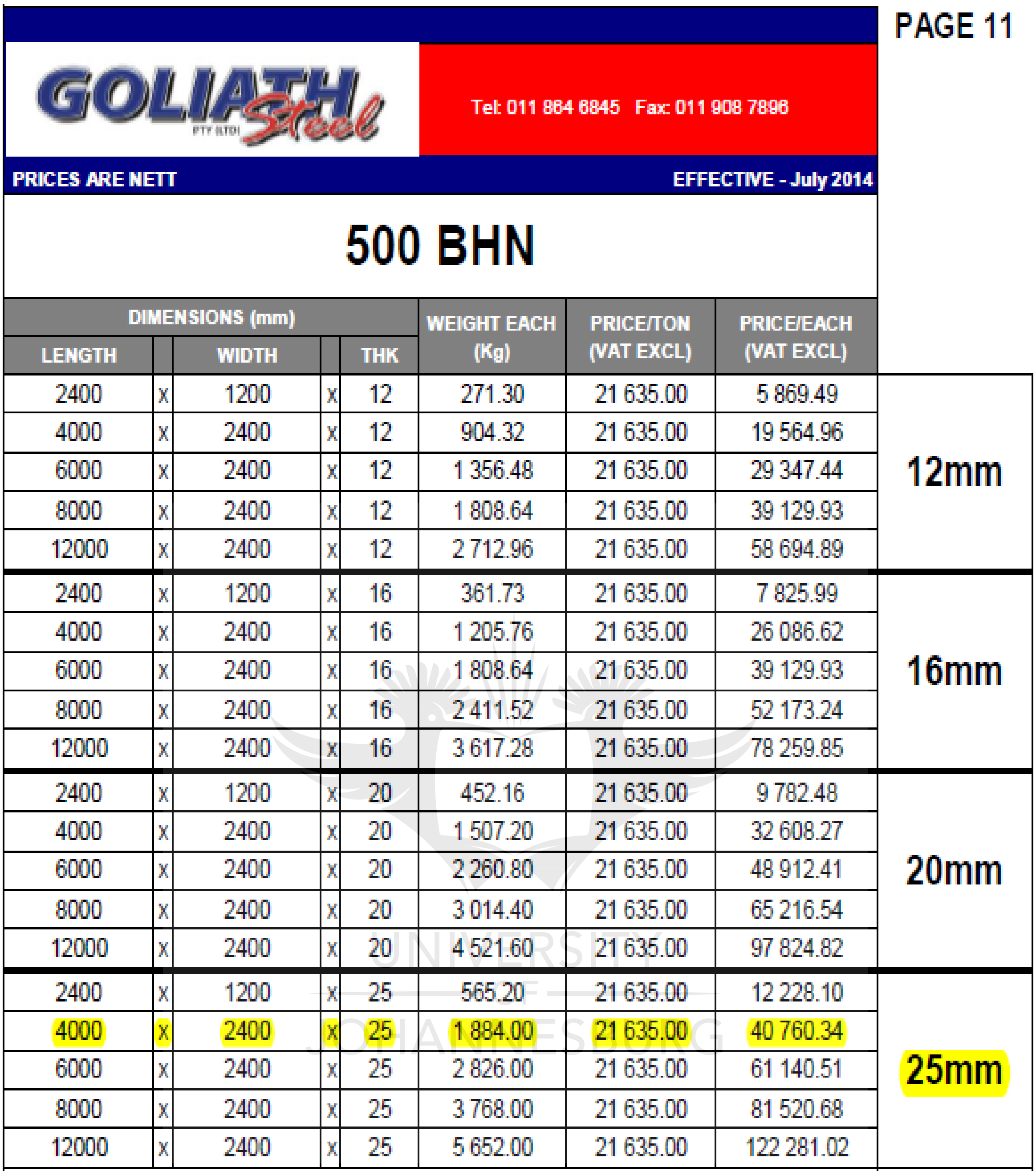
Tel: 011 864 6845 Fax: 011 908 7896

PRICES ARE NETT

EFFECTIVE - July 2014

400BHN

DIMENSIONS (mm)					WEIGHT EACH (Kg)	PRICE/TON (VAT EXCL)	PRICE/EACH (VAT EXCL)	
LENGTH		WIDTH		THK				
12000	x	2400	x	6	1 356.48	20 844.00	28 274.47	6mm
2400	x	1200	x	8	180.86	19 610.00	3 546.74	8mm
4000	x	2400	x	8	602.88	19 610.00	11 822.48	
6000	x	2400	x	8	904.32	19 610.00	17 733.72	
8000	x	2400	x	8	1 205.76	19 610.00	23 644.95	
12000	x	2400	x	8	1 808.64	19 610.00	35 467.43	
2400	x	1200	x	10	226.08	18 885.00	4 269.52	10mm
4000	x	2400	x	10	753.60	18 885.00	14 231.74	
6000	x	2400	x	10	1 130.40	18 885.00	21 347.60	
8000	x	2400	x	10	1 507.20	18 885.00	28 463.47	
12000	x	2400	x	10	2 260.80	18 885.00	42 695.21	
2400	x	1200	x	12	271.30	18 885.00	5 123.42	12mm
4000	x	2400	x	12	904.32	18 885.00	17 078.08	
6000	x	2400	x	12	1 356.48	18 885.00	25 617.12	
8000	x	2400	x	12	1 808.64	18 885.00	34 156.17	
12000	x	2400	x	12	2 712.96	18 885.00	51 234.25	
2400	x	1200	x	16	361.73	18 885.00	6 831.23	16mm
4000	x	2400	x	16	1 205.76	18 885.00	22 770.78	
6000	x	2400	x	16	1 808.64	18 885.00	34 156.17	
8000	x	2400	x	16	2 411.52	18 885.00	45 541.56	
12000	x	2400	x	16	3 617.28	18 885.00	68 312.33	
2400	x	1200	x	20	452.16	18 885.00	8 539.04	20mm
4000	x	2400	x	20	1 507.20	18 885.00	28 463.47	
6000	x	2400	x	20	2 260.80	18 885.00	42 695.21	
8000	x	2400	x	20	3 014.40	18 885.00	56 926.94	
12000	x	2400	x	20	4 521.60	18 885.00	85 390.42	
2400	x	1200	x	25	565.20	18 885.00	10 673.80	25mm
4000	x	2400	x	25	1 884.00	18 885.00	35 579.34	
6000	x	2400	x	25	2 826.00	18 885.00	53 369.01	
8000	x	2400	x	25	3 768.00	18 885.00	71 158.68	
12000	x	2400	x	25	5 652.00	18 885.00	106 738.02	



PAGE 11

GOLIATH Steel PTY (LTD)

Tel: 011 864 6845 Fax: 011 908 7896

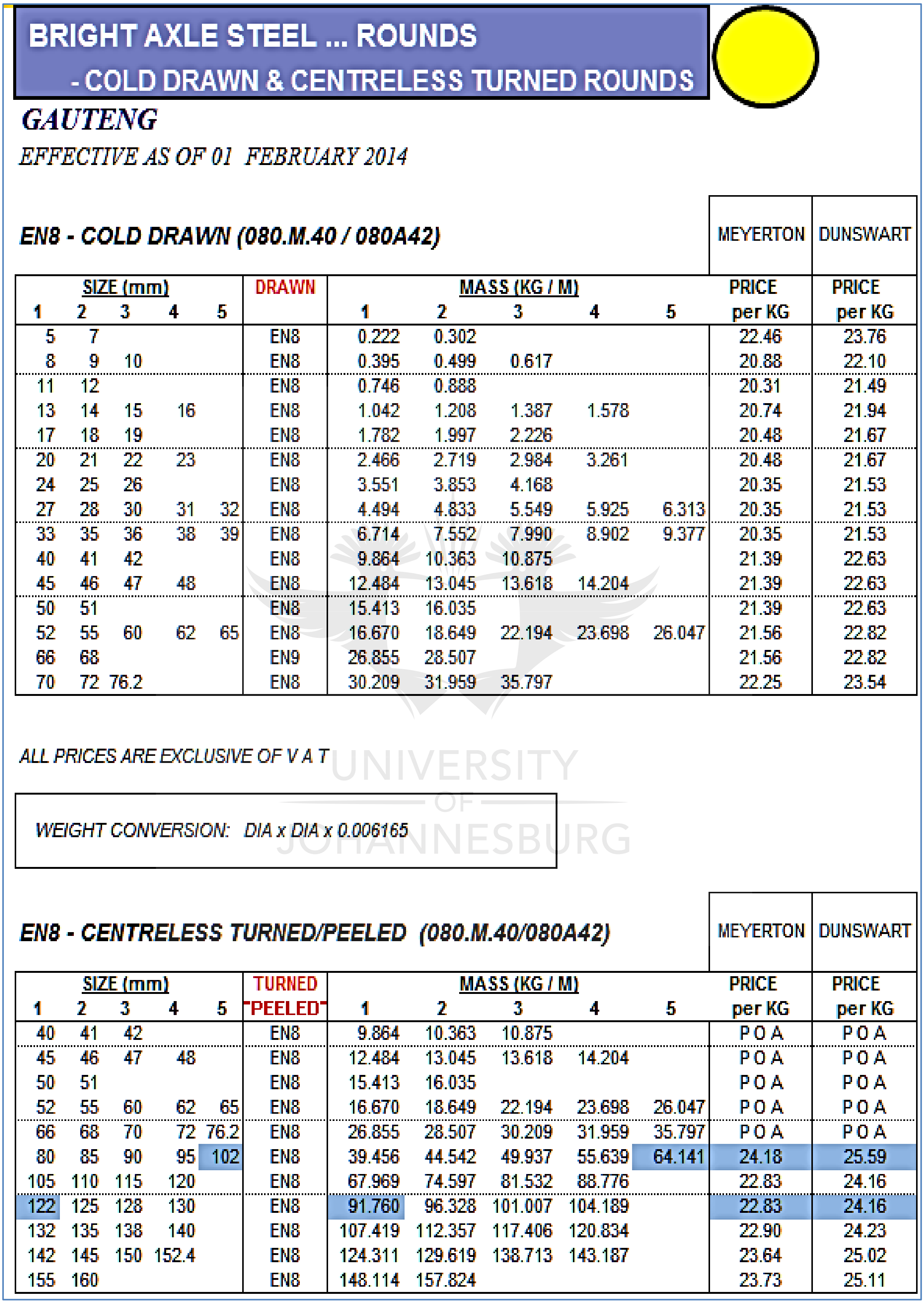
PRICES ARE NETT

EFFECTIVE - July 2014

500 BHN

DIMENSIONS (mm)					WEIGHT EACH (Kg)	PRICE/TON (VAT EXCL)	PRICE/EACH (VAT EXCL)	
LENGTH		WIDTH		THK				
2400	x	1200	x	12	271.30	21 635.00	5 869.49	12mm
4000	x	2400	x	12	904.32	21 635.00	19 564.96	
6000	x	2400	x	12	1 356.48	21 635.00	29 347.44	
8000	x	2400	x	12	1 808.64	21 635.00	39 129.93	
12000	x	2400	x	12	2 712.96	21 635.00	58 694.89	
2400	x	1200	x	16	361.73	21 635.00	7 825.99	16mm
4000	x	2400	x	16	1 205.76	21 635.00	26 086.62	
6000	x	2400	x	16	1 808.64	21 635.00	39 129.93	
8000	x	2400	x	16	2 411.52	21 635.00	52 173.24	
12000	x	2400	x	16	3 617.28	21 635.00	78 259.85	
2400	x	1200	x	20	452.16	21 635.00	9 782.48	20mm
4000	x	2400	x	20	1 507.20	21 635.00	32 608.27	
6000	x	2400	x	20	2 260.80	21 635.00	48 912.41	
8000	x	2400	x	20	3 014.40	21 635.00	65 216.54	
12000	x	2400	x	20	4 521.60	21 635.00	97 824.82	
2400	x	1200	x	25	565.20	21 635.00	12 228.10	25mm
4000	x	2400	x	25	1 884.00	21 635.00	40 760.34	
6000	x	2400	x	25	2 826.00	21 635.00	61 140.51	
8000	x	2400	x	25	3 768.00	21 635.00	81 520.68	
12000	x	2400	x	25	5 652.00	21 635.00	122 281.02	

EN8 (080M40)



BRIGHT AXLE STEEL ... ROUNDS
- COLD DRAWN & CENTRELESS TURNED ROUNDS

GAUTENG

EFFECTIVE AS OF 01 FEBRUARY 2014

EN8 - COLD DRAWN (080.M.40 / 080A42)

											MEYERTON	DUNSWART
SIZE (mm)					DRAWN	MASS (KG / M)					PRICE	PRICE
1	2	3	4	5		1	2	3	4	5	per KG	per KG
5	7				EN8	0.222	0.302				22.46	23.76
8	9	10			EN8	0.395	0.499	0.617			20.88	22.10
11	12				EN8	0.746	0.888				20.31	21.49
13	14	15	16		EN8	1.042	1.208	1.387	1.578		20.74	21.94
17	18	19			EN8	1.782	1.997	2.226			20.48	21.67
20	21	22	23		EN8	2.466	2.719	2.984	3.261		20.48	21.67
24	25	26			EN8	3.551	3.853	4.168			20.35	21.53
27	28	30	31	32	EN8	4.494	4.833	5.549	5.925	6.313	20.35	21.53
33	35	36	38	39	EN8	6.714	7.552	7.990	8.902	9.377	20.35	21.53
40	41	42			EN8	9.864	10.363	10.875			21.39	22.63
45	46	47	48		EN8	12.484	13.045	13.618	14.204		21.39	22.63
50	51				EN8	15.413	16.035				21.39	22.63
52	55	60	62	65	EN8	16.670	18.649	22.194	23.698	26.047	21.56	22.82
66	68				EN9	26.855	28.507				21.56	22.82
70	72	76.2			EN8	30.209	31.959	35.797			22.25	23.54

ALL PRICES ARE EXCLUSIVE OF V A T

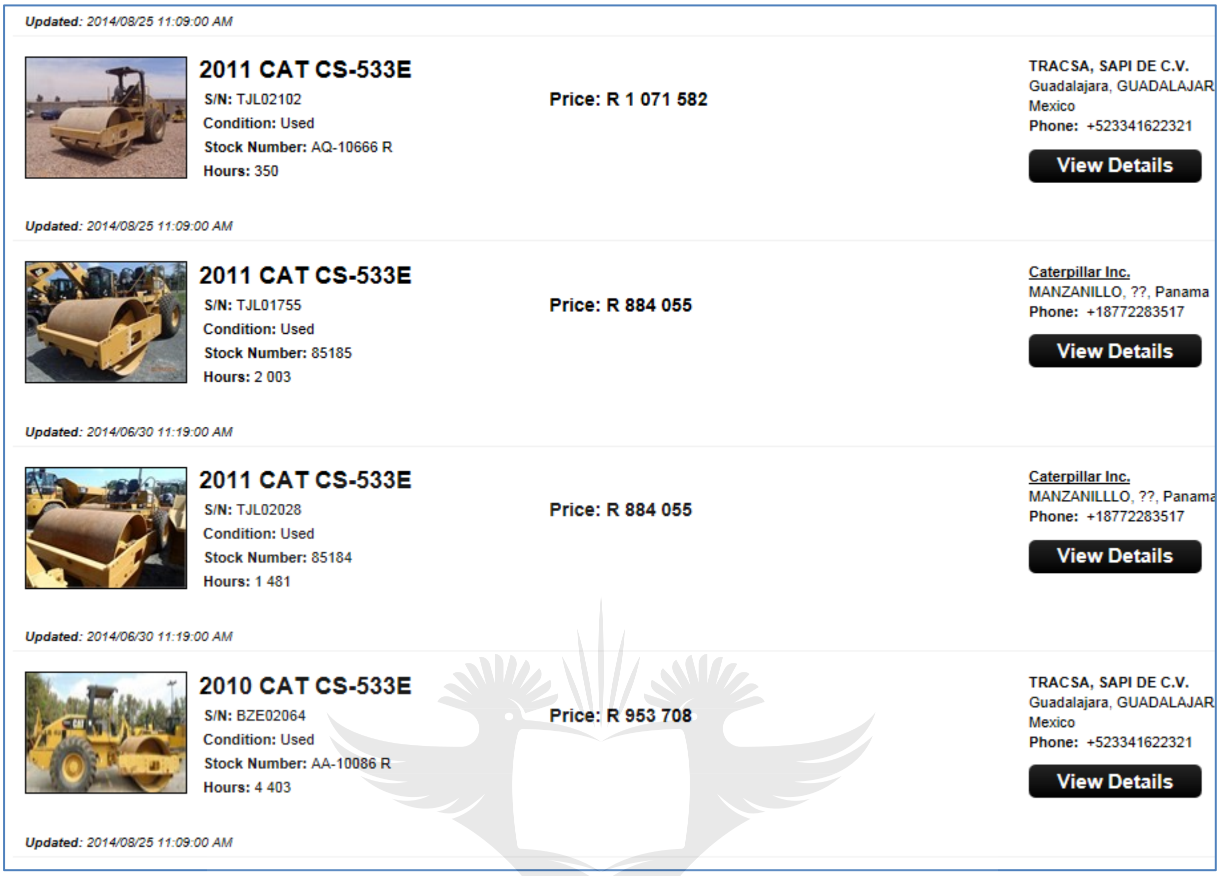
WEIGHT CONVERSION: DIA x DIA x 0.006165

EN8 - CENTRELESS TURNED/PEELED (080.M.40/080A42)

											MEYERTON	DUNSWART
SIZE (mm)					TURNED	MASS (KG / M)					PRICE	PRICE
1	2	3	4	5	"PEELED"	1	2	3	4	5	per KG	per KG
40	41	42			EN8	9.864	10.363	10.875			P O A	P O A
45	46	47	48		EN8	12.484	13.045	13.618	14.204		P O A	P O A
50	51				EN8	15.413	16.035				P O A	P O A
52	55	60	62	65	EN8	16.670	18.649	22.194	23.698	26.047	P O A	P O A
66	68	70	72	76.2	EN8	26.855	28.507	30.209	31.959	35.797	P O A	P O A
80	85	90	95	102	EN8	39.456	44.542	49.937	55.639	64.141	24.18	25.59
105	110	115	120		EN8	67.969	74.597	81.532	88.776		22.83	24.16
122	125	128	130		EN8	91.760	96.328	101.007	104.189		22.83	24.16
132	135	138	140		EN8	107.419	112.357	117.406	120.834		22.90	24.23
142	145	150	152.4		EN8	124.311	129.619	138.713	143.187		23.64	25.02
155	160				EN8	148.114	157.824				23.73	25.11

Appendix 45: Vibratory road roller prices.

Caterpillar Road Roller Prices



Updated: 2014/08/25 11:09:00 AM

2011 CAT CS-533E
S/N: TJL02102
Condition: Used
Stock Number: AQ-10666 R
Hours: 350

Price: R 1 071 582

TRACSA, SAPI DE C.V.
Guadalajara, GUADALAJAR
Mexico
Phone: +523341622321

View Details

Updated: 2014/08/25 11:09:00 AM

2011 CAT CS-533E
S/N: TJL01755
Condition: Used
Stock Number: 85185
Hours: 2 003

Price: R 884 055

Caterpillar Inc.
MANZANILLO, ??, Panama
Phone: +18772283517

View Details

Updated: 2014/06/30 11:19:00 AM

2011 CAT CS-533E
S/N: TJL02028
Condition: Used
Stock Number: 85184
Hours: 1 481

Price: R 884 055

Caterpillar Inc.
MANZANILLLO, ??, Panama
Phone: +18772283517

View Details

Updated: 2014/06/30 11:19:00 AM

2010 CAT CS-533E
S/N: BZE02064
Condition: Used
Stock Number: AA-10086 R
Hours: 4 403

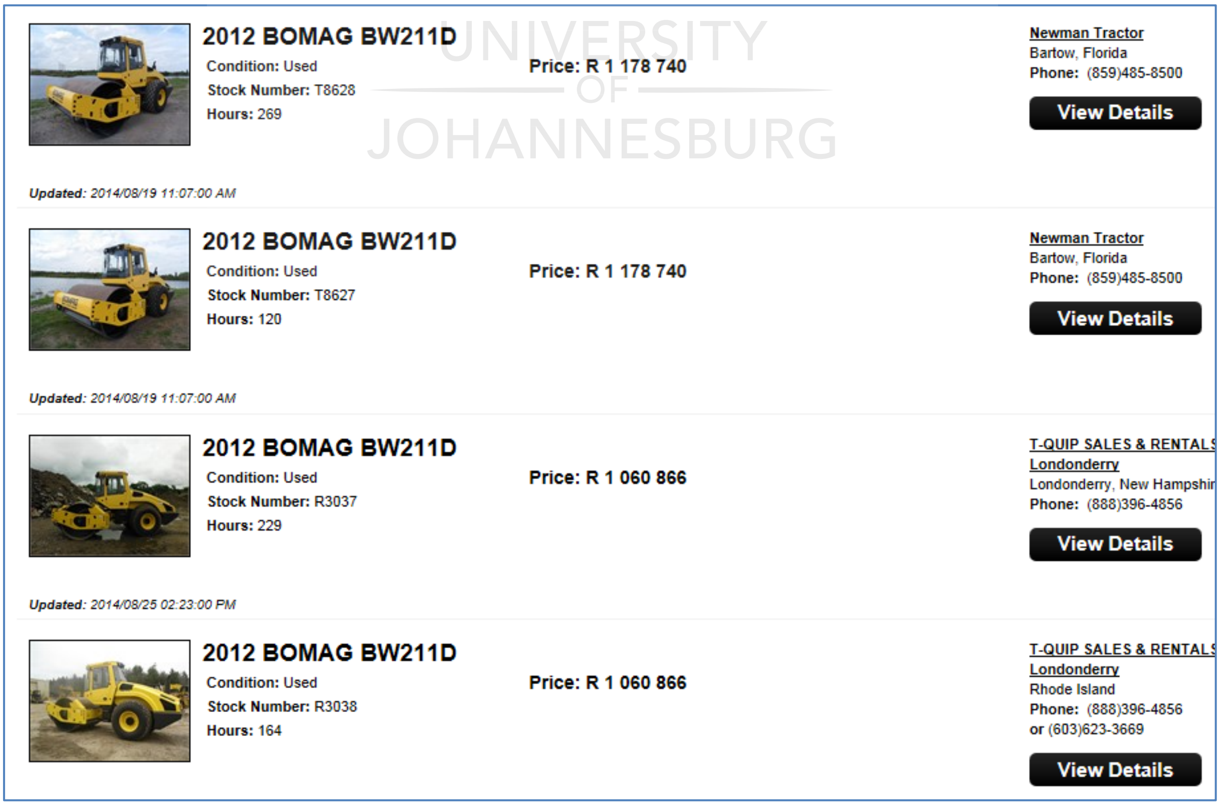
Price: R 953 708

TRACSA, SAPI DE C.V.
Guadalajara, GUADALAJAR
Mexico
Phone: +523341622321

View Details

Updated: 2014/08/25 11:09:00 AM

Bomag Road Roller Prices



2012 BOMAG BW211D
Condition: Used
Stock Number: T8628
Hours: 269

Price: R 1 178 740

Newman Tractor
Bartow, Florida
Phone: (859)485-8500

View Details

Updated: 2014/08/19 11:07:00 AM

2012 BOMAG BW211D
Condition: Used
Stock Number: T8627
Hours: 120

Price: R 1 178 740

Newman Tractor
Bartow, Florida
Phone: (859)485-8500

View Details

Updated: 2014/08/19 11:07:00 AM

2012 BOMAG BW211D
Condition: Used
Stock Number: R3037
Hours: 229

Price: R 1 060 866

T-QUIP SALES & RENTALS
Londonderry
Londonderry, New Hampshir
Phone: (888)396-4856

View Details

Updated: 2014/08/25 02:23:00 PM

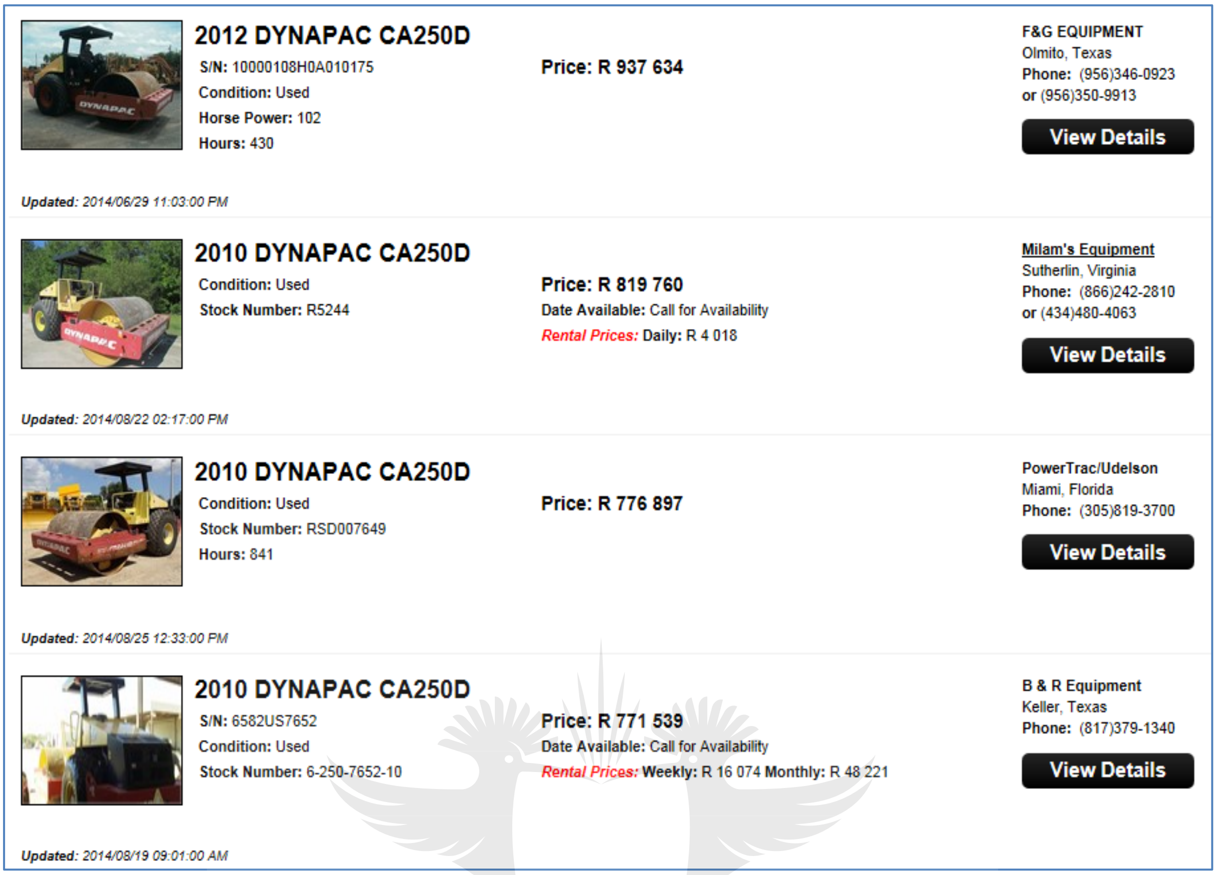
2012 BOMAG BW211D
Condition: Used
Stock Number: R3038
Hours: 164

Price: R 1 060 866

T-QUIP SALES & RENTALS
Londonderry
Rhode Island
Phone: (888)396-4856
or (603)623-3669

View Details

Dynapac Road Roller Prices



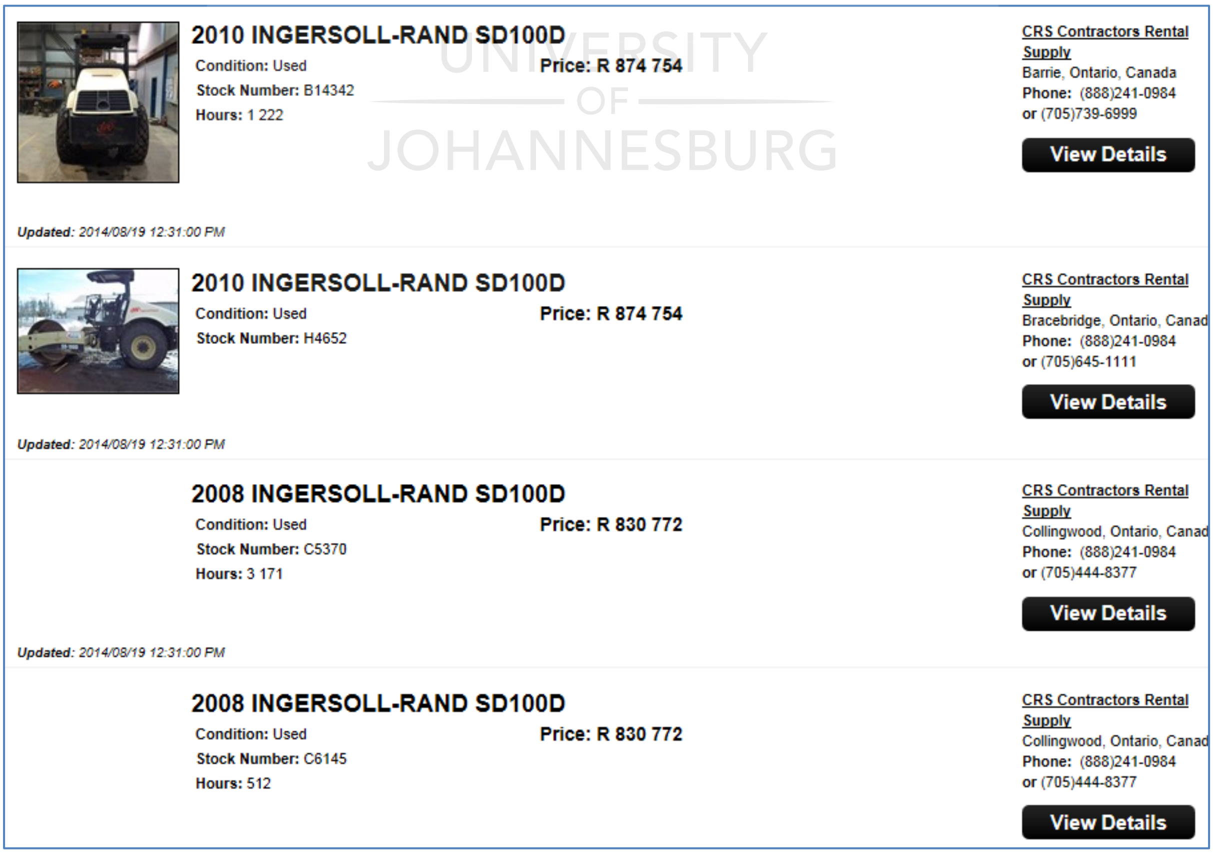
2012 DYNAPAC CA250D
S/N: 10000108H0A010175
Condition: Used
Horse Power: 102
Hours: 430
Price: R 937 634
F&G EQUIPMENT
Olmito, Texas
Phone: (956)346-0923
or (956)350-9913
Updated: 2014/06/29 11:03:00 PM

2010 DYNAPAC CA250D
Condition: Used
Stock Number: R5244
Price: R 819 760
Date Available: Call for Availability
Rental Prices: Daily: R 4 018
Milam's Equipment
Sutherlin, Virginia
Phone: (866)242-2810
or (434)480-4063
Updated: 2014/08/22 02:17:00 PM

2010 DYNAPAC CA250D
Condition: Used
Stock Number: RSD007649
Hours: 841
Price: R 776 897
PowerTrac/Udelson
Miami, Florida
Phone: (305)819-3700
Updated: 2014/08/25 12:33:00 PM

2010 DYNAPAC CA250D
S/N: 6582US7652
Condition: Used
Stock Number: 6-250-7652-10
Price: R 771 539
Date Available: Call for Availability
Rental Prices: Weekly: R 16 074 Monthly: R 48 221
B & R Equipment
Keller, Texas
Phone: (817)379-1340
Updated: 2014/08/19 09:01:00 AM

Ingersoll-rand Road Roller Prices



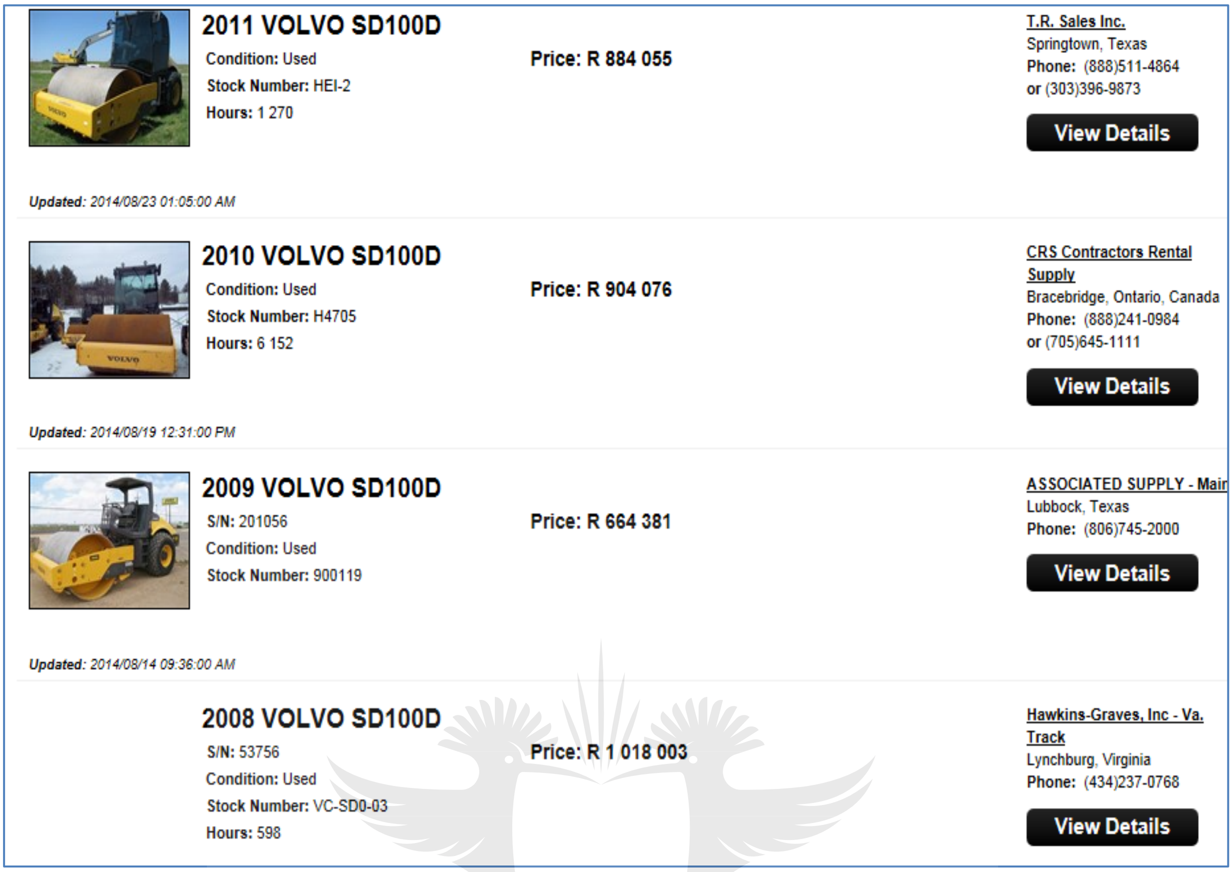
2010 INGERSOLL-RAND SD100D
Condition: Used
Stock Number: B14342
Hours: 1 222
Price: R 874 754
CRS Contractors Rental Supply
Barrie, Ontario, Canada
Phone: (888)241-0984
or (705)739-6999
Updated: 2014/08/19 12:31:00 PM

2010 INGERSOLL-RAND SD100D
Condition: Used
Stock Number: H4652
Price: R 874 754
CRS Contractors Rental Supply
Bracebridge, Ontario, Canad
Phone: (888)241-0984
or (705)645-1111
Updated: 2014/08/19 12:31:00 PM

2008 INGERSOLL-RAND SD100D
Condition: Used
Stock Number: C5370
Hours: 3 171
Price: R 830 772
CRS Contractors Rental Supply
Collingwood, Ontario, Canad
Phone: (888)241-0984
or (705)444-8377
Updated: 2014/08/19 12:31:00 PM

2008 INGERSOLL-RAND SD100D
Condition: Used
Stock Number: C6145
Hours: 512
Price: R 830 772
CRS Contractors Rental Supply
Collingwood, Ontario, Canad
Phone: (888)241-0984
or (705)444-8377

Volvo Road Roller Prices



2011 VOLVO SD100D

Condition: Used

Stock Number: HEI-2

Hours: 1 270

Price: R 884 055

T.R. Sales Inc.
Springtown, Texas
Phone: (888)511-4864
or (303)396-9873

Updated: 2014/08/23 01:05:00 AM

2010 VOLVO SD100D

Condition: Used

Stock Number: H4705

Hours: 6 152

Price: R 904 076

CRS Contractors Rental Supply
Bracebridge, Ontario, Canada
Phone: (888)241-0984
or (705)645-1111

Updated: 2014/08/19 12:31:00 PM

2009 VOLVO SD100D

S/N: 201056

Condition: Used

Stock Number: 900119

Price: R 664 381

ASSOCIATED SUPPLY - Mair
Lubbock, Texas
Phone: (806)745-2000

Updated: 2014/08/14 09:36:00 AM

2008 VOLVO SD100D

S/N: 53756

Condition: Used

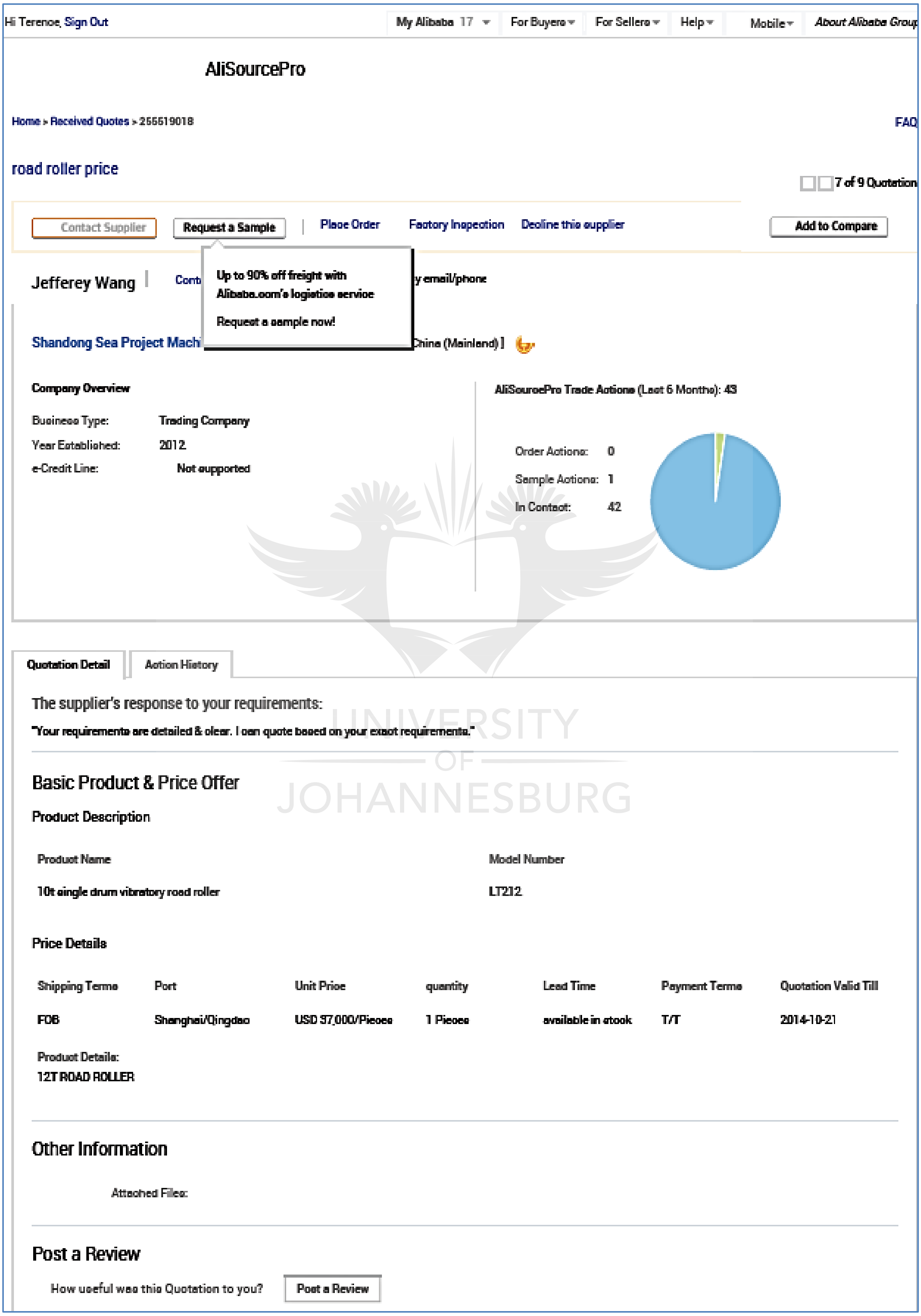
Stock Number: VC-SD0-03

Hours: 598

Price: R 1 018 003

Hawkins-Graves, Inc - Va. Track
Lynchburg, Virginia
Phone: (434)237-0768

Appendix 46: Road Roller Quotes from China (Brand new machine).



road roller price

Up to 90% off freight with Alibaba.com's logistics service

Request a sample now!

Jefferey Wang | Cont… y email/phone

Shandong Sea Project Machi… China (Mainland)]

Company Overview

Business Type: Trading Company
Year Established: 2012
e-Credit Line: Not supported

AliSourcePro Trade Actions (Last 6 Months): 43

Order Actions:	0
Sample Actions:	1
In Contact:	42

The supplier's response to your requirements:

"Your requirements are detailed & clear. I can quote based on your exact requirements."

Basic Product & Price Offer

Product Description

Product Name	Model Number
10t single drum vibratory road roller	LT212

Price Details

Shipping Terms	Port	Unit Price	quantity	Lead Time	Payment Terms	Quotation Valid Till
FOB	Shanghai/Qingdao	USD 37,000/Pieces	1 Pieces	available in stock	T/T	2014-10-21

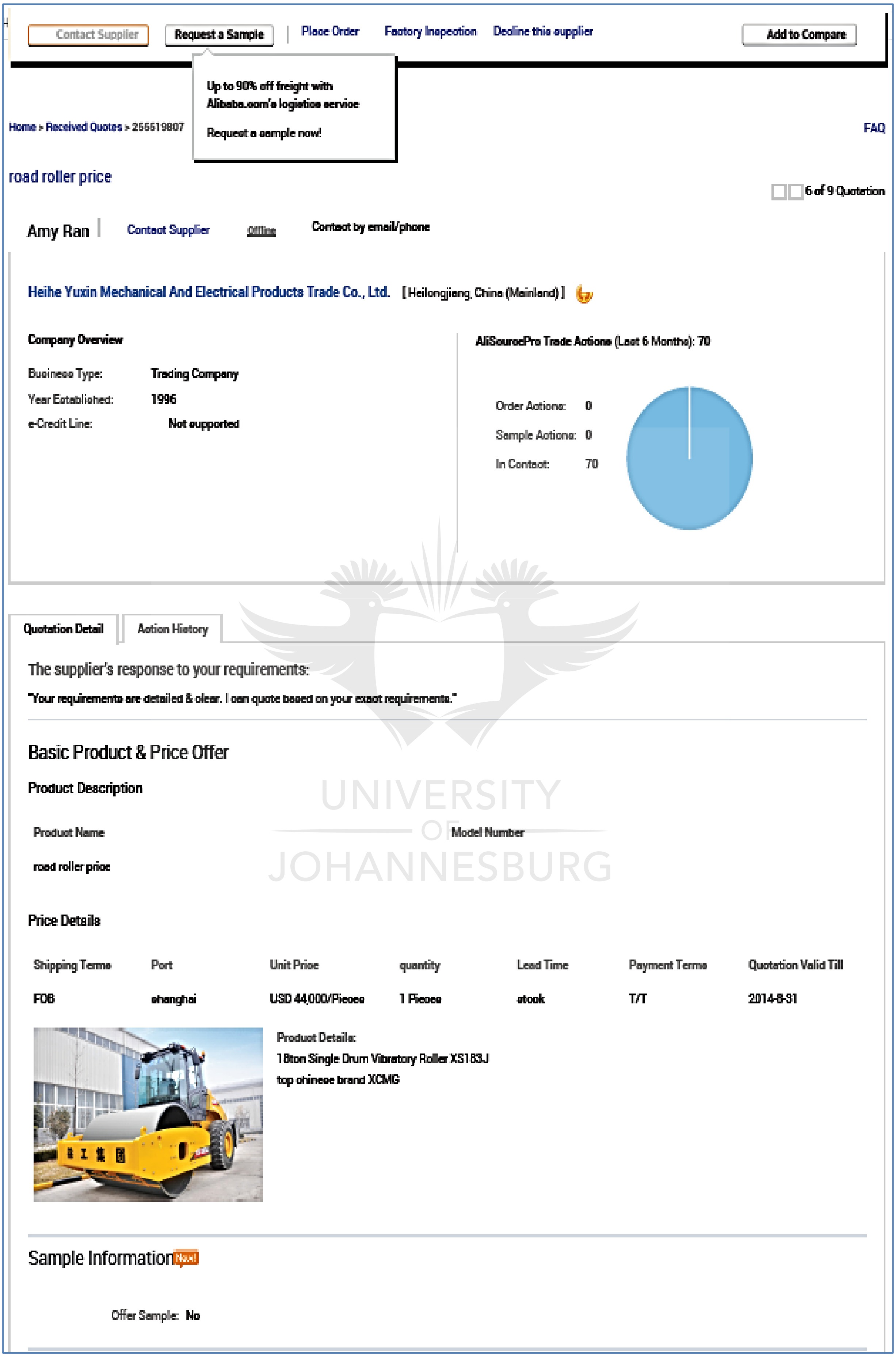
Product Details:
12T ROAD ROLLER

Other Information

Attached Files:

Post a Review

How useful was this Quotation to you?



Up to 90% off freight with Alibaba.com's logistics service

Request a sample now!

road roller price

Amy Ran | Contact Supplier Offline Contact by email/phone

Heihe Yuxin Mechanical And Electrical Products Trade Co., Ltd. [Heilongjiang, China (Mainland)]

Company Overview

Business Type: Trading Company

Year Established: 1996

e-Credit Line: Not supported

AliSourcePro Trade Actions (Last 6 Months): 70

Order Actions: 0

Sample Actions: 0

In Contact: 70

The supplier's response to your requirements:

"Your requirements are detailed & clear. I can quote based on your exact requirements."

Basic Product & Price Offer

Product Description

Product Name	Model Number
road roller price	

Price Details

Shipping Terms	Port	Unit Price	quantity	Lead Time	Payment Terms	Quotation Valid Till
FOB	shanghai	USD 44,000/Pieces	1 Pieces	stock	T/T	2014-8-31

Product Details:
18ton Single Drum Vibratory Roller XS183J
top chinese brand XCMG

Sample Information New!

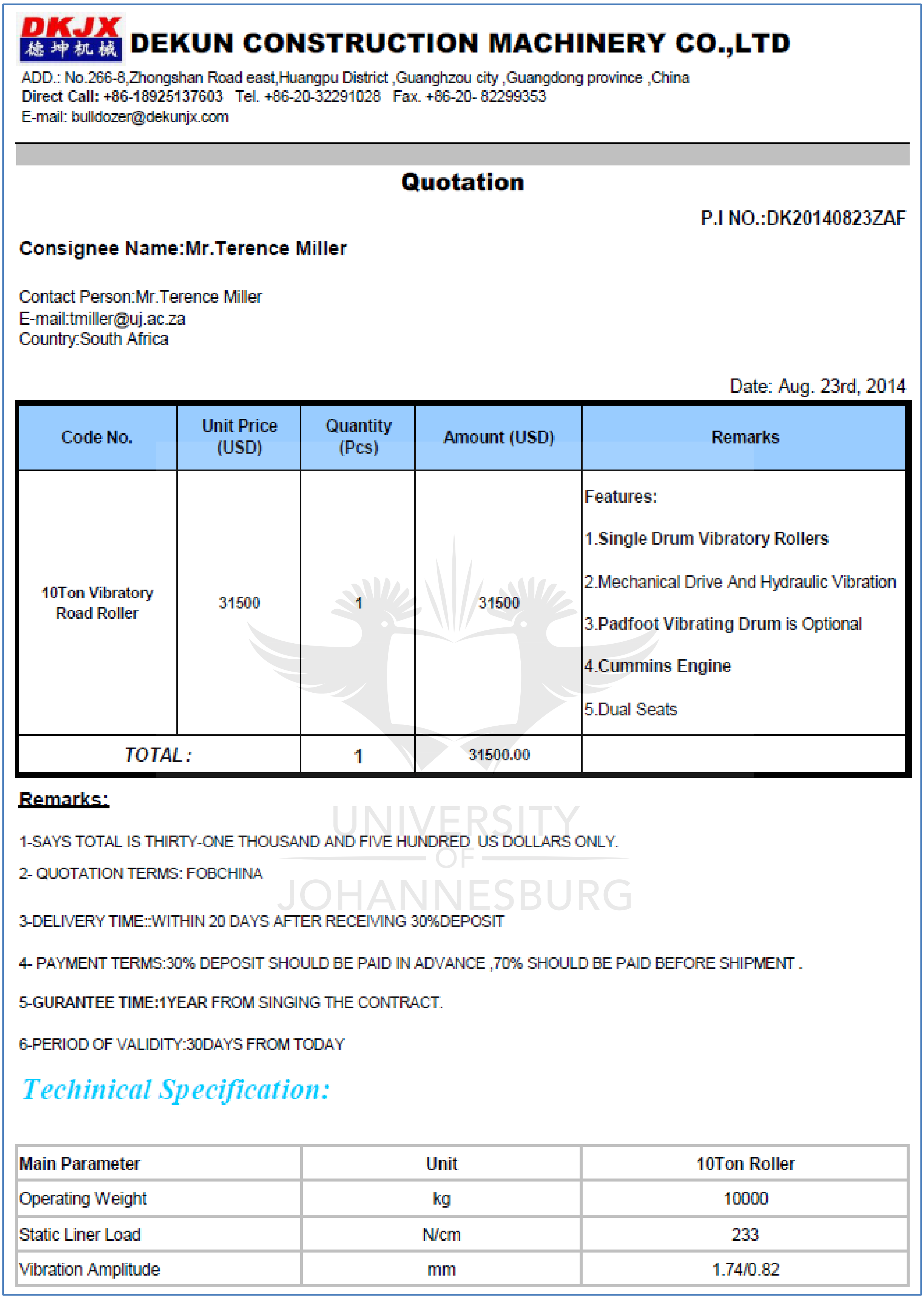
Offer Sample: No

Wuhan Kudat Industry & Trade Co., Ltd.
www.kudatchina.com Tel: 86-27-87747620 Fax: 86-27-87747760

LSS1001 Single Drum Road Roller with Mechanical control

Model		LSS1001
Operating mass	kg	10000
Static linear load	N/cm	233
Vibration amplitude	mm	1.74/0.82
Vibration frequency	Hz	32
Centrifugal force	kN	198/93
Travel speed	km/h	10.5
Gradeability	%	25
Turning radius	mm	6800
Drum width	mm	2100
Drum diameter	mm	1500
Wheelbase	mm	2900
Ground clearance	mm	386
Diesel engine model		4BTA3.9
Diesel engine power	kW	80
Overall dimensions	mm	5500×2300×3000

**Add: Room 401-3, E1 Building Optical Valley Software Park, East Lake High-Tech.
Development Zone, Wuhan 430073 China**



DKJX 德坤机械 **DEKUN CONSTRUCTION MACHINERY CO.,LTD**

ADD.: No.266-8,Zhongshan Road east,Huangpu District ,Guanghzou city ,Guangdong province ,China
Direct Call: +86-18925137603 Tel. +86-20-32291028 Fax. +86-20- 82299353
E-mail: bulldozer@dekunjx.com

Quotation

P.I NO.:DK20140823ZAF

Consignee Name:Mr.Terence Miller

Contact Person:Mr.Terence Miller
E-mail:tmiller@uj.ac.za
Country:South Africa

Date: Aug. 23rd, 2014

Code No.	Unit Price (USD)	Quantity (Pcs)	Amount (USD)	Remarks
10Ton Vibratory Road Roller	31500	1	31500	Features: 1.Single Drum Vibratory Rollers 2.Mechanical Drive And Hydraulic Vibration 3.Padfoot Vibrating Drum is Optional 4.Cummins Engine 5.Dual Seats
TOTAL :		1	31500.00	

Remarks:

1-SAYS TOTAL IS THIRTY-ONE THOUSAND AND FIVE HUNDRED US DOLLARS ONLY.

2- QUOTATION TERMS: FOBCHINA

3-DELIVERY TIME::WITHIN 20 DAYS AFTER RECEIVING 30%DEPOSIT

4- PAYMENT TERMS:30% DEPOSIT SHOULD BE PAID IN ADVANCE ,70% SHOULD BE PAID BEFORE SHIPMENT .

5-GURANTEE TIME:1YEAR FROM SINGING THE CONTRACT.

6-PERIOD OF VALIDITY:30DAYS FROM TODAY

Techinical Specification:

Main Parameter	Unit	10Ton Roller
Operating Weight	kg	10000
Static Liner Load	N/cm	233
Vibration Amplitude	mm	1.74/0.82

Quote by email (China)

From: Vivian Yang [mailto:whkudat@gmail.com]
Sent: 25 August 2014 08:01 AM
To: Miller, Terence
Subject: Re: RFQ quotation of single drum road roller

Dear Terence Miller,

Glad to search your inquiry about road roller. We have specialized in this field since year 1990.

Based on your requirements, I recommend you our LSS1001 single drum road roller with 10tons operating weight. Please find attachment about specification for your reference.

LSS1001: USD 32,460/unit FOB Shanghai

Warranty: One year and life long after sale service

Payment Terms: T/T in advance

Delivery Time: 30 days after receipt deposit

Price Validity: 30 days

Shipment: 40GP: 1unit

If you have any other needs, such as shipping cost, just kindly suggest your destination port.

Looking forward to hearing from you soon.

Best Regards

Miss Vivian (Sale Manager)

Wuhan Kudat Industry & Trade Co.,ltd
Skype: yueqiuqian

Tel: 0086-27-8774 7620

Website: www.kudatchina.com

Add: Room 401-3, E1 Building, Optical Valley Software Park,East Lake High-Tech. Development Zone
Wuhan 430073 China

2014-08-25 15:56 GMT+08:00 Miller, Terence <tmiller@uj.ac.za>:

Dear Vivian, please advise shipping cost for your ten ton road roller.

Shipping requirements Address;

Country : South Africa

City : Johannesburg

Suburb : Elandsfontein

Regards

Terence Miller

Dear Terence Miller,

Thanks for your prompt reply.
LSS1001: USD 32,460/unit FOB Shanghai

Shipping cost: USD 2,800 for 40GP to Johannesburg. It need about 30 days to arrive at this port.

So the total price of LSS1001 single drum road roller is USD 35,260/unit C&F Johannesburg, South Africa.

Shipping cost validity day: 10 days
If you have any other needs, please let me know.

Hope we can have a chance to cooperate.

Waiting for your feedback.

Best Regards

Miss Vivian (Sale Manager)

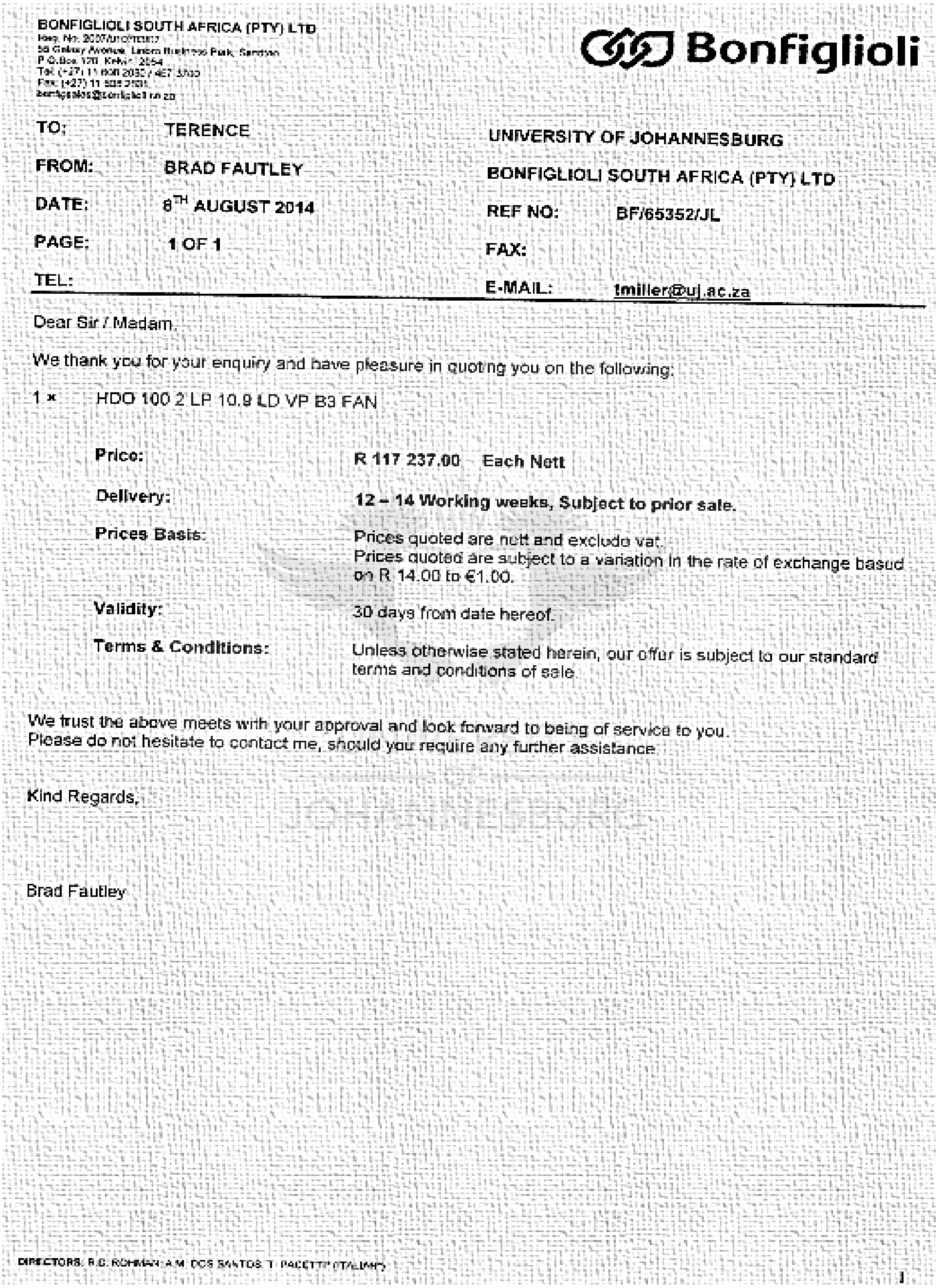
Wuhan Kudat Industry & Trade Co.,ltd
Skype: yueqiuqian
Tel: 0086-27-8774 7620

Website: www.kudatchina.com

Add: Room 401-3, E1 Building, Optical Valley Software Park,East Lake High-Tech. Development Zone Wuhan 430073 China

Appendix 47: Power Transmission quotes.

Bonfiglioli (Gearbox quote)



BONFIGLIOLI SOUTH AFRICA (PTY) LTD

Bonfiglioli

TO: TERENCE
UNIVERSITY OF JOHANNESBURG
FROM: BRAD FAUTLEY
BONFIGLIOLI SOUTH AFRICA (PTY) LTD
DATE: 8TH AUGUST 2014
REF NO: BF/65352/JL
PAGE: 1 OF 1
FAX:
TEL:
E-MAIL: tmiller@uj.ac.za

Dear Sir / Madam,

We thank you for your enquiry and have pleasure in quoting you on the following:

1 × HDO 100 2 LP 10.9 LD VP B3 FAN

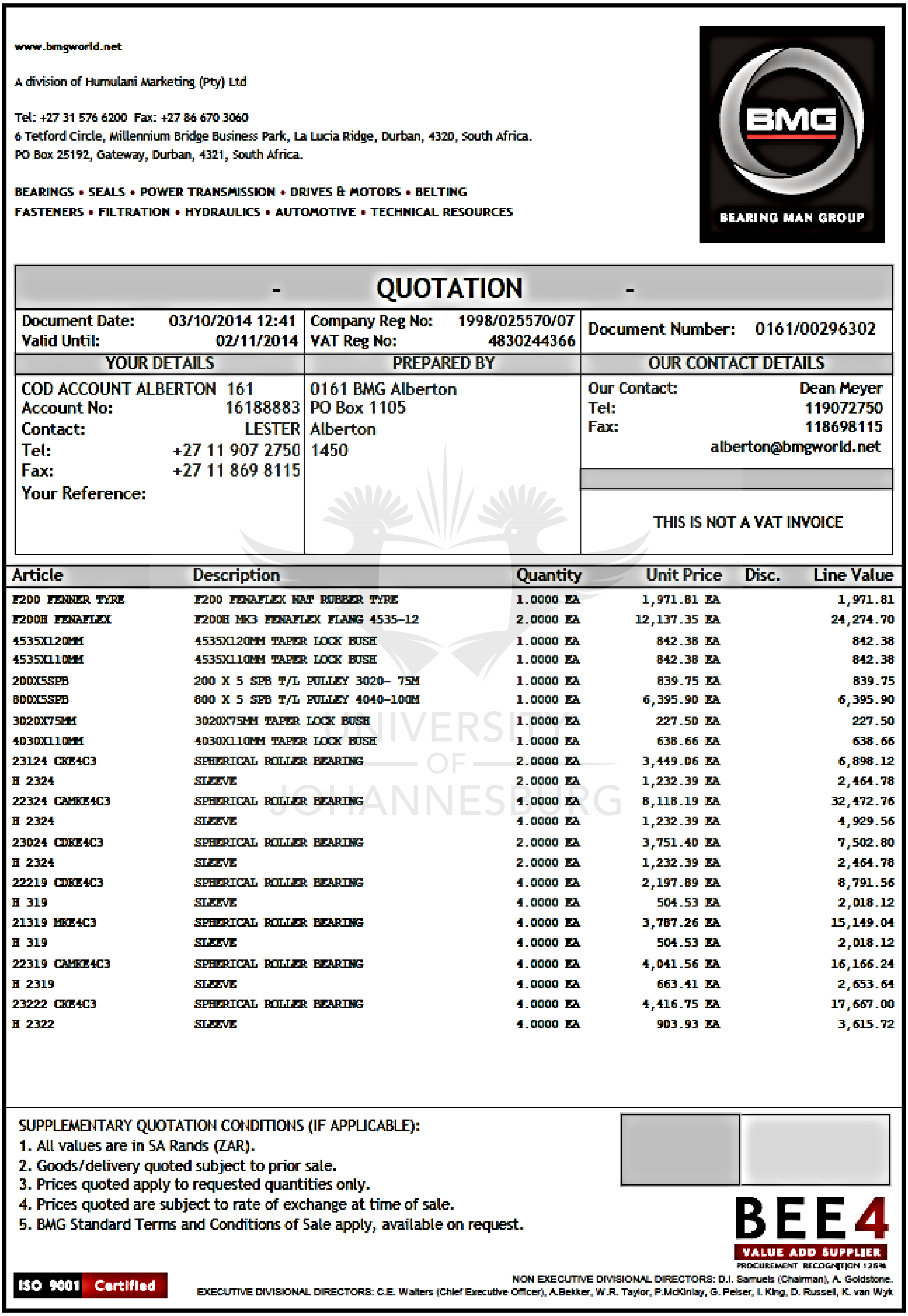
Price:	**R 117 237.00 Each Nett**
Delivery:	**12 – 14 Working weeks, Subject to prior sale.**
Prices Basis:	Prices quoted are nett and exclude vat. Prices quoted are subject to a variation in the rate of exchange based on R 14.00 to €1.00.
Validity:	30 days from date hereof.
Terms & Conditions:	Unless otherwise stated herein, our offer is subject to our standard terms and conditions of sale.

We trust the above meets with your approval and look forward to being of service to you.
Please do not hesitate to contact me, should you require any further assistance.

Kind Regards,

Brad Fautley

Bearing Man (Tyre Coupling, Friction Wedge Belt Drive & NSK Spherical Roller Bearings Quote.)



www.bmgworld.net

A division of Humulani Marketing (Pty) Ltd

Tel: +27 31 576 6200 Fax: +27 86 670 3060
6 Tetford Circle, Millennium Bridge Business Park, La Lucia Ridge, Durban, 4320, South Africa.
PO Box 25192, Gateway, Durban, 4321, South Africa.

BEARINGS • SEALS • POWER TRANSMISSION • DRIVES & MOTORS • BELTING
FASTENERS • FILTRATION • HYDRAULICS • AUTOMOTIVE • TECHNICAL RESOURCES

BMG
BEARING MAN GROUP

- QUOTATION -

Document Date: 03/10/2014 12:41 Valid Until: 02/11/2014	Company Reg No: 1998/025570/07 VAT Reg No: 4830244366	Document Number: 0161/00296302
YOUR DETAILS	**PREPARED BY**	**OUR CONTACT DETAILS**
COD ACCOUNT ALBERTON 161 Account No: 16188883 Contact: LESTER Tel: +27 11 907 2750 Fax: +27 11 869 8115 Your Reference:	0161 BMG Alberton PO Box 1105 Alberton 1450	Our Contact: Dean Meyer Tel: 119072750 Fax: 118698115 alberton@bmgworld.net THIS IS NOT A VAT INVOICE

Article	Description	Quantity	Unit Price	Disc.	Line Value
F200 FENNER TYRE	F200 FENAFLEX NAT RUBBER TYRE	1.0000 EA	1,971.81 EA		1,971.81
F200H FENAFLEX	F200H MK3 FENAFLEX FLANG 4535-12	2.0000 EA	12,137.35 EA		24,274.70
4535X120MM	4535X120MM TAPER LOCK BUSH	1.0000 EA	842.38 EA		842.38
4535X110MM	4535X110MM TAPER LOCK BUSH	1.0000 EA	842.38 EA		842.38
200X5SPB	200 X 5 SPB T/L PULLEY 3020- 75M	1.0000 EA	839.75 EA		839.75
800X5SPB	800 X 5 SPB T/L PULLEY 4040-100M	1.0000 EA	6,395.90 EA		6,395.90
3020X75MM	3020X75MM TAPER LOCK BUSH	1.0000 EA	227.50 EA		227.50
4030X110MM	4030X110MM TAPER LOCK BUSH	1.0000 EA	638.66 EA		638.66
23124 CKE4C3	SPHERICAL ROLLER BEARING	2.0000 EA	3,449.06 EA		6,898.12
H 2324	SLEEVE	2.0000 EA	1,232.39 EA		2,464.78
22324 CAMKE4C3	SPHERICAL ROLLER BEARING	4.0000 EA	8,118.19 EA		32,472.76
H 2324	SLEEVE	4.0000 EA	1,232.39 EA		4,929.56
23024 CDKE4C3	SPHERICAL ROLLER BEARING	2.0000 EA	3,751.40 EA		7,502.80
H 2324	SLEEVE	2.0000 EA	1,232.39 EA		2,464.78
22219 CDKE4C3	SPHERICAL ROLLER BEARING	4.0000 EA	2,197.89 EA		8,791.56
H 319	SLEEVE	4.0000 EA	504.53 EA		2,018.12
21319 MKE4C3	SPHERICAL ROLLER BEARING	4.0000 EA	3,787.26 EA		15,149.04
H 319	SLEEVE	4.0000 EA	504.53 EA		2,018.12
22319 CAMKE4C3	SPHERICAL ROLLER BEARING	4.0000 EA	4,041.56 EA		16,166.24
H 2319	SLEEVE	4.0000 EA	663.41 EA		2,653.64
23222 CKE4C3	SPHERICAL ROLLER BEARING	4.0000 EA	4,416.75 EA		17,667.00
H 2322	SLEEVE	4.0000 EA	903.93 EA		3,615.72

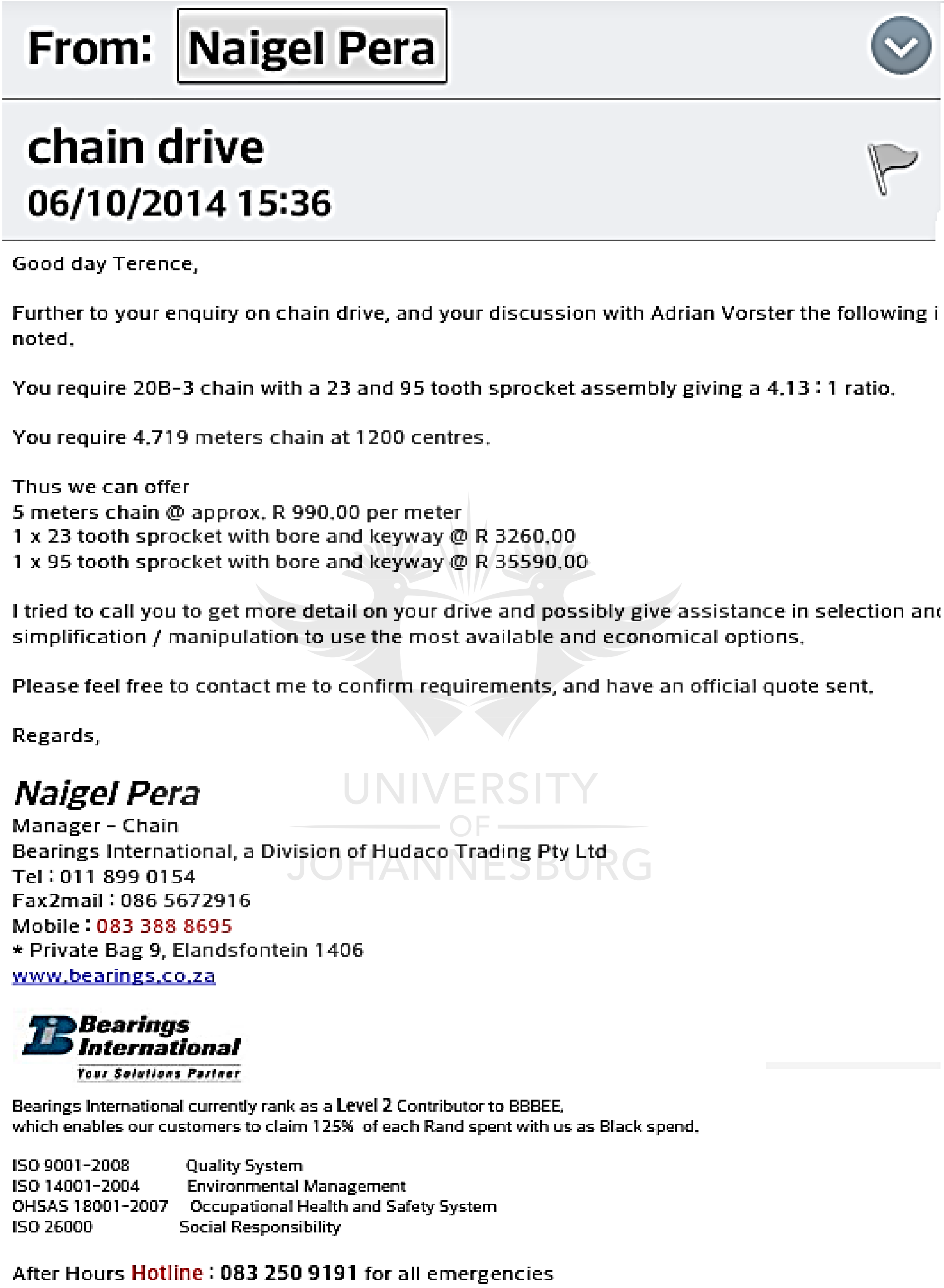
SUPPLEMENTARY QUOTATION CONDITIONS (IF APPLICABLE):
1. All values are in SA Rands (ZAR).
2. Goods/delivery quoted subject to prior sale.
3. Prices quoted apply to requested quantities only.
4. Prices quoted are subject to rate of exchange at time of sale.
5. BMG Standard Terms and Conditions of Sale apply, available on request.

BEE 4
VALUE ADD SUPPLIER
PROCUREMENT RECOGNITION 125%

ISO 9001 Certified

NON EXECUTIVE DIVISIONAL DIRECTORS: D.I. Samuels (Chairman), A. Goldstone.
EXECUTIVE DIVISIONAL DIRECTORS: C.E. Walters (Chief Executive Officer), A.Bekker, W.R. Taylor, P.McKinlay, G. Pelser, I. King, D. Russell, K. van Wyk

Bearings International (20B Chain drive quote)



From: **Naigel Pera**

chain drive

06/10/2014 15:36

Good day Terence,

Further to your enquiry on chain drive, and your discussion with Adrian Vorster the following i noted.

You require 20B-3 chain with a 23 and 95 tooth sprocket assembly giving a 4.13 : 1 ratio.

You require 4.719 meters chain at 1200 centres.

Thus we can offer
5 meters chain @ approx. R 990.00 per meter
1 x 23 tooth sprocket with bore and keyway @ R 3260.00
1 x 95 tooth sprocket with bore and keyway @ R 35590.00

I tried to call you to get more detail on your drive and possibly give assistance in selection an
simplification / manipulation to use the most available and economical options.

Please feel free to contact me to confirm requirements, and have an official quote sent.

Regards,

Naigel Pera
Manager - Chain
Bearings International, a Division of Hudaco Trading Pty Ltd
Tel : 011 899 0154
Fax2mail : 086 5672916
Mobile : 083 388 8695
* Private Bag 9, Elandsfontein 1406
www.bearings.co.za

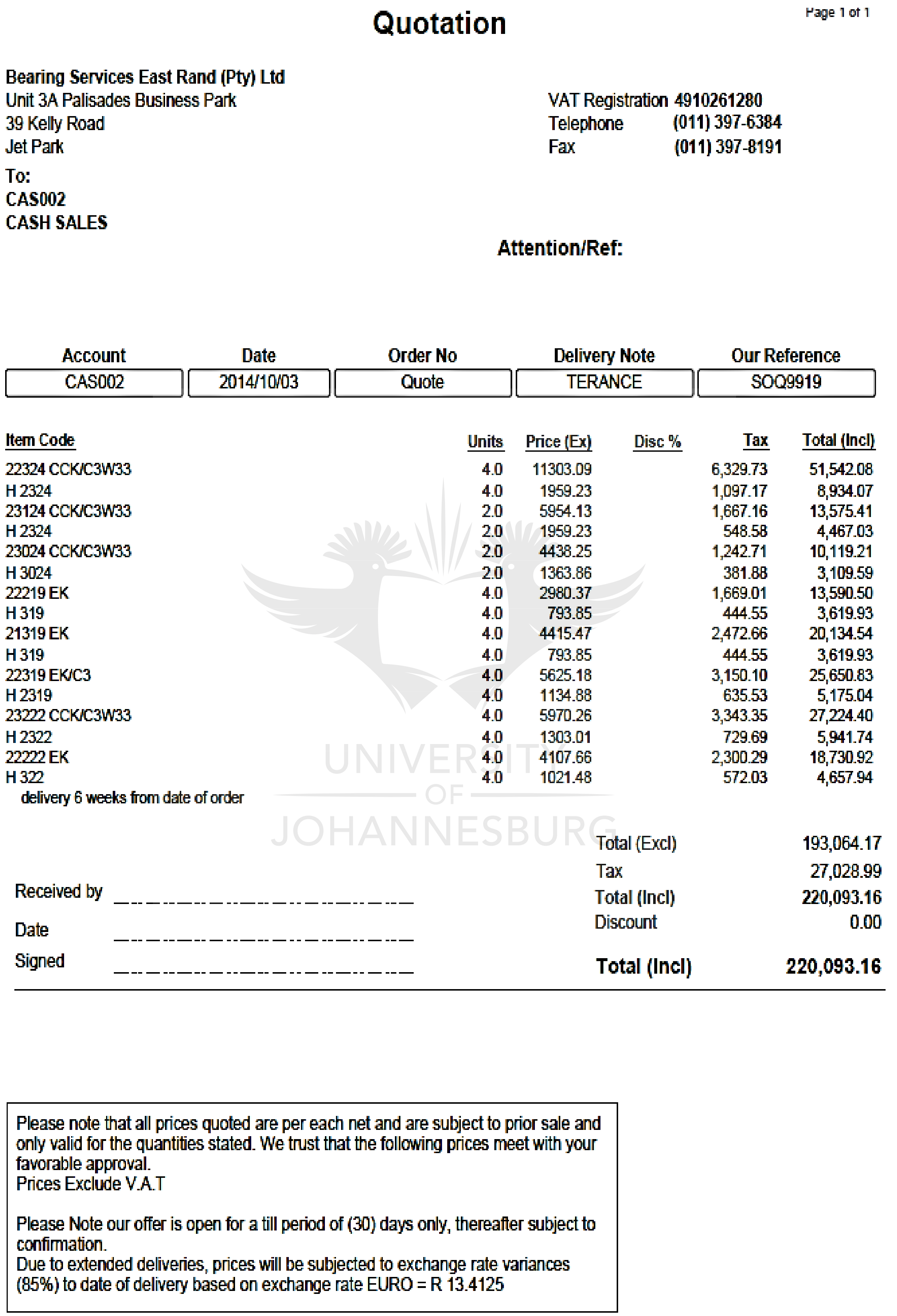
Bearings International
Your Solutions Partner

Bearings International currently rank as a Level 2 Contributor to BBBEE,
which enables our customers to claim 125% of each Rand spent with us as Black spend.

ISO 9001-2008	Quality System
ISO 14001-2004	Environmental Management
OHSAS 18001-2007	Occupational Health and Safety System
ISO 26000	Social Responsibility

After Hours Hotline : **083 250 9191** for all emergencies

SKF (Bearings Quote)



Page 1 of 1

Quotation

Bearing Services East Rand (Pty) Ltd
Unit 3A Palisades Business Park
39 Kelly Road
Jet Park

VAT Registration **4910261280**
Telephone **(011) 397-6384**
Fax **(011) 397-8191**

To:
CAS002
CASH SALES

Attention/Ref:

Account	Date	Order No	Delivery Note	Our Reference
CAS002	2014/10/03	Quote	TERANCE	SOQ9919

Item Code	Units	Price (Ex)	Disc %	Tax	Total (Incl)
22324 CCK/C3W33	4.0	11303.09		6,329.73	51,542.08
H 2324	4.0	1959.23		1,097.17	8,934.07
23124 CCK/C3W33	2.0	5954.13		1,667.16	13,575.41
H 2324	2.0	1959.23		548.58	4,467.03
23024 CCK/C3W33	2.0	4438.25		1,242.71	10,119.21
H 3024	2.0	1363.86		381.88	3,109.59
22219 EK	4.0	2980.37		1,669.01	13,590.50
H 319	4.0	793.85		444.55	3,619.93
21319 EK	4.0	4415.47		2,472.66	20,134.54
H 319	4.0	793.85		444.55	3,619.93
22319 EK/C3	4.0	5625.18		3,150.10	25,650.83
H 2319	4.0	1134.88		635.53	5,175.04
23222 CCK/C3W33	4.0	5970.26		3,343.35	27,224.40
H 2322	4.0	1303.01		729.69	5,941.74
22222 EK	4.0	4107.66		2,300.29	18,730.92
H 322	4.0	1021.48		572.03	4,657.94

delivery 6 weeks from date of order

Total (Excl)	193,064.17
Tax	27,028.99
Total (Incl)	220,093.16
Discount	0.00
Total (Incl)	**220,093.16**

Received by ______________________

Date ______________________

Signed ______________________

Please note that all prices quoted are per each net and are subject to prior sale and only valid for the quantities stated. We trust that the following prices meet with your favorable approval.
Prices Exclude V.A.T

Please Note our offer is open for a till period of (30) days only, thereafter subject to confirmation.
Due to extended deliveries, prices will be subjected to exchange rate variances (85%) to date of delivery based on exchange rate EURO = R 13.4125

Appendix 48: Communication.

Drive shaft output power

From: Miller, Terence [mailto:tmiller@uj.ac.za]

Sent: 15 April 2013 03:08 PM

To: Higino Chris GRM ZFPO

Subject: T7232 Specs.

Hi Chris

I spoke to you on Thursday regarding the T7232 transmission.
Can you supply me with the following:

PTO Output power:
PTO Output shaft diameter(Crown tip)
PTO Output shaft diameter(Crown root)
PTO torque

What type of prop shaft do you recommend for this type of drive (i.e.power, torque etc.)?
Can the PTO reverse the rotation of the drive shaft and at what speed.

Thank you very much.

Kind Regards
Terence Miller

From: Chris.Higino@zf.com [mailto:Chris.Higino@zf.com]
Sent: 29 April 2013 12:03 PM
To: Miller, Terence
Subject: RE: T7232 Specs.

Hello Terence

Please see attached documents and remarks:

PTO Output power:
(PTO torque)
Transmission input power T-7232 PVS: 134kW (2100r/min engine)
PTO Output power approx. 120kW (2100r/min engine)

PTO Output shaft diameter(Crown tip)
PTO Output shaft diameter(Crown root)
For the T-7232 PVS transmission are four different shafts available.

According ISO 500 are three shafts available:
Standard PTO output shaft:
Type I = 1 3/8” 6 teeth straight splines
Optional PTO output shaft:
Type II = 1 3/8” 21 teeth involute splines
Type III = 1 ¾” 20 teeth involute splines

Recommended PTO power for PTO output shaft according ISO 500-1.
Dimensions PTO output shaft according ISO 500-3.

According SAE is one shaft available:
Optional PTO output shaft:
Profil 1= 1 ¾” 6 teeth straight splines SAE 6C similar to standard 6 B

US-PTO:
For the T-7232 PVS is also an US-PTO version available. The different between the Standard and the US-Version is an additional hall sensor for detecting what kind of shaft is mounted. For the US-PTO are the Type I and II available. Both shafts have an different machining in the area of the hall sensor.

What type of prop shaft do you recommend for this type of drive(i.e.power, torque etc.)?
ISO 500 recommends in the power range of 115 to 275kW Type II shaft. But the common shaft for the most tractor implements is Type I these days. For that reason the standard on a T-7232 PVS transmission is Type I shaft.

Can the PTO reverse the rotation of the drive shaft and at what speed
Rotation in reverse isn´t possible.

I hope that this information is what you are looking for
Please be free to contact me should you have any further questions

Mit freundlichen Grüßen/Kind regards,

Chris Higino
Technical Representative: Off-Road Vehicles

ZF Services South Africa (Pty) Ltd
170 Herman Street, Meadowdale, Ext. 3, Germiston
P.O. Box 2098, Kempton Park, 1620
Phone +27 11 457-0000, Fax +086 750 8580, Mobile+ 27 72 205-9234
chris.higino@zf.com

Appendix 49: Stress Analysis.

All stress analysis done in this section is done on the basis of yield stress of which all is well below the yield point. This is to accommodate fatigue that is caused by the cyclical stresses caused by vibration in the various components.

All stress analysis reports are developed by using the Ansys package in Autodesk Inventor 2013.

1 Compactor Roller Drum Stress Analysis Report

Project

Part Number	Compacting Drum
Designer	Terence Miller

Status

Design Status	Work In Progress

Physical

Material	Steel, High Strength Low Alloy
Density	7.84 g/cm^3
Mass	2123.57 kg
Area	23020100 mm^2
Volume	270864000 mm^3
Centre of Gravity	x=1122.42 mm y=-1138.16 mm z=0 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

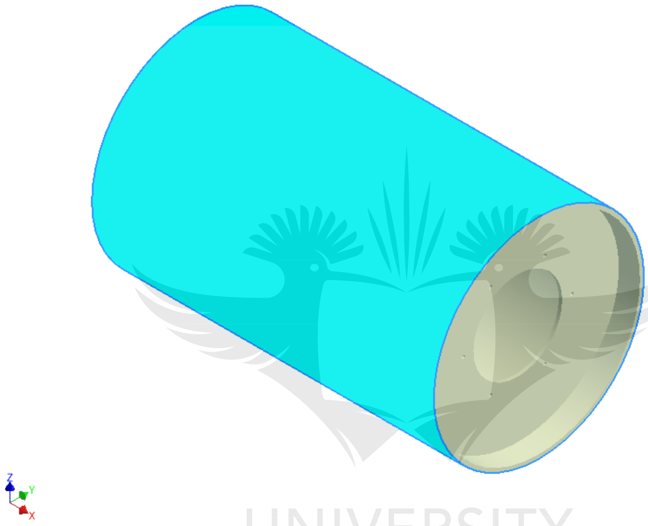
Material(s)

Name	Steel, High Strength, Low Alloy	
General	Mass Density	7.84 g/cm^3
	Yield Strength	275.8 MPa
	Ultimate Tensile Strength	448 MPa
Stress	Young's Modulus	200 GPa
	Poisson's Ratio	0.287 ul
	Shear Modulus	77.7001 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	47 W/(m K)
	Specific Heat	420 J/(kg c)
Part Name(s)	Compacting Drum	

Force:1

Load Type	Force
Magnitude	229300.000 N
Vector X	0.000 N
Vector Y	229300.000 N
Vector Z	0.000 N

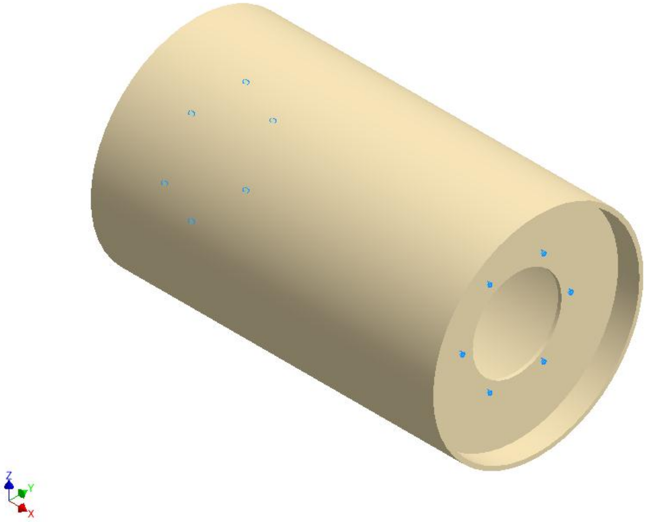
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

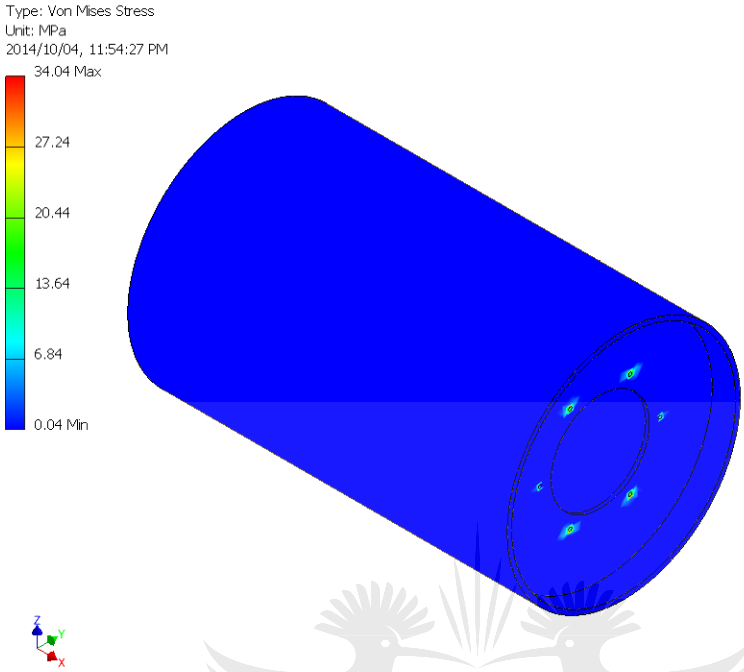
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	229300 N	0 N	0 N m	0 N m
		-229300 N		0 N m
		0 N		0 N m

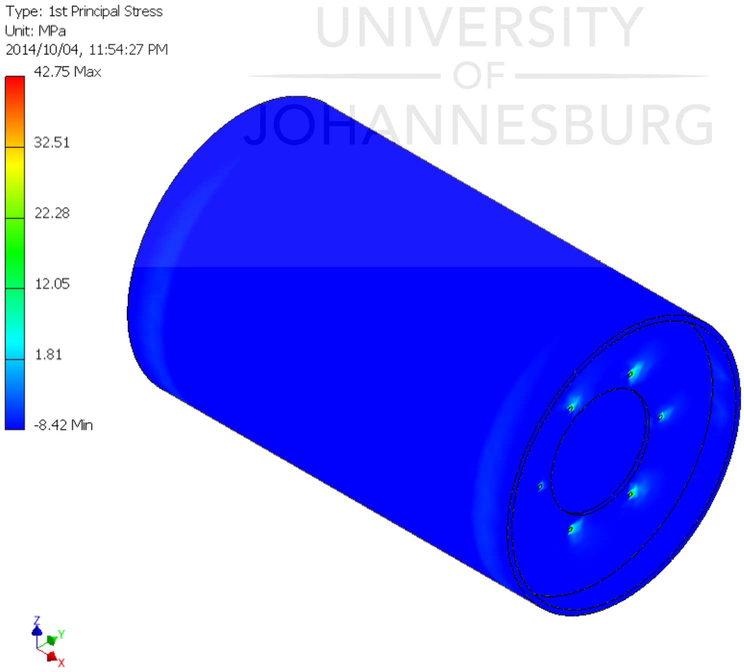
Result Summary

Name	Minimum	Maximum
Volume	270864000 mm^3	
Mass	2123.57 kg	
Von Mises Stress	0.0428494 MPa	34.0421 MPa
1st Principal Stress	-8.42339 MPa	42.7478 MPa
3rd Principal Stress	-42.7408 MPa	8.40683 MPa
Displacement	0 mm	0.0280554 mm
Safety Factor	8.10174 ul	15 ul

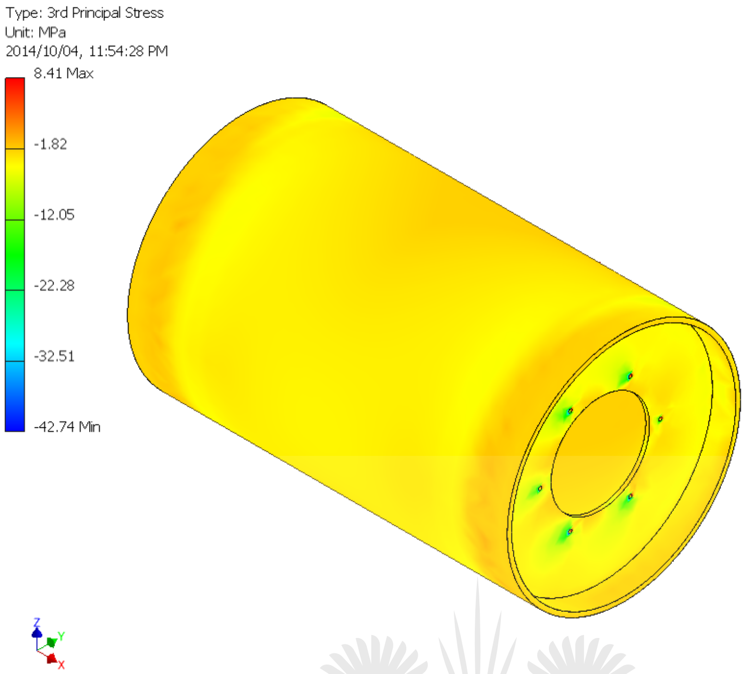
Von Mises Stress



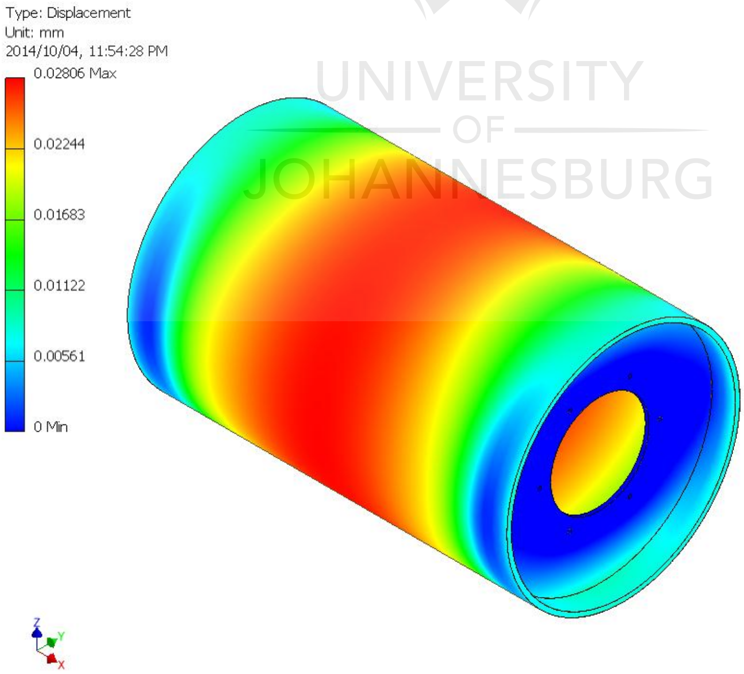
1st Principal Stress



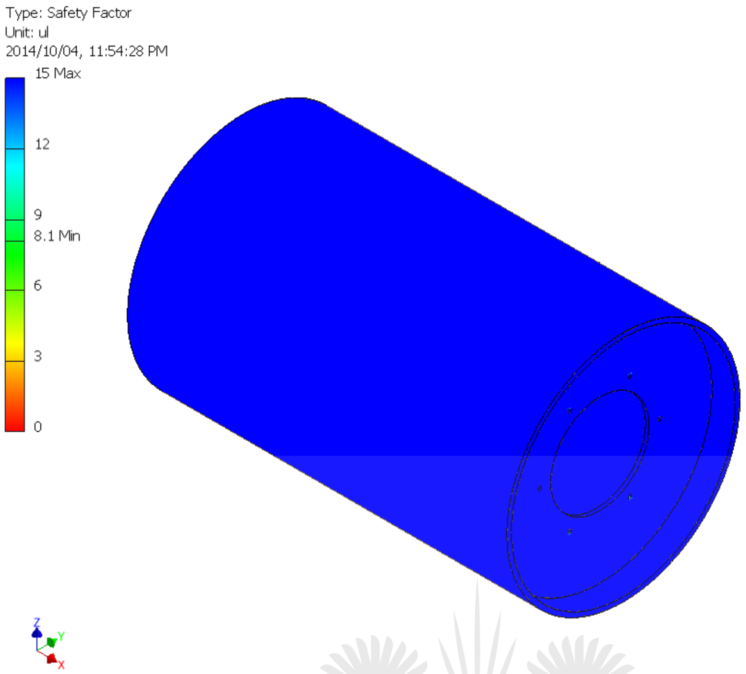
3rd Principal Stress



Displacement



Safety Factor



Actual factor of safety for Hardox 500 BHN = 1250/42.75 = 29.24

2. Eccentric Weight

Project

Part Number	Eccentric Weight
Designer	Terence Miller
Cost	R 0,00

Status

Design Status	Work In Progress

Physical

Material	Steel
Density	7.85 g/cm^3
Mass	25.7095 kg
Area	297295 mm^2
Volume	3275100 mm^3
Centre of Gravity	x=0 mm y=85.0942 mm z=-0.213741 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014/10/30, 02:35 AM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

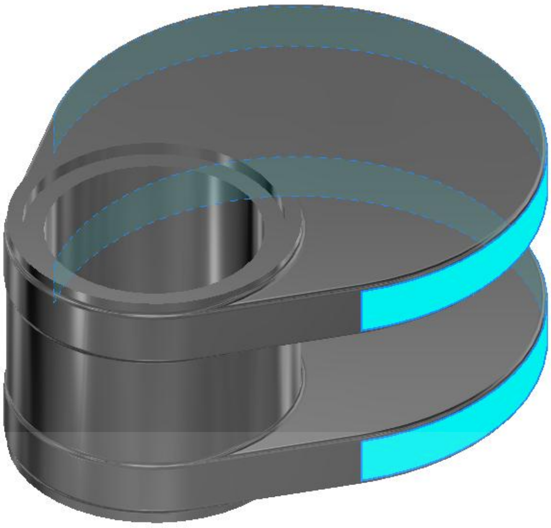
Material(s)

Name	Steel	
General	Mass Density	7.85 g/cm^3
	Yield Strength	207 MPa
	Ultimate Tensile Strength	345 MPa
Stress	Young's Modulus	210 GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	80.7692 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	56 W/(m K)
	Specific Heat	460 J/(kg c)
Part Name(s)	Eccentric Weight A	

Force:1

Load Type	Force
Magnitude	228300.000 N
Vector X	0.000 N
Vector Y	228300.000 N
Vector Z	0.000 N

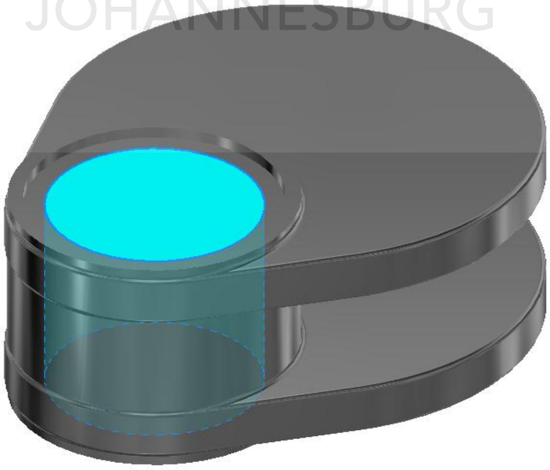
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

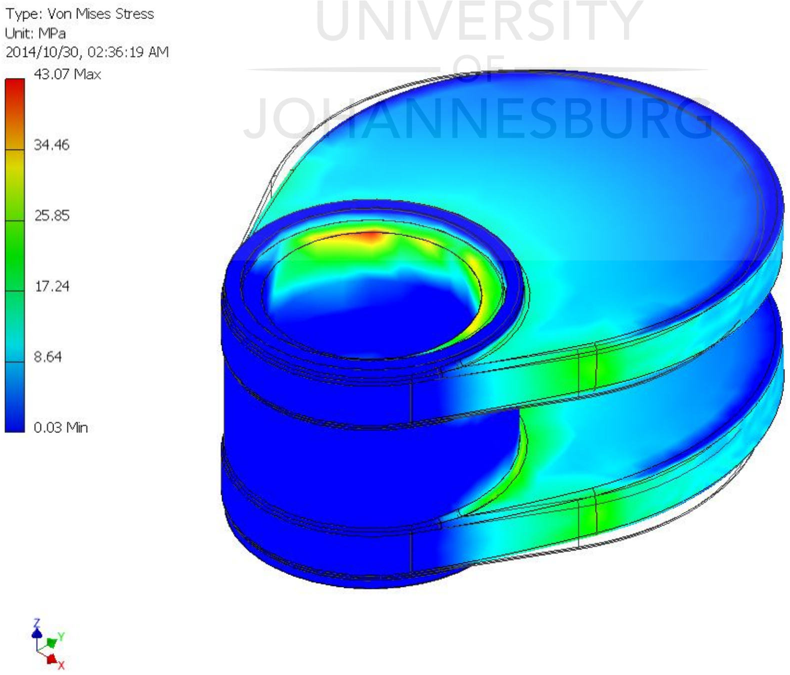
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Pin Constraint:1	114151 N	0 N	28.6031 N m	-28.6031 N m
		-114151 N		0 N m
		0 N		0 N m
Fixed Constraint:1	114151 N	0 N	28.6031 N m	-28.6031 N m
		-114151 N		0 N m
		0 N		0 N m

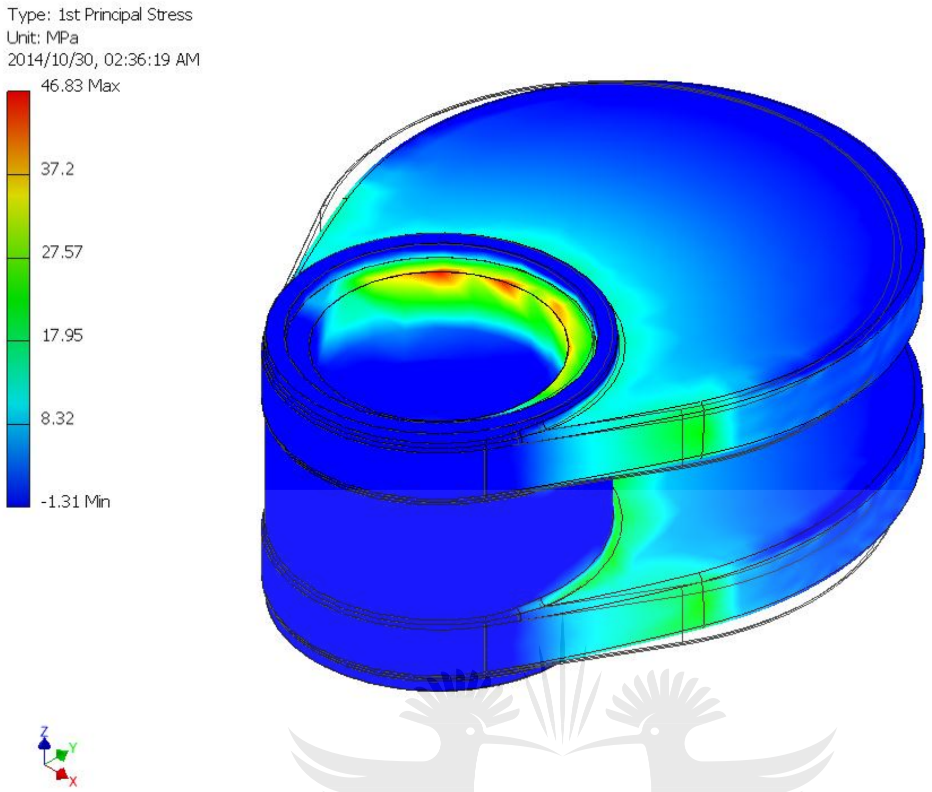
Result Summary

Name	Minimum	Maximum
Volume	3275090 mm^3	
Mass	25.7094 kg	
Von Mises Stress	0.0259774 MPa	43.0732 MPa
1st Principal Stress	-1.30559 MPa	46.8275 MPa
3rd Principal Stress	-13.3327 MPa	10.2008 MPa
Displacement	0 mm	0.0140495 mm
Safety Factor	4.80577 ul	15 ul

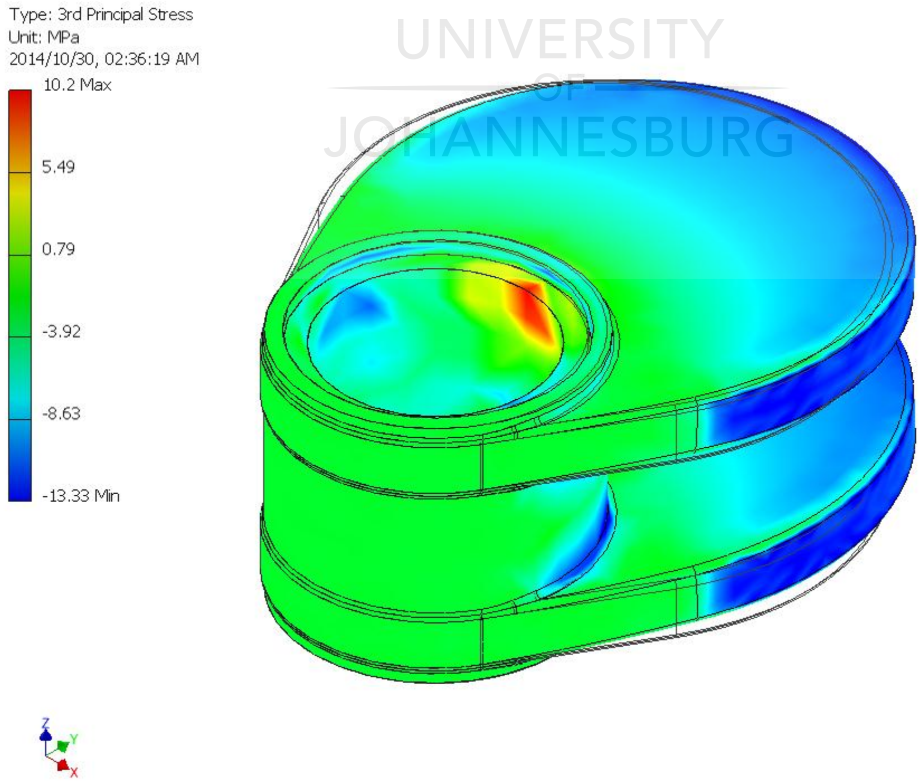
Von Mises Stress



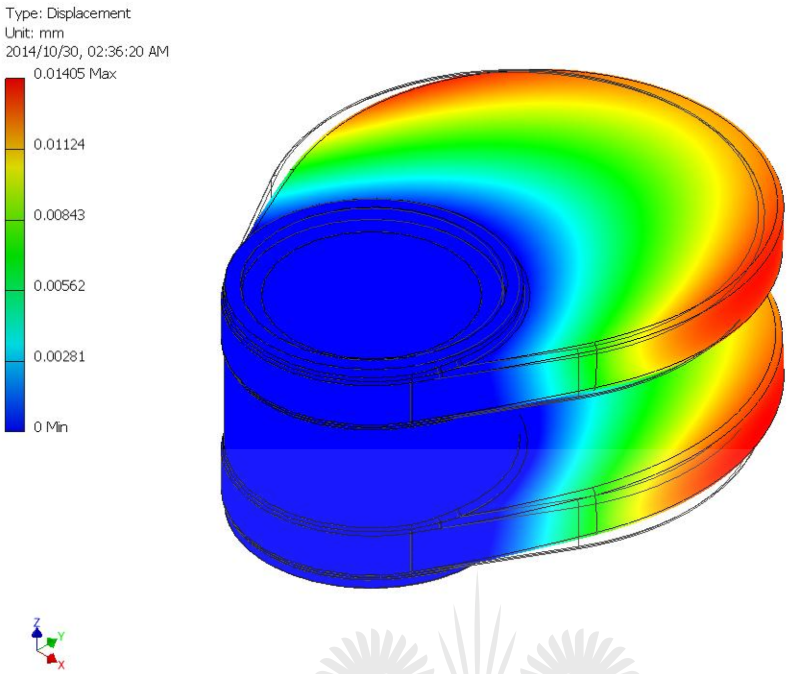
1st Principal Stress



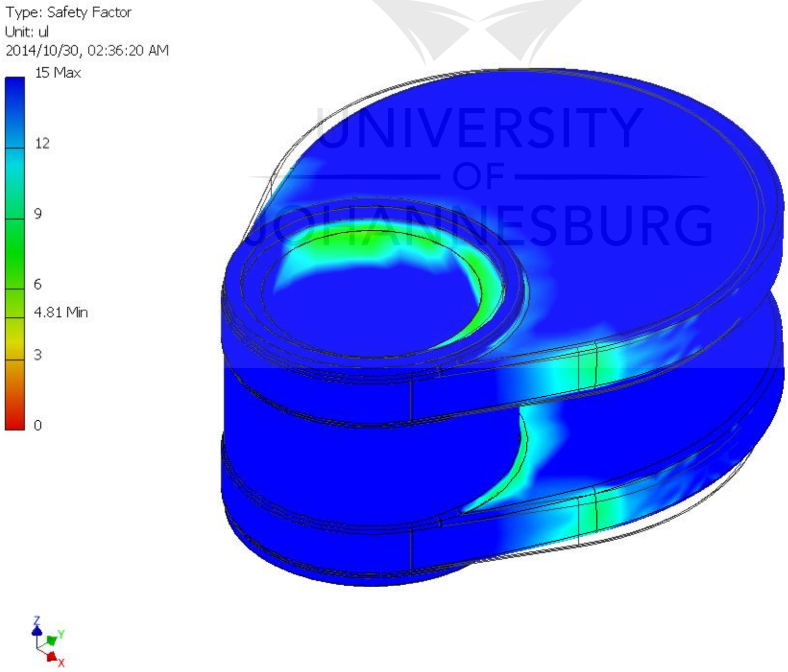
3rd Principal Stress



Displacement



Safety Factor



NB: The safety factor of 4.81 is an acceptable value**.**

3. Bearing Housing Flange Half

Summary

Author	Terence Miller

Project

Part Number	Bearing Housing Flange Half
Designer	Terence miller
Cost	R 0,00

Status

Design Status	Work In Progress

Physical

Material	Iron, Cast
Density	7.25 g/cm^3
Mass	92.9851 kg
Area	1455480 mm^2
Volume	12825500 mm^3
Centre of Gravity	x=0 mm y=0 mm z=79.4093 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014/11/26, 03:36 AM
Detect and Eliminate Rigid Body Modes	No

Mesh settings

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

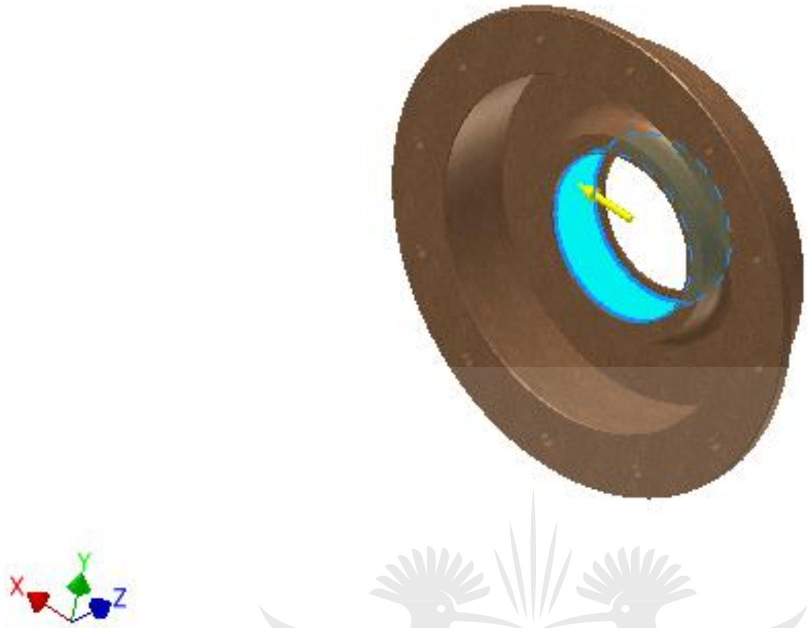
Name	Iron, Cast	
General	Mass Density	7.25 g/cm^3
	Yield Strength	200 MPa
	Ultimate Tensile Strength	276 MPa
Stress	Young's Modulus	120.5 GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	46.3462 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	50 W/(m K)
	Specific Heat	540 J/(kg c)
Part Name(s)	Bearing Mounting & Housing Flange	

Operating conditions

Bearing Load:1

1	Bearing Load
Magnitude	113546.000 N
Vector X	113546.000 N
Vector Y	0.000 N
Vector Z	0.000 N

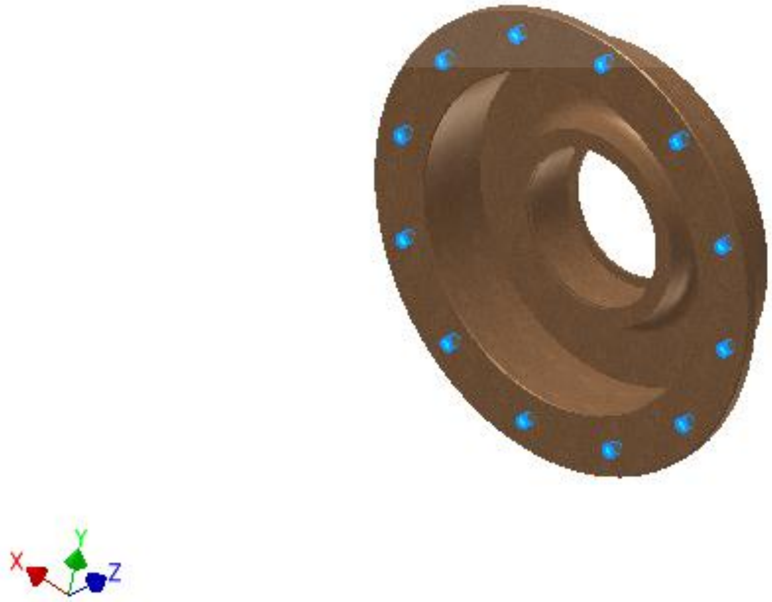
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

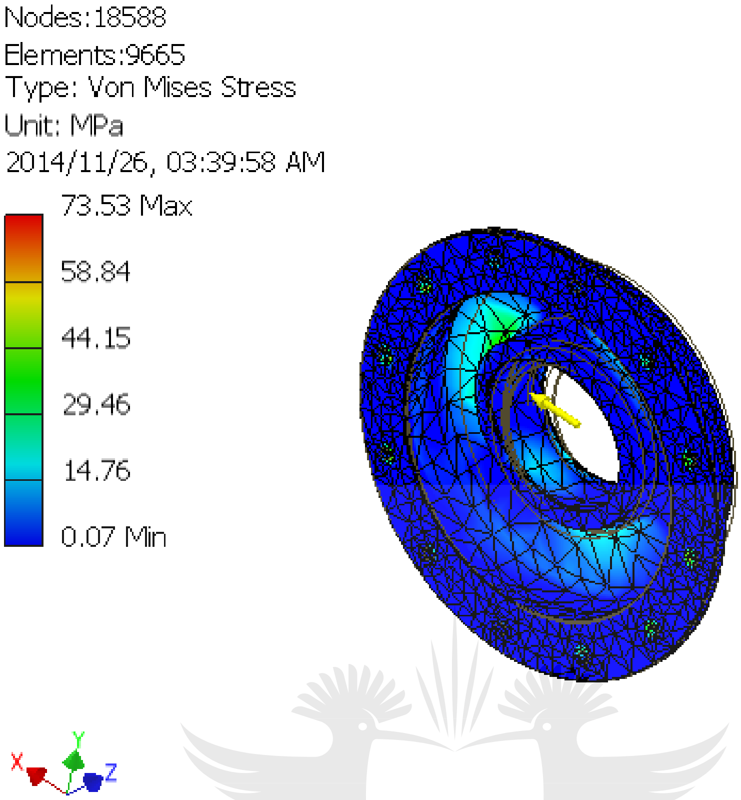
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	113546 N	-113546 N	12660.5 N m	0 N m
		0 N		-12660.5 N m
		0 N		0 N m

Result Summary

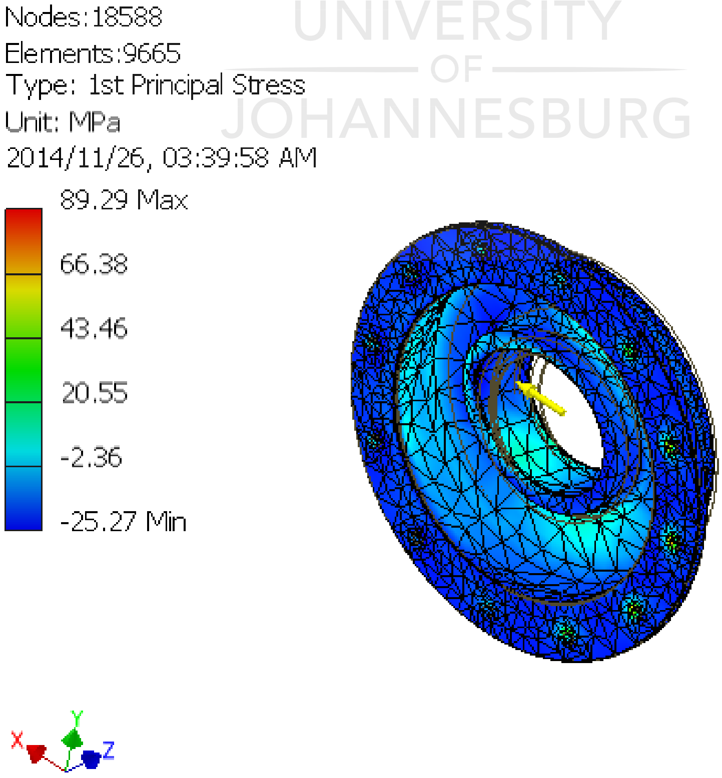
Name	Minimum	Maximum
Volume	12825500 mm^3	
Mass	92.9851 kg	
Von Mises Stress	0.0716276 MPa	73.5311 MPa
1st Principal Stress	-25.2722 MPa	89.2889 MPa
3rd Principal Stress	-99.306 MPa	21.1493 MPa
Displacement	0 mm	0.160846 mm
Safety Factor	2.71994 ul	15 ul

Von Mises Stress



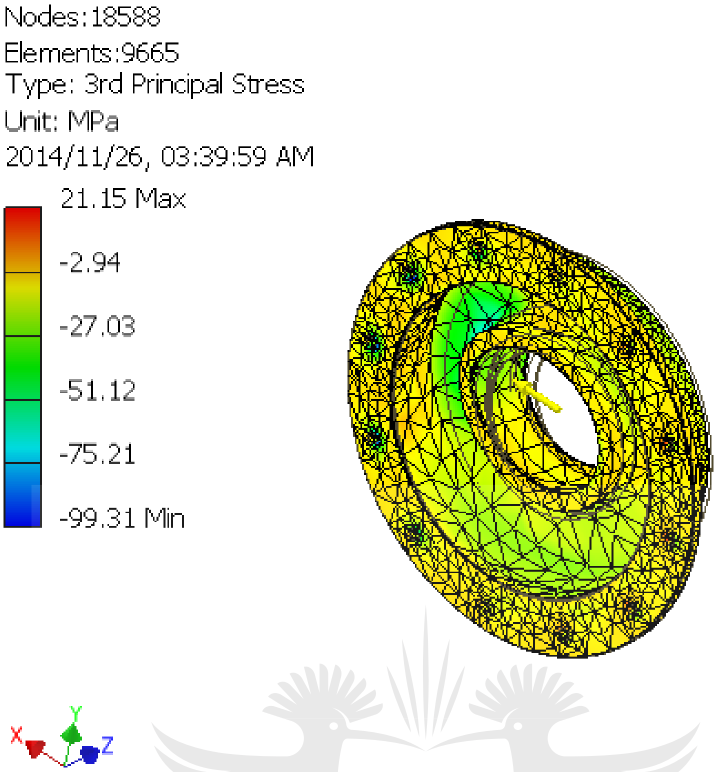
Nodes:18588
Elements:9665
Type: Von Mises Stress
Unit: MPa
2014/11/26, 03:39:58 AM

1st Principal Stress

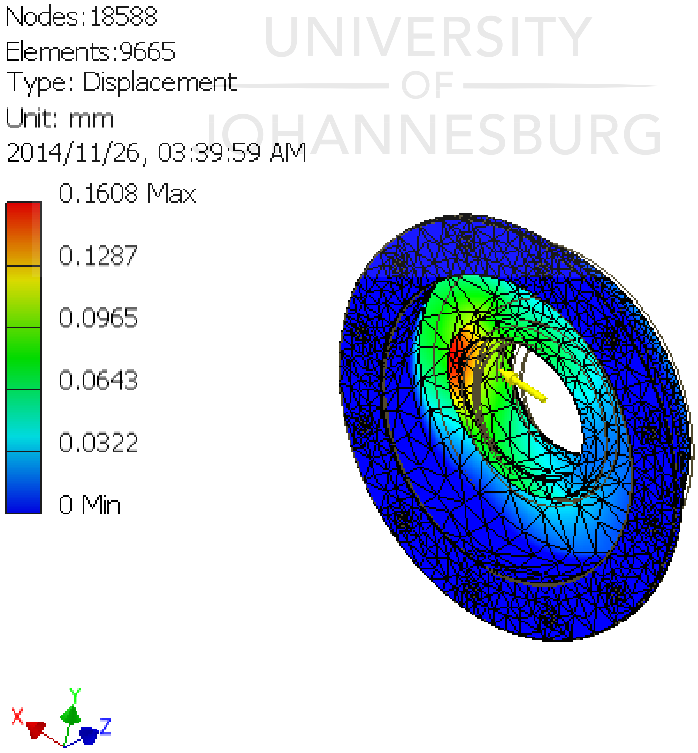


Nodes:18588
Elements:9665
Type: 1st Principal Stress
Unit: MPa
2014/11/26, 03:39:58 AM

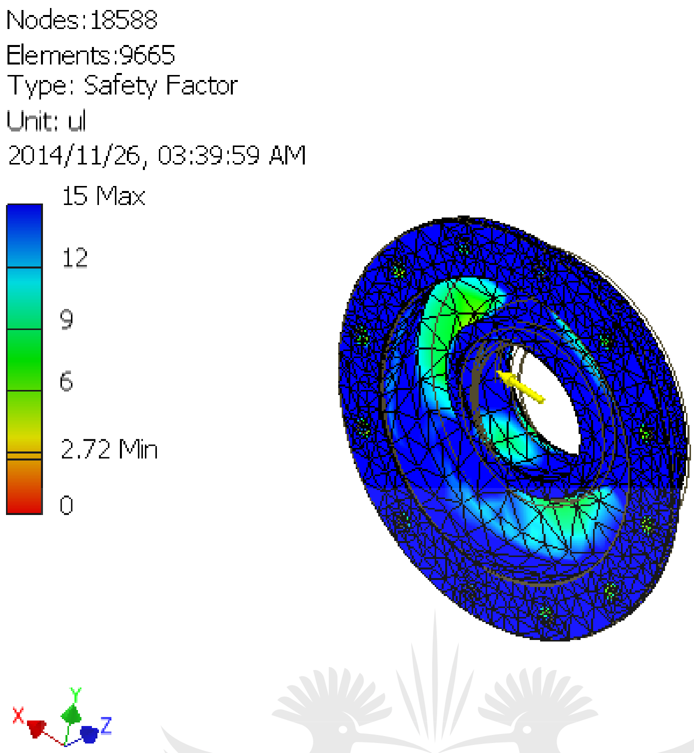
3rd Principal Stress



Displacement



Safety Factor



The Factor of safety is 2.72 minimum when using cast iron. This is the cheapest option to very large complex shapes, machining will take a lot of time and labour.

4. Frame

Summary

Author	Terence Miller

Project

Part Number	Frame
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

Physical

Material	Weldox 500
Density	1 g/cm^3
Mass	106.752 kg
Area	10576400 mm^2
Volume	106752000 mm^3
Centre of Gravity	x=-739.027 mm y=-512.804 mm z=204.059 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014/10/30, 12:52 AM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

Name	Weldox 500	
General	Mass Density	1 g/cm^3
	Yield Strength	500 MPa
	Ultimate Tensile Strength	0 MPa
Stress	Young's Modulus	207 GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	79.6154 GPa
Stress Thermal	Expansion Coefficient	0.0000001 ul/c
	Thermal Conductivity	0.001 W/(m K)

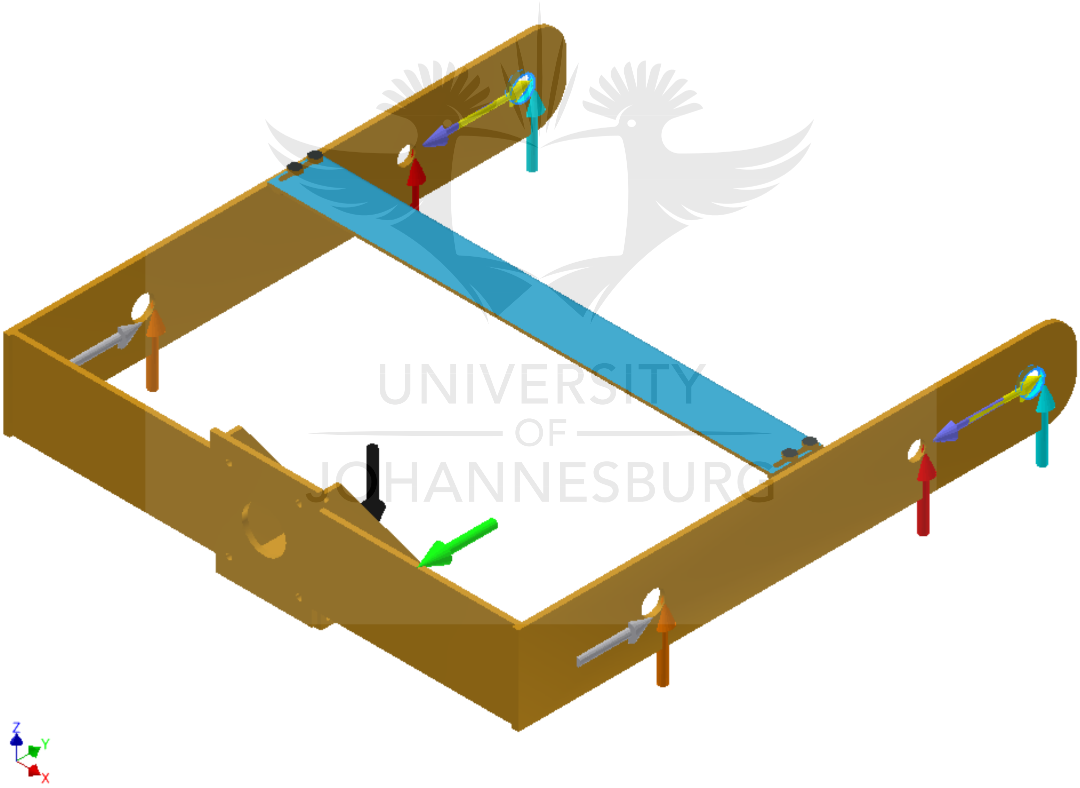
	Specific Heat	100 J/(kg c)
Part Name(s)	Frame	

Operating conditions

Force:1

Load Type	Force
Magnitude	5000.000 N
Vector X	0.000 N
Vector Y	5000.000 N
Vector Z	0.000 N

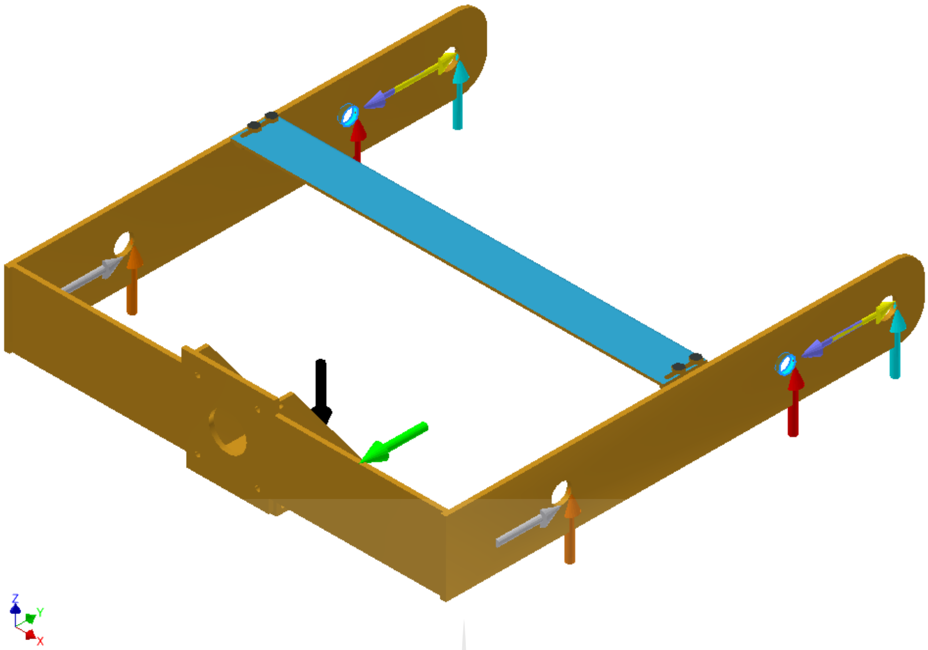
Selected Face(s)



Force:2

Load Type	Force
Magnitude	4000.000 N
Vector X	0.000 N
Vector Y	-4000.000 N
Vector Z	0.000 N

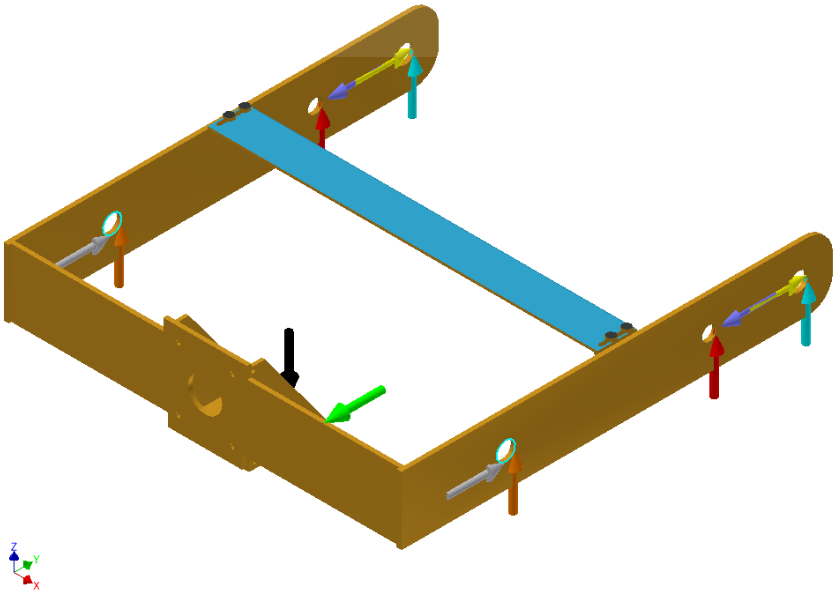
Selected Face(s)



Force:3

Load Type	Force
Magnitude	8300.000 N
Vector X	0.000 N
Vector Y	8300.000 N
Vector Z	0.000 N

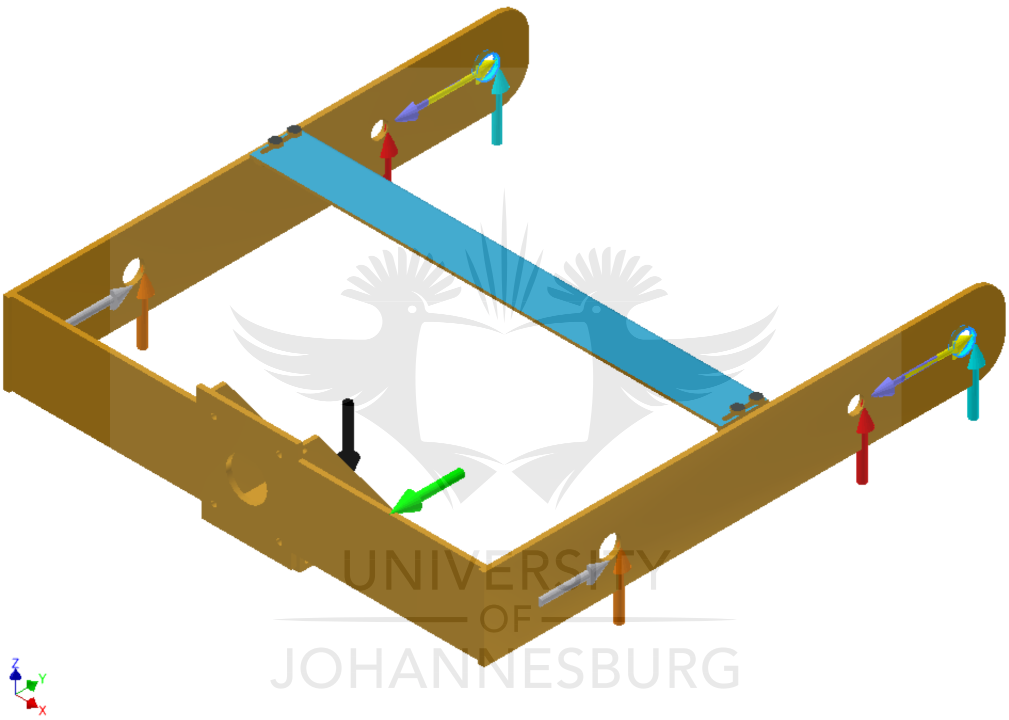
Selected Face(s)



Force:4

Load Type	Force
Magnitude	38055.000 N
Vector X	0.000 N
Vector Y	0.000 N
Vector Z	38055.000 N

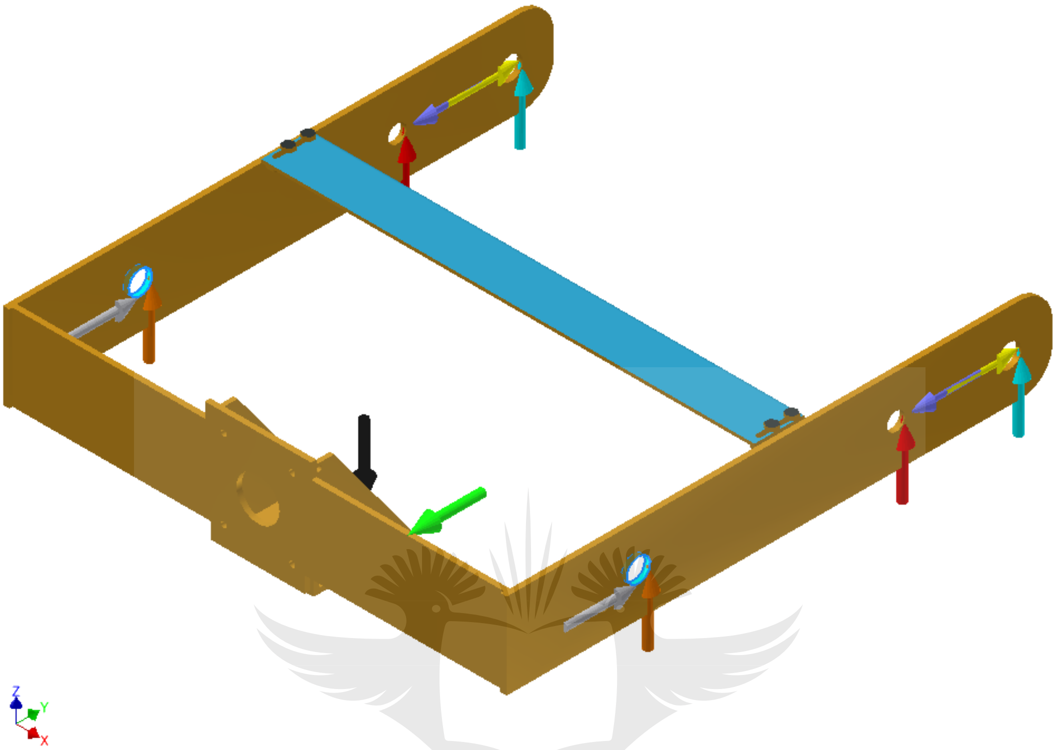
Selected Face(s)



Force:5

Load Type	Force
Magnitude	23331.600 N
Vector X	0.000 N
Vector Y	0.000 N
Vector Z	23331.600 N

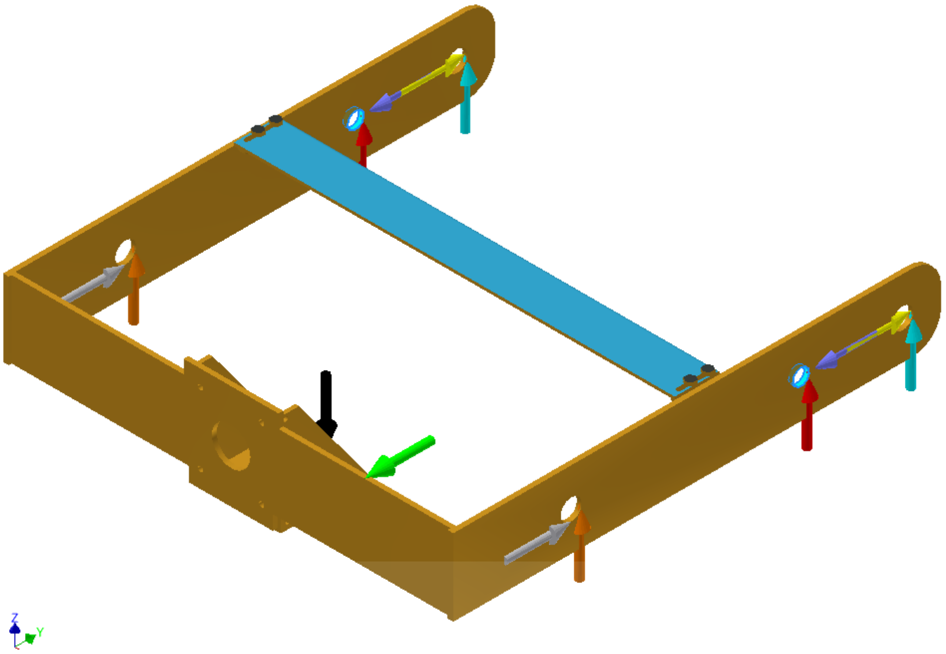
Selected Face(s)



Force:6

Load Type	Force
Magnitude	17499.000 N
Vector X	0.000 N
Vector Y	0.000 N
Vector Z	17499.000 N

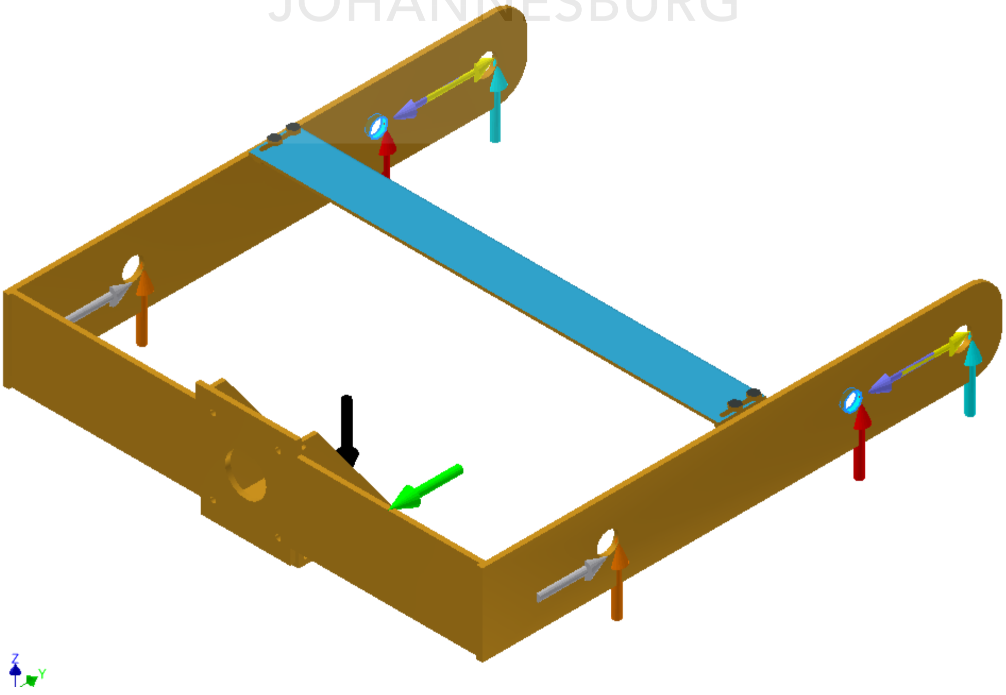
Selected Face(s)



Force:7

Load Type	Force
Magnitude	8300.000 N
Vector X	0.000 N
Vector Y	-8300.000 N
Vector Z	0.000 N

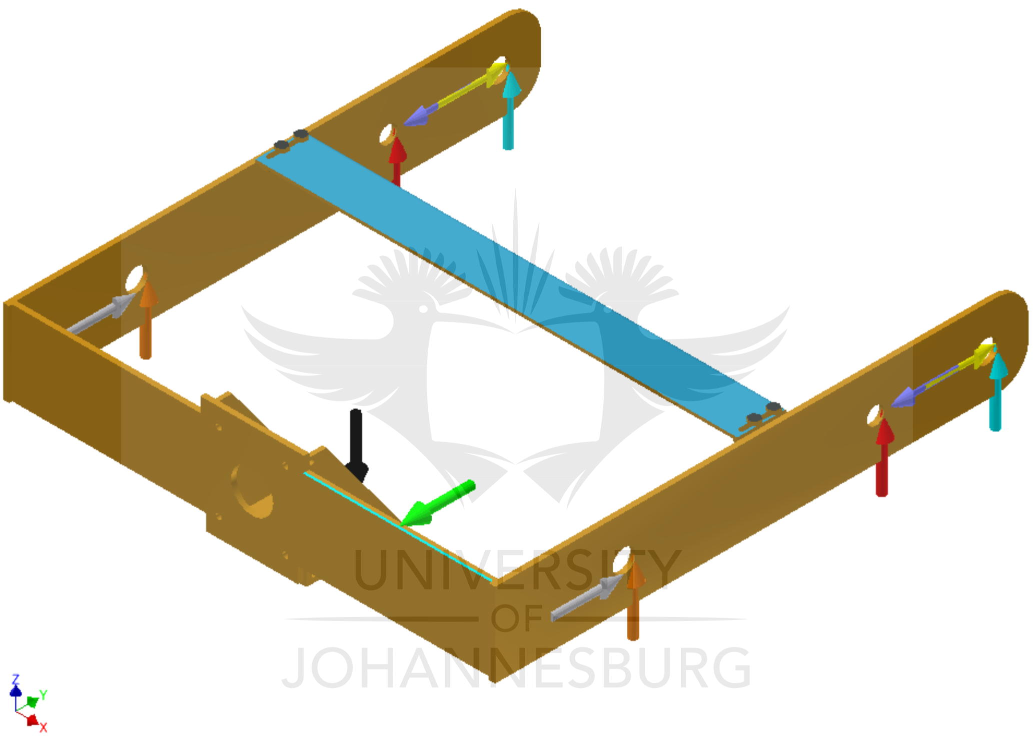
Selected Face(s)



Force:8

Load Type	Force
Magnitude	6250.000 N
Vector X	0.000 N
Vector Y	-6250.000 N
Vector Z	0.000 N

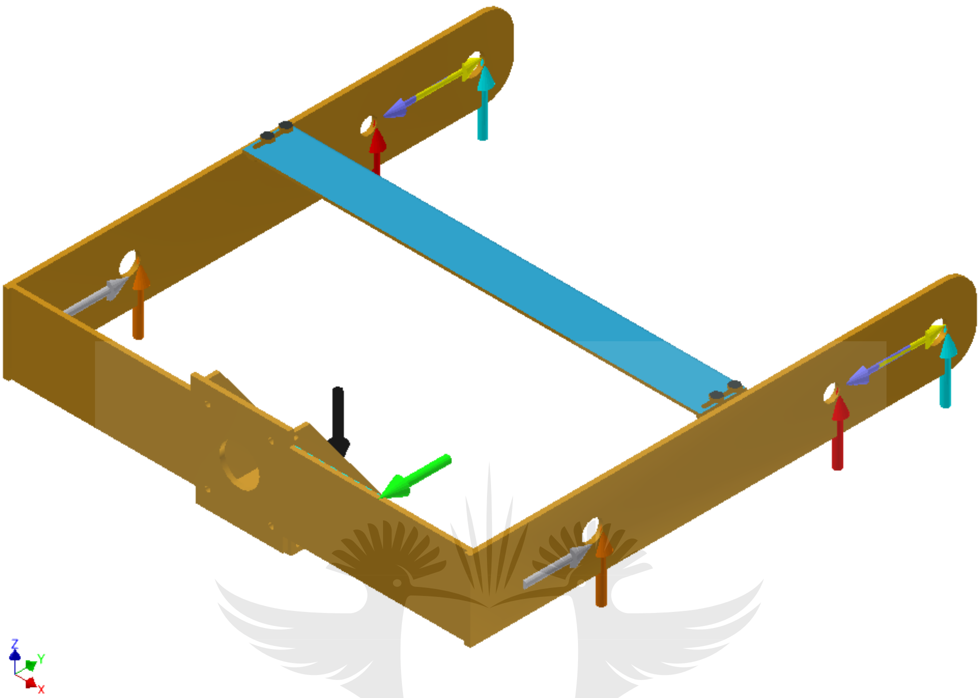
Selected Face(s)



Force:9

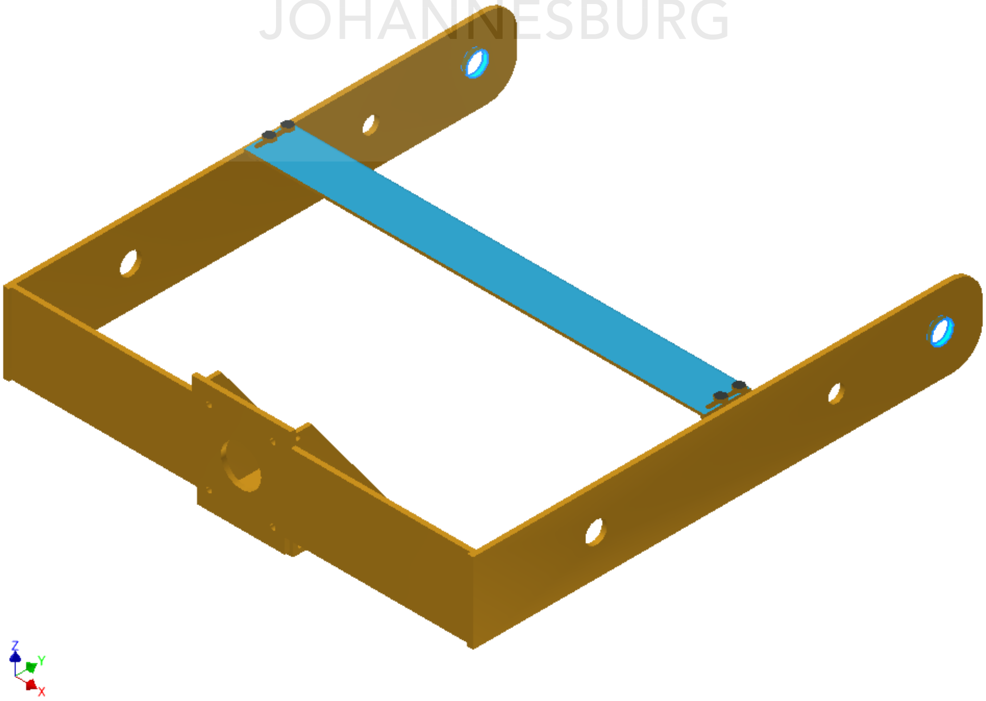
Load Type	Force
Magnitude	8000.000 N
Vector X	0.000 N
Vector Y	0.000 N
Vector Z	-8000.000 N

Selected Face(s)



Fixed Constraint:1

Selected Face(s)



Results

Reaction Force and Moment on Constraints

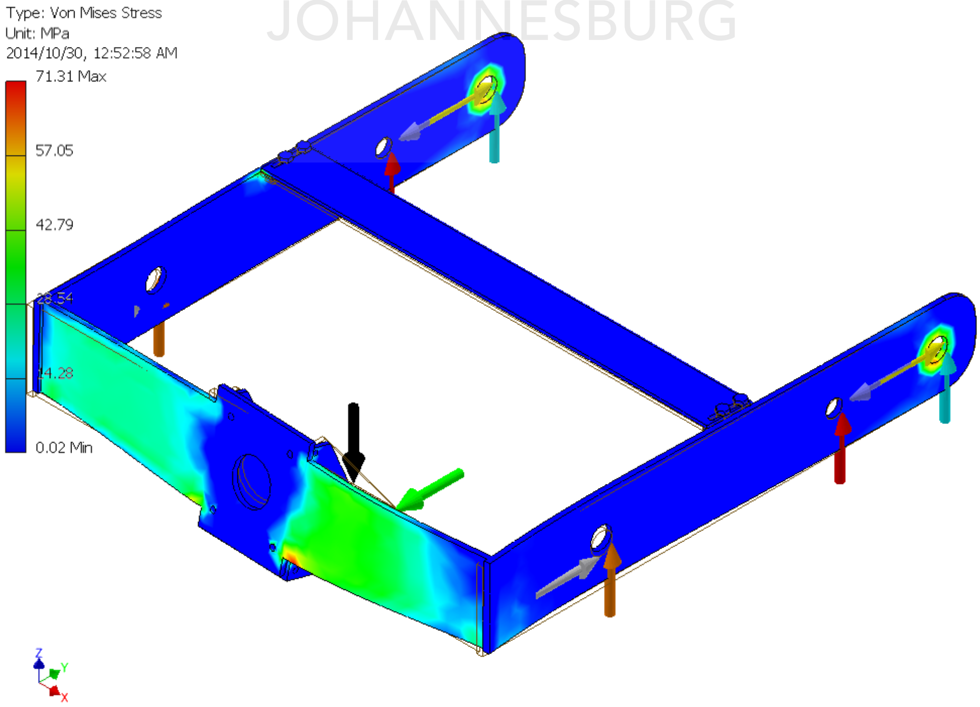
Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	71079.8 N	0 N	33972.3 N m	33708 N m
		5250 N		0 N m
		-70885.6 N		4229.49 N m

Result Summary

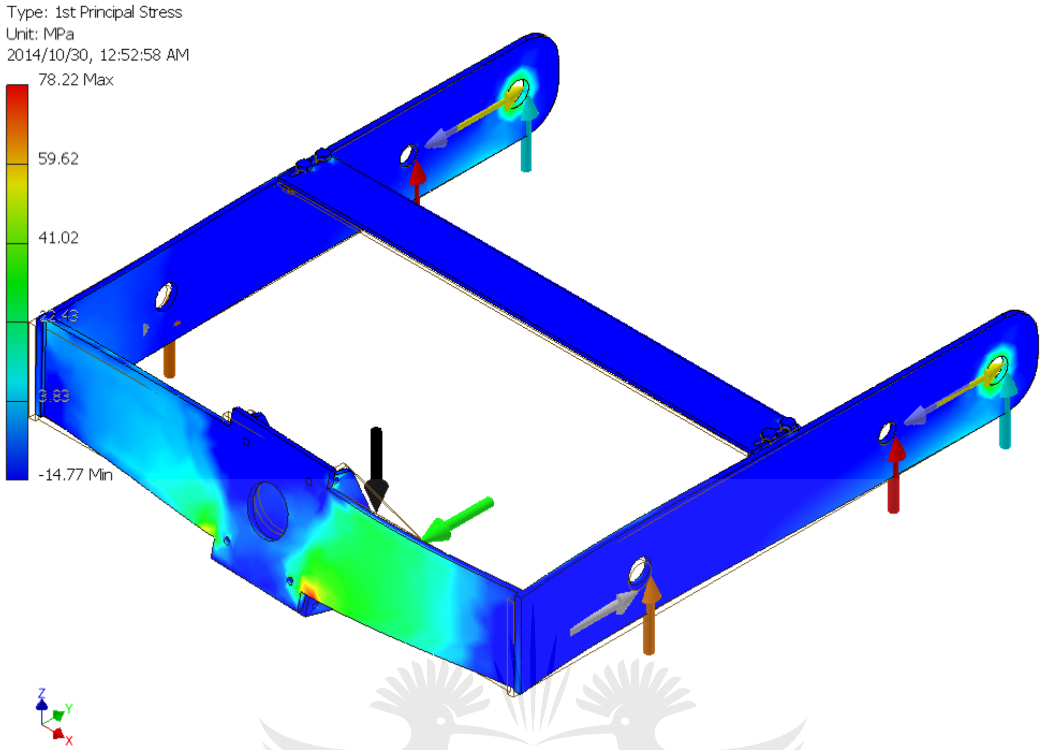
Name	Minimum	Maximum
Mass	106.752 kg	
Von Mises Stress	0.0200071 MPa	71.3093 MPa
1st Principal Stress	-14.7705 MPa	78.2183 MPa
3rd Principal Stress	-62.509 MPa	10.8138 MPa
Displacement	0 mm	6.08212 mm
Safety Factor	7.01171 ul	15 ul

Figures

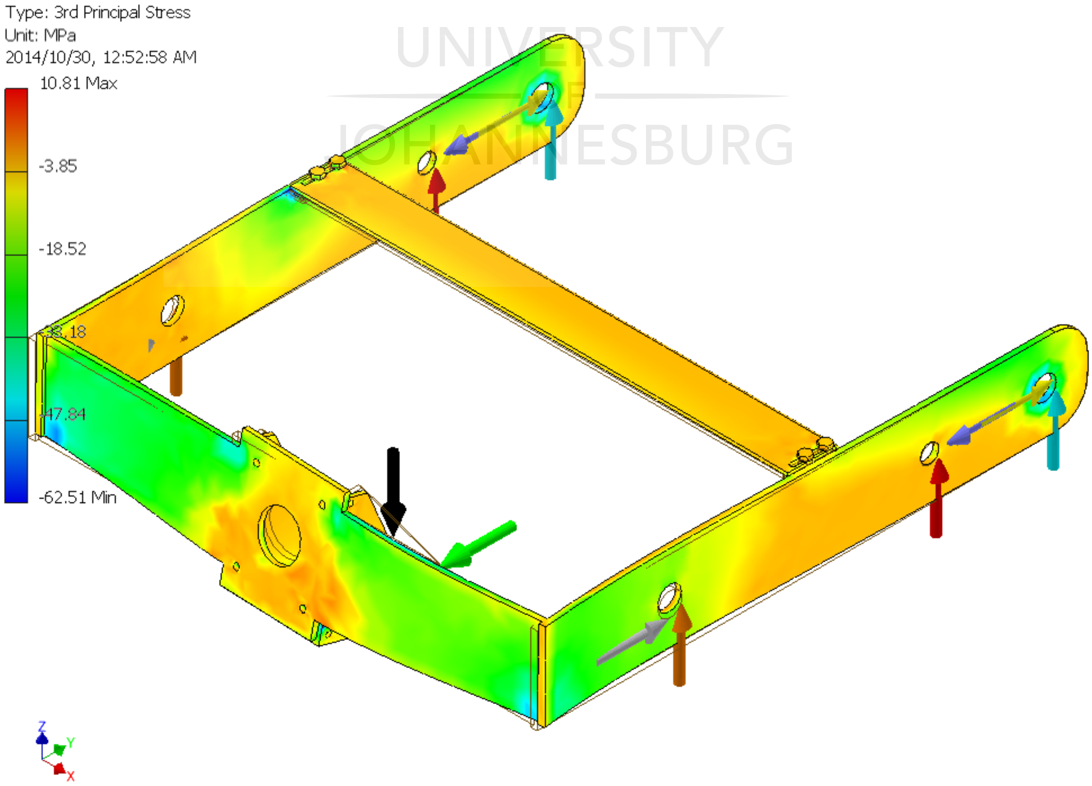
Von Mises Stress



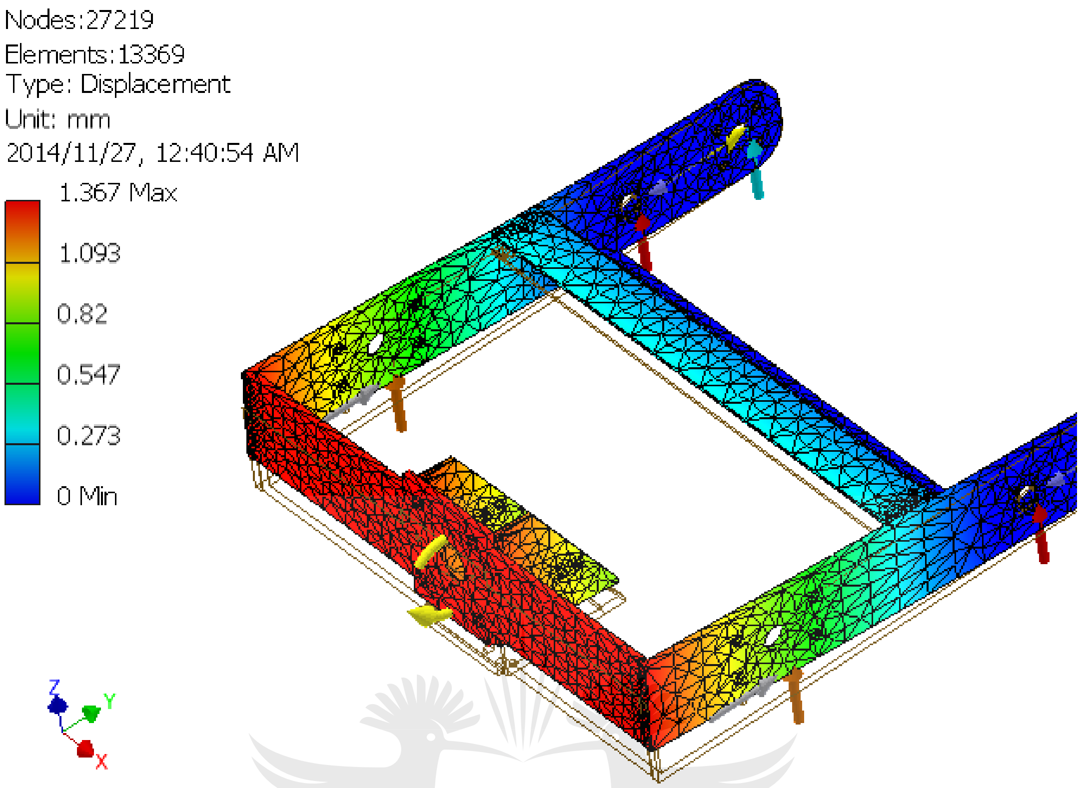
1st Principal Stress



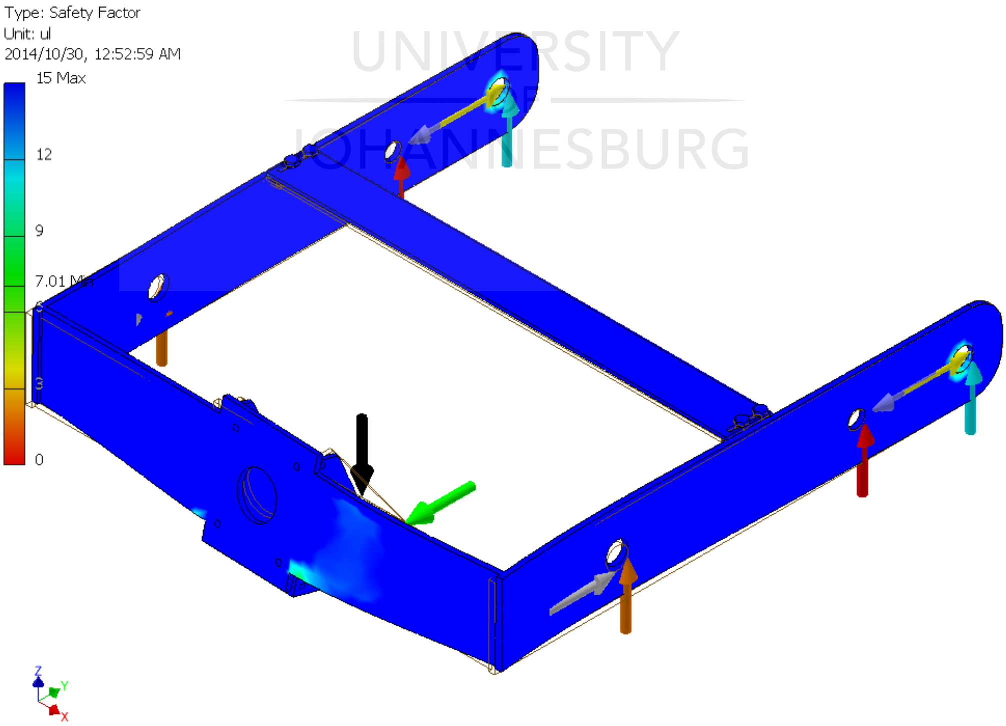
3rd Principal Stress



Displacement



Safety Factor



Final factor of safety is 7 for the frame. This indicates the ability of the frame to withstand vibration.

5 FS frame bearing support bracket.

Project Info (iProperties)

Summary

Author	Terence Miller

Project

Part Number	
Designer	Terence Miller
Cost	R 0,00
Date Created	2014/11/12

Status

Design Status	Work In Progress

Physical

Material	Steel, Carbon
Density	7.87 g/cm^3
Mass	20.4351 kg
Area	386218 mm^2
Volume	2596590 mm^3
Centre of Gravity	x=-587.39 mm y=-101.866 mm z=59.2429 mm

Simulation:1

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014/11/26, 11:34 AM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

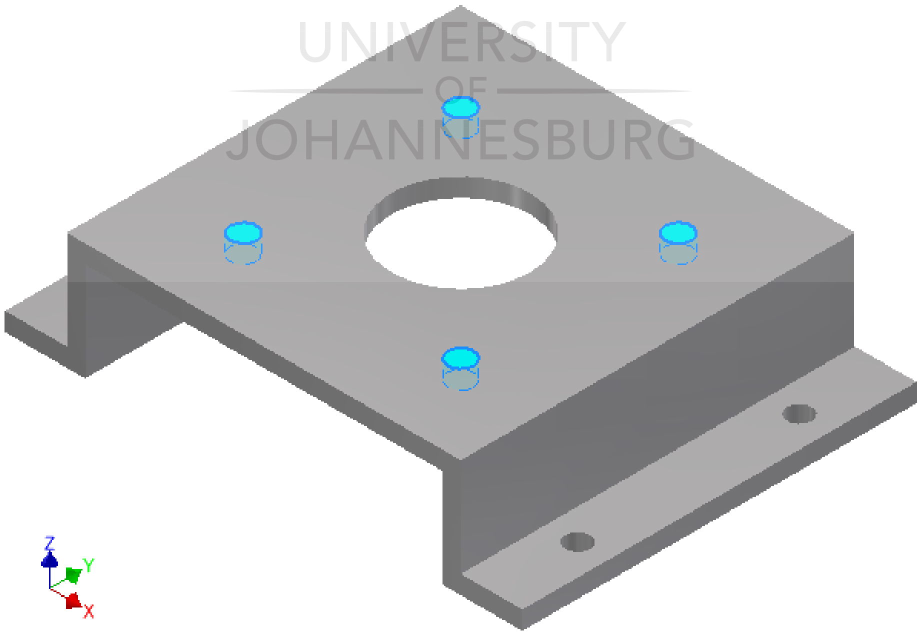
Material(s)

Name	Steel, Carbon	
General	Mass Density	7.87 g/cm^3
	Yield Strength	350 MPa
	Ultimate Tensile Strength	420 MPa
Stress	Young's Modulus	200 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	77.5194 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	52 W/(m K)
	Specific Heat	486 J/(kg c)
Part Name(s)	FS bearing support plate	

Force:1

Load Type	Force
Magnitude	20735.100 N
Vector X	0.000 N
Vector Y	20735.100 N
Vector Z	0.000 N

Selected Face(s)

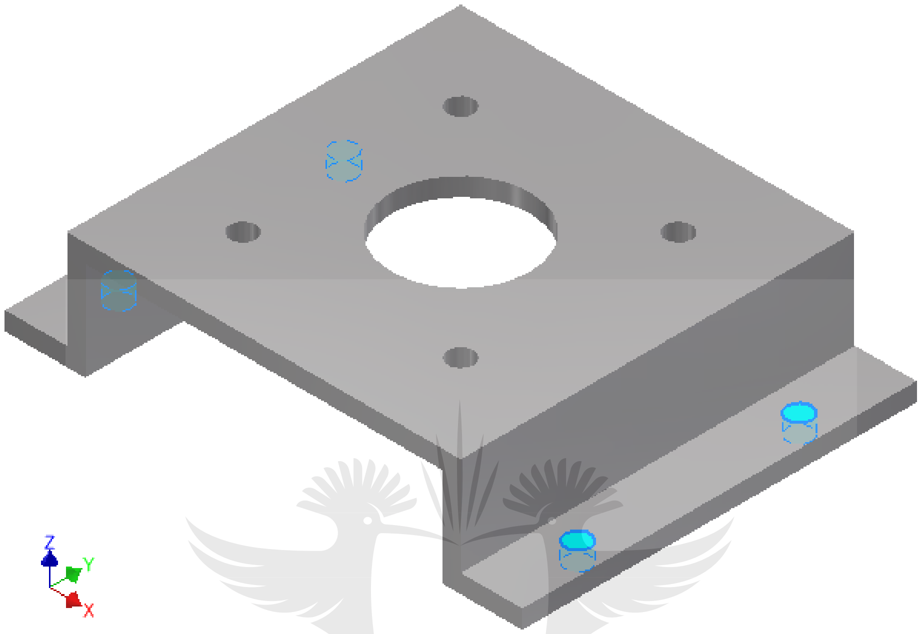


Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)

3



Results

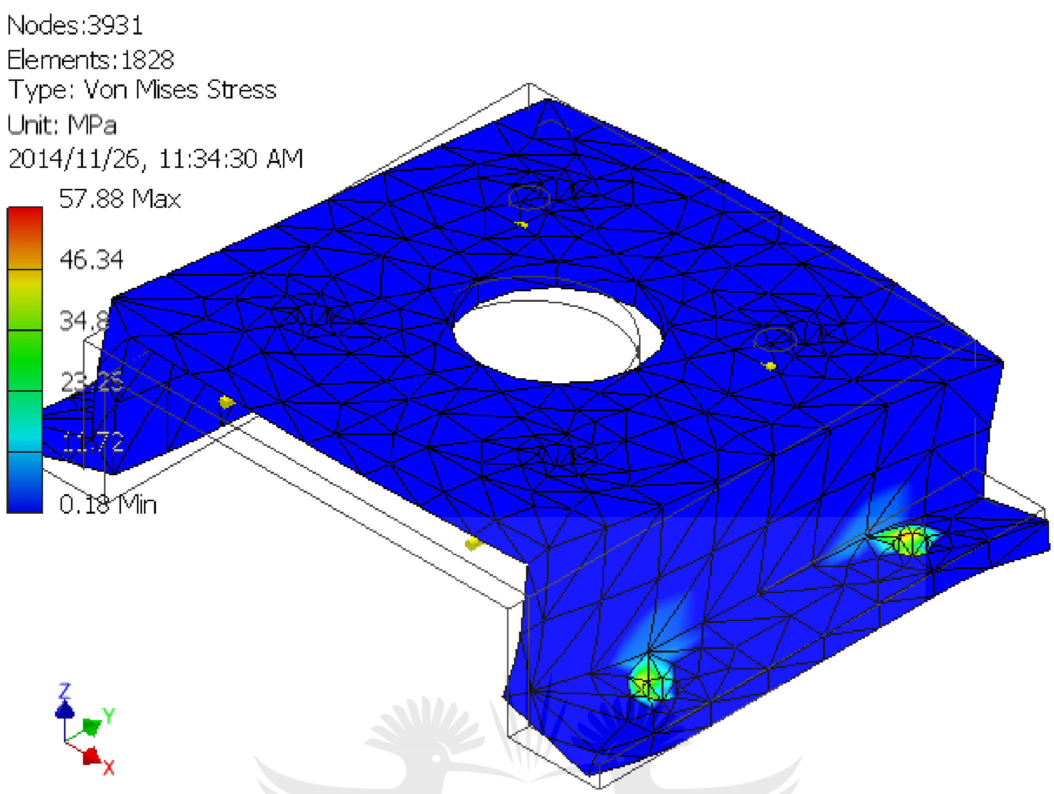
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	20735.1 N	0 N	1658.85 N m	1658.85 N m
		-20735.1 N		0 N m
		0 N		0 N m

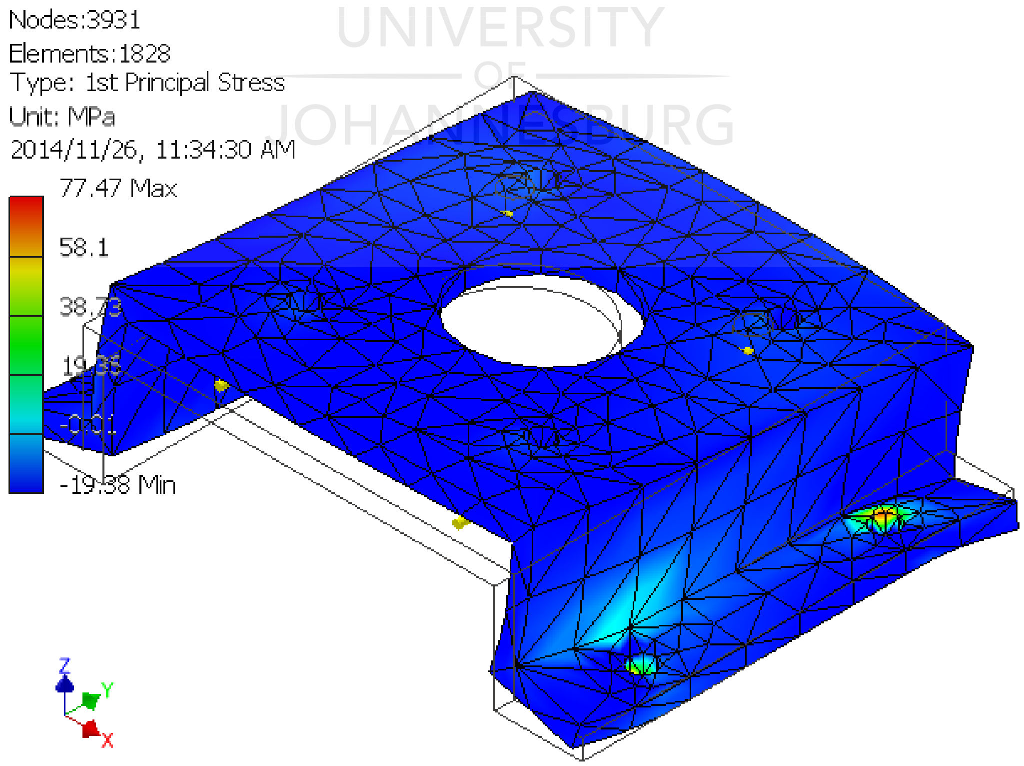
Result Summary

Name	Minimum	Maximum
Volume	2596590 mm^3	
Mass	20.4352 kg	
Von Mises Stress	0.177598 MPa	57.8769 MPa
1st Principal Stress	-19.3774 MPa	77.4688 MPa
3rd Principal Stress	-78.4149 MPa	18.3994 MPa
Displacement	0 mm	0.0161506 mm
Safety Factor	6.04732 ul	15 ul

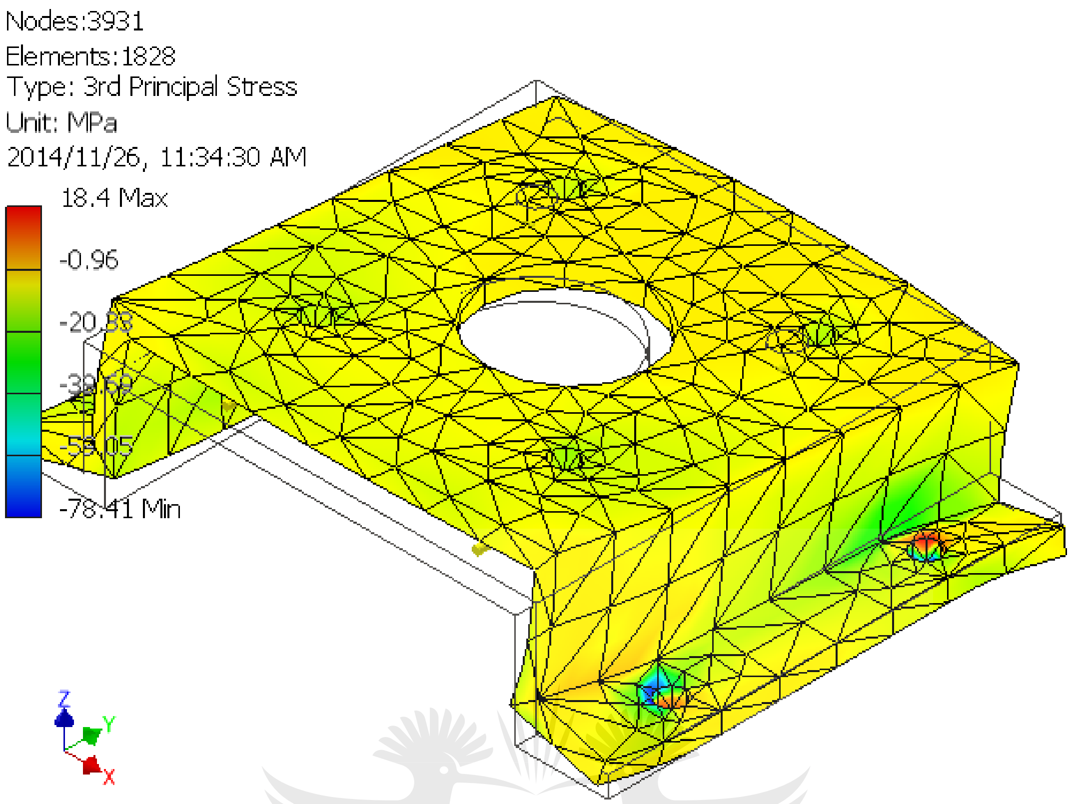
Von Mises Stress



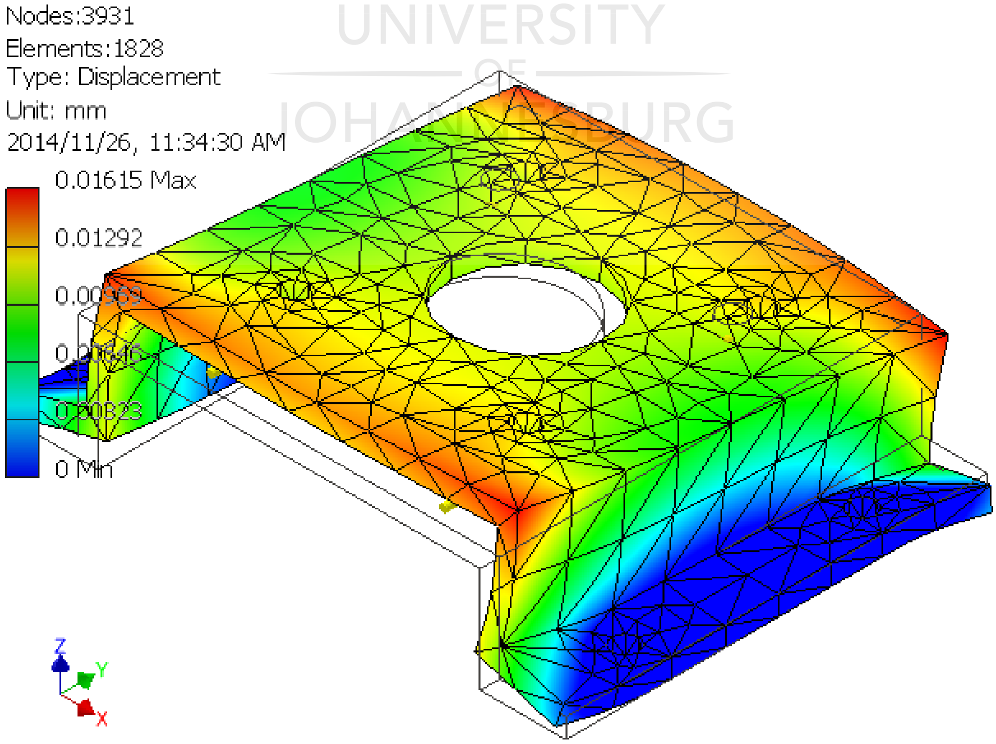
1st Principal Stress



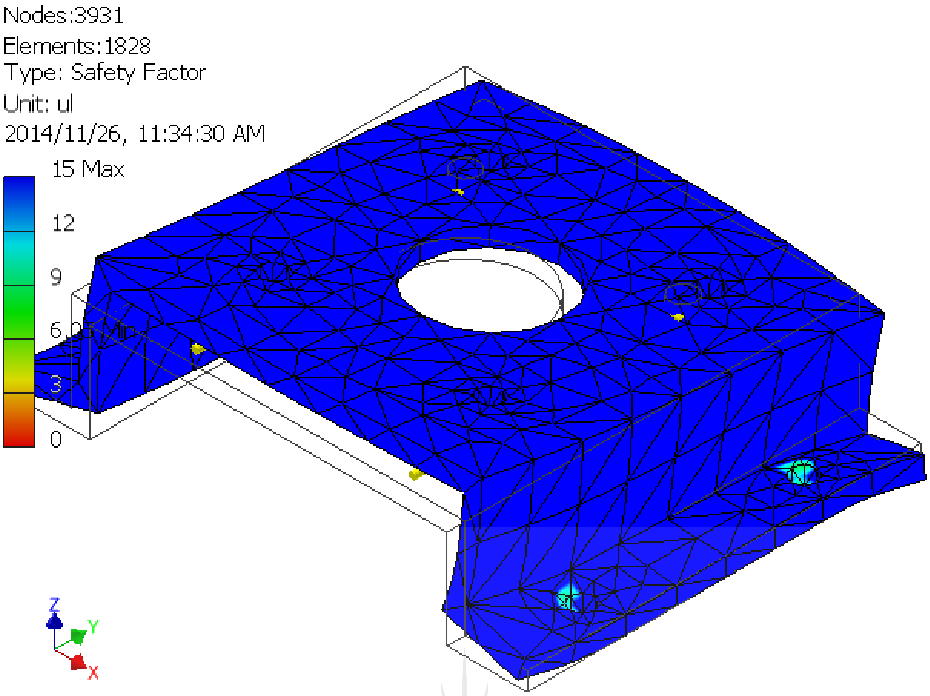
3rd Principal Stress



Displacement



Safety Factor



6 Gearbox top support bracket

Summary

Author	Terence Miller

Project

Part Number	Gearbox top support bracket.
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

Physical

Material	Steel, Carbon
Density	7.87 g/cm^3
Mass	32.5994 kg
Area	450065 mm^2
Volume	4142230 mm^3
Centre of Gravity	x=423.62 mm y=-65.6103 mm z=0 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

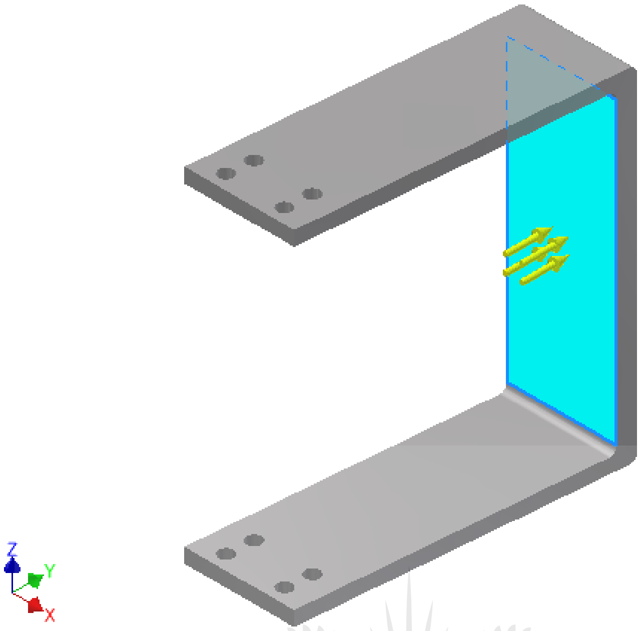
Name	Steel, Carbon	
General	Mass Density	7.87 g/cm^3
	Yield Strength	350 MPa
	Ultimate Tensile Strength	420 MPa
Stress	Young's Modulus	200 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	77.5194 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	52 W/(m K)
	Specific Heat	486 J/(kg c)
Part Name(s)	Gearbox top support bracket.	

Operating conditions

Pressure:1

Load Type	Pressure
Magnitude	0.465 MPa

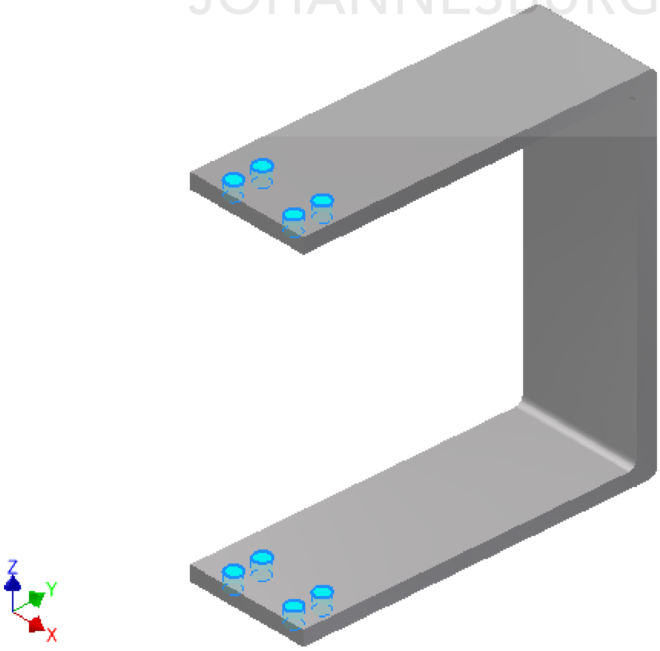
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

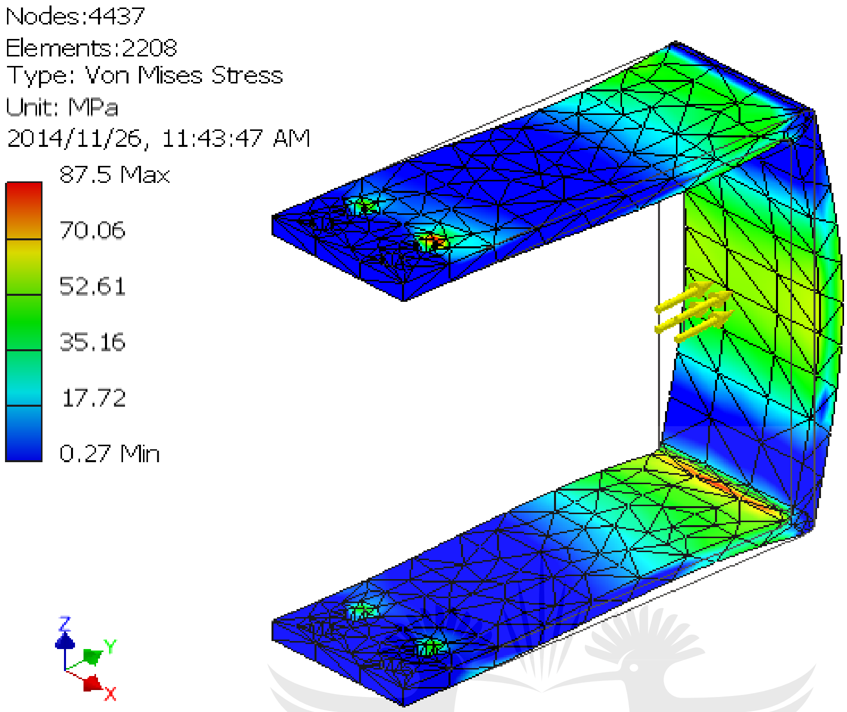
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	28604.9 N	0 N	919.935 N m	0 N m
		-28604.9 N		0 N m
		0 N		-919.935 N m

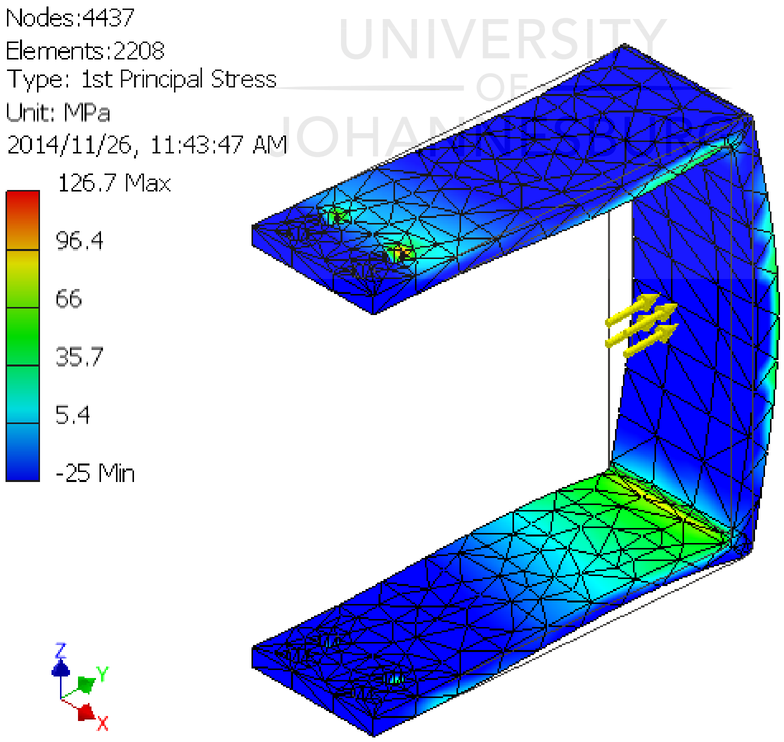
Result Summary

Name	Minimum	Maximum
Volume	4142230 mm^3	
Mass	32.5994 kg	
Von Mises Stress	0.271706 MPa	87.5037 MPa
1st Principal Stress	-24.9669 MPa	126.716 MPa
3rd Principal Stress	-88.9259 MPa	33.7765 MPa
Displacement	0 mm	0.517504 mm
Safety Factor	3.99983 ul	15 ul

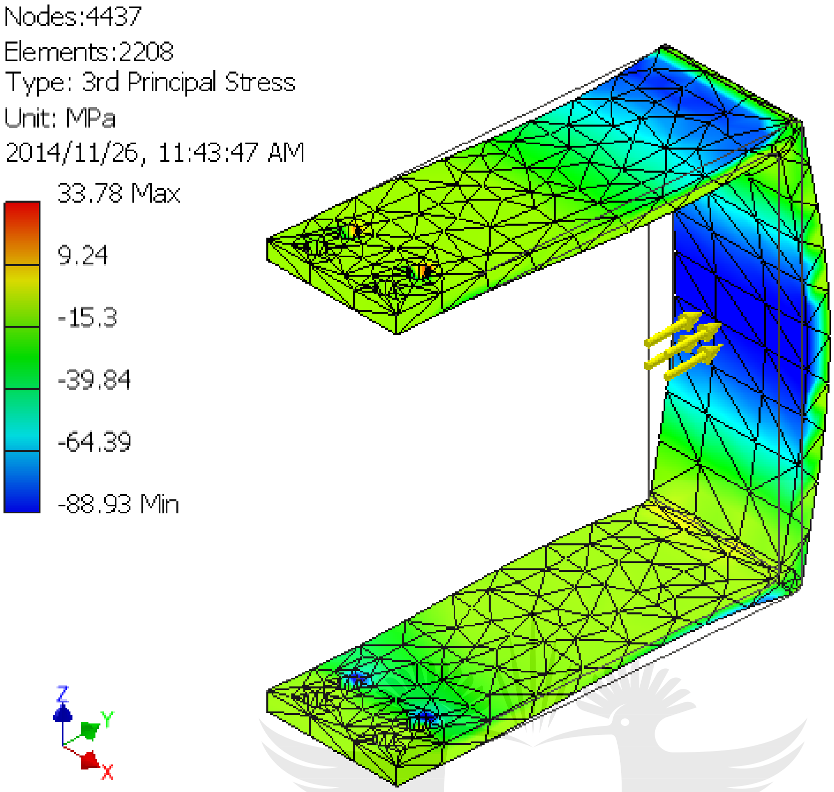
Von Mises Stress



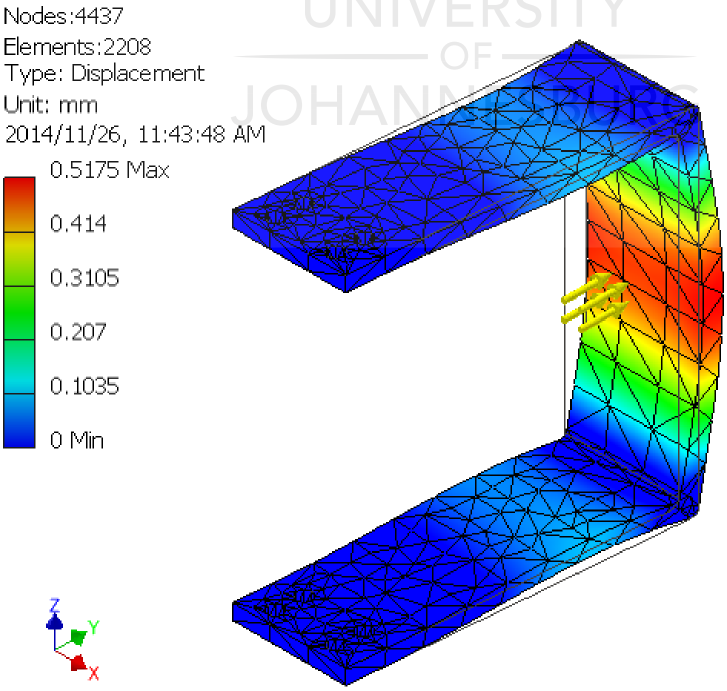
1st Principal Stress



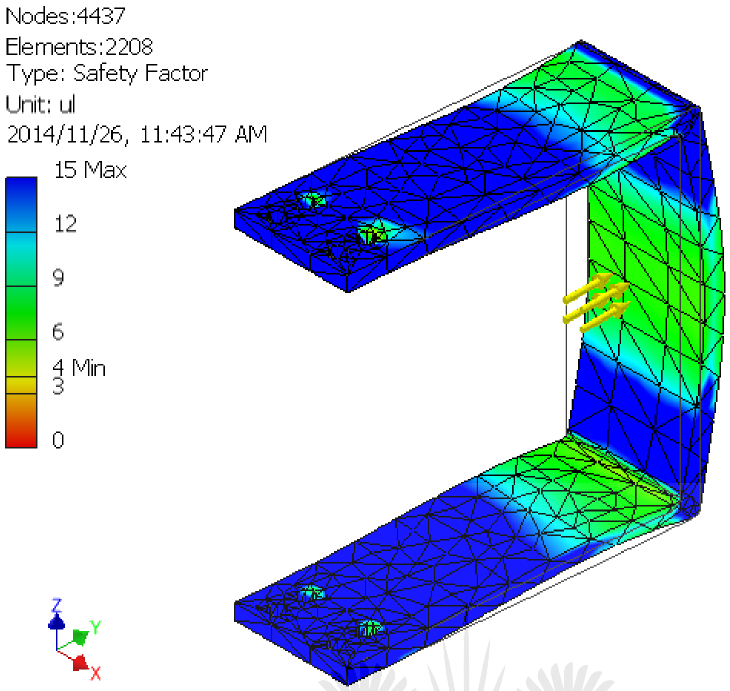
3rd Principal Stress



Displacement



Safety Factor



Factor of safety is 4 for carbon steel.

7 Bearing FSNL_619_TL__85__21319_K_ support bracket

Summary

Author	Terence Miller

Project

Part Number	Bearing FSNL_619_TL__85__21319_K_ support bracket
Designer	Terence Miller
Cost	R 0,00
Date Created	2014/11/10

Status

Design Status	Work In Progress

Physical

Material	Steel, Mild
Density	7.86 g/cm^3
Mass	49.7865 kg
Area	867834 mm^2
Volume	6334160 mm^3
Centre of Gravity	x=44 mm y=205 mm z=222.592 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014/11/11, 10:20 PM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

Name	Steel, Mild

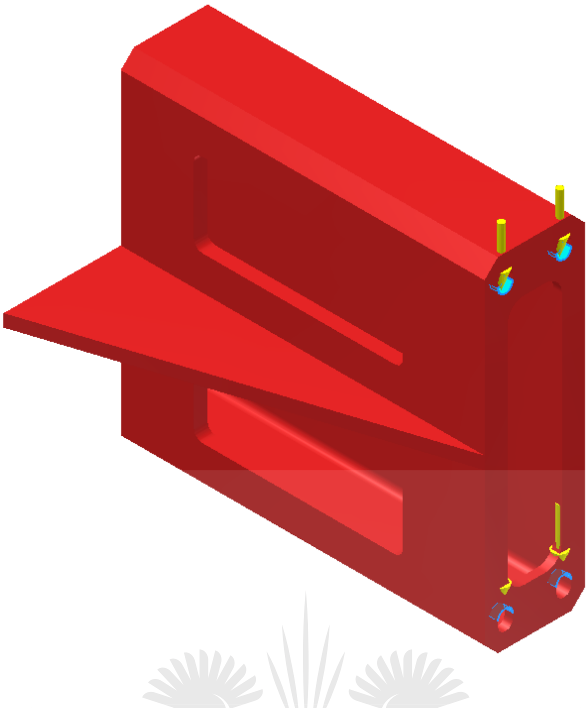
General	Mass Density	7.86 g/cm^3
	Yield Strength	207 MPa
	Ultimate Tensile Strength	345 MPa
Stress	Young's Modulus	220 GPa
	Poisson's Ratio	0.275 ul
	Shear Modulus	86.2745 GPa
Stress Thermal	Expansion Coefficient	0.000012 ul/c
	Thermal Conductivity	56 W/(m K)
	Specific Heat	460 J/(kg c)
Part Name(s)	Bearing FSNL_619_TL__85__21319_K_ support bracket	

Operating conditions

Force:1

Load Type	Force
Magnitude	1500.000 N
Vector X	0.000 N
Vector Y	-1500.000 N
Vector Z	0.000 N

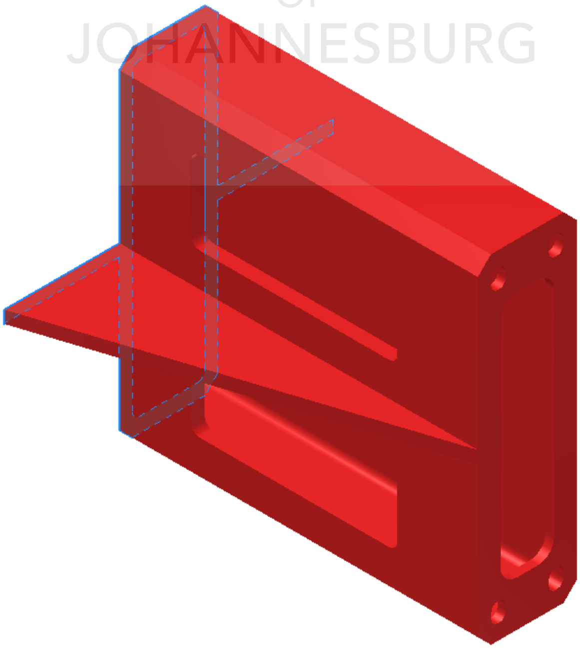
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

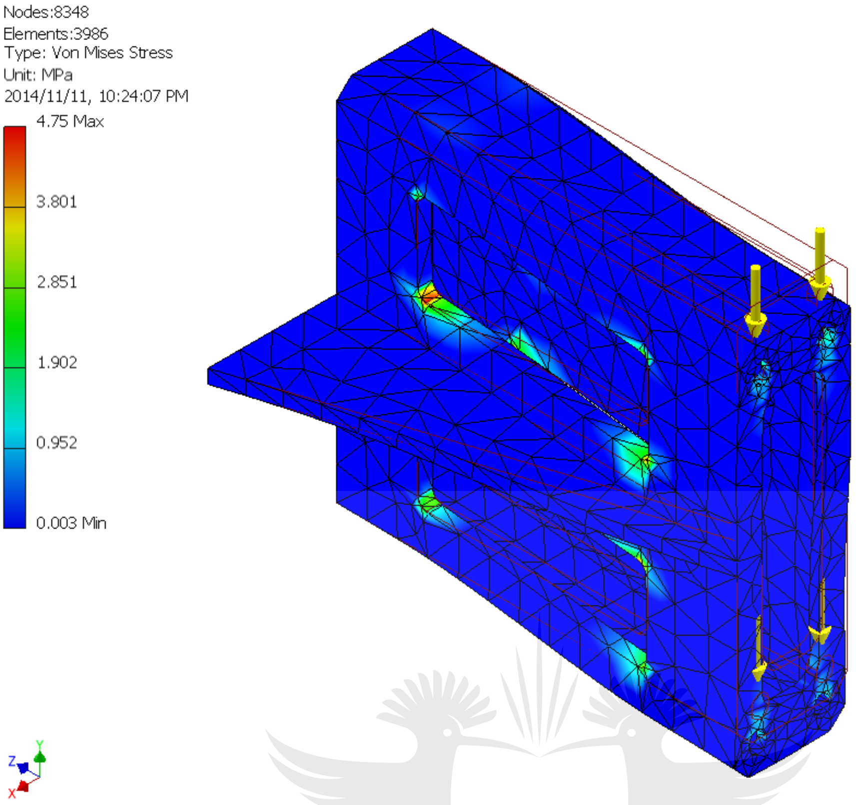
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	1500 N	0 N	645 N m	645 N m
		1500 N		0 N m
		0 N		0 N m

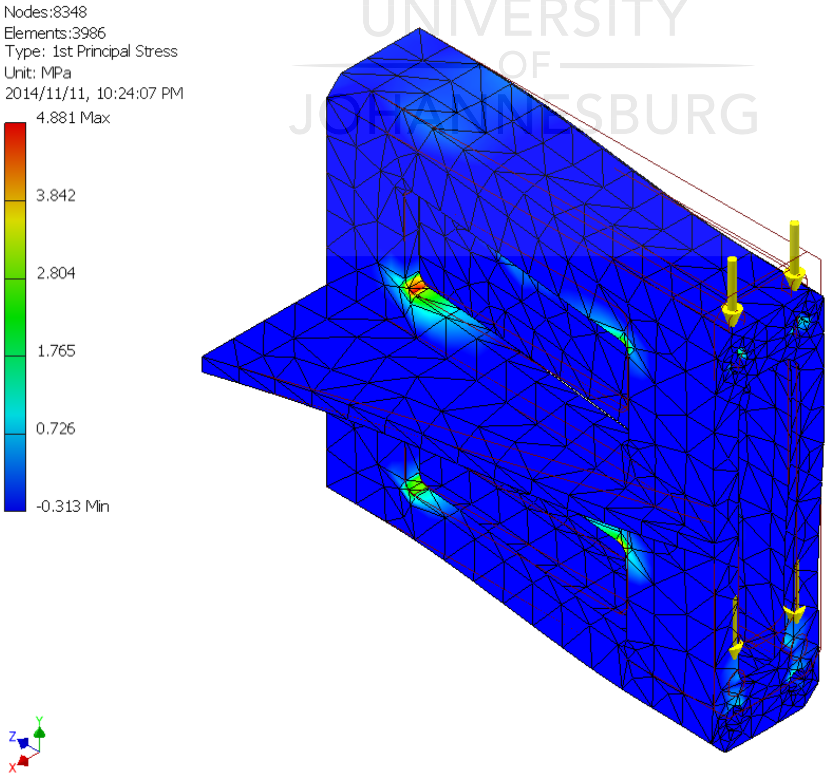
Result Summary

Name	Minimum	Maximum
Volume	6334160 mm^3	
Mass	49.7865 kg	
Von Mises Stress	0.00303555 MPa	4.75035 MPa
1st Principal Stress	-0.312519 MPa	4.88122 MPa
3rd Principal Stress	-5.08467 MPa	0.316901 MPa
Displacement	0 mm	0.00612997 mm
Safety Factor	15 ul	15 ul

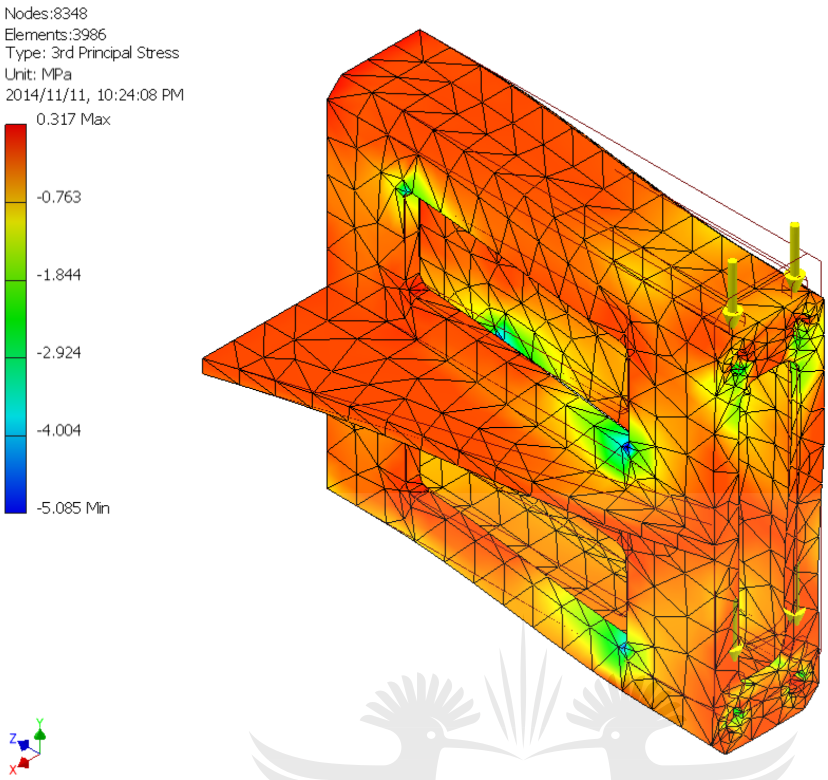
Von Mises Stress



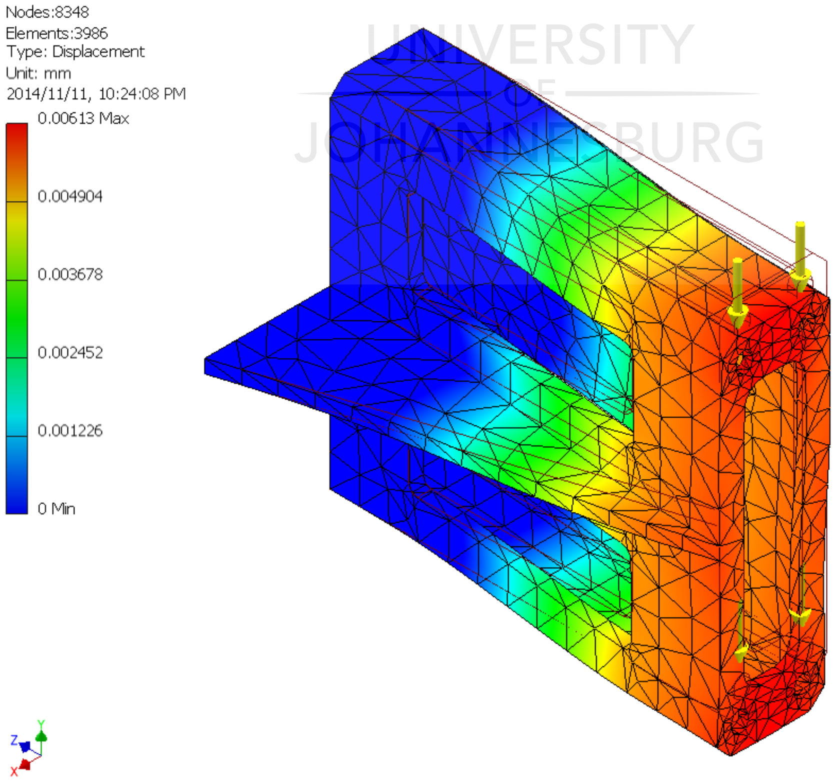
1st Principal Stress



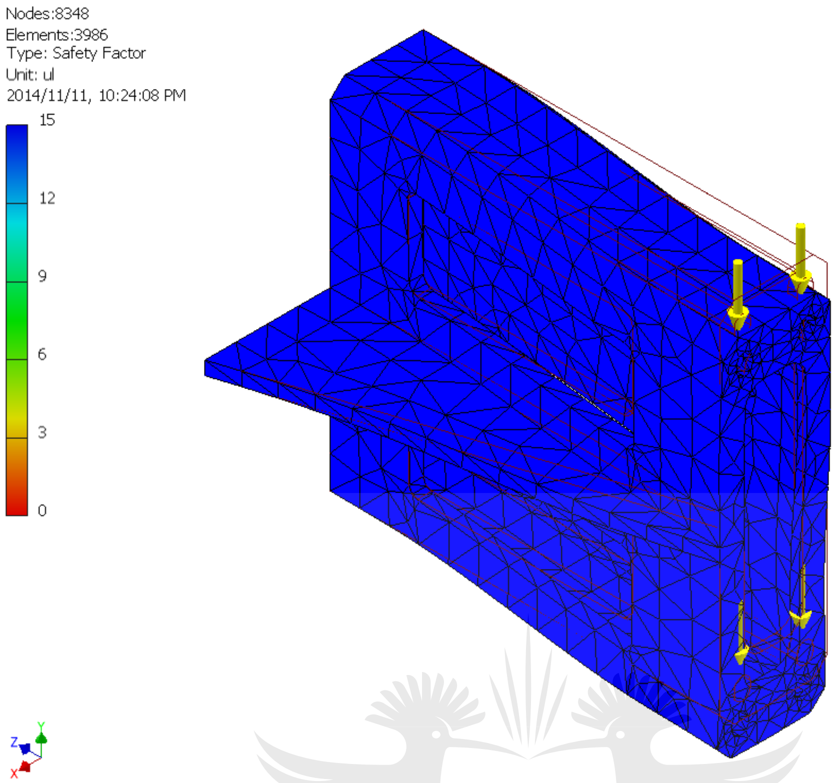
3rd Principal Stress



Displacement



Safety Factor



8 FS & MS bearing half A.

Summary

Author	Terence Miller

Project

Part Number	FS bearing half A
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

Physical

Material	Cast Iron
Density	7.25 g/cm^3
Mass	14.6182 kg
Area	196231 mm^2
Volume	2058890 mm^3
Centre of Gravity	x=-65.0071 mm y=-0.00000740472 mm z=0.00000916708 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

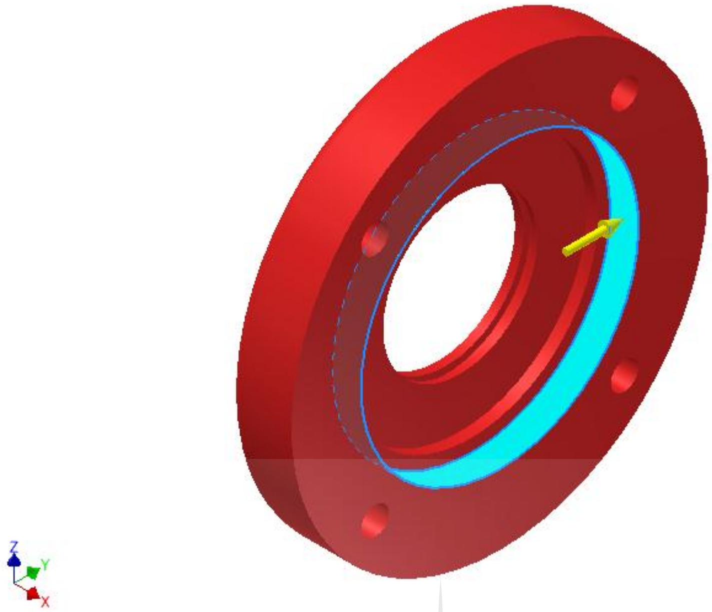
Name	Cast Iron	
General	Mass Density	7.25 g/cm^3
	Yield Strength	332 MPa
	Ultimate Tensile Strength	464 MPa
Stress	Young's Modulus	168 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	65.1163 GPa
Stress Thermal	Expansion Coefficient	0.000014 ul/c
	Thermal Conductivity	21 W/(m K)
	Specific Heat	540 J/(kg c)
Part Name(s)	FS bearing half A	

Operating conditions

Bearing Load:1

Load Type	Bearing Load
Magnitude	39169.000 N
Vector X	0.000 N
Vector Y	39169.000 N
Vector Z	0.000 N

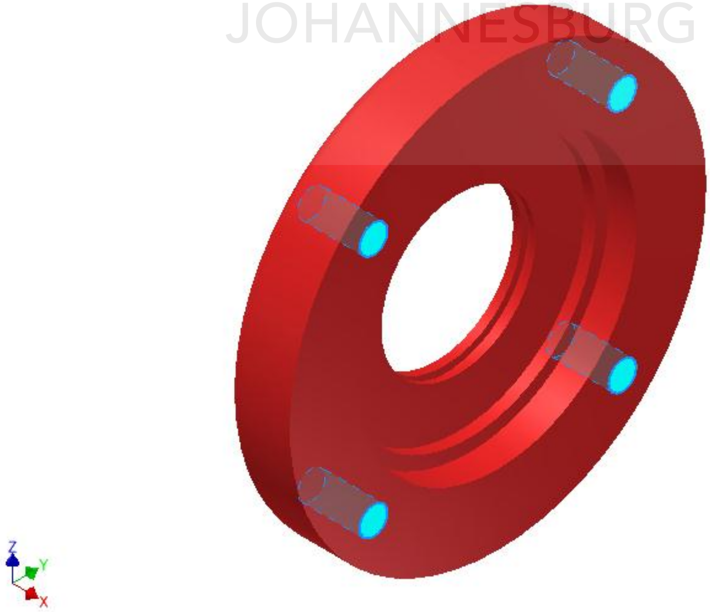
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

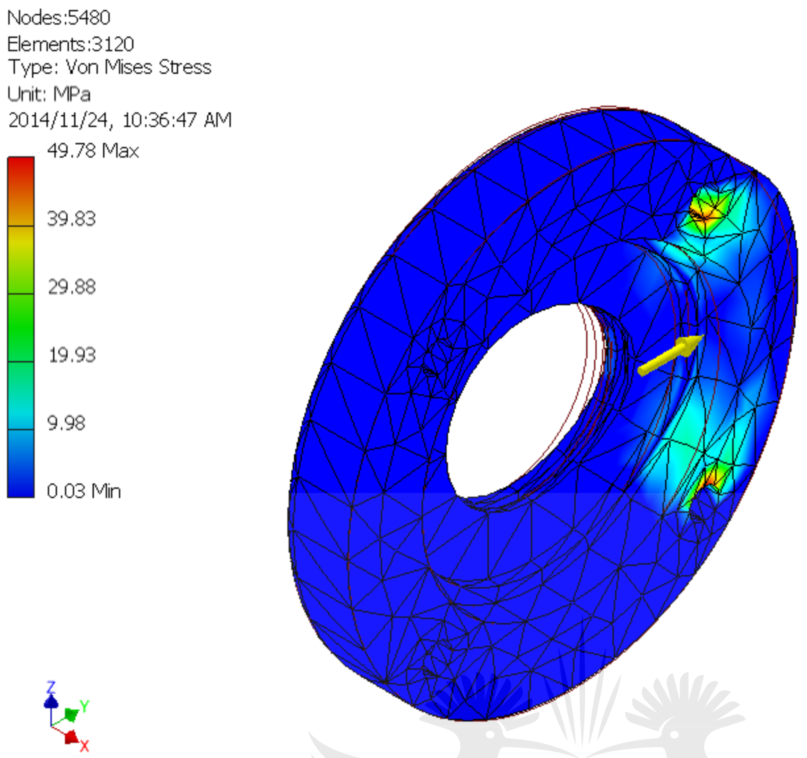
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	39169 N	0 N	470.218 N m	0 N m
		-39169 N		0 N m
		0 N		-470.218 N m

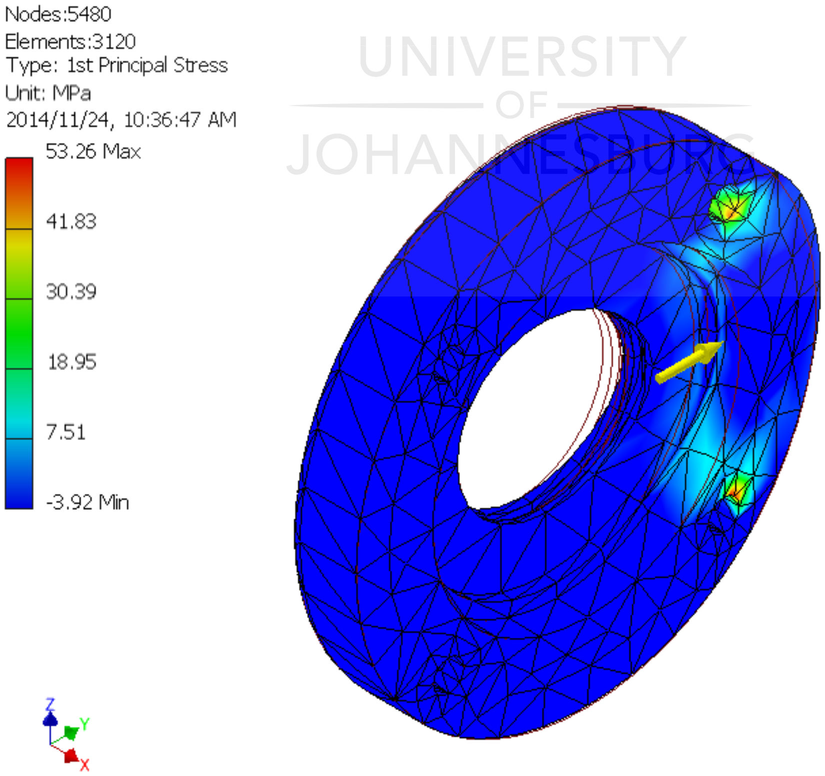
Result Summary

Name	Minimum	Maximum
Volume	2058890 mm^3	
Mass	14.6182 kg	
Von Mises Stress	0.0274267 MPa	49.7795 MPa
1st Principal Stress	-3.92417 MPa	53.2625 MPa
3rd Principal Stress	-37.1108 MPa	7.38059 MPa
Displacement	0 mm	0.014395 mm
Safety Factor	6.66941 ul	15 ul

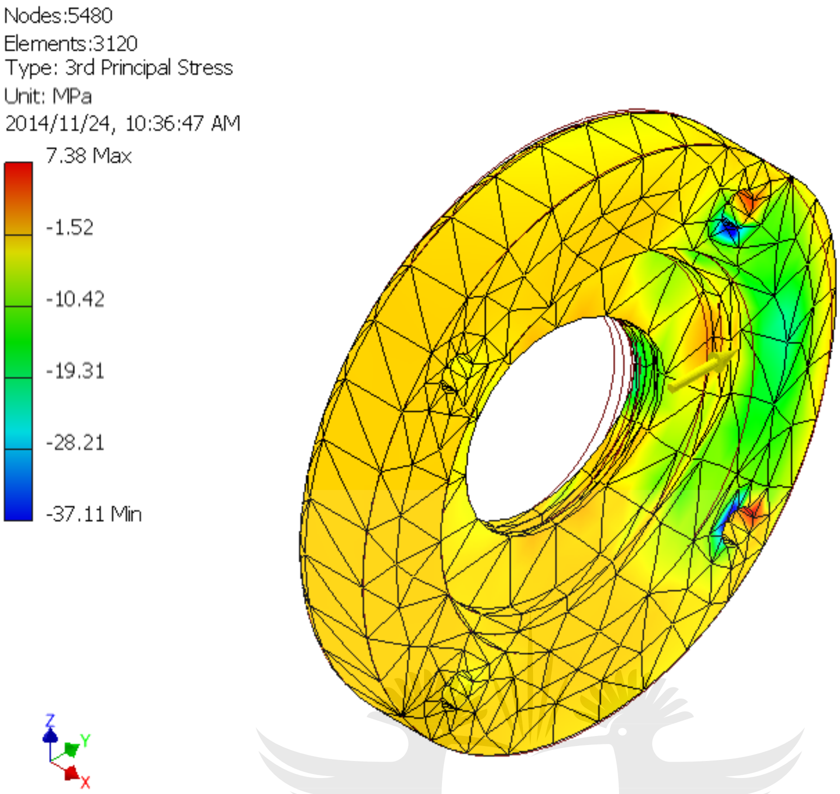
Von Mises Stress



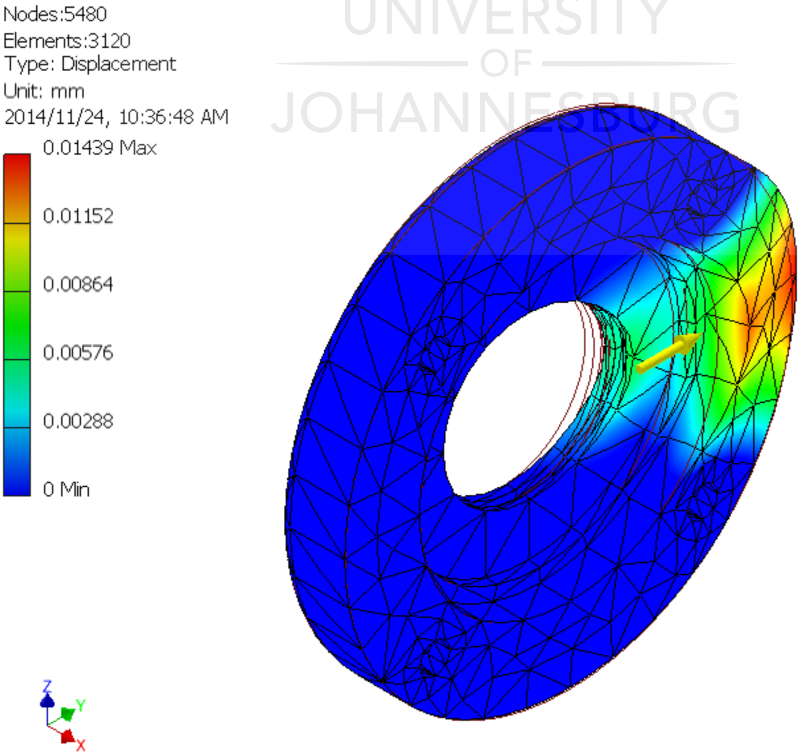
1st Principal Stress



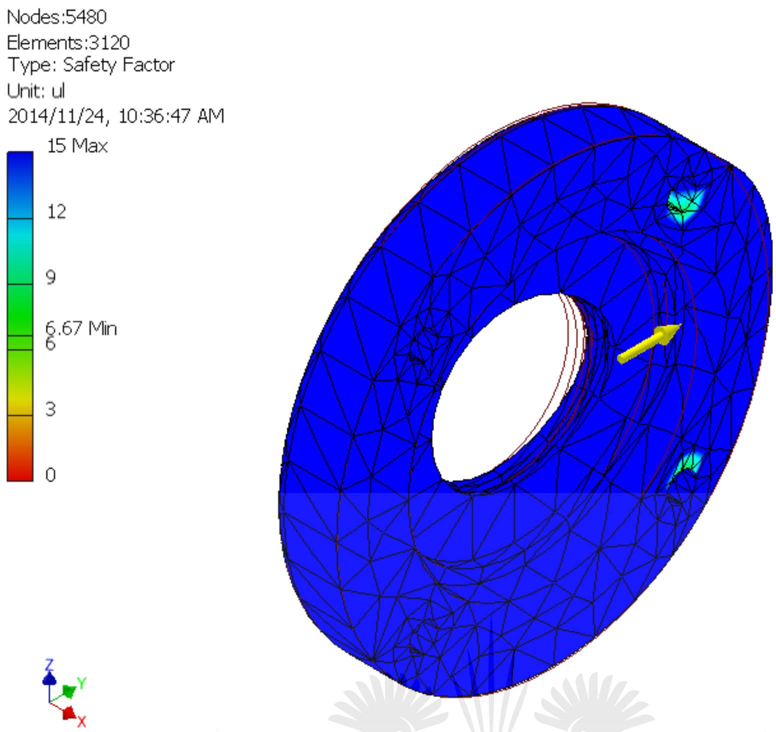
3rd Principal Stress



Displacement



Safety Factor



9 Main/FS bearing half B

Summary

Author	Terence Miller

Project

Part Number	FS bearing half B
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

Physical

Material	Cast Iron
Density	7.25 g/cm^3
Mass	18.6148 kg
Area	220896 mm^2
Volume	2621800 mm^3
Centre of Gravity	x=-61.4972 mm y=0 mm z=0 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

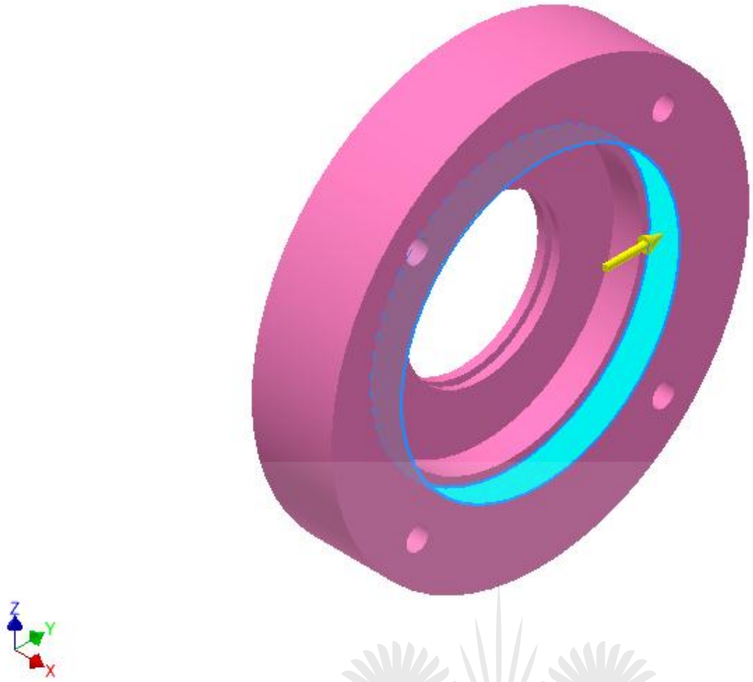
Material(s)

Name	Cast Iron	
General	Mass Density	7.25 g/cm^3
	Yield Strength	332 MPa
	Ultimate Tensile Strength	464 MPa
Stress	Young's Modulus	168 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	65.1163 GPa
Stress Thermal	Expansion Coefficient	0.000014 ul/c
	Thermal Conductivity	21 W/(m K)
	Specific Heat	540 J/(kg c)
Part Name(s)	FS bearing half B	

Bearing Load:1

Load Type	Bearing Load
Magnitude	20735.100 N
Vector X	0.000 N
Vector Y	20735.100 N
Vector Z	0.000 N

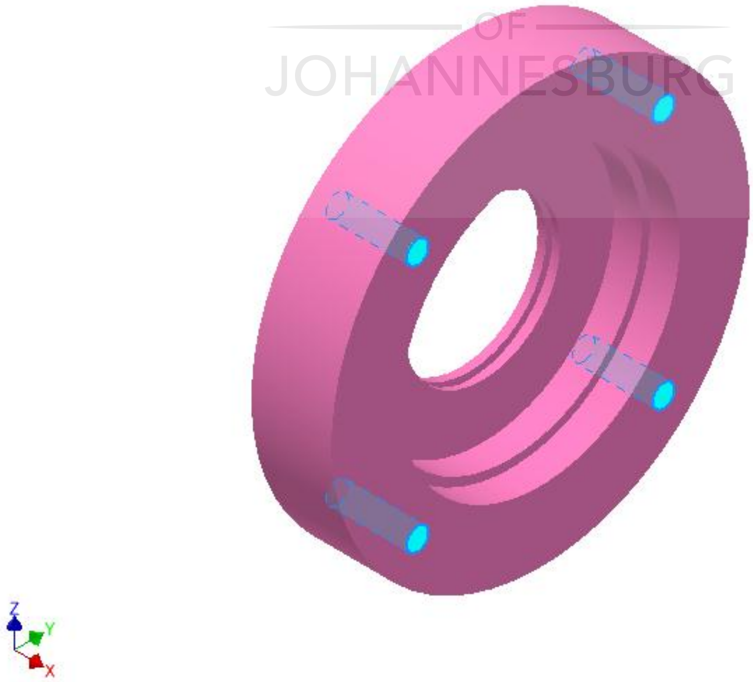
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



Results

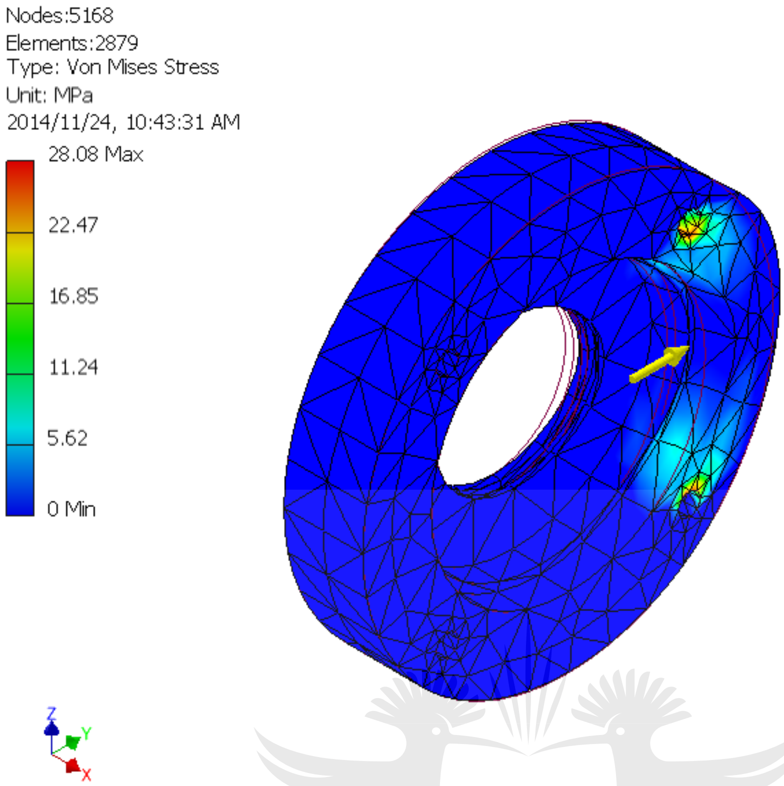
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	20735.1 N	0 N	373.121 N m	0 N m
		-20735.1 N		0 N m
		0 N		-373.121 N m

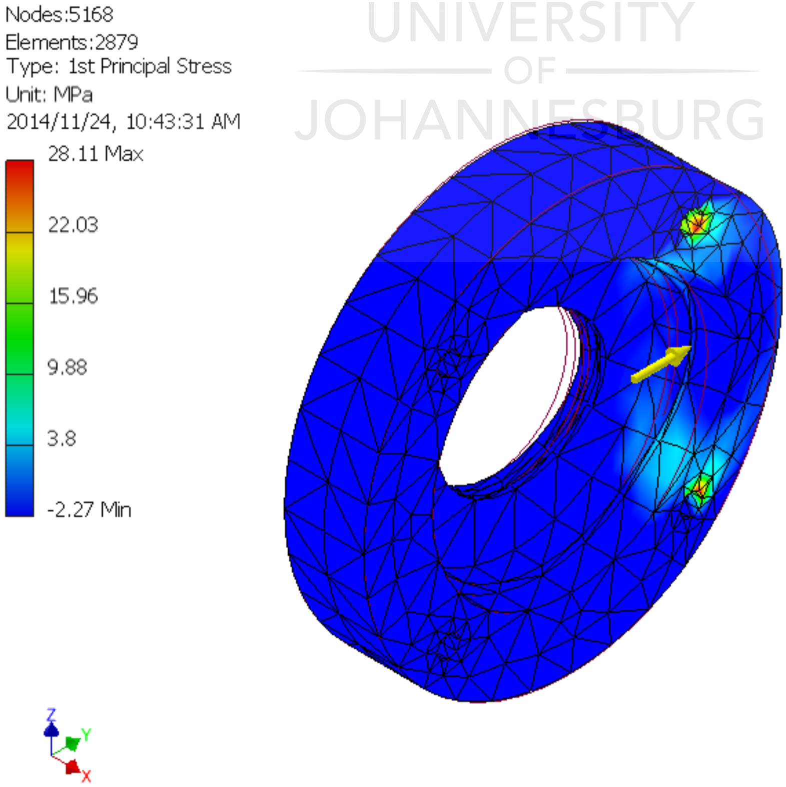
Result Summary

Name	Minimum	Maximum
Volume	2621800 mm^3	
Mass	18.6148 kg	
Von Mises Stress	0.00469358 MPa	28.0805 MPa
1st Principal Stress	-2.27336 MPa	28.112 MPa
3rd Principal Stress	-19.5085 MPa	4.62783 MPa
Displacement	0 mm	0.00749429 mm
Safety Factor	11.8231 ul	15 ul

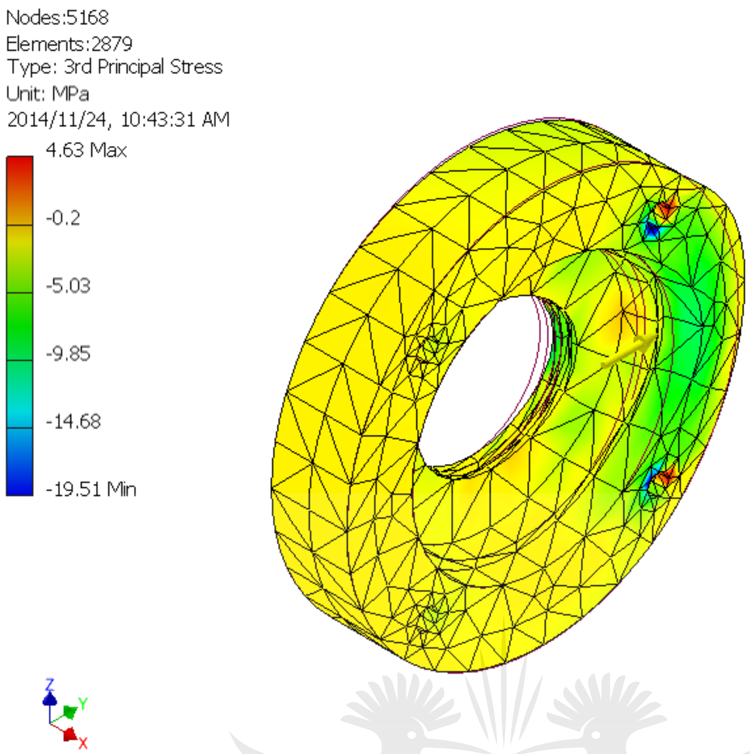
Von Mises Stress



1st Principal Stress

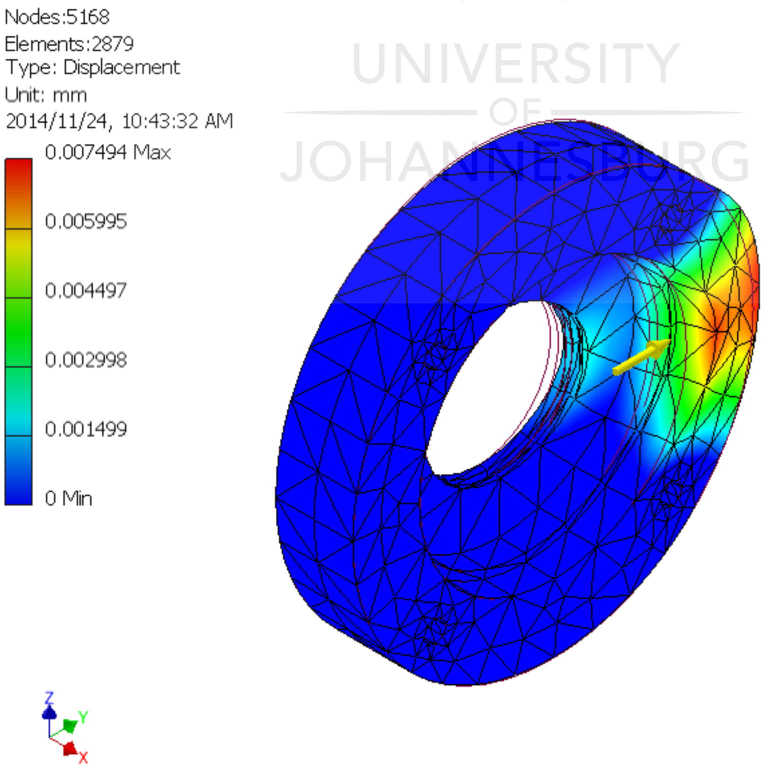


3rd Principal Stress



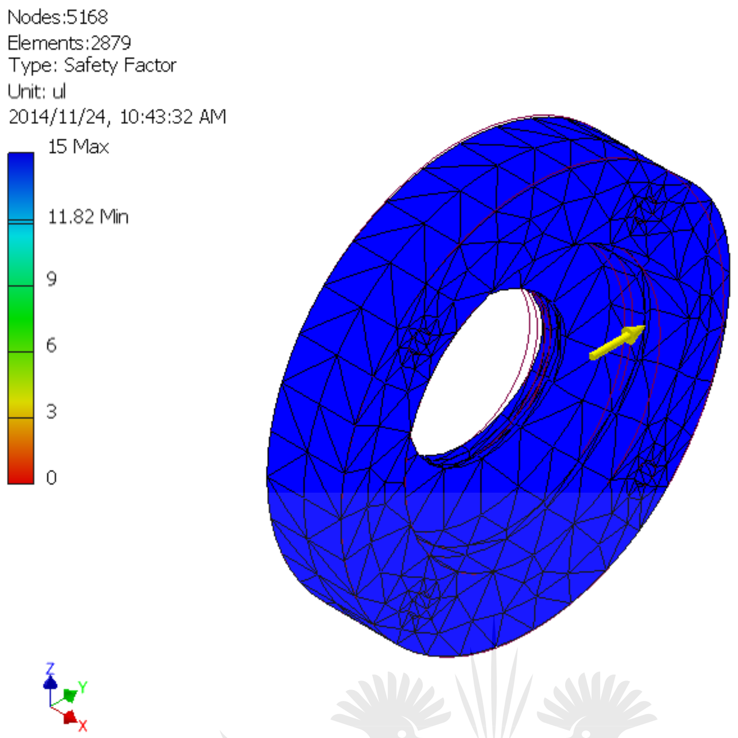
Nodes:5168
Elements:2879
Type: 3rd Principal Stress
Unit: MPa
2014/11/24, 10:43:31 AM

Displacement



Nodes:5168
Elements:2879
Type: Displacement
Unit: mm
2014/11/24, 10:43:32 AM

Safety Factor



10 IS bearing half A.

Summary

Author	Terence Miller

Project

Part Number	IS bearing half B
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

Physical

Material	Cast Iron
Density	7.25 g/cm^3

Mass	11.9575 kg
Area	159606 mm^2
Volume	1684160 mm^3
Centre of Gravity	x=-66.3101 mm y=0.00000545974 mm z=0.00000545974 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

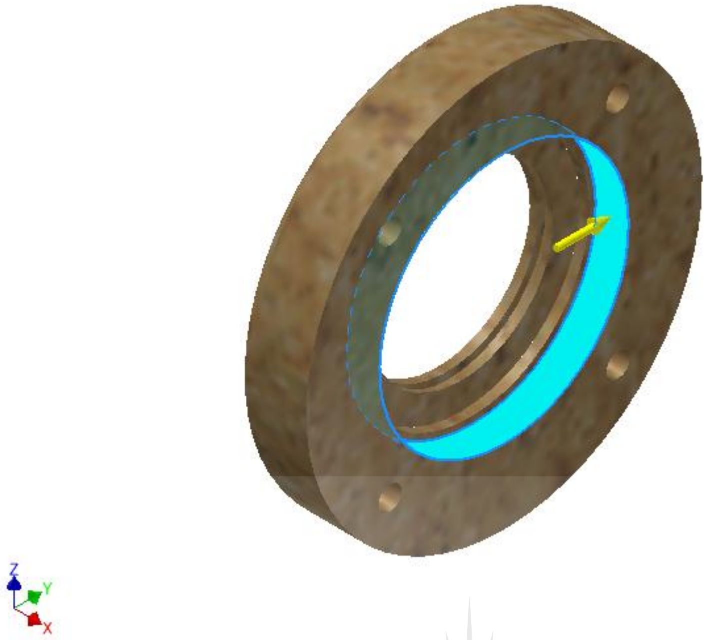
Material(s)

Name	Cast Iron	
General	Mass Density	7.25 g/cm^3
	Yield Strength	332 MPa
	Ultimate Tensile Strength	464 MPa
Stress	Young's Modulus	168 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	65.1163 GPa
Stress Thermal	Expansion Coefficient	0.000014 ul/c
	Thermal Conductivity	21 W/(m K)
	Specific Heat	540 J/(kg c)
Part Name(s)	IS bearing half A	

Bearing Load:1

Load Type	Bearing Load
Magnitude	15367.000 N
Vector X	0.000 N
Vector Y	15367.000 N
Vector Z	0.000 N

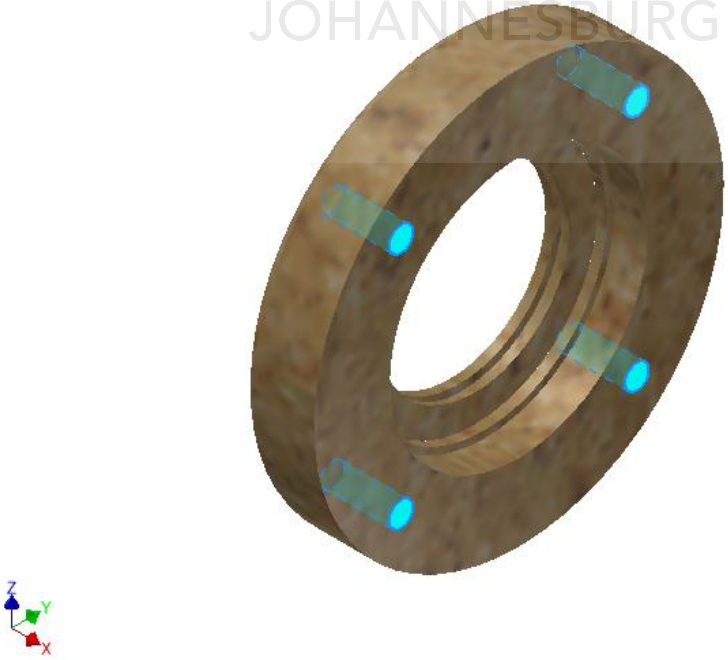
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)

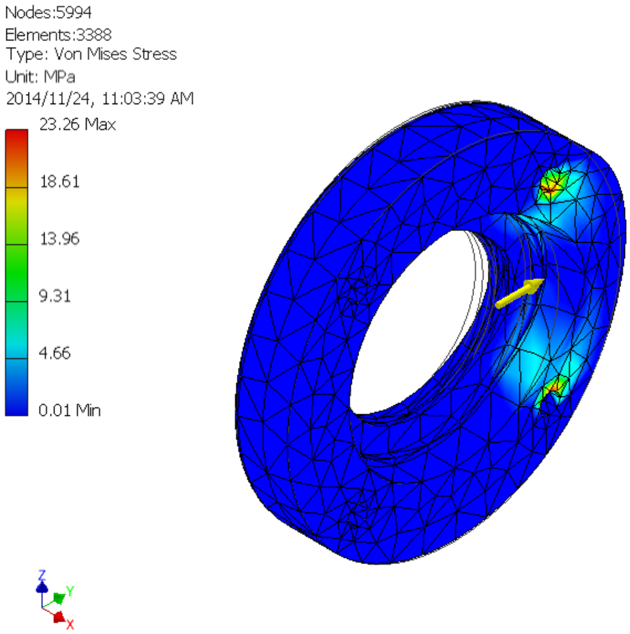


Results

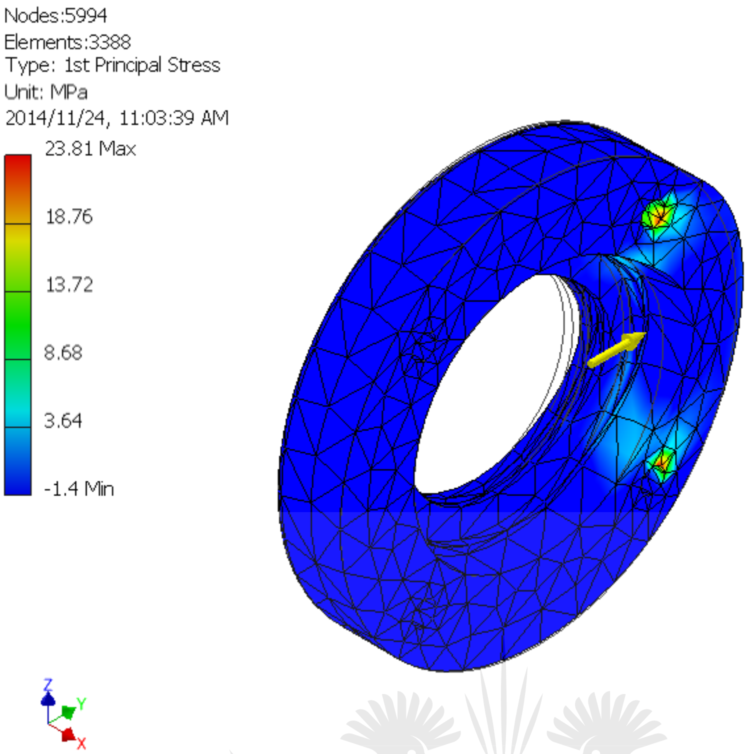
Reaction Force and Moment on Constraints

<table>
<tr><th rowspan="2">Constraint Name</th><th colspan="2">Reaction Force</th><th colspan="2">Reaction Moment</th></tr>
<tr><th>Magnitude</th><th>Component (X,Y,Z)</th><th>Magnitude</th><th>Component (X,Y,Z)</th></tr>
<tr><td>
<table>
<tr><th>Name</th><th>Minimum</th><th>Maximum</th></tr>
<tr><td>Volume</td><td colspan="2">1684160 mm^3</td></tr>
<tr><td>Mass</td><td colspan="2">11.9575 kg</td></tr>
<tr><td>Von Mises Stress</td><td>0.00651196 MPa</td><td>23.2595 MPa</td></tr>
<tr><td>1st Principal Stress</td><td>-1.40395 MPa</td><td>23.8067 MPa</td></tr>
<tr><td>3rd Principal Stress</td><td>-16.3811 MPa</td><td>3.62911 MPa</td></tr>
<tr><td>Displacement</td><td>0 mm</td><td>0.00530352 mm</td></tr>
<tr><td>Safety Factor</td><td>14.2737 ul</td><td>15 ul</td></tr>
</table>
Fixed Constraint:1</td><td>15367 N</td><td>0 N
-15367 N
0 N</td><td>172.897 N m</td><td>0 N m
0 N m
-172.897 N m</td></tr>
</table>

Von Mises Stress

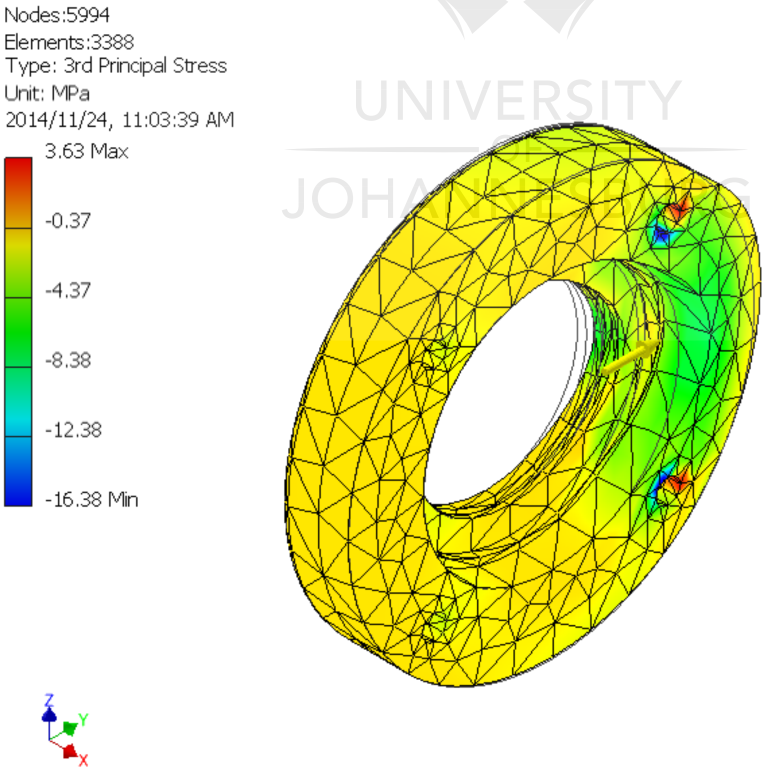


1st Principal Stress



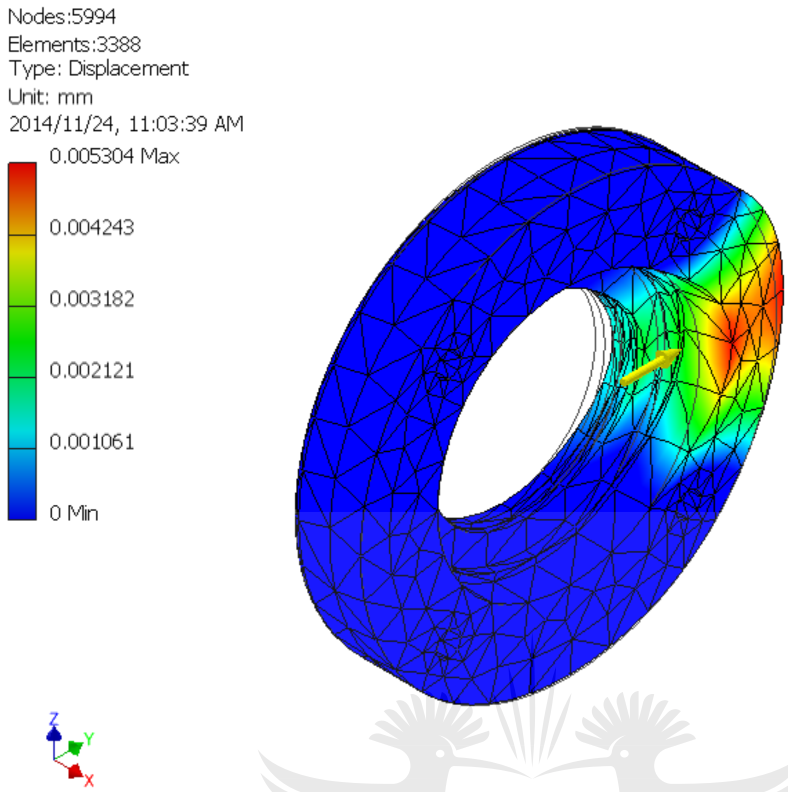
Nodes:5994
Elements:3388
Type: 1st Principal Stress
Unit: MPa
2014/11/24, 11:03:39 AM

3rd Principal Stress

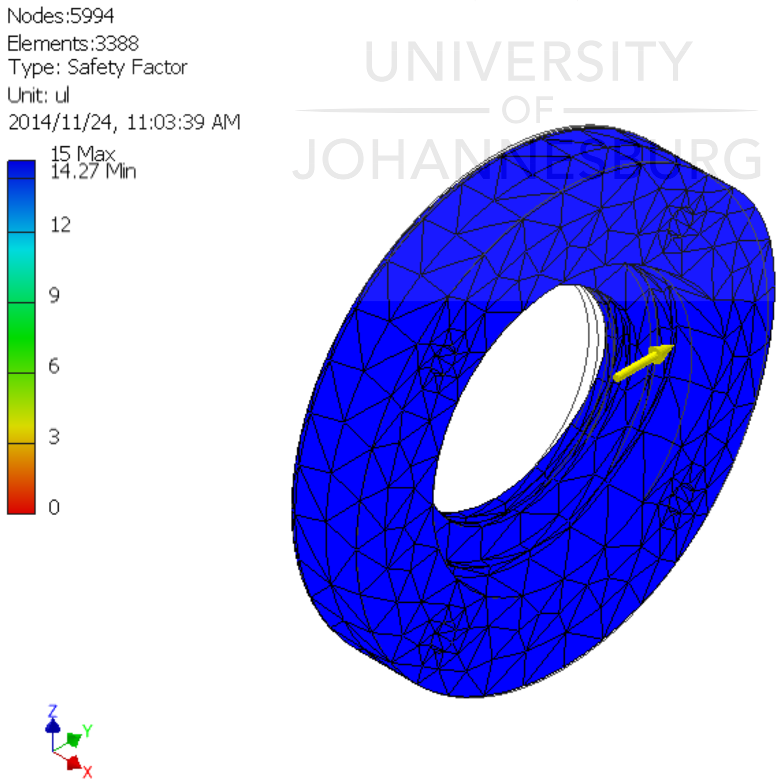


Nodes:5994
Elements:3388
Type: 3rd Principal Stress
Unit: MPa
2014/11/24, 11:03:39 AM

Displacement



Safety Factor



11 IS bearing half B

Summary

Author	Terence Miller

Project

Part Number	IS bearing half B.
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

Physical

Material	Cast Iron
Density	7.25 g/cm^3
Mass	8.107 kg
Area	162448 mm^2
Volume	1141830 mm^3
Centre of Gravity	x=-50.8304 mm y=0.00000474227 mm z=0.00000474227 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014/11/24, 11:12 AM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

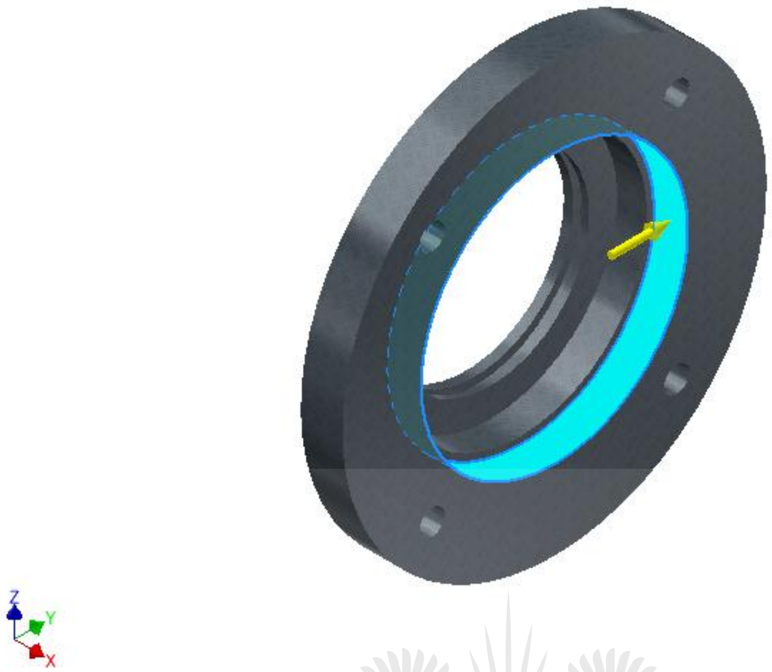
Material(s)

Name	Cast Iron	
General	Mass Density	7.25 g/cm^3
	Yield Strength	332 MPa
	Ultimate Tensile Strength	464 MPa
Stress	Young's Modulus	168 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	65.1163 GPa
Stress Thermal	Expansion Coefficient	0.000014 ul/c
	Thermal Conductivity	21 W/(m K)
	Specific Heat	540 J/(kg c)
Part Name(s)	IS bearing half B.	

Bearing Load:1

Load Type	Bearing Load
Magnitude	15367.000 N
Vector X	0.000 N
Vector Y	15367.000 N
Vector Z	0.000 N

Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)

Results

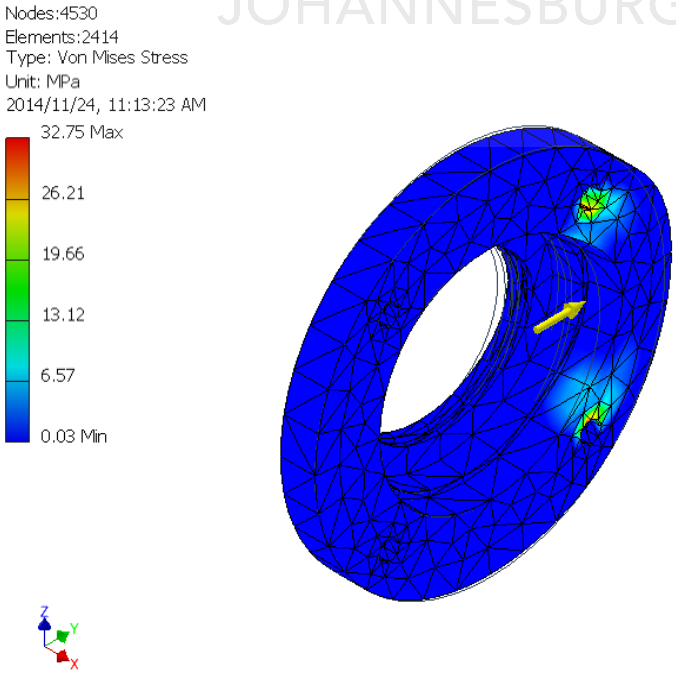
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	15367 N	0 N	38.4644 N m	0 N m
		-15367 N		0 N m
		0 N		-38.4644 N m

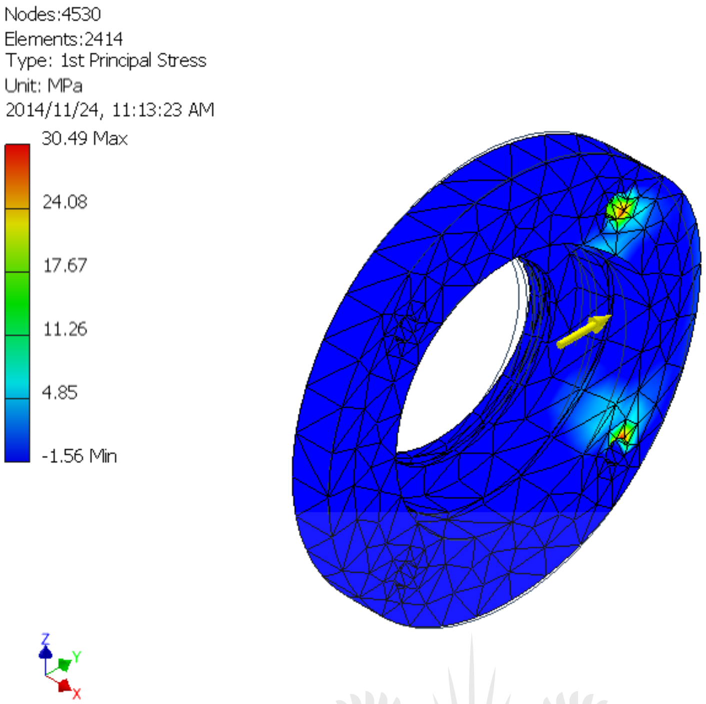
Result Summary

Name	Minimum	Maximum
Volume	1141830 mm^3	
Mass	8.107 kg	
Von Mises Stress	0.028118 MPa	32.7521 MPa
1st Principal Stress	-1.56454 MPa	30.4887 MPa
3rd Principal Stress	-20.6018 MPa	4.22402 MPa
Displacement	0 mm	0.00689757 mm
Safety Factor	10.1368 ul	15 ul

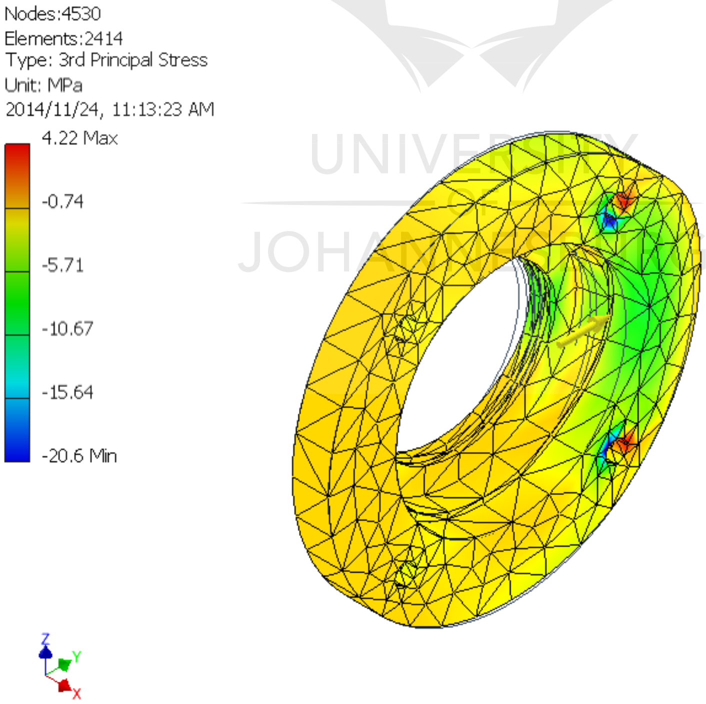
Von Mises Stress



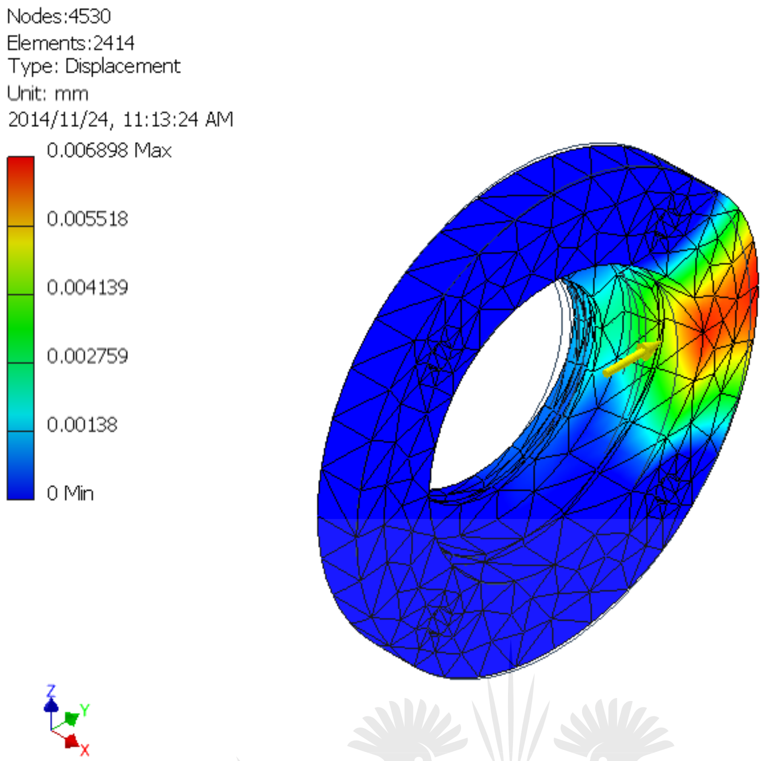
1st Principal Stress



3rd Principal Stress

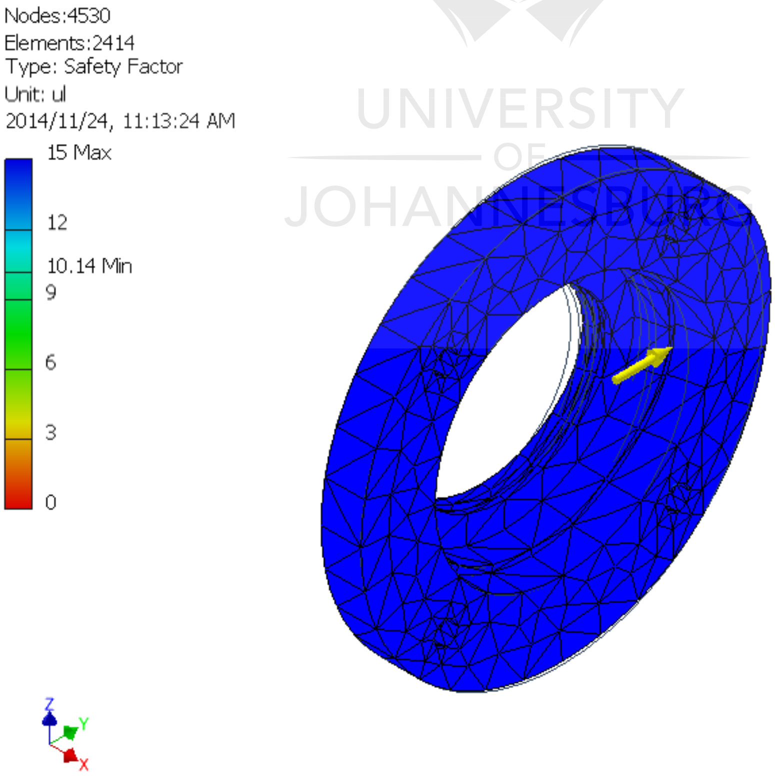


Displacement



Nodes:4530
Elements:2414
Type: Displacement
Unit: mm
2014/11/24, 11:13:24 AM

Safety Factor



Nodes:4530
Elements:2414
Type: Safety Factor
Unit: ul
2014/11/24, 11:13:24 AM

12 IS frame bearing half AA

Summary

Author	Terence Miller

Project

Part Number	IS bearing half AA
Designer	Terence Miller
Cost	R 0,00
Date Created	2014

Status

Design Status	Work In Progress

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

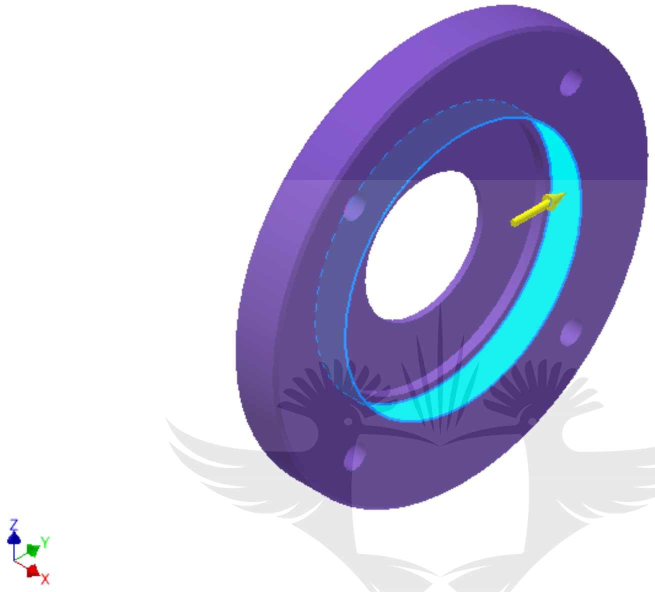
Material(s)

Material	Cast Iron
Density	7.25 g/cm^3
Mass	8.107 kg
Area	162448 mm^2
Volume	1141830 mm^3
Centre of Gravity	x=-50.8304 mm y=0.00000474227 mm z=0.00000474227 mm

Bearing Load:1

Load Type	Bearing Load
Magnitude	20000.000 N
Vector X	0.000 N
Vector Y	20000.000 N
Vector Z	0.000 N

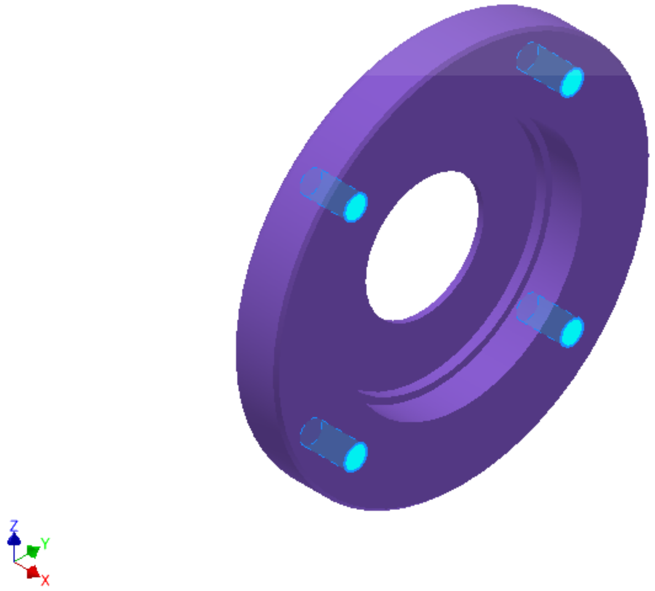
Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)



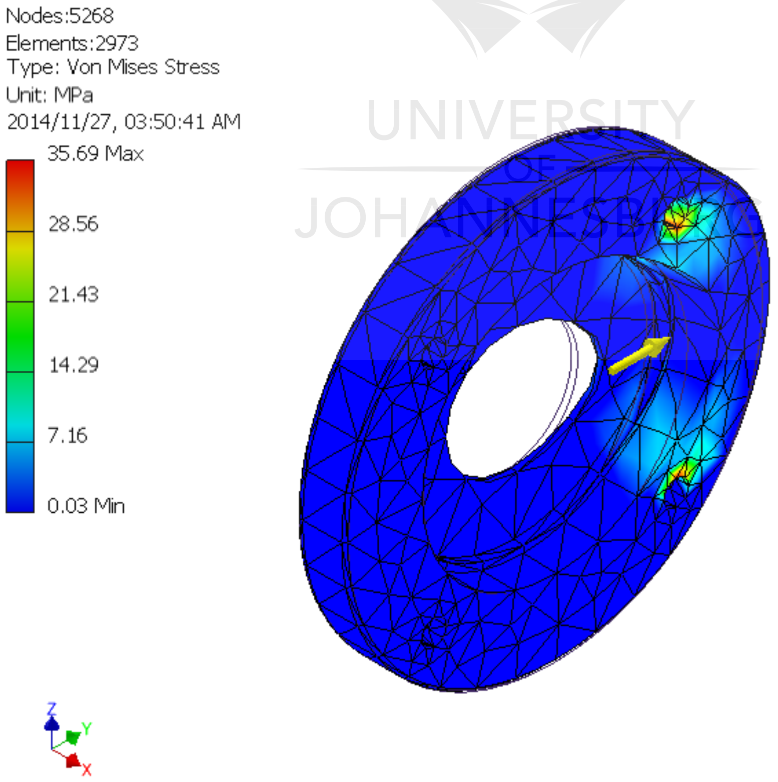
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	20000 N	0 N	104.989 N m	0 N m
		-20000 N		0 N m
		0 N		-104.989 N m

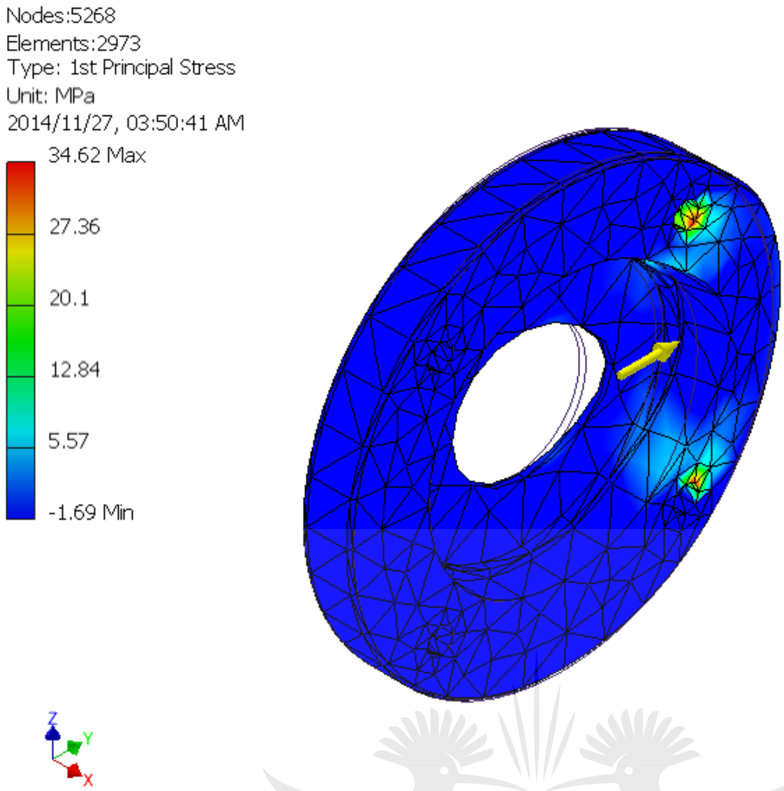
Result Summary

Name	Minimum	Maximum
Volume	1157210 mm^3	
Mass	9.10721 kg	
Von Mises Stress	0.0253044 MPa	35.6916 MPa
1st Principal Stress	-1.68666 MPa	34.6192 MPa
3rd Principal Stress	-24.4621 MPa	5.27614 MPa
Displacement	0 mm	0.00695986 mm
Safety Factor	9.80623 ul	15 ul

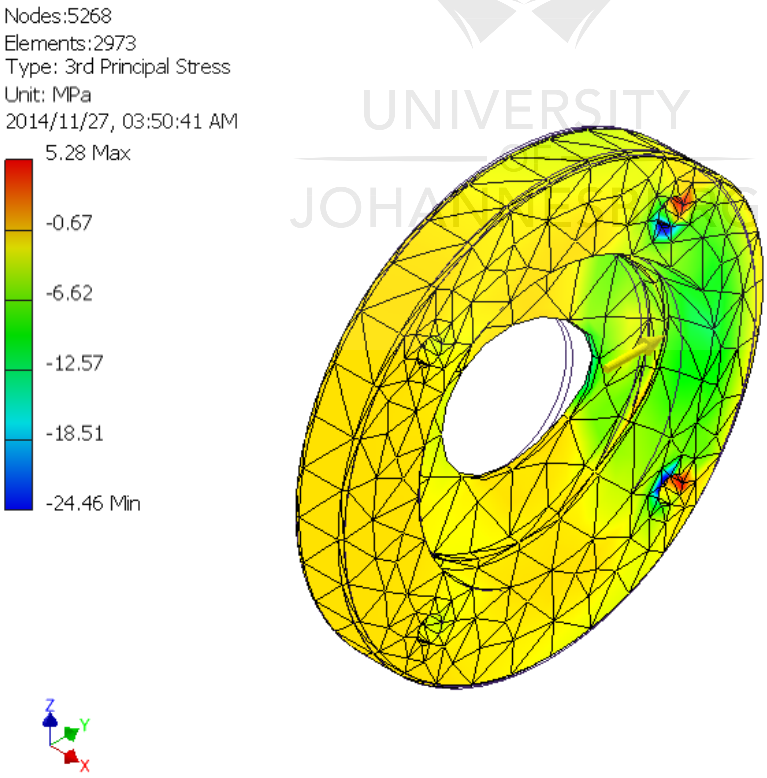
Von Mises Stress



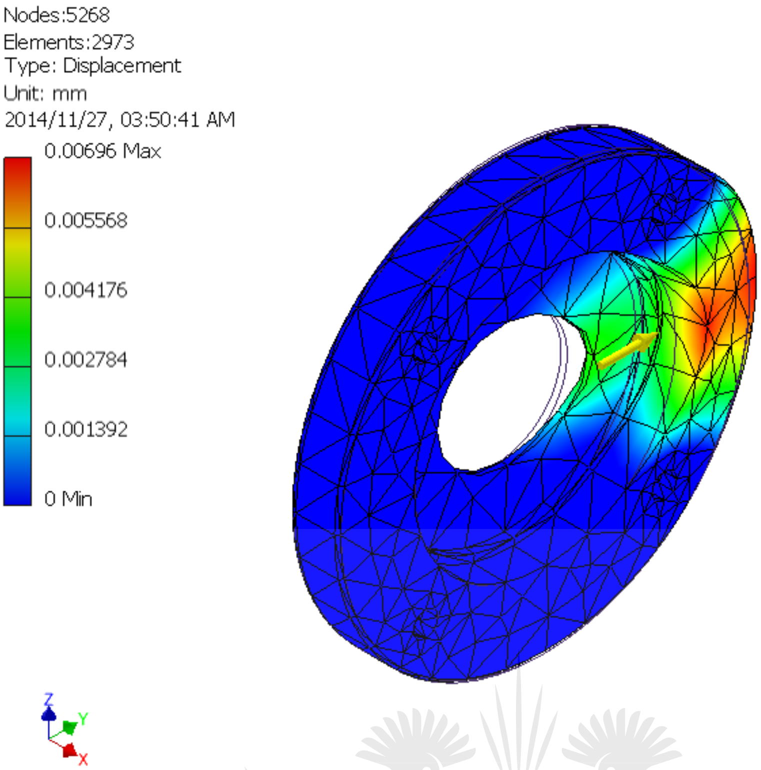
1st Principal Stress



3rd Principal Stress

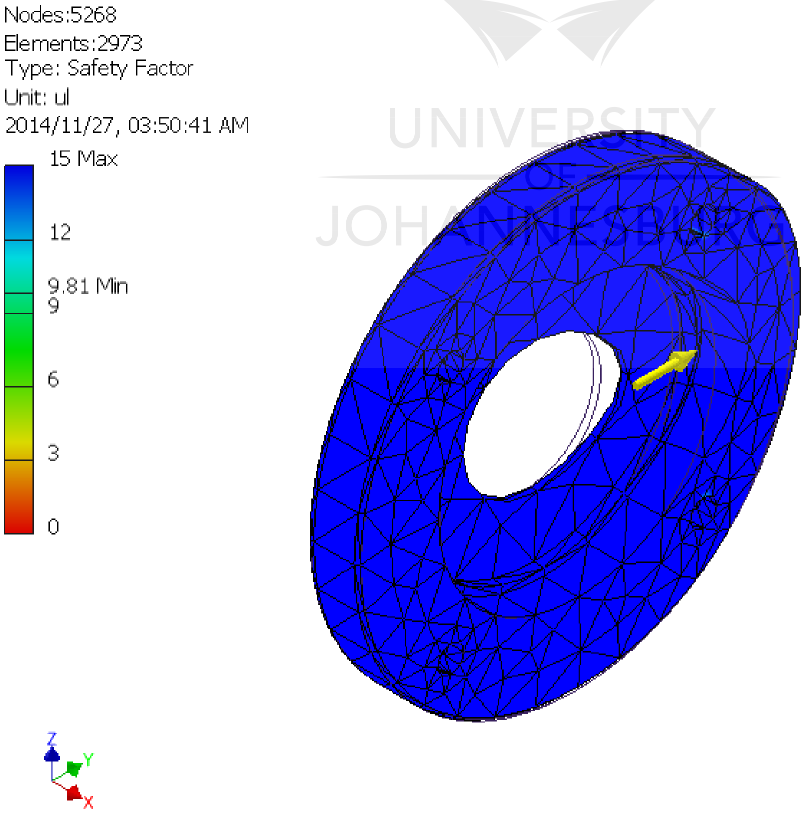


Displacement



Nodes:5268
Elements:2973
Type: Displacement
Unit: mm
2014/11/27, 03:50:41 AM

Safety Factor



Nodes:5268
Elements:2973
Type: Safety Factor
Unit: ul
2014/11/27, 03:50:41 AM

13 IS frame bearing half BB

Summary

Author	Terence Miller

Project

Part Number	IS bearing half BB
Designer	Terence Miller
Cost	R 0,00
Date Created	2014/11/11

Status

Design Status	Work In Progress

Physical

Material	Cast Iron
Density	7.25 g/cm^3
Mass	11.6757 kg
Area	169133 mm^2
Volume	1644460 mm^3
Centre of Gravity	x=-53.8215 mm y=0.00000565367 mm z=0.00000565367 mm

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	2014
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

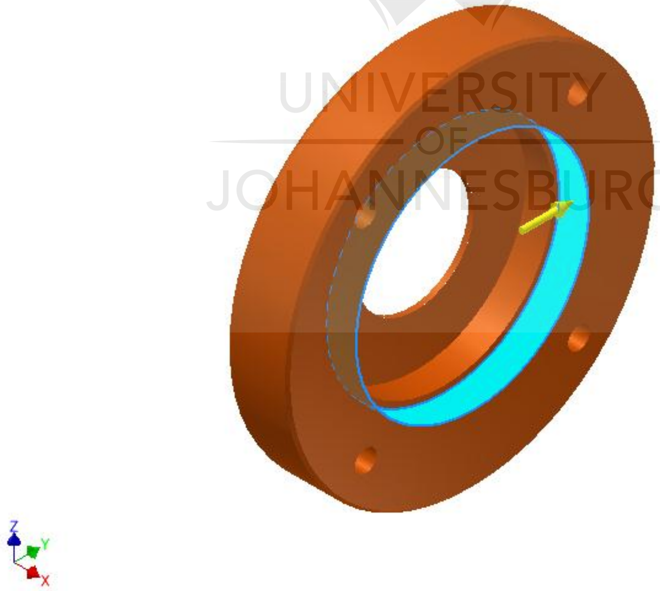
Material(s)

Name	Cast Iron	
General	Mass Density	7.25 g/cm^3
	Yield Strength	332 MPa
	Ultimate Tensile Strength	464 MPa
Stress	Young's Modulus	168 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	65.1163 GPa
Stress Thermal	Expansion Coefficient	0.000014 ul/c
	Thermal Conductivity	21 W/(m K)
	Specific Heat	540 J/(kg c)
Part Name(s)	Intermedaite shaft bearing half BB	

Bearing Load:1

Load Type	Bearing Load
Magnitude	20000.000 N
Vector X	0.000 N
Vector Y	20000.000 N
Vector Z	0.000 N

Selected Face(s)



Fixed Constraint:1

Constraint Type	Fixed Constraint

Selected Face(s)

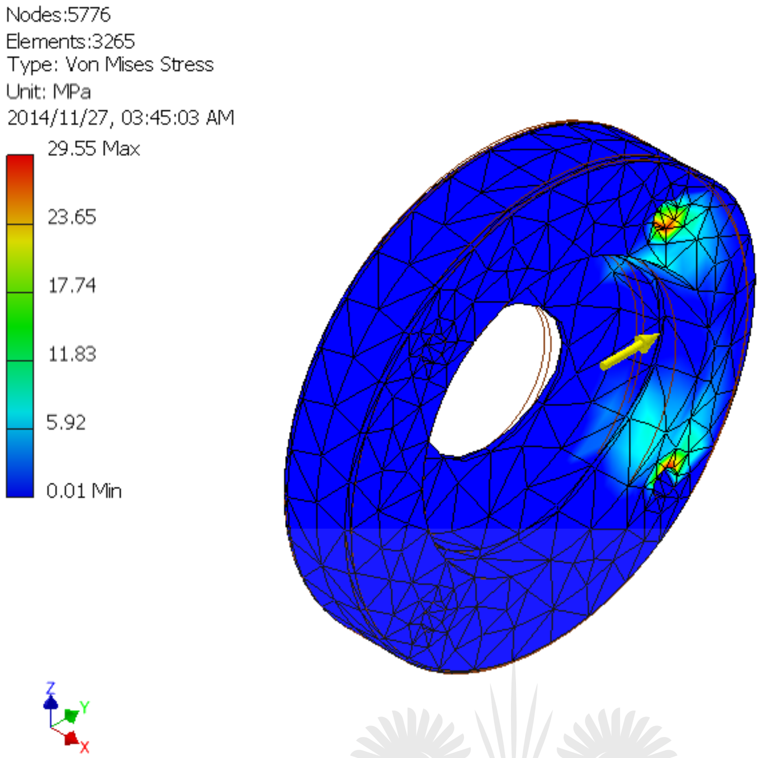
Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	20000 N	0 N	240.005 N m	0 N m
		-20000 N		0 N m
		0 N		-240.005 N m

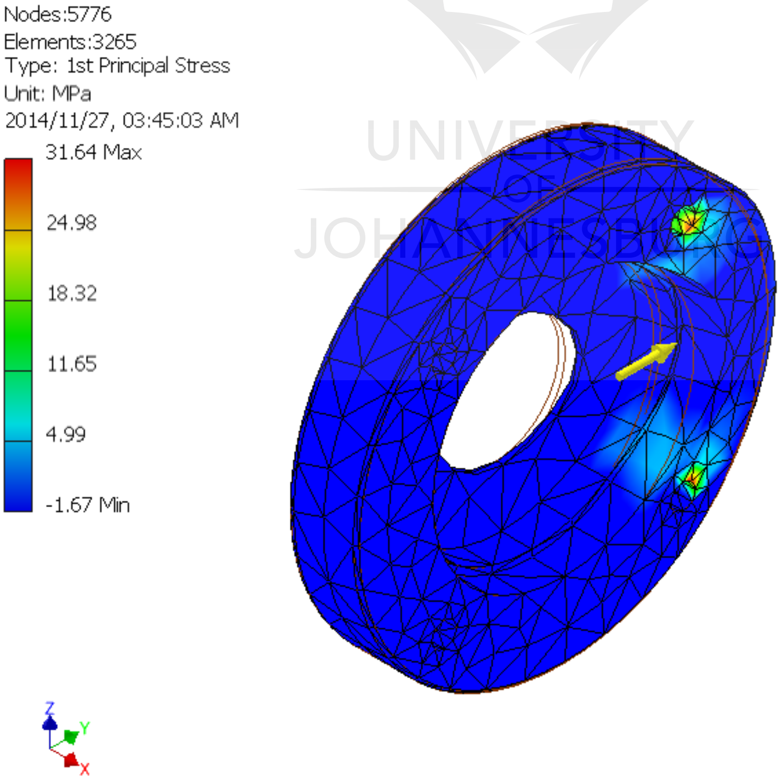
Result Summary

Name	Minimum	Maximum
Volume	1644460 mm^3	
Mass	11.6757 kg	
Von Mises Stress	0.0137044 MPa	29.553 MPa
1st Principal Stress	-1.67114 MPa	31.6424 MPa
3rd Principal Stress	-21.1898 MPa	4.85788 MPa
Displacement	0 mm	0.00722243 mm
Safety Factor	11.2341 ul	15 ul

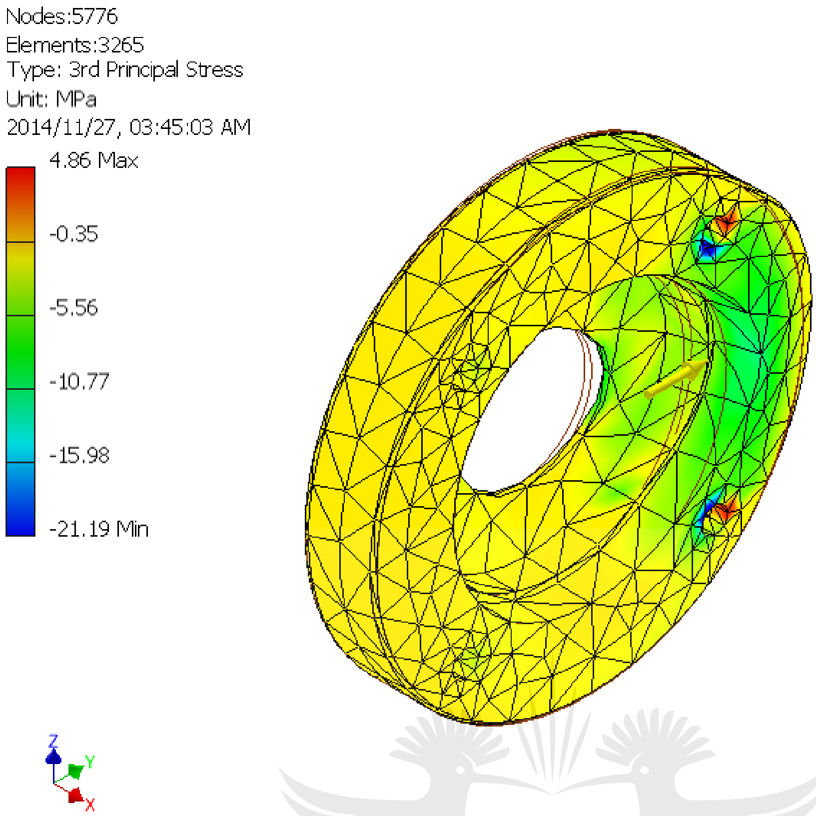
Von Mises Stress



1st Principal Stress

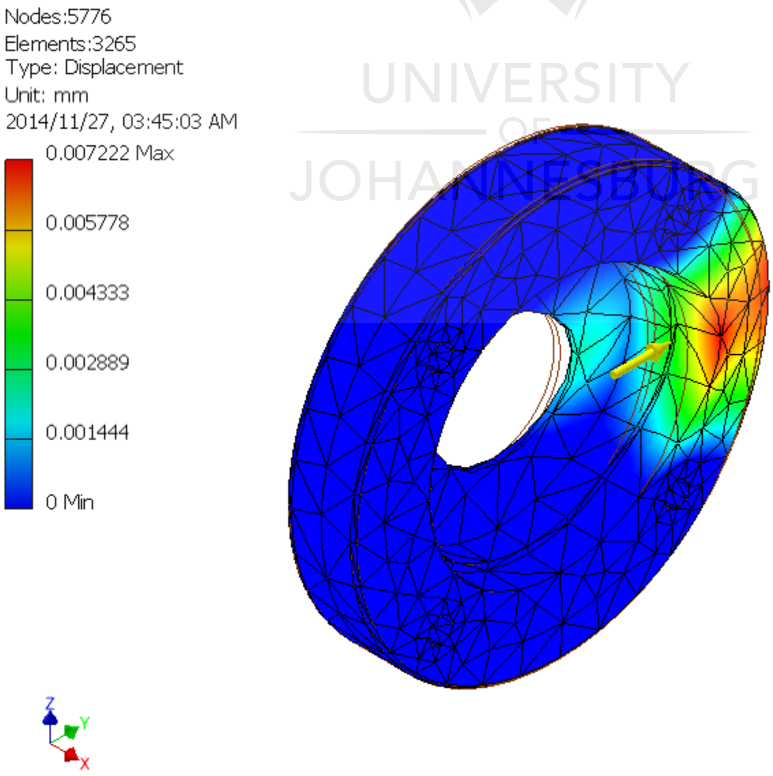


3rd Principal Stress



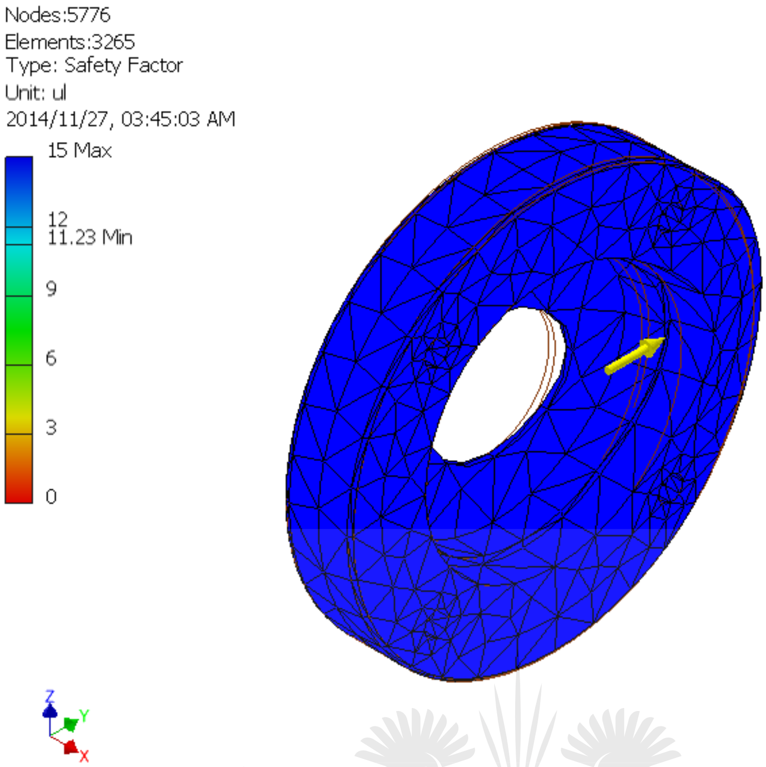
Nodes:5776
Elements:3265
Type: 3rd Principal Stress
Unit: MPa
2014/11/27, 03:45:03 AM

Displacement

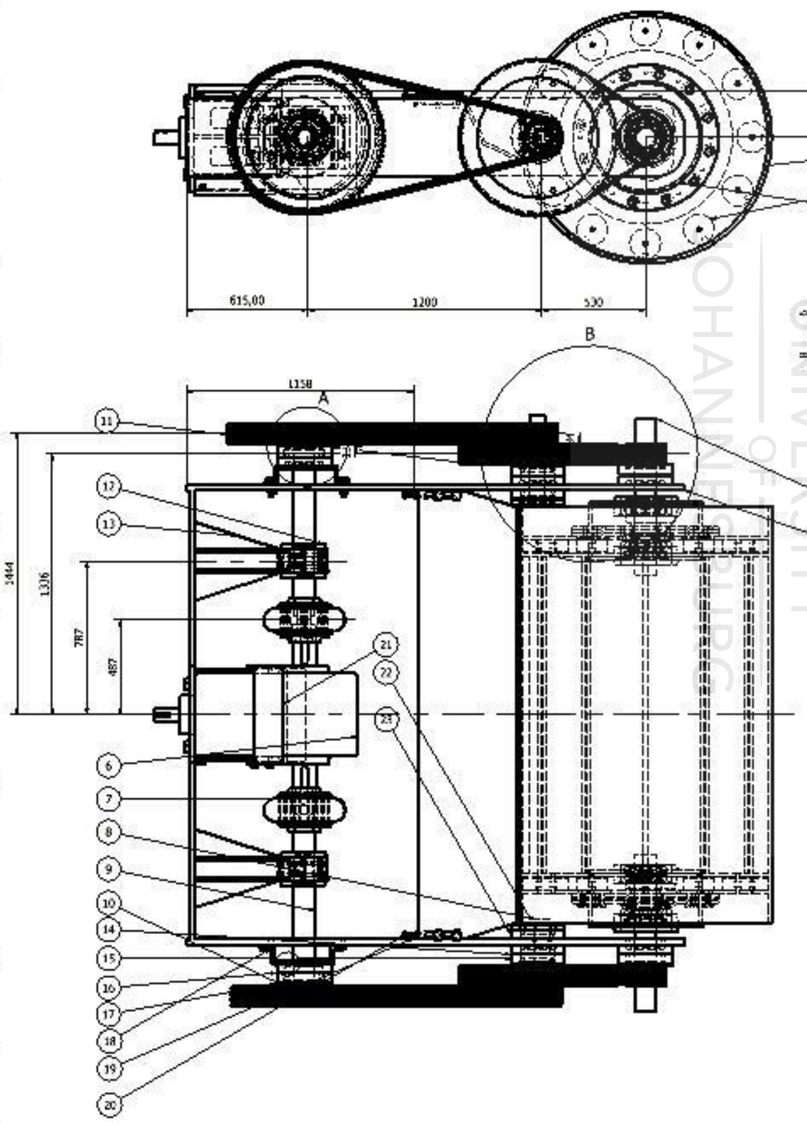


Nodes:5776
Elements:3265
Type: Displacement
Unit: mm
2014/11/27, 03:45:03 AM

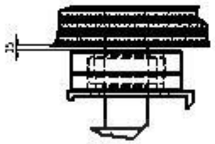
Safety Factor



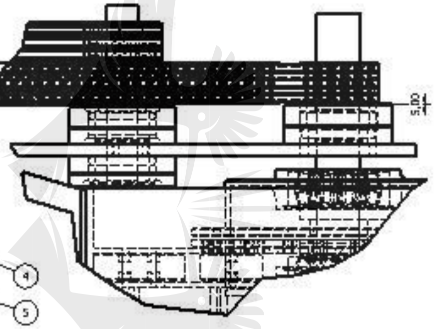
Appendix 50: Engineering manufacturing drawings.



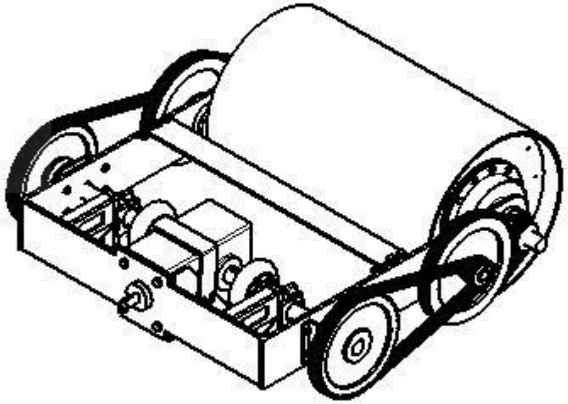
A(1:6)



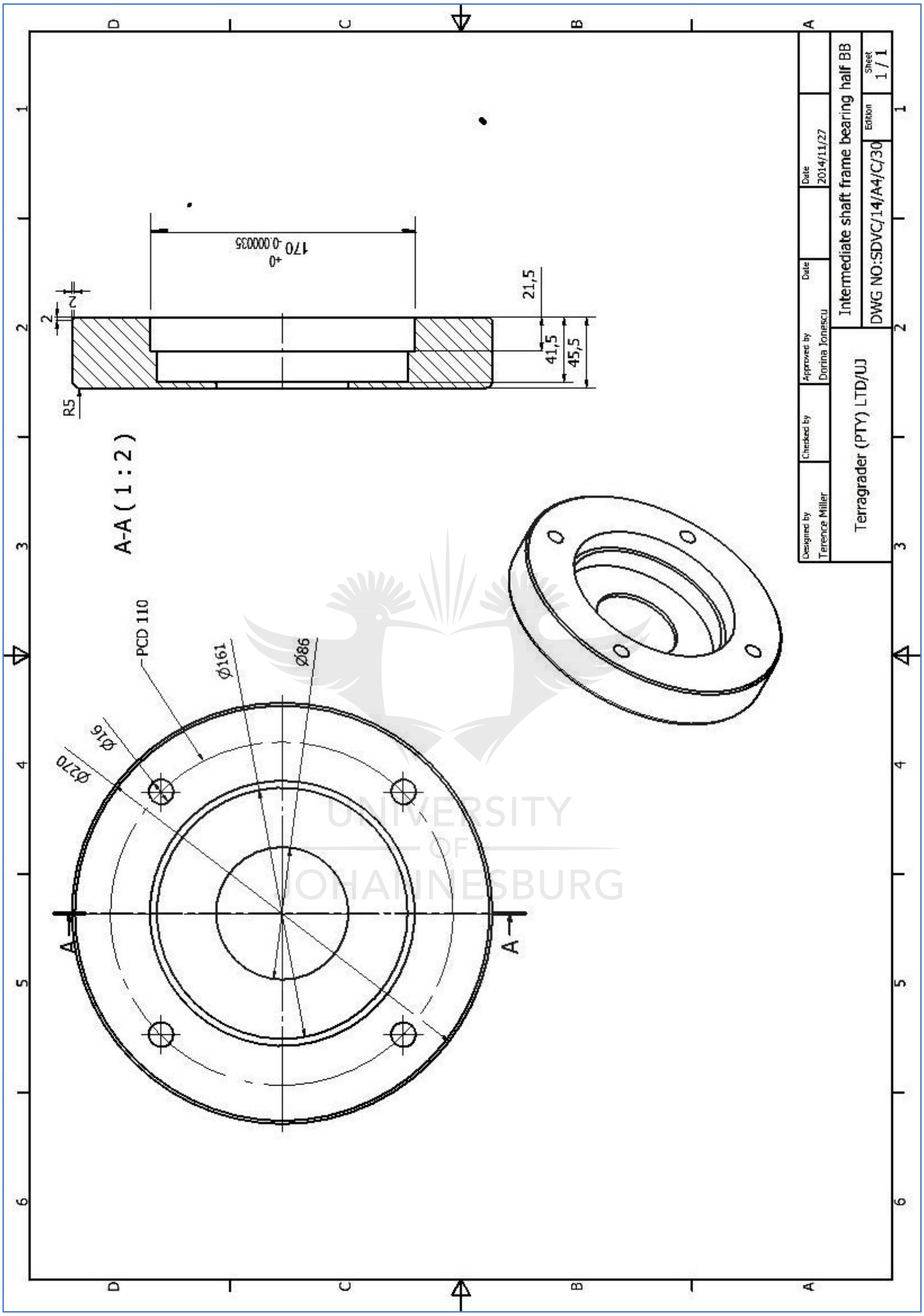
B(1:6)

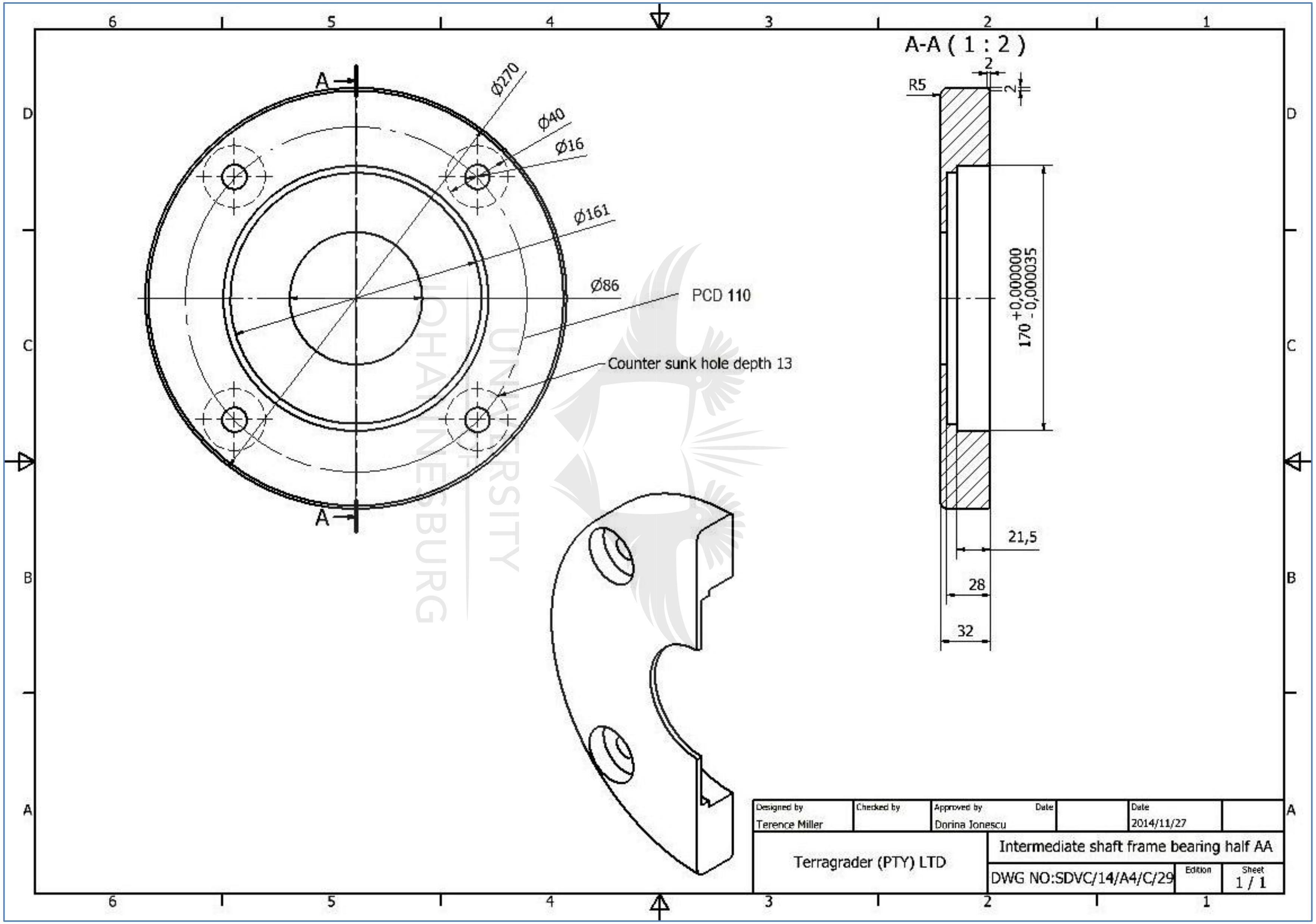


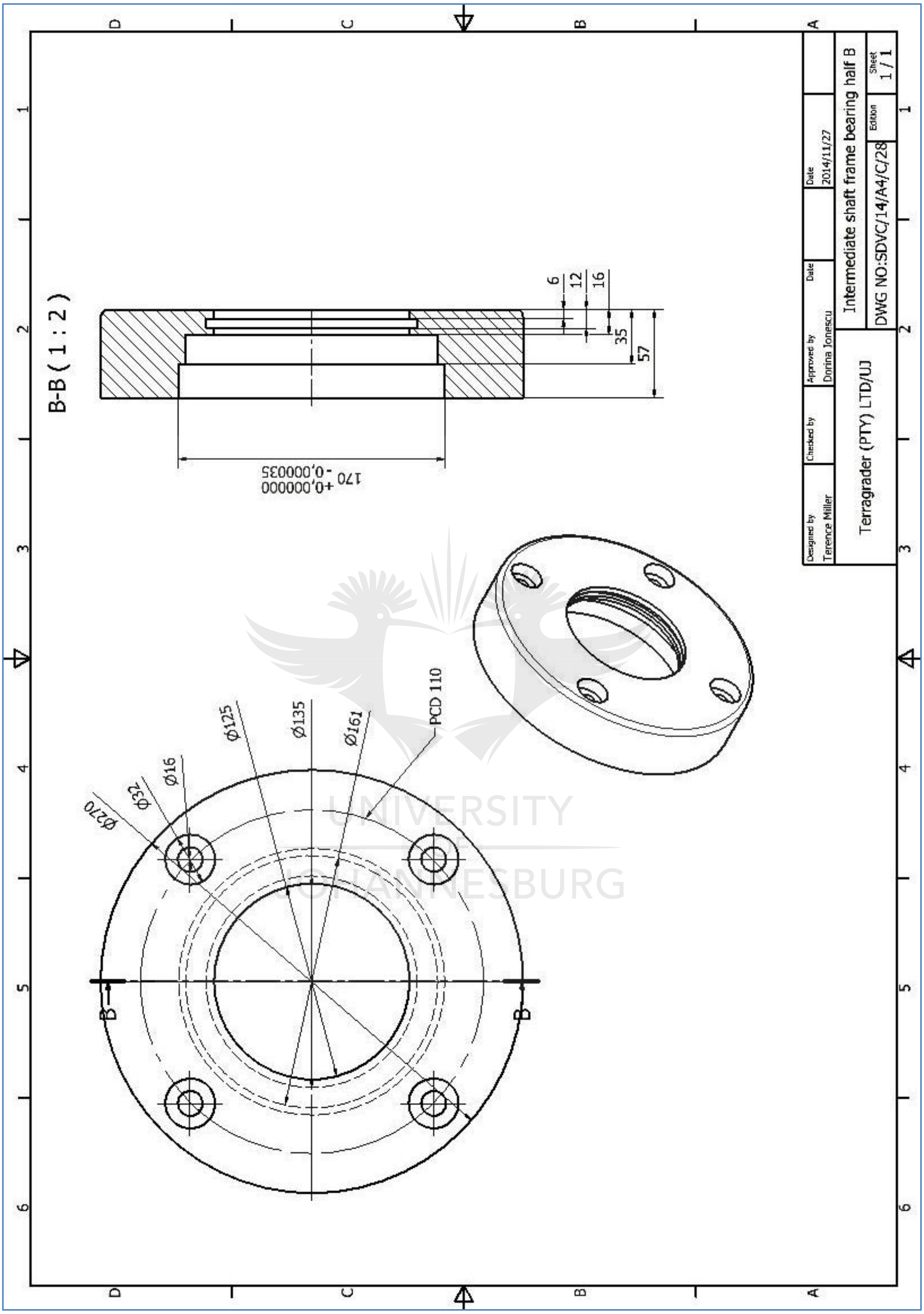
PARTS LIST			
ITEM	QTY	PART NUMBER	Material
1	1	Vibratory drum	Hardox 500 & Weldox 500
2	24	Vibration Isolator Assembley	Rubber
3	2	Eccentric Weight Housing	Cast Iron
4	2	Main Shaft	Steel 080M40
5	1	Frame	Weldox 500
6	1	HDO Gearbox	Standard
7	2	F200 Fenner-flex coupling	Rubber
8	2	Intermediate Shaft	Steel 080M40
9	2	First Shaft	Steel 080M40
10	2	Chain Drive.	Standard
11	2	Belt Drive.	Standard
12	2	Bearing FSNL_619_TL__85__213 19_K_	Housing (Spherical roller bearing)
13	2	Bearing FSNL_619_TL__85__213 19_K_ support bracket	Carbon Steel
14	2	Intermedaite shaft bearing half B	Cast Iron
15	2	Intermediate shaft bearing half A,	Cast Iron
16	4	Main/First shaft bearing half A	Cast Iron
17	4	Main/First shaft bearing half B	Cast Iron
18	2	First shaft bearing support plate	Carbon Steel
19	2	Scraper plate adjustment screw support lug	Carbon Steel
20	2	Scraper plate adjustment screw	GR 4,8
21	1	Gearbox top support bracket.	Carbon Steel
22	2	Intermedaite shaft bearing half AA	Cast Iron
23	2	Intermedaite shaft bearing half BB	Cast Iron
24		Bolts & Screws	All bolts and screws should be of GR 8,8 (Vibration)

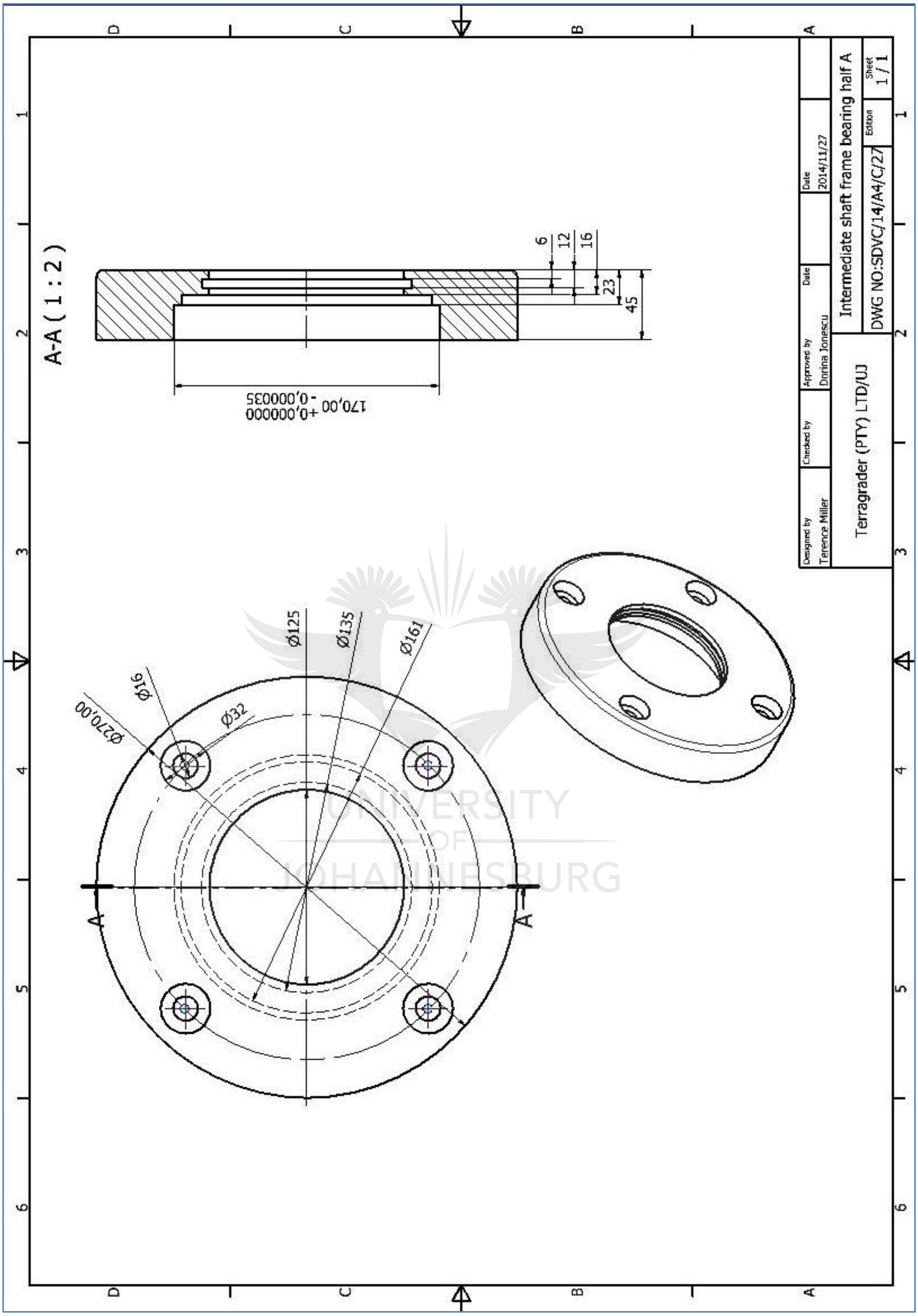


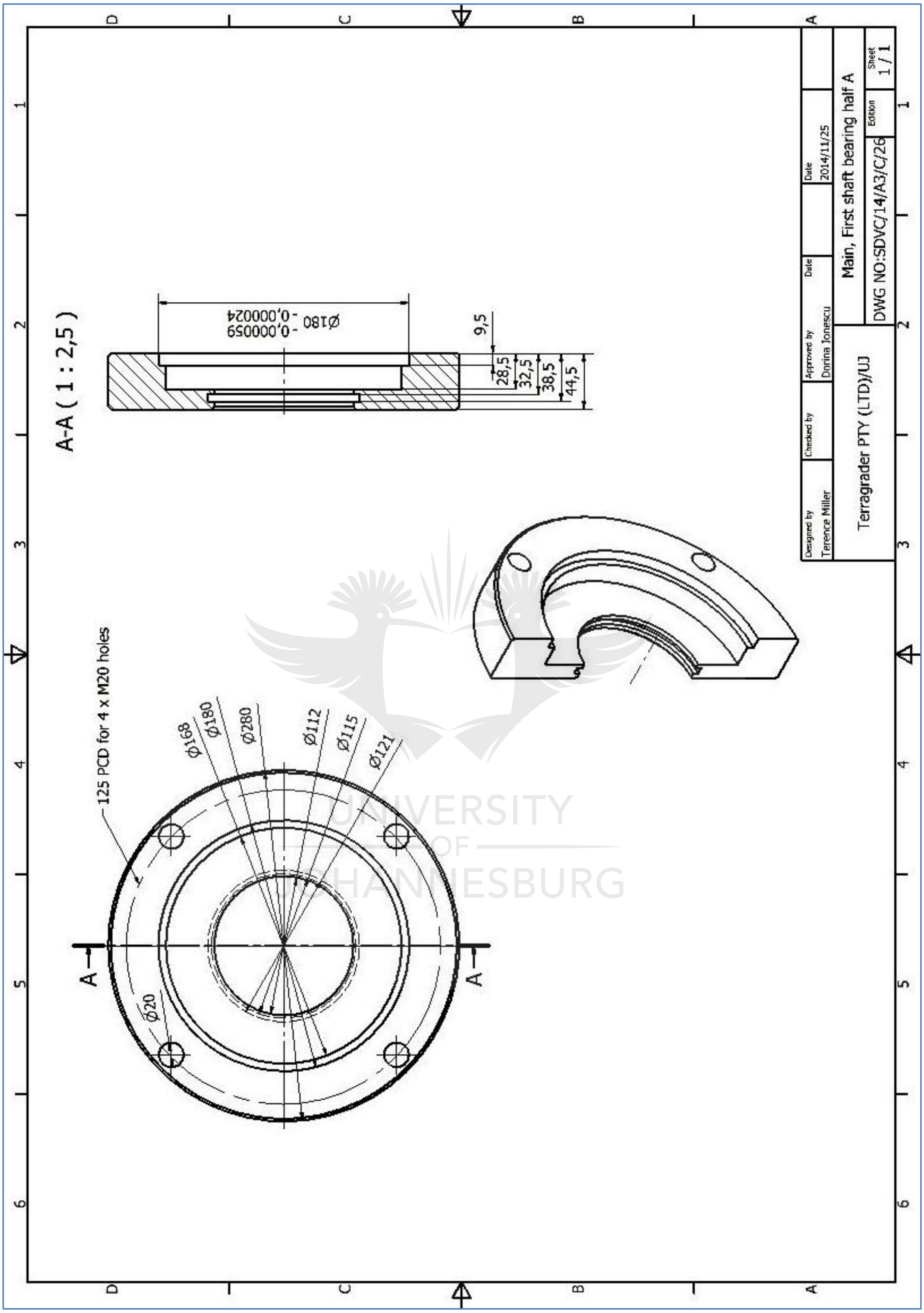
Designed by Terrance Mills	Checked by	Approved by Dunn Smidt	Date 2014/11/27	
Terragrader(PTY) LTD/UJ	Mechanical advantaged vibratory rollerAssy.			
	DWG NO:SDVC/14/A1/GA/1	Edition 1	Sheet 1 / 1	

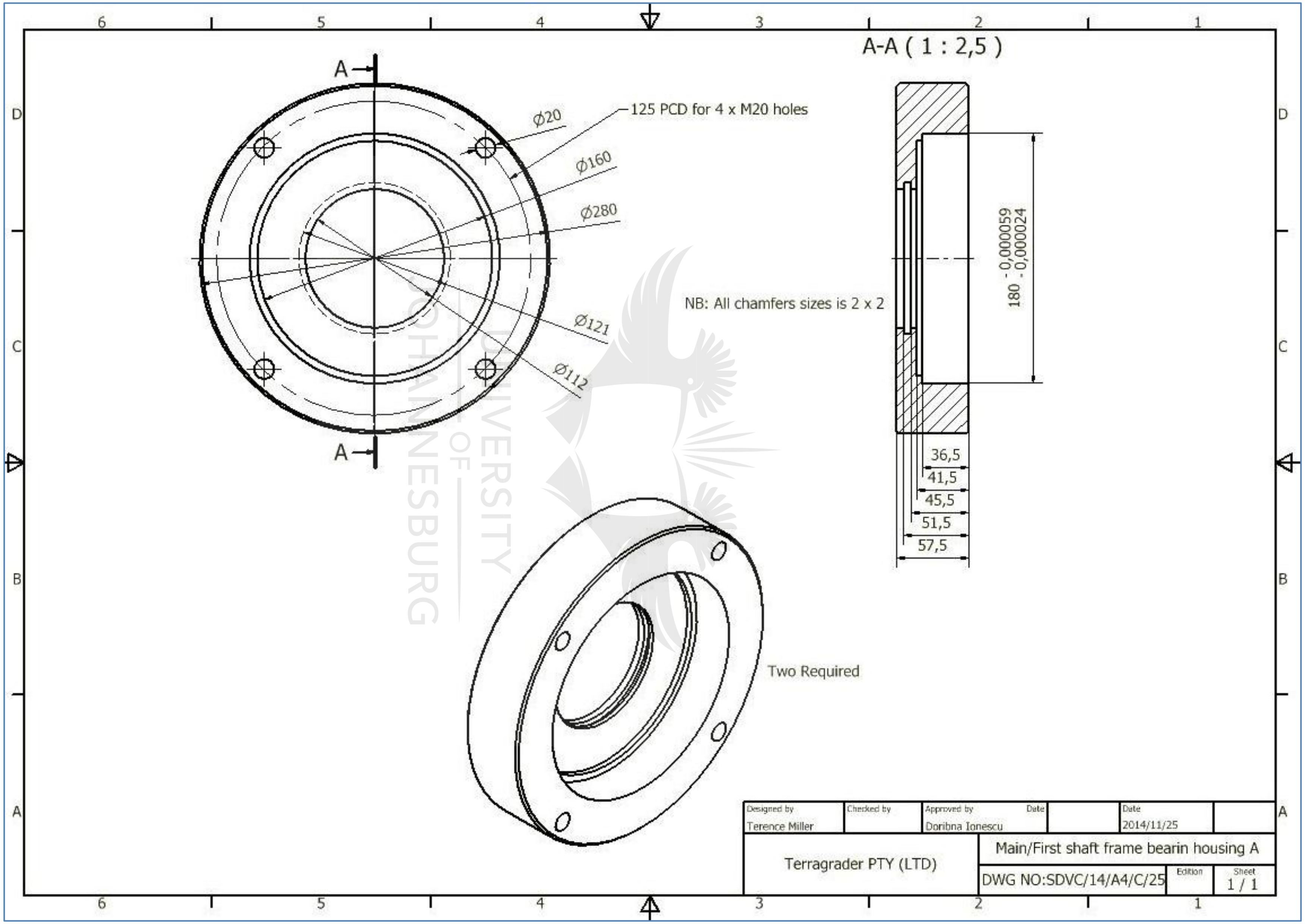

A-A (1 : 2)
170 +0 -0.000035
2
2
R5
21,5
41,5
45,5
PCD 110
Ø161
Ø86
Ø16
Ø270
A
A
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014/11/27
Terragrader (PTY) LTD/UJ
Intermediate shaft frame bearing half BB
DWG NO:SDVC/14/A4/C/30
Edition
1
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6

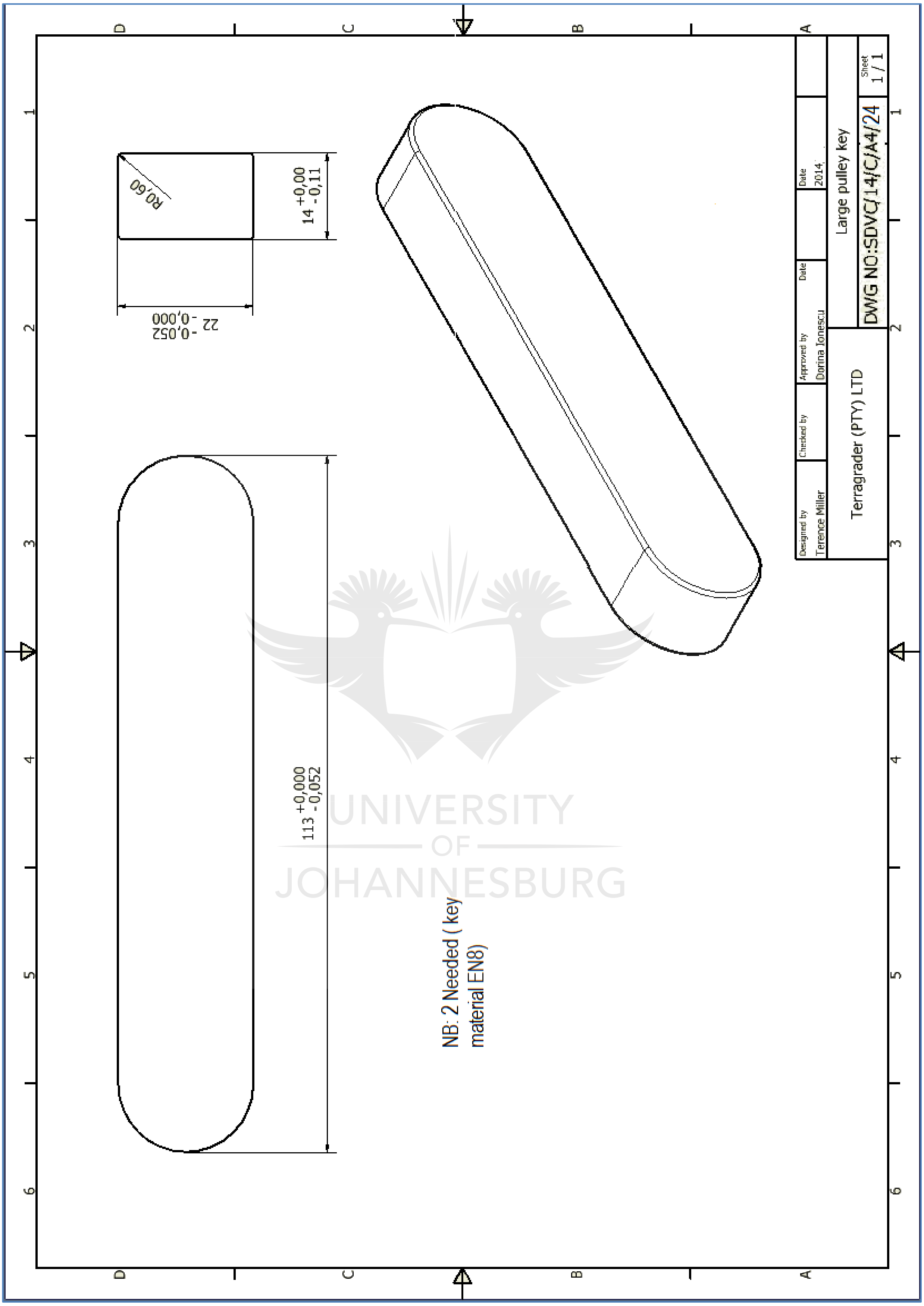

A-A (1 : 2)
A
A
Ø270
Ø40
Ø16
Ø161
Ø86
PCD 110
Counter sunk hole depth 13
R5
2
2
170 +0,000000 -0,000035
21,5
28
32
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014/11/27
Terragrader (PTY) LTD
Intermediate shaft frame bearing half AA
DWG NO:SDVC/14/A4/C/29
Edition
Sheet
1 / 1
1
2
3
4
5
6
A
B
C
D

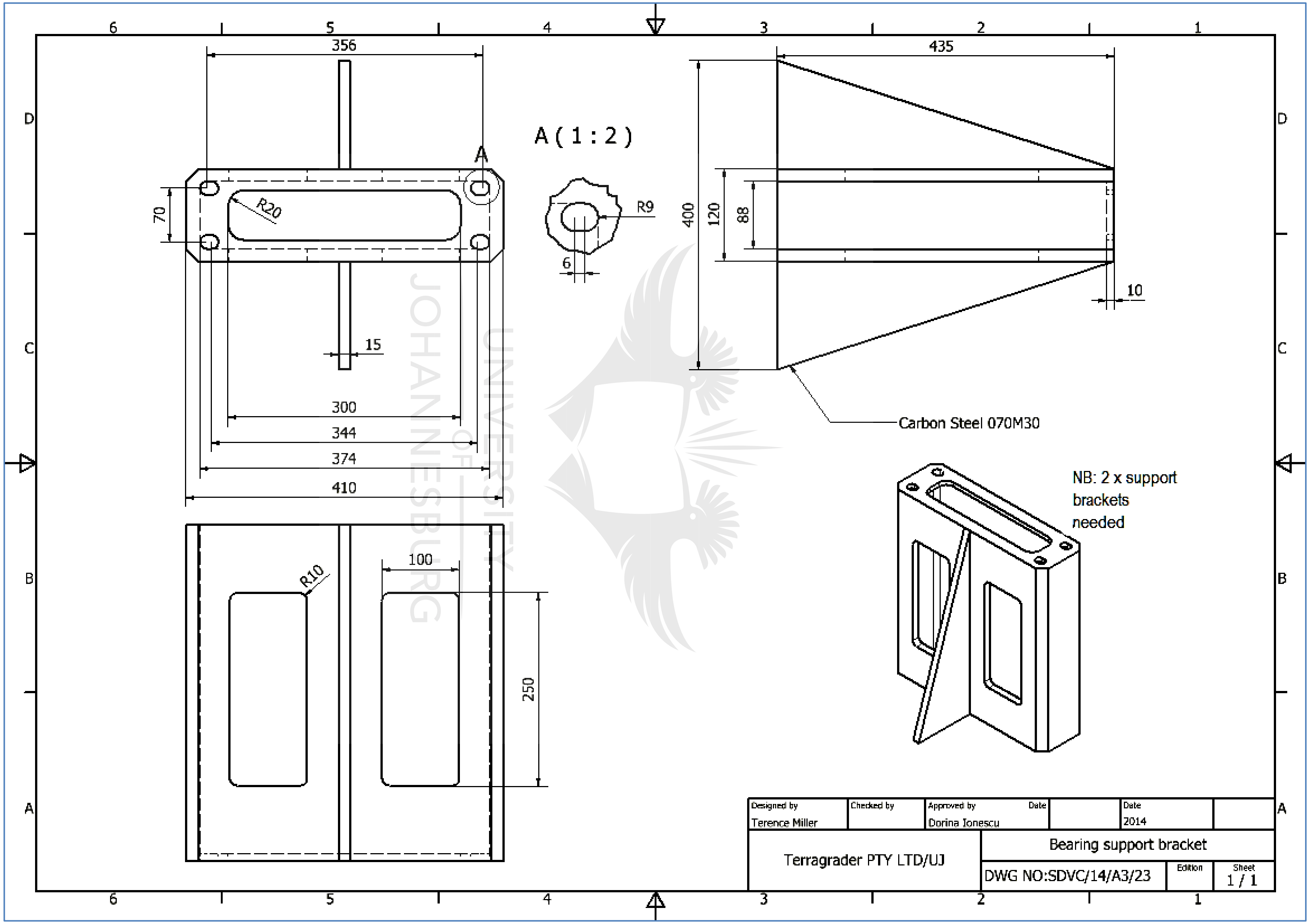

B-B (1 : 2)
6
12
16
35
57
170 +0,000000 -0,000035
PCD 110
Ø125
Ø135
Ø161
Ø16
Ø32
Ø270
B
B
A
B
C
D
1
2
3
4
5
6
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014/11/27
Intermediate shaft frame bearing half B
Terragrader (PTY) LTD/UJ
DWG NO:SDVC/14/A4/C/28
Edition
Sheet
1 / 1

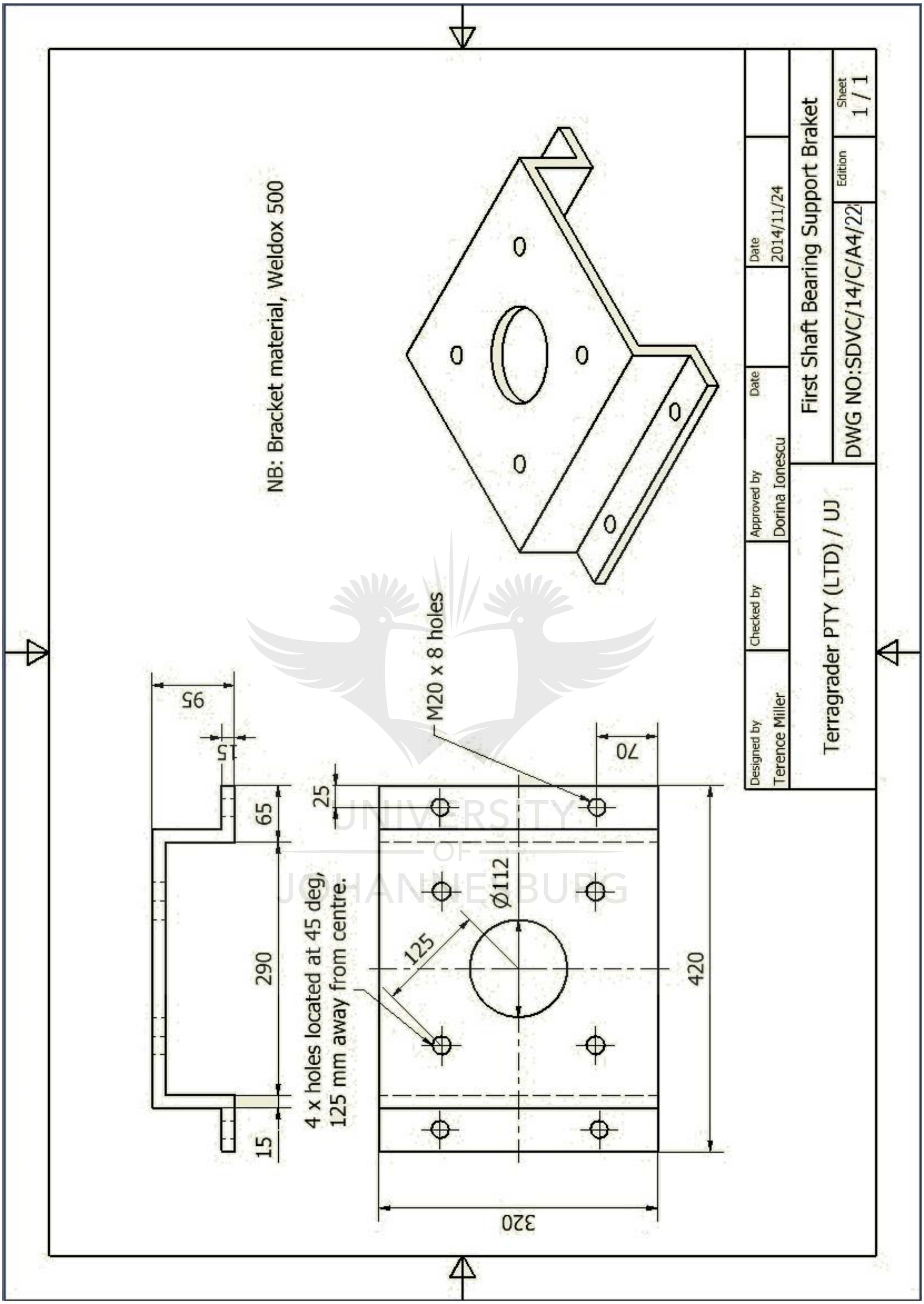

A-A (1 : 2)
6
12
16
23
45
170,00 +0,000000 -0,0000035
Ø125
Ø135
Ø161
Ø16
Ø32
Ø270,00
A
A
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014/11/27
Terragrader (PTY) LTD/UJ
Intermediate shaft frame bearing half A
DWG NO:SDVC/14/A4/C/27
Edition
1
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6

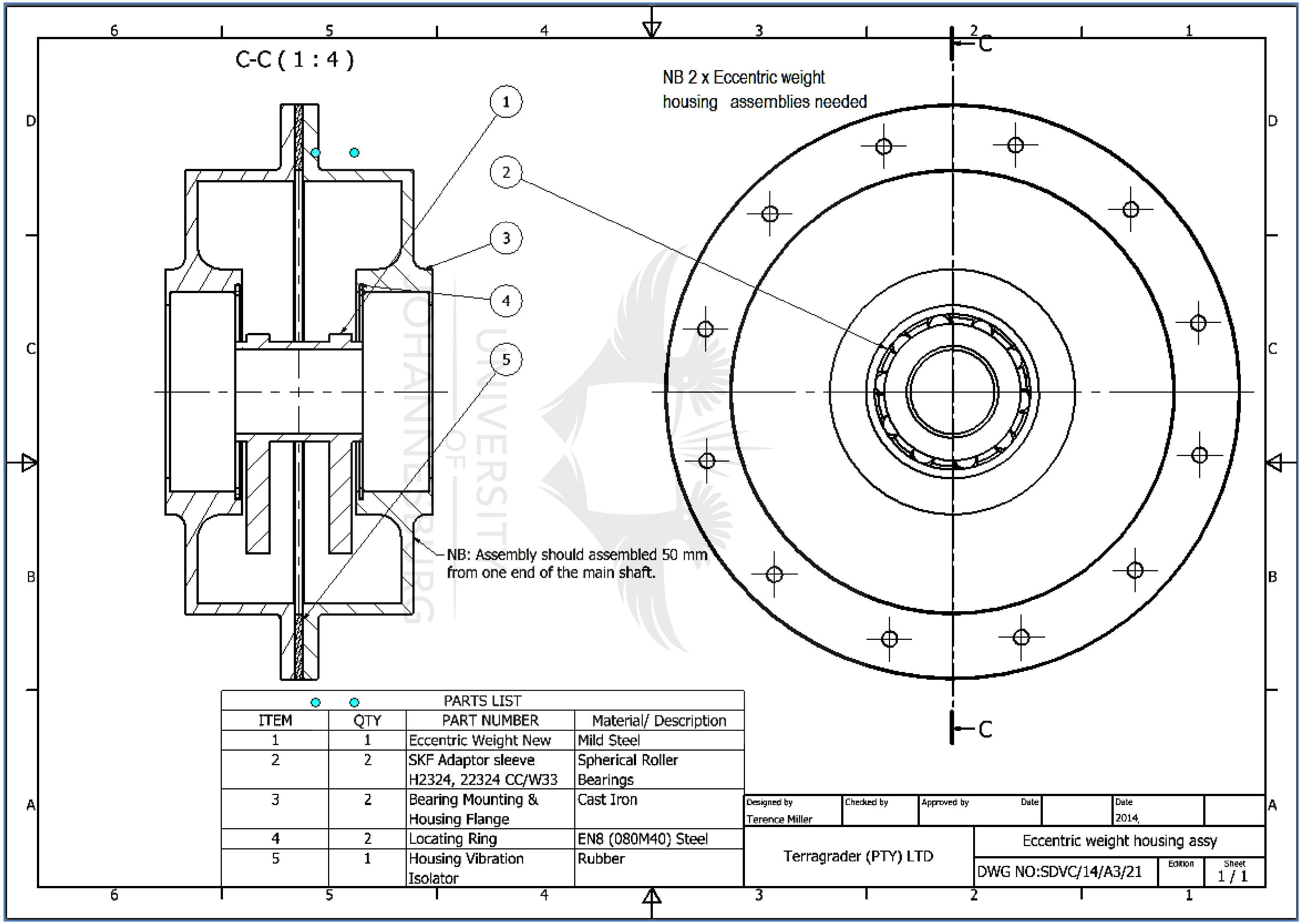

A-A (1 : 2,5)
Ø180 -0,000059 -0,000024
9,5
28,5
32,5
38,5
44,5
125 PCD for 4 x M20 holes
Ø168
Ø180
Ø280
Ø112
Ø115
Ø121
Ø20
A
A
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014/11/25
Terragrader PTY (LTD)/UJ
Main, First shaft bearing half A
DWG NO:SDVC/14/A3/C/26
Edition
1
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6


A-A (1 : 2,5)
A
A
Ø20
125 PCD for 4 x M20 holes
Ø160
Ø280
Ø121
Ø112
NB: All chamfers sizes is 2 x 2
180 - 0,000059 - 0,000024
36,5
41,5
45,5
51,5
57,5
Two Required
Designed by
Terence Miller
Checked by
Approved by
Doribna Ionescu
Date
Date
2014/11/25
Terragrader PTY (LTD)
Main/First shaft frame bearin housing A
DWG NO:SDVC/14/A4/C/25
Edition
Sheet
1 / 1
6
5
4
3
2
1
D
C
B
A


R0,60
14 +0,00 -0,11
22 -0,000 -0,052
113 +0,000 -0,052
NB: 2 Needed (key material EN8)
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Terragrader (PTY) LTD
Large pulley key
DWG NO:SDVC/14/C/A4/24
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6


6
5
4
3
2
1
D
C
B
A
356
A
70
R20
15
300
344
374
410
A (1 : 2)
R9
6
435
400
120
88
10
Carbon Steel 070M30
NB: 2 x support brackets needed
R10
100
250
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Terragrader PTY LTD/UJ
Bearing support bracket
DWG NO:SDVC/14/A3/23
Edition
Sheet
1 / 1


NB: Bracket material, Weldox 500
M20 x 8 holes
4 x holes located at 45 deg, 125 mm away from centre.
95
15
25
65
290
15
420
320
70
125
Ø112
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014/11/24
First Shaft Bearing Support Braket
Terragrader PTY (LTD) / UJ
DWG NO:SDVC/14/C/A4/22
Edition
Sheet
1 / 1



NB 2 x Eccentric weight housing assemblies needed

NB: Assembly should assembled 50 mm from one end of the main shaft.

PARTS LIST

ITEM	QTY	PART NUMBER	Material/ Description
1	1	Eccentric Weight New	Mild Steel
2	2	SKF Adaptor sleeve H2324, 22324 CC/W33	Spherical Roller Bearings
3	2	Bearing Mounting & Housing Flange	Cast Iron
4	2	Locating Ring	EN8 (080M40) Steel
5	1	Housing Vibration Isolator	Rubber

Designed by	Checked by	Approved by	Date	Date
Terence Miller				2014,

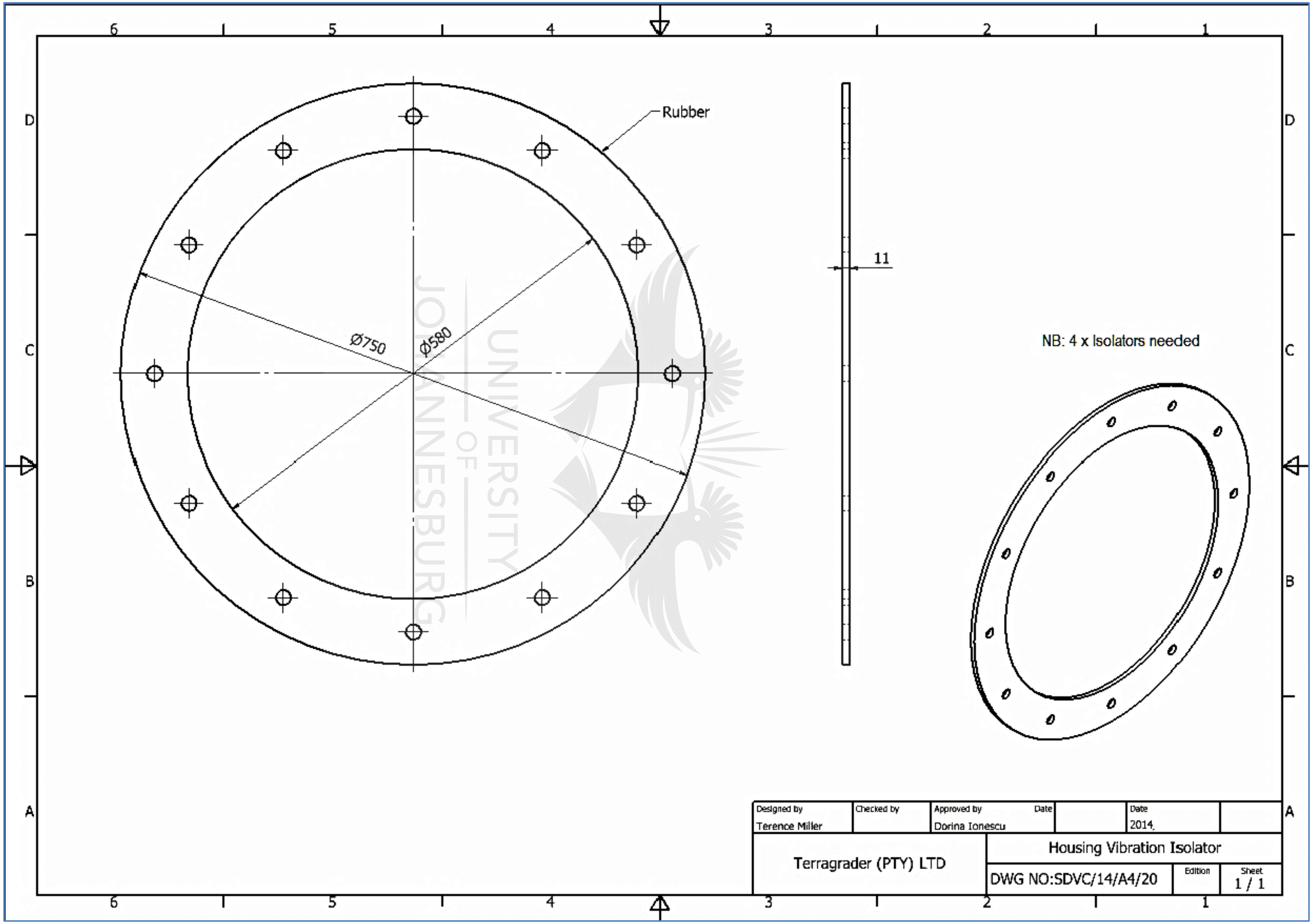
Terragrader (PTY) LTD

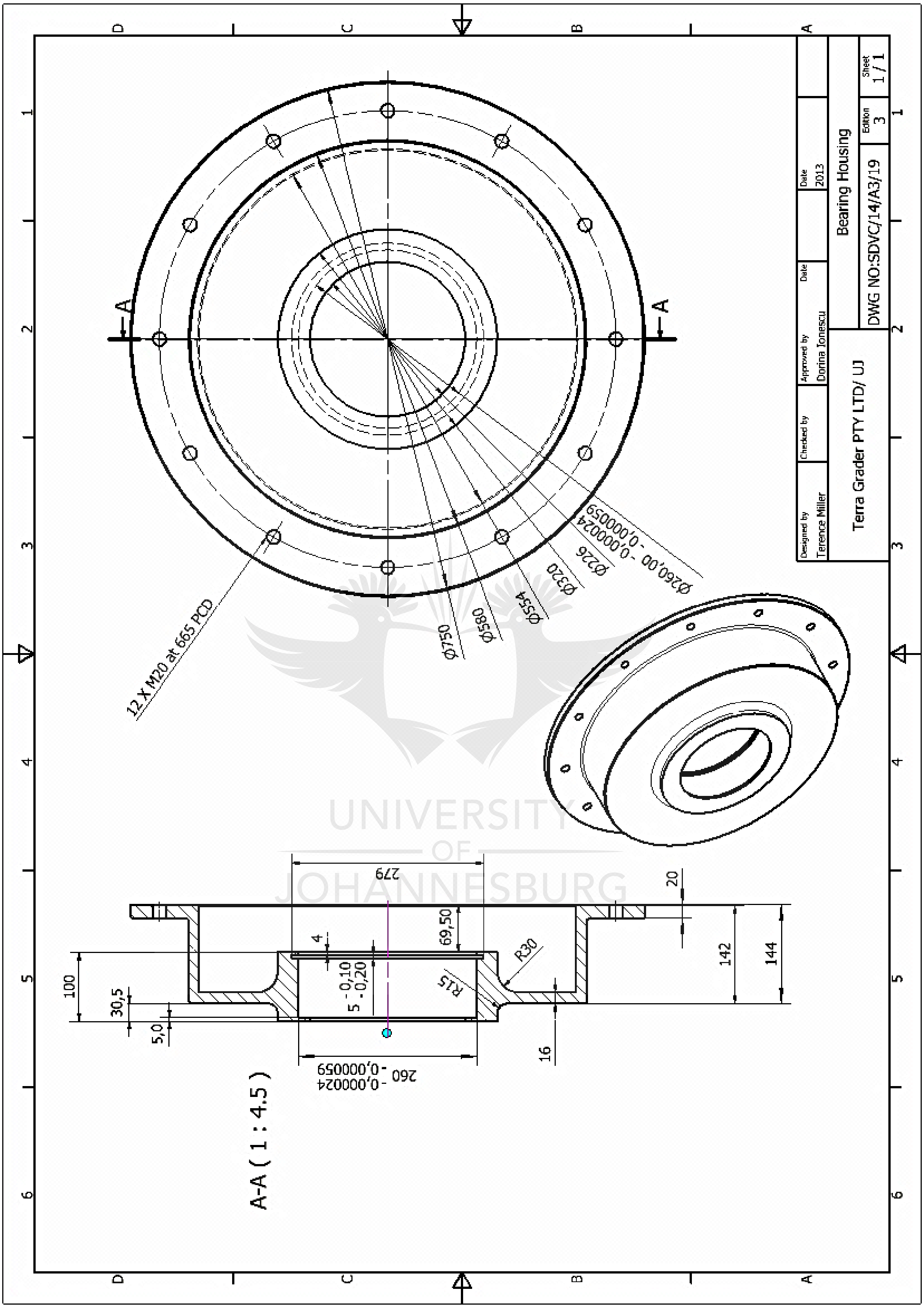
Eccentric weight housing assy

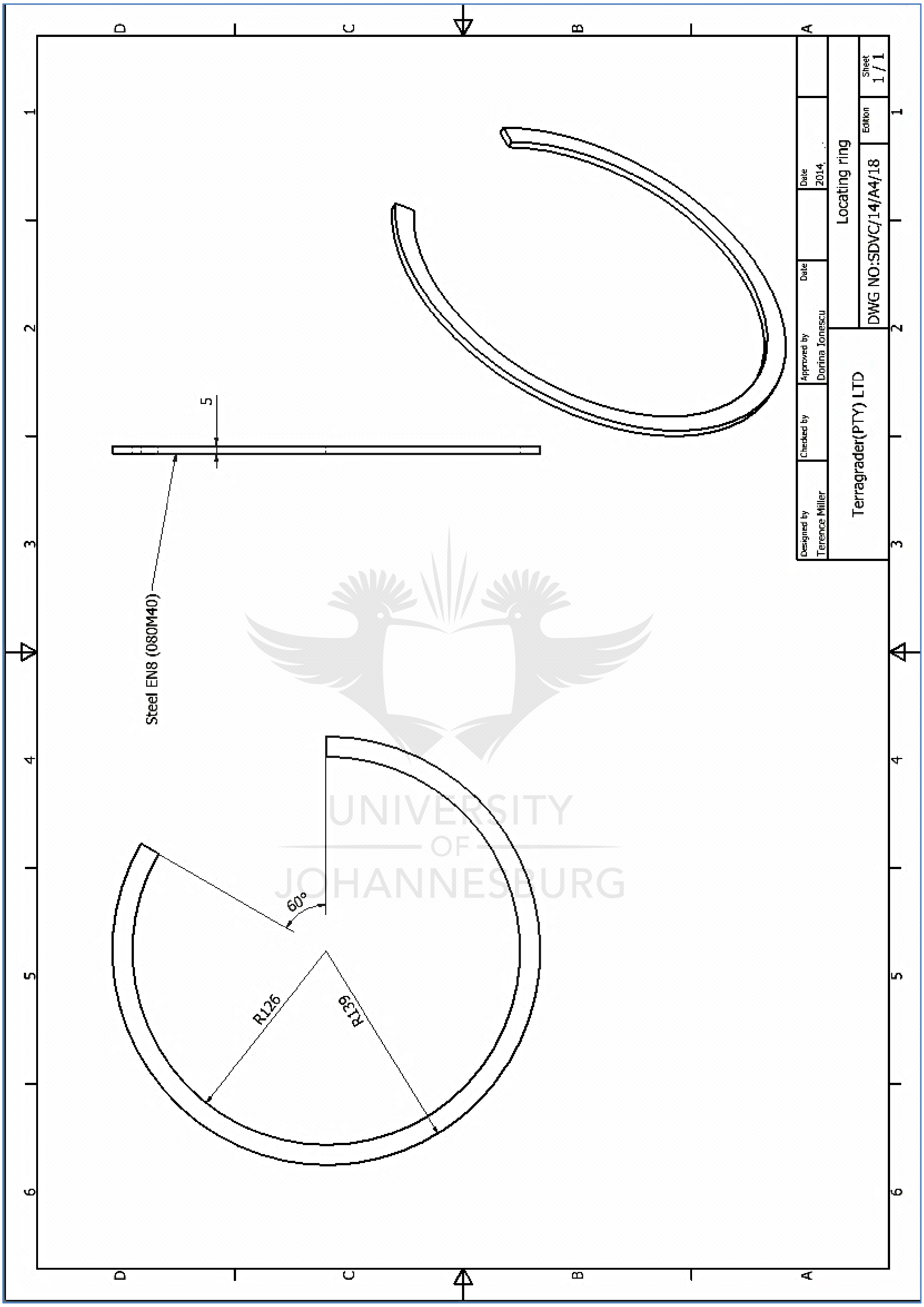
DWG NO:SDVC/14/A3/21

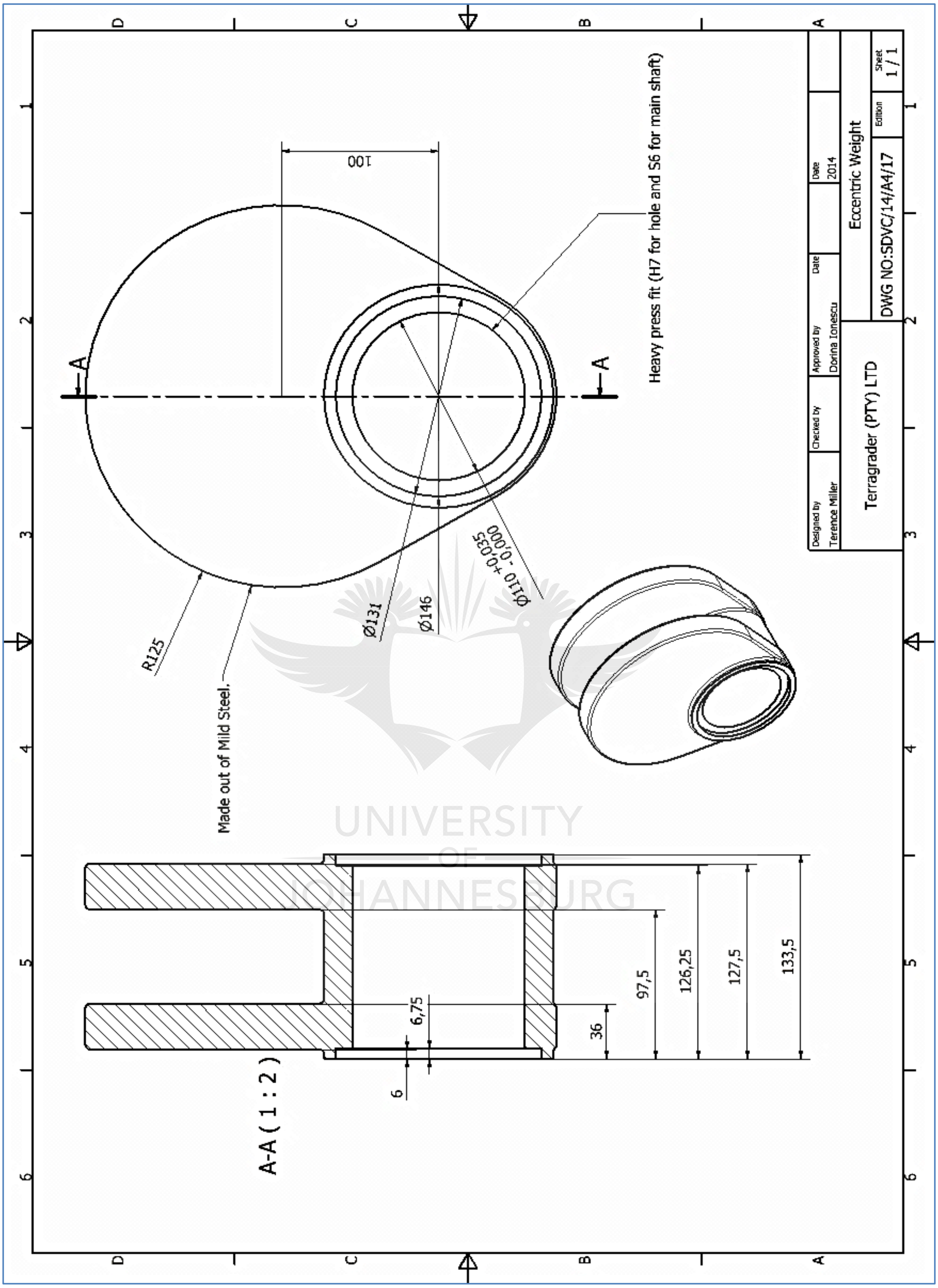
Edition

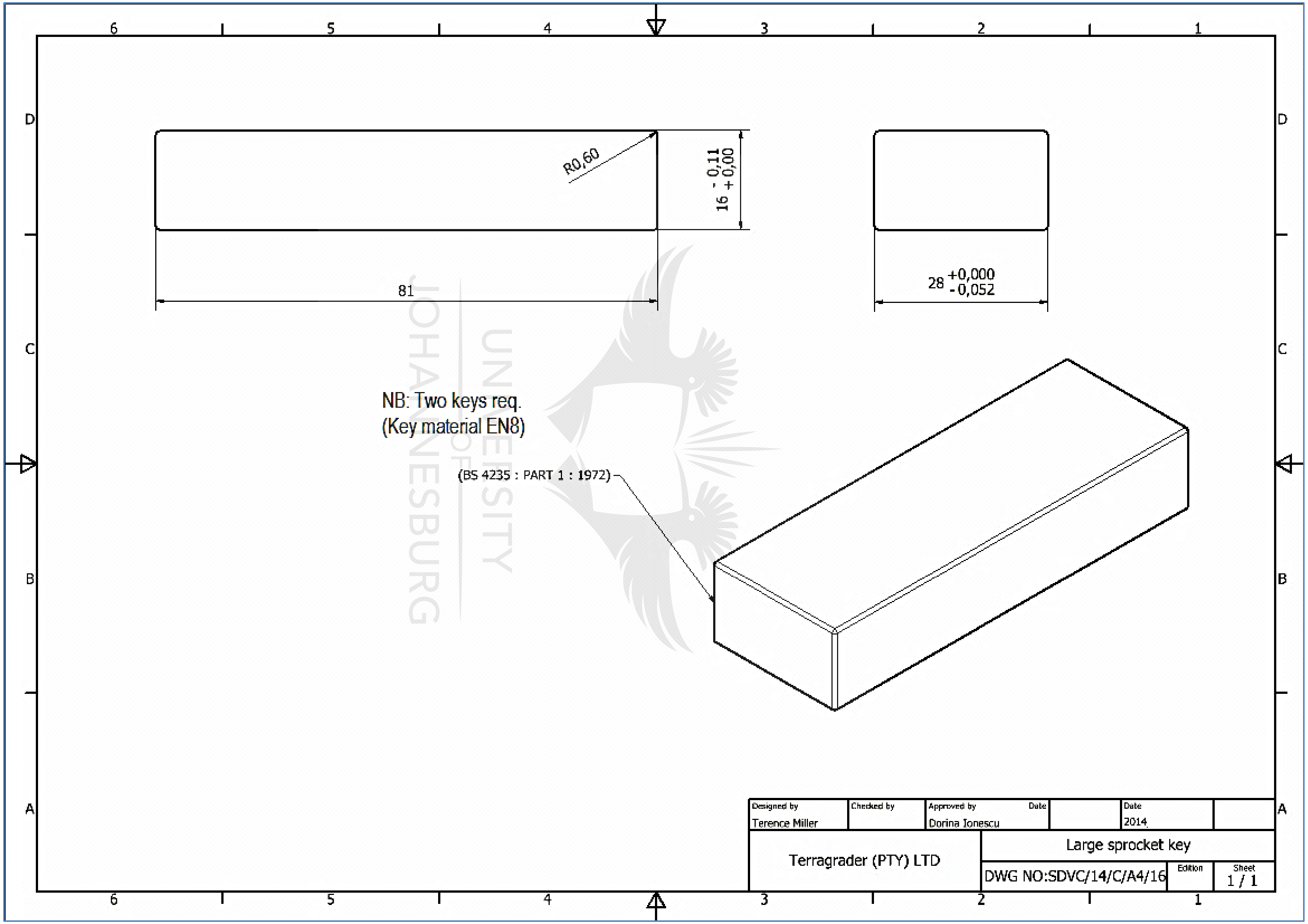
Sheet 1 / 1

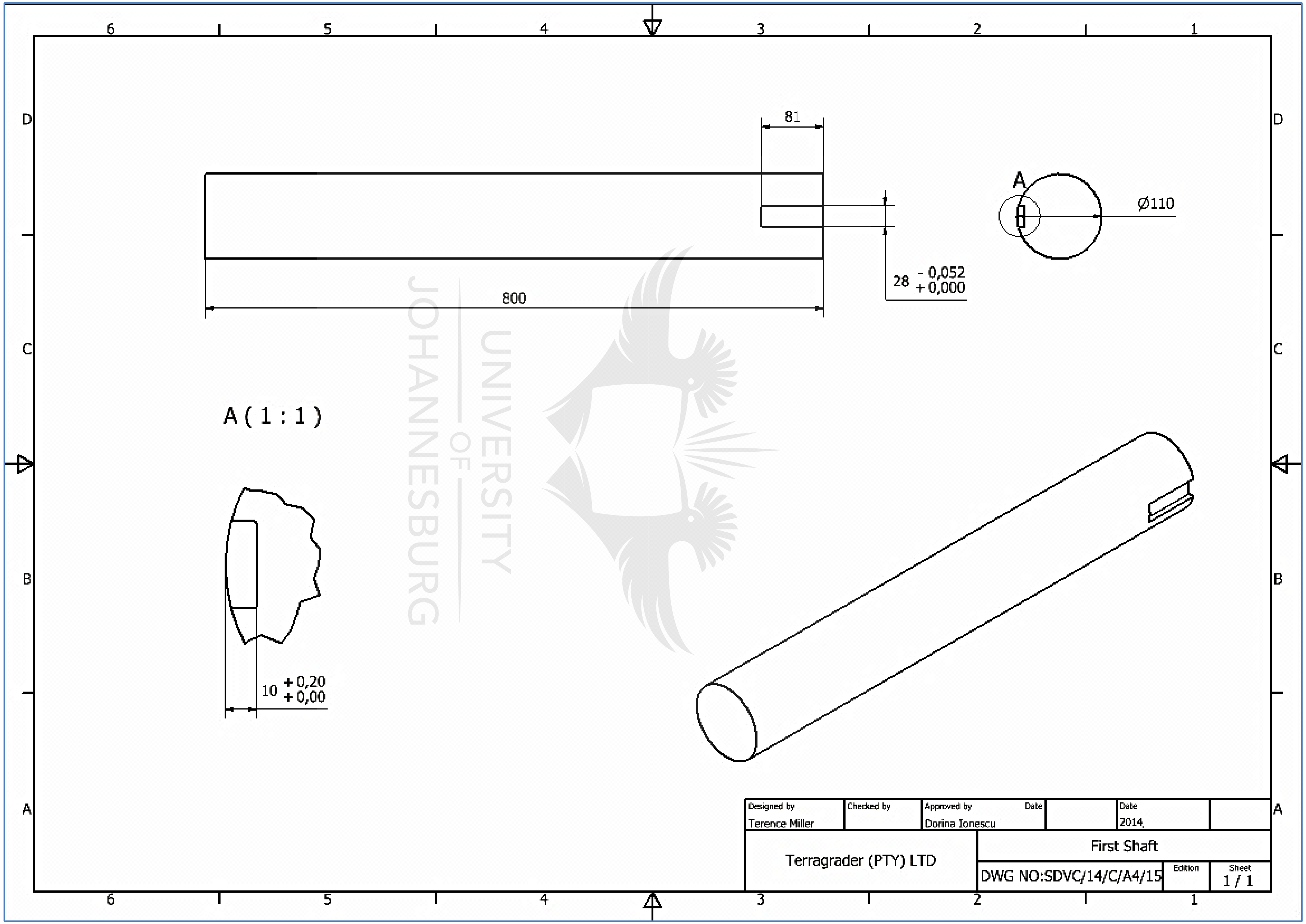

6
5
4
3
2
1
D
C
B
A
Rubber
Ø750
Ø580
11
NB: 4 x Isolators needed
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014.
Terragrader (PTY) LTD
Housing Vibration Isolator
DWG NO:SDVC/14/A4/20
Edition
Sheet
1 / 1

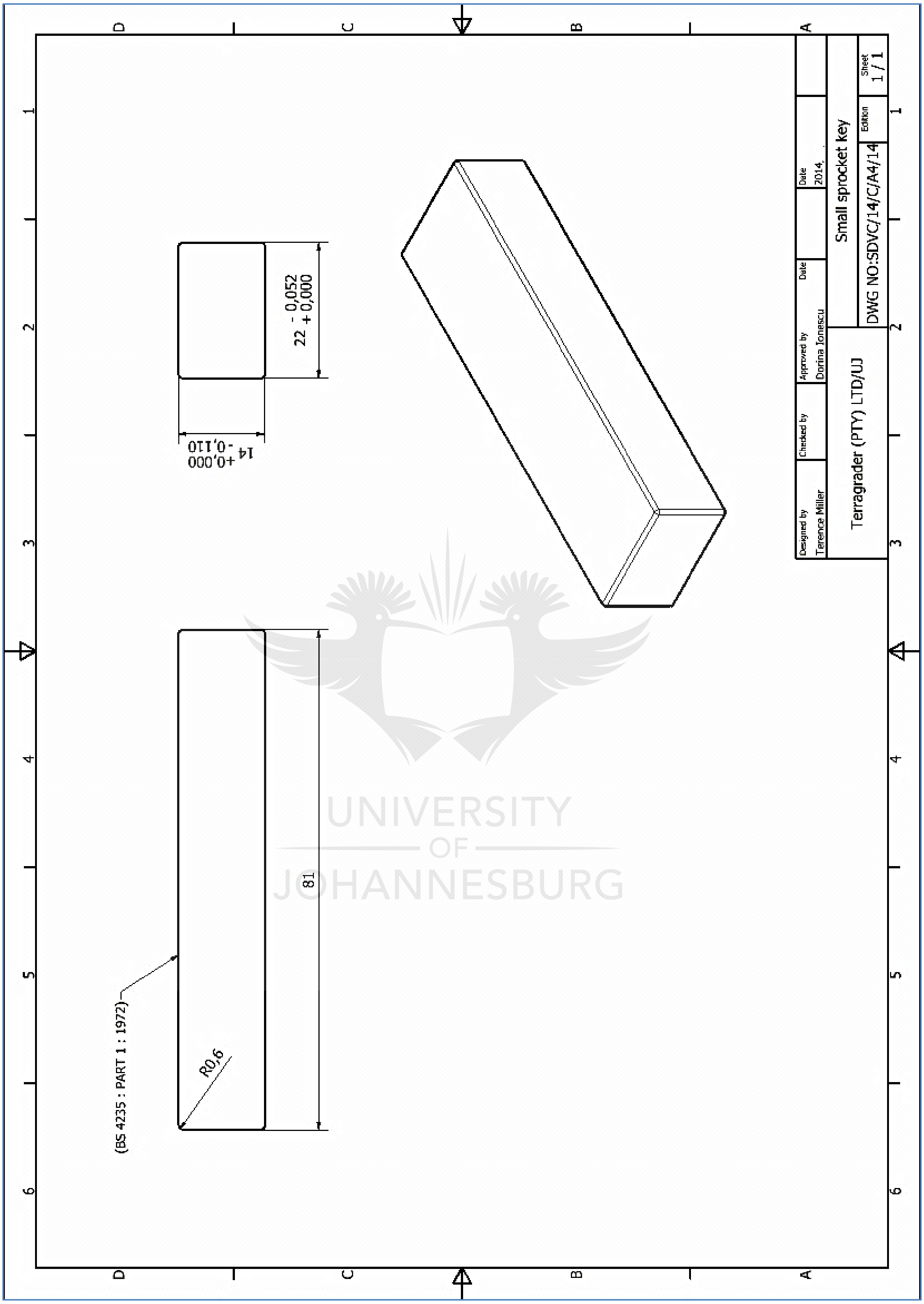

A
A
Ø750
Ø580
Ø554
Ø320
Ø226
Ø260,00 - 0,000024 - 0,000059
12 X M20 at 665 PCD
A-A (1 : 4.5)
279
69,50
4
5 - 0,10 - 0,20
R30
R15
20
142
144
100
30,5
5,0
16
260 - 0,000024 - 0,000059
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2013
Bearing Housing
Edition
3
Sheet
1 / 1
DWG NO:SDVC/14/A3/19
Terra Grader PTY LTD/ UJ
A
B
C
D
1
2
3
4
5
6

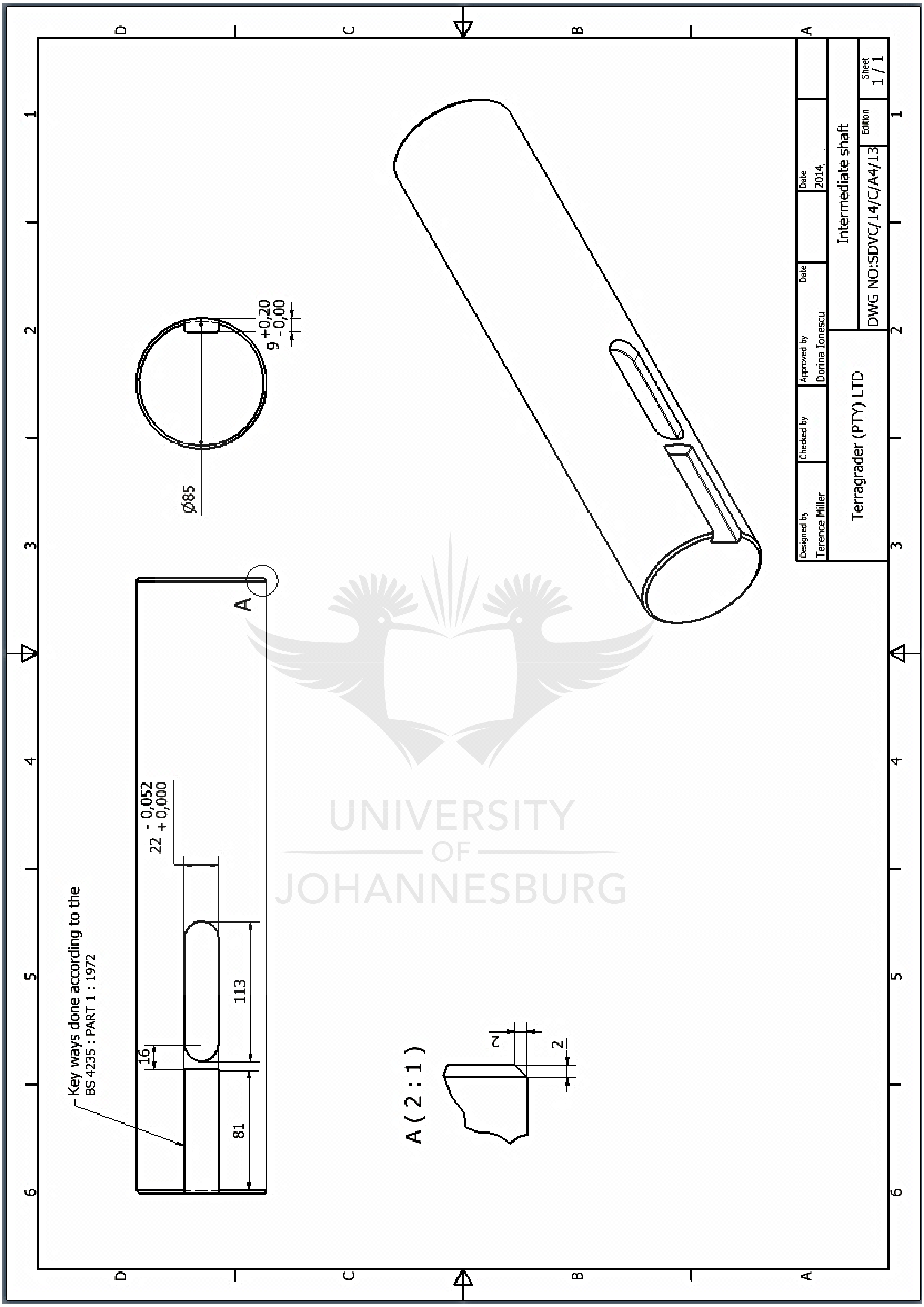

Steel EN8 (080M40)
5
60°
R126
R139
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Locating ring
Terragrader(PTY) LTD
DWG NO:SDVC/14/A4/18
Edition
1
Sheet
1 / 1

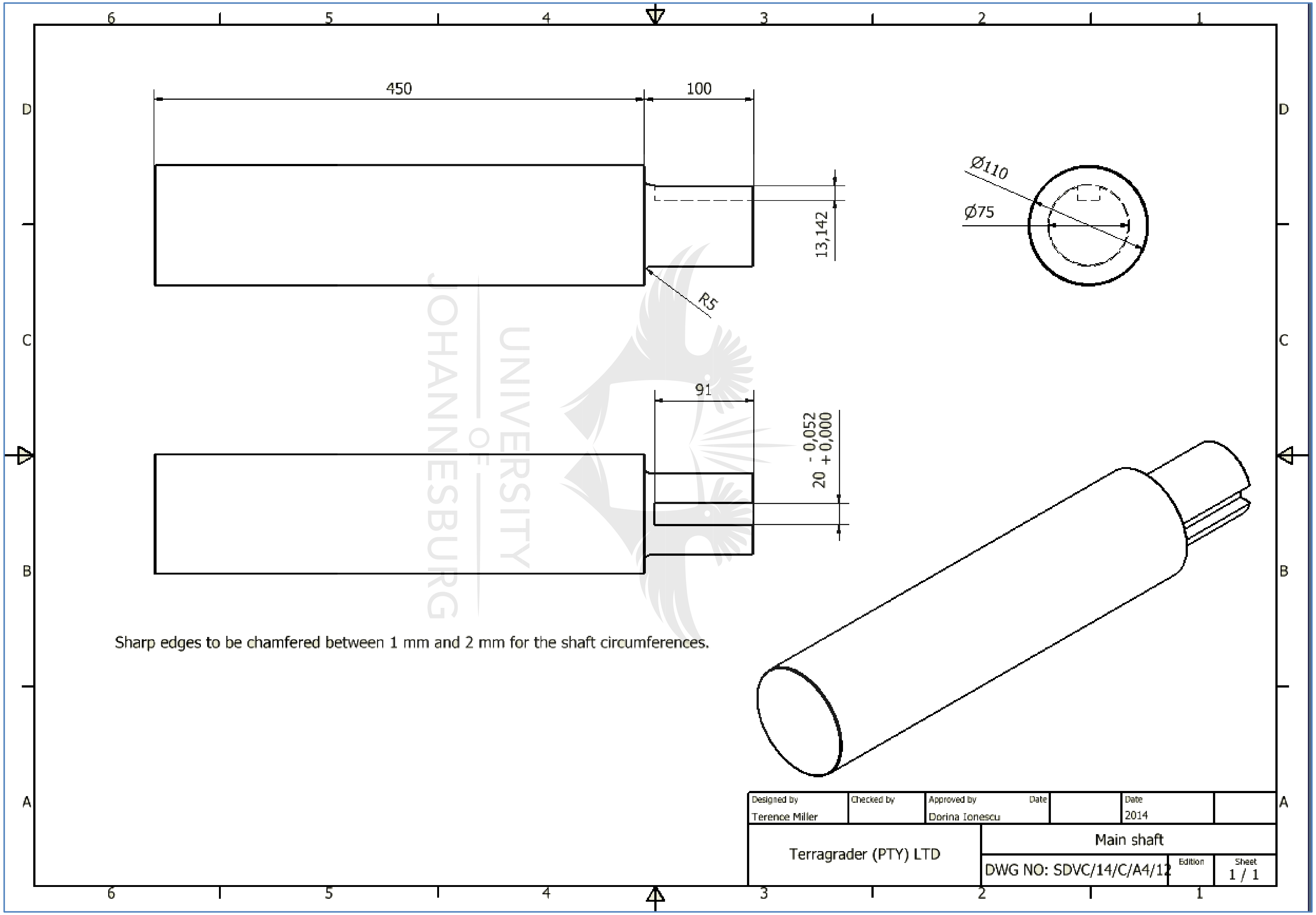

100
Heavy press fit (H7 for hole and S6 for main shaft)
A
A
R125
Made out of Mild Steel.
Ø131
Ø146
Ø110 +0,035 -0,0000
A-A (1 : 2)
6
6,75
36
97,5
126,25
127,5
133,5
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Terragrader (PTY) LTD
Eccentric Weight
DWG NO:SDVC/14/A4/17
Edition
1
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6


6
5
4
3
2
1
D
C
B
A
R0,60
16 - 0,11 + 0,00
81
28 +0,000 -0,052
NB: Two keys req.
(Key material EN8)
(BS 4235 : PART 1 : 1972)
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Terragrader (PTY) LTD
Large sprocket key
DWG NO:SDVC/14/C/A4/16
Edition
Sheet
1 / 1

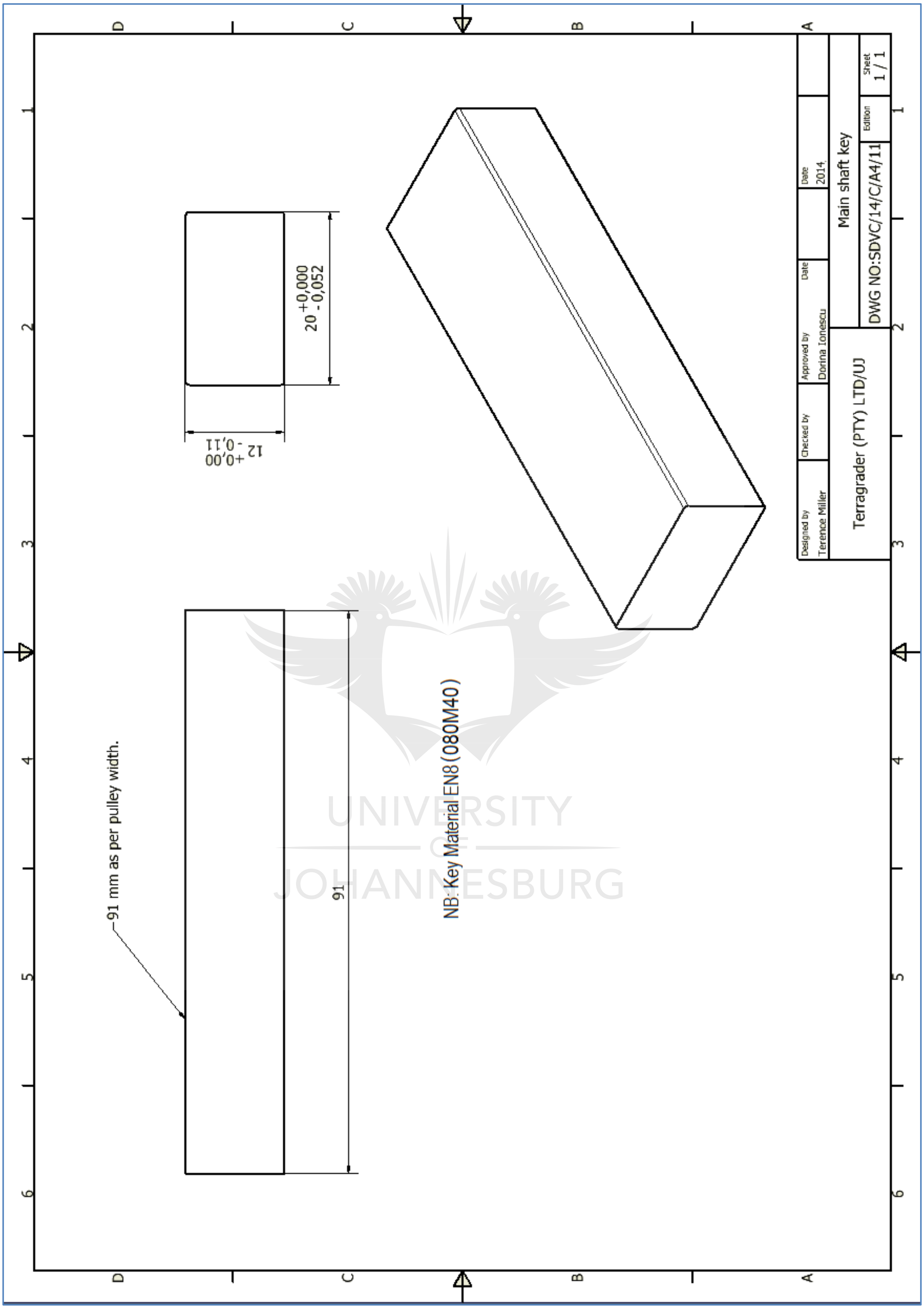

6
5
4
3
2
1
D
C
B
A
81
800
28 - 0,052 + 0,000
A
Ø110
A (1 : 1)
10 + 0,20 + 0,00
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Terragrader (PTY) LTD
First Shaft
DWG NO:SDVC/14/C/A4/15
Edition
Sheet
1 / 1

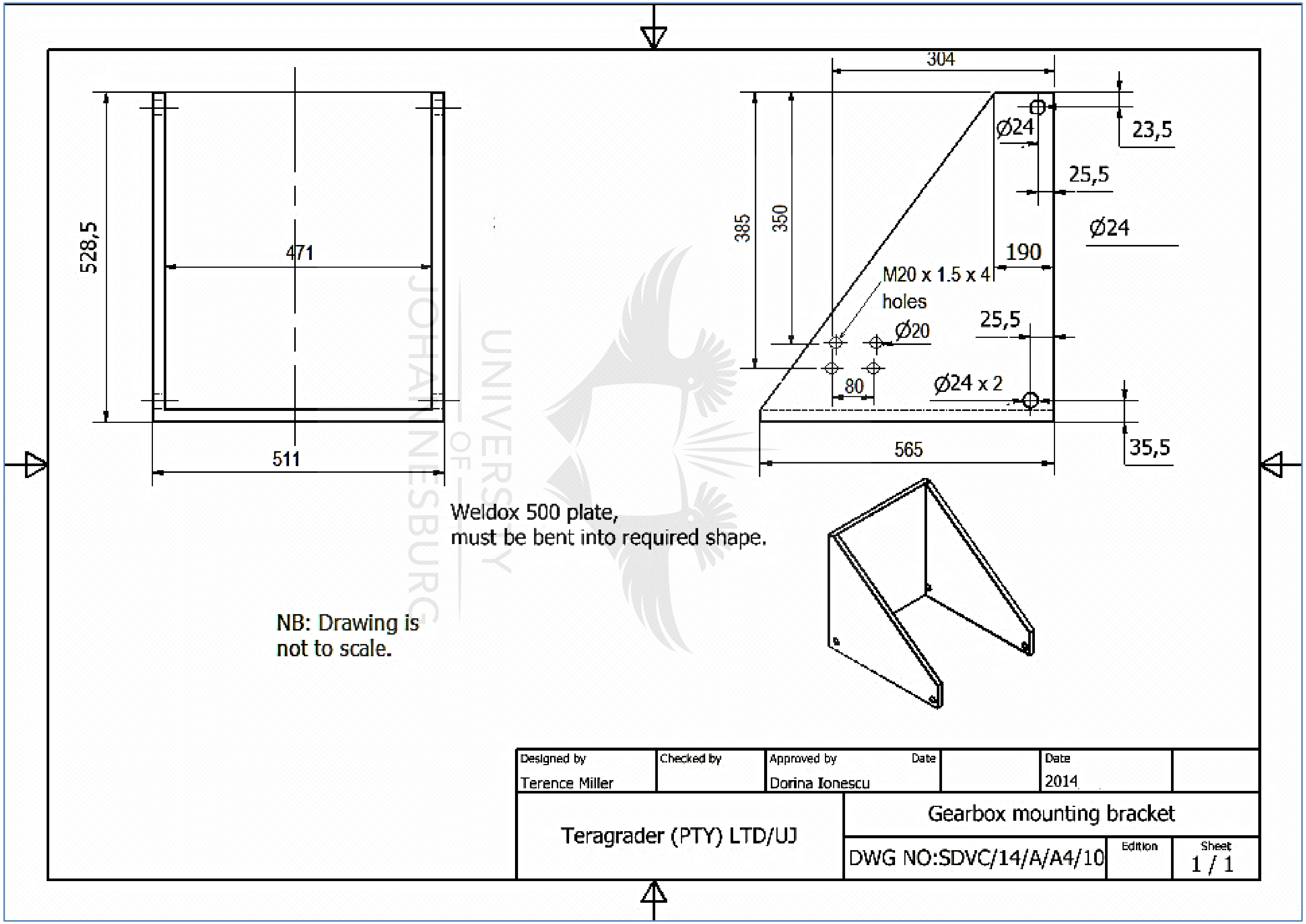

22 - 0,052
+ 0,000
14 +0,000
-0,110
81
R0,6
(BS 4235 : PART 1 : 1972)
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Terragrader (PTY) LTD/UJ
Small sprocket key
DWG NO:SDVC/14/C/A4/14
Edition
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6


Key ways done according to the
BS 4235 : PART 1 : 1972
22 - 0,052
+0,000
113
81
16
A
Ø85
9 +0,20
-0,00
A (2 : 1)
2
2
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014,
Terragrader (PTY) LTD
Intermediate shaft
DWG NO:SDVC/14/C/A4/13
Edition
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6



Sharp edges to be chamfered between 1 mm and 2 mm for the shaft circumferences.


D
C
B
A
1
2
3
4
5
6
20 +0,000 -0,052
12 +0,00 -0,11
91
91 mm as per pulley width.
NB: Key Material EN8 (080M40)
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Main shaft key
Terragrader (PTY) LTD/UJ
DWG NO:SDVC/14/C/A4/11
Edition
Sheet
1 / 1



Weldox 500 plate,
must be bent into required shape.

NB: Drawing is
not to scale.

Designed by
Terence Miller

Checked by

Approved by
Dorina Ionescu

Date

Date
2014

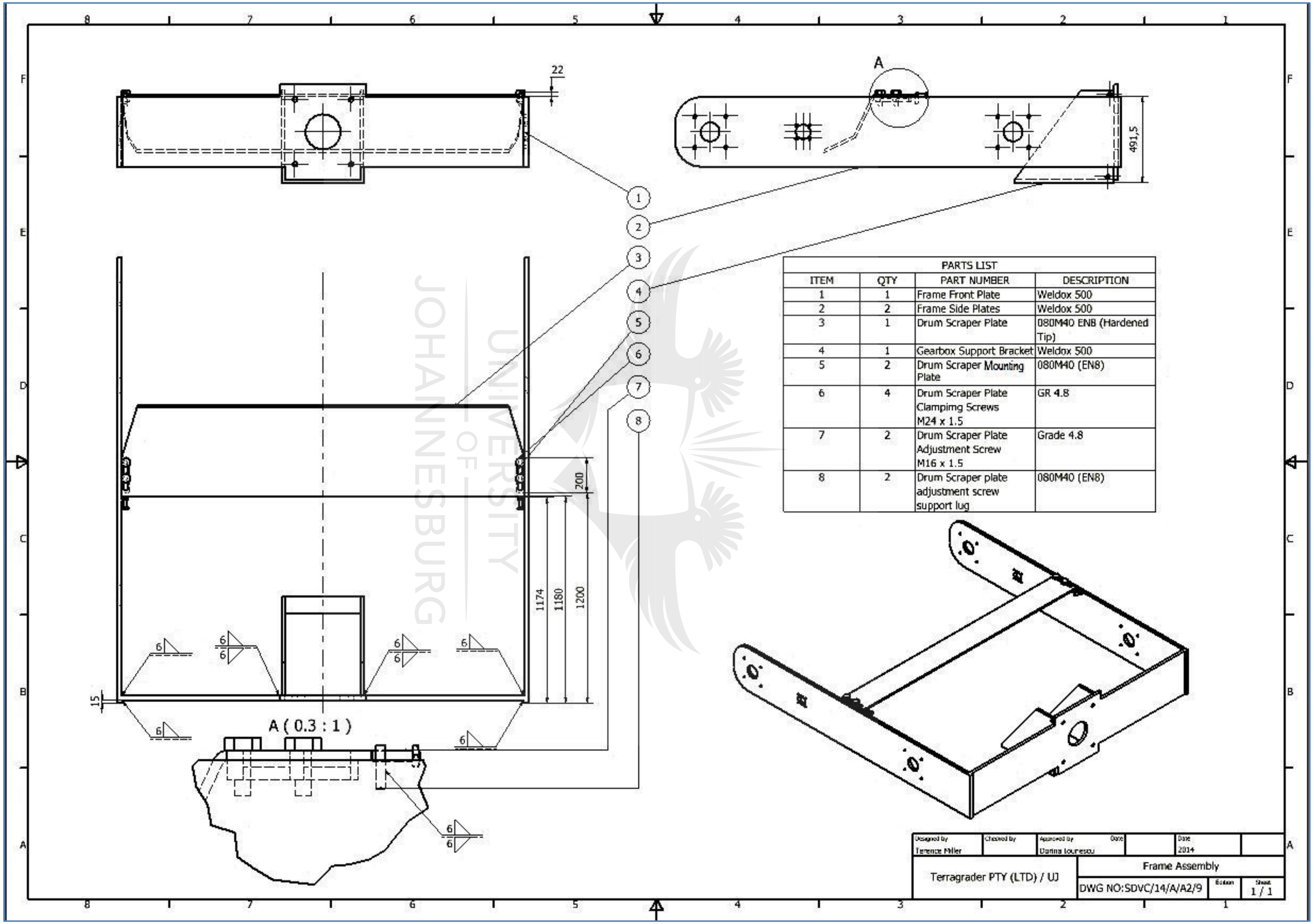
Teragrader (PTY) LTD/UJ

Gearbox mounting bracket

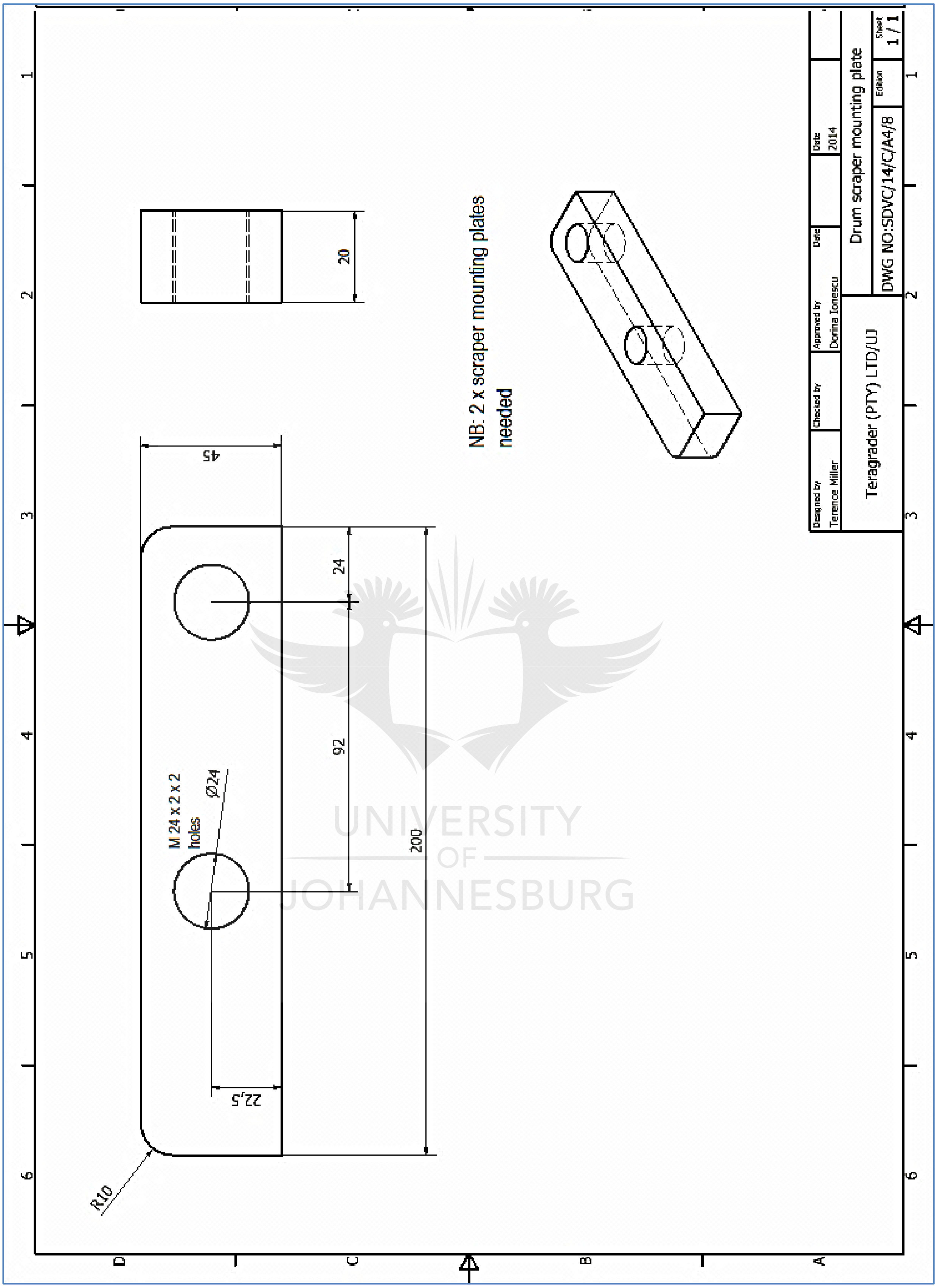
DWG NO:SDVC/14/A/A4/10

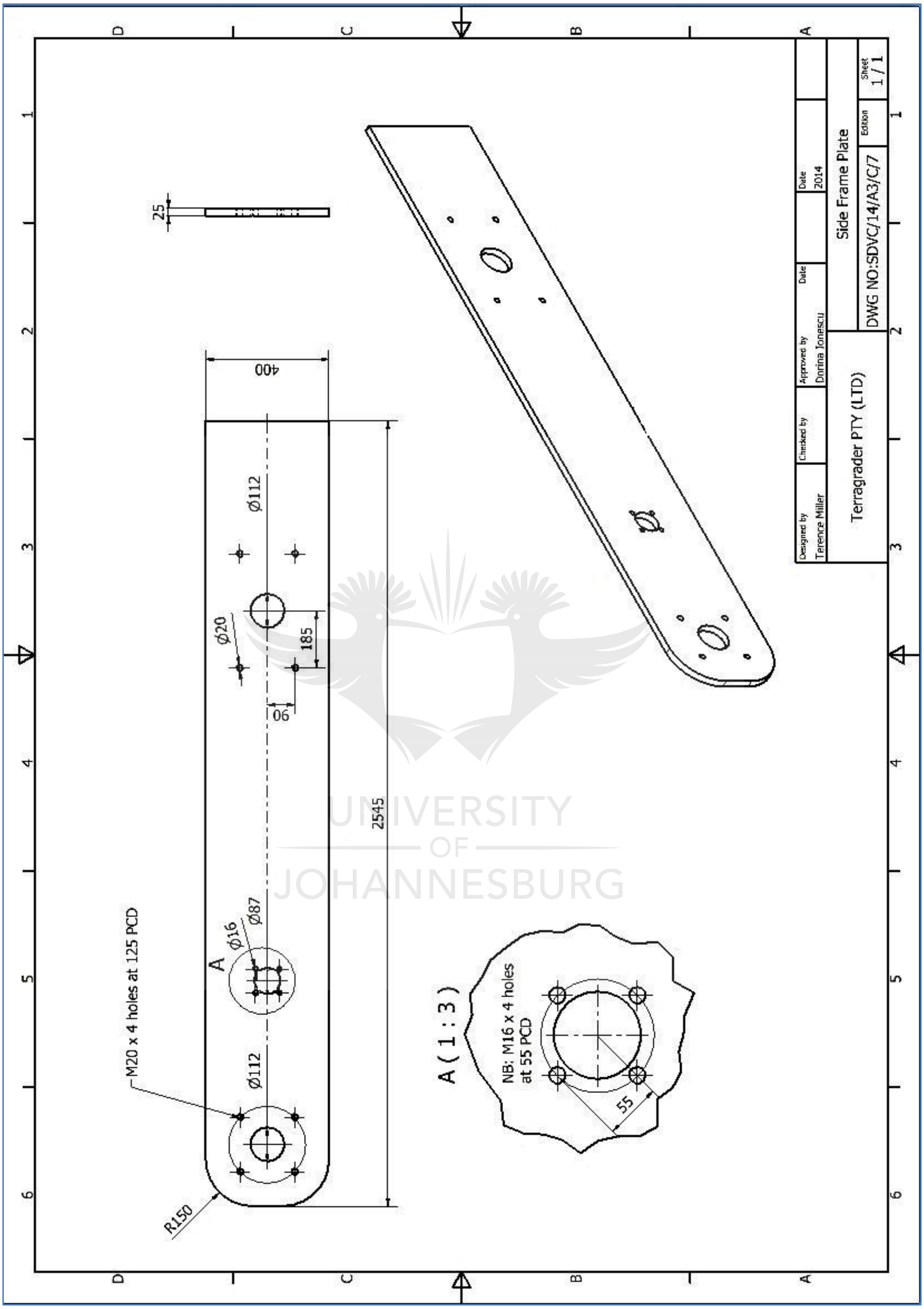
Edition

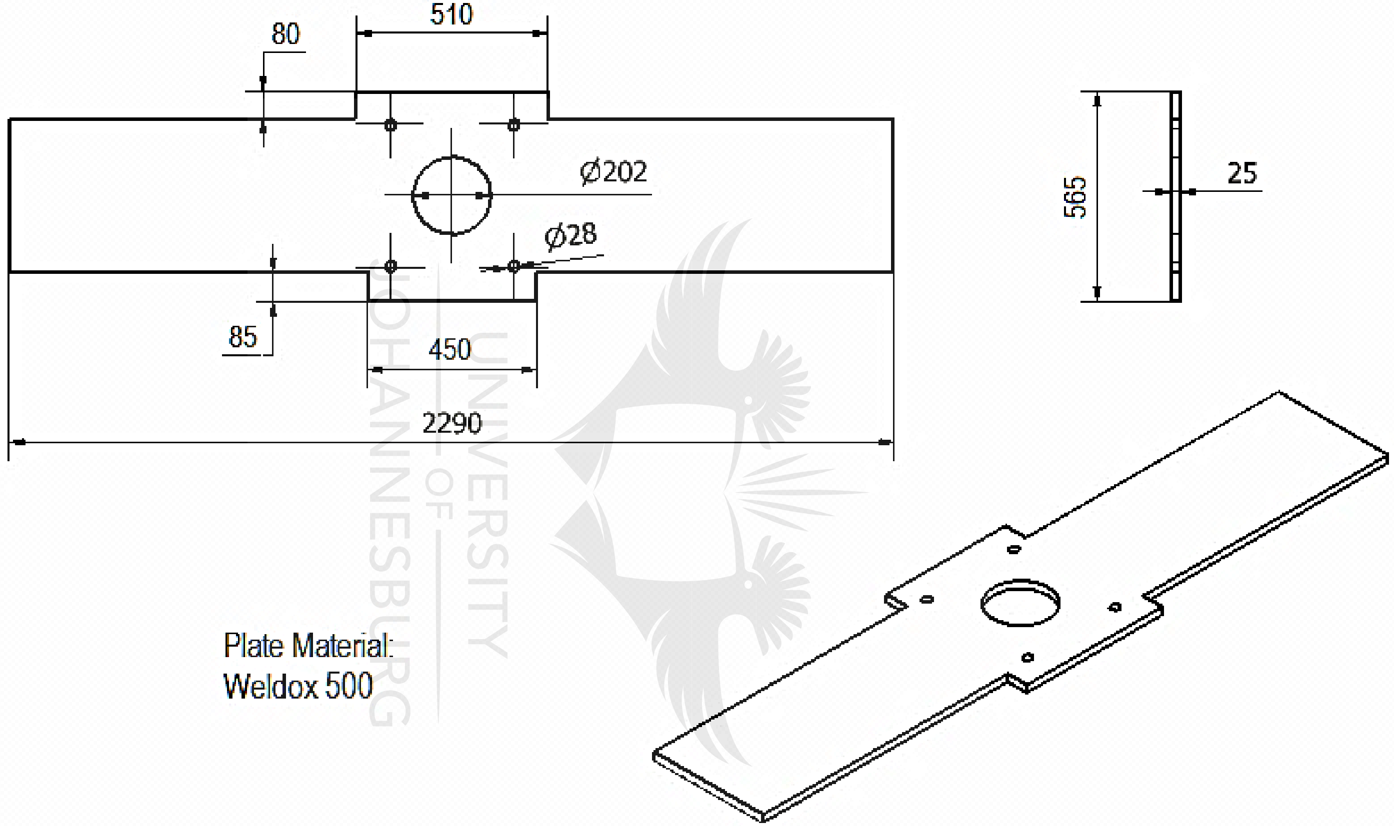
Sheet
1 / 1



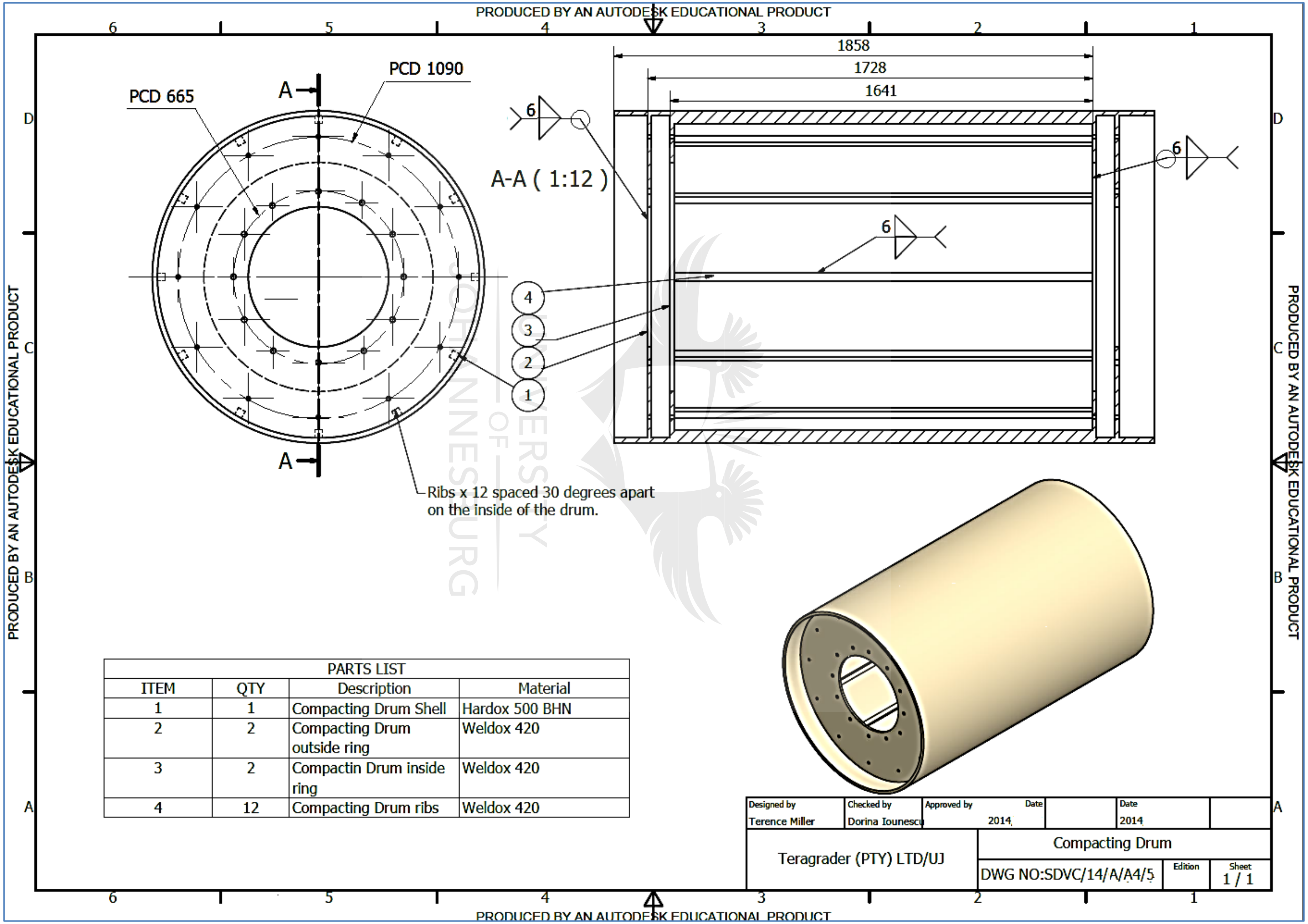
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	Frame Front Plate	Weldox 500
2	2	Frame Side Plates	Weldox 500
3	1	Drum Scraper Plate	080M40 EN8 (Hardened Tip)
4	1	Gearbox Support Bracket	Weldox 500
5	2	Drum Scraper Mounting Plate	080M40 (EN8)
6	4	Drum Scraper Plate Clampimg Screws M24 x 1.5	GR 4.8
7	2	Drum Scraper Plate Adjustment Screw M16 x 1.5	Grade 4.8
8	2	Drum Scraper plate adjustment screw support lug	080M40 (EN8)


NB: 2 x scraper mounting plates needed
M 24 x 2 x 2 holes
Ø24
R10
22,5
45
20
24
92
200
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Drum scraper mounting plate
Teragrader (PTY) LTD/UJ
DWG NO:SDVC/14/C/A4/8
Edition
Sheet
1 / 1
1
2
3
4
5
6
A
B
C
D


25
400
Ø112
Ø20
185
90
2545
A
Ø16
Ø87
Ø112
M20 x 4 holes at 125 PCD
R150
A (1 : 3)
NB: M16 x 4 holes at 55 PCD
55
Designed by
Terence Miller
Checked by
Approved by
Dorina Ionescu
Date
Date
2014
Side Frame Plate
Terragrader PTY (LTD)
DWG NO:SDVC/14/A3/C/7
Edition
1
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6



Designed by Terence Miller	Checked by	Approved by Dorina Ionescu	Date		Date 2014	
Teragrader (PTY) LTD/UJ			Frame front plate			
			DWG NO:SDVC/14/A/A4/6		Edition	Sheet 1 / 1



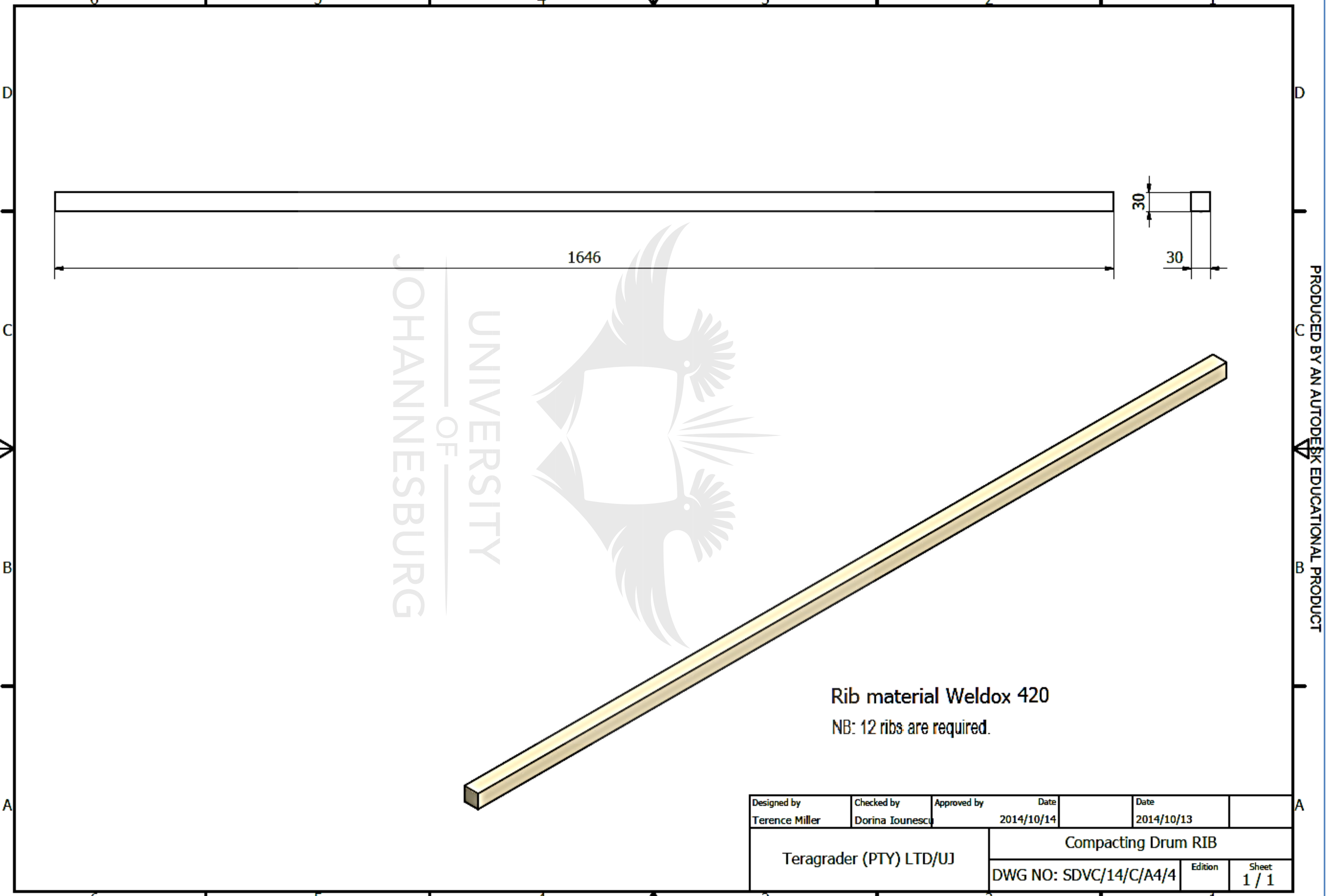
PARTS LIST			
ITEM	QTY	Description	Material
1	1	Compacting Drum Shell	Hardox 500 BHN
2	2	Compacting Drum outside ring	Weldox 420
3	2	Compactin Drum inside ring	Weldox 420
4	12	Compacting Drum ribs	Weldox 420

Designed by	Checked by	Approved by	Date	Date
Terence Miller	Dorina Iounescu		2014	2014

Teragrader (PTY) LTD/UJ

Compacting Drum

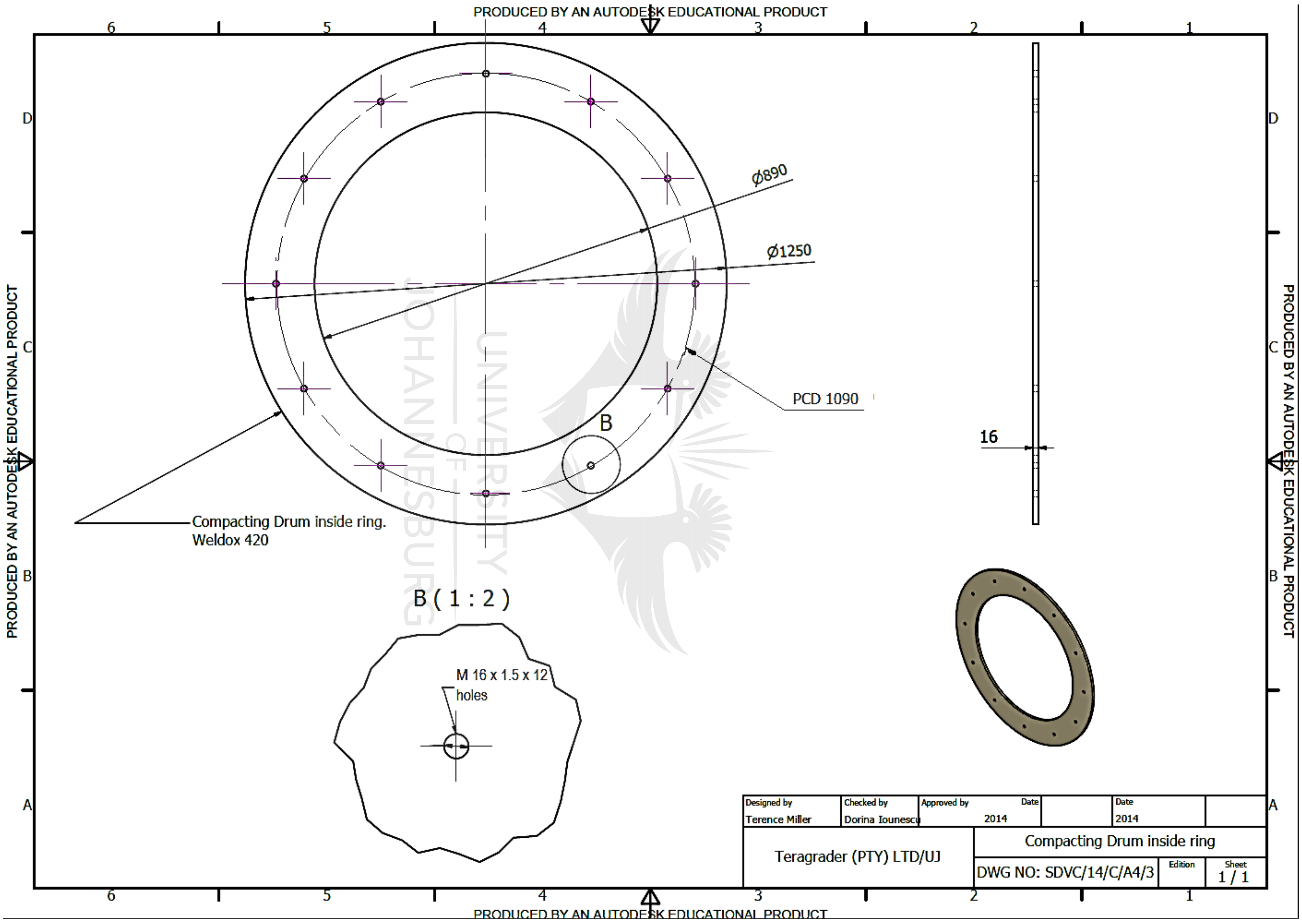
DWG NO:SDVC/14/A/A4/5
Edition
Sheet 1 / 1

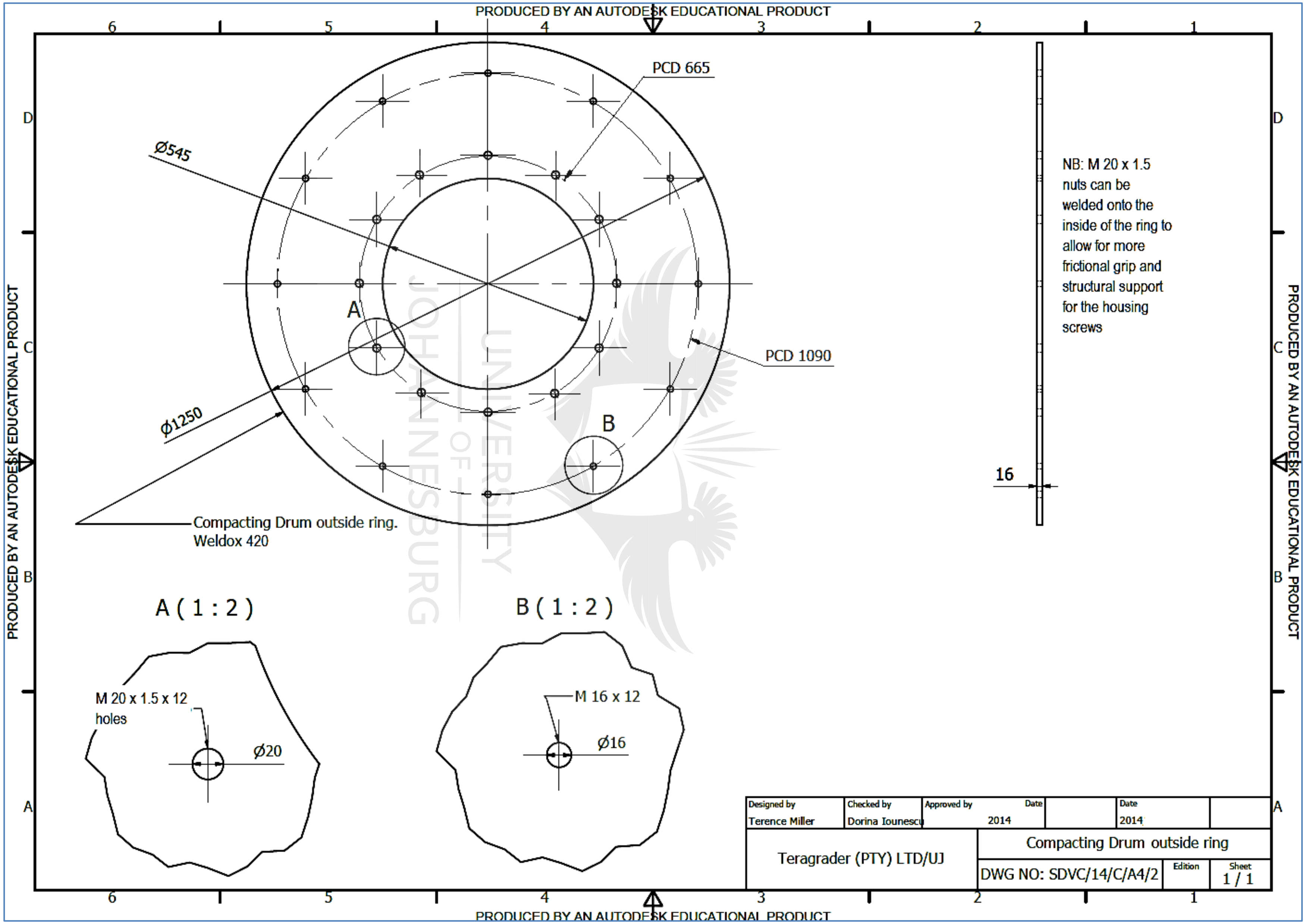


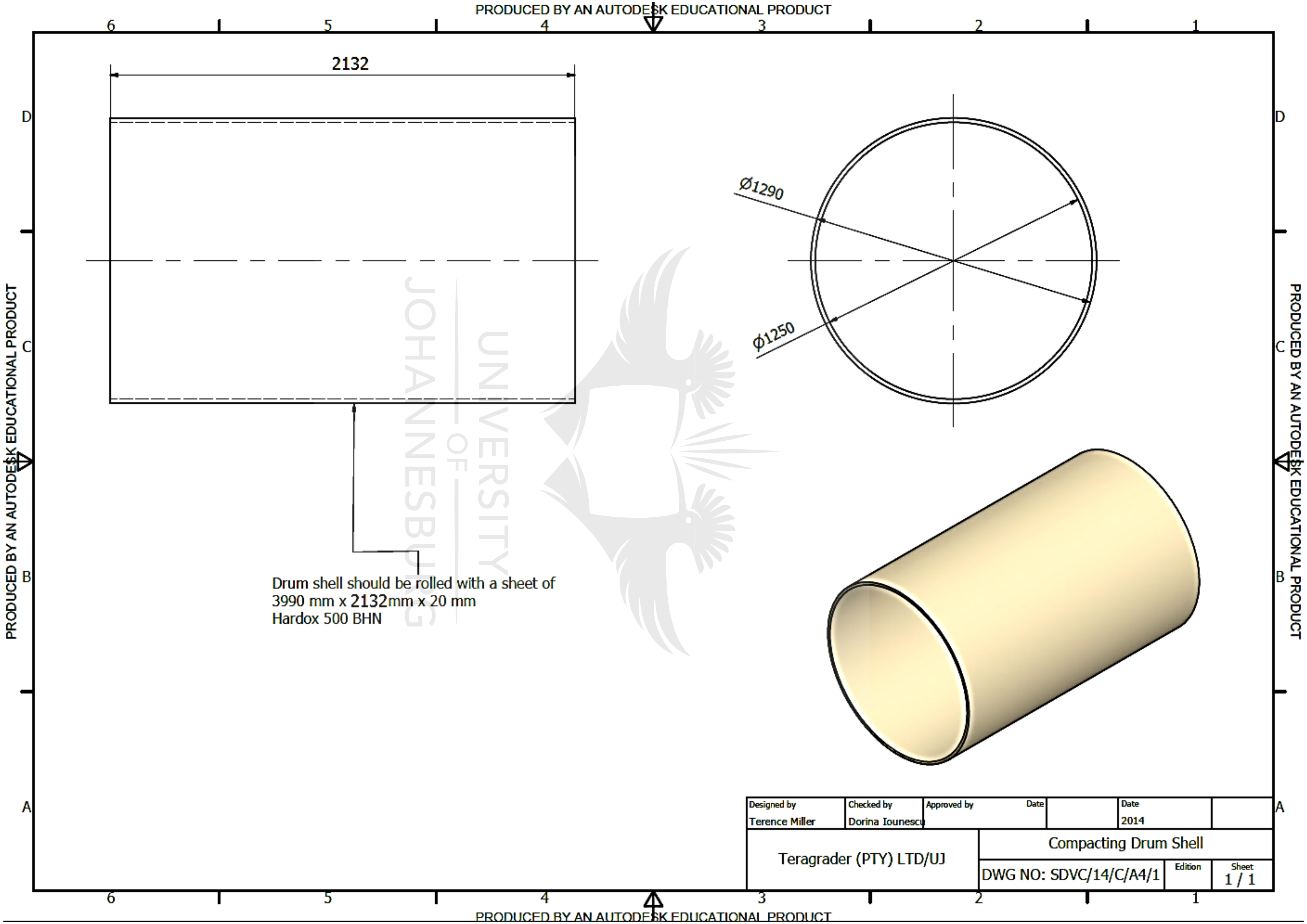
Rib material Weldox 420

NB: 12 ribs are required.

Designed by	Checked by	Approved by	Date	Date	
Terence Miller	Dorina Iounescu		2014/10/14	2014/10/13	
Teragrader (PTY) LTD/UJ		Compacting Drum RIB			
		DWG NO: SDVC/14/C/A4/4		Edition	Sheet 1 / 1


PRODUCED BY AN AUTODESK EDUCATIONAL PRODUCT
Ø890
Ø1250
PCD 1090
B
16
Compacting Drum inside ring.
Weldox 420
B (1 : 2)
M 16 x 1.5 x 12
holes
Designed by
Terence Miller
Checked by
Dorina Iounescu
Approved by
Date
2014
Date
2014
Teragrader (PTY) LTD/UJ
Compacting Drum inside ring
DWG NO: SDVC/14/C/A4/3
Edition
Sheet
1 / 1
A
B
C
D
1
2
3
4
5
6


PRODUCED BY AN AUTODESK EDUCATIONAL PRODUCT
6
5
4
3
2
1
D
C
B
A
PCD 665
Ø545
PCD 1090
A
B
Ø1250
Compacting Drum outside ring.
Weldox 420
NB: M 20 x 1.5
nuts can be
welded onto the
inside of the ring to
allow for more
frictional grip and
structural support
for the housing
screws
16
A (1 : 2)
M 20 x 1.5 x 12
holes
Ø20
B (1 : 2)
M 16 x 12
Ø16
Designed by
Terence Miller
Checked by
Dorina Iounescu
Approved by
Date
2014
Date
2014
Teragrader (PTY) LTD/UJ
Compacting Drum outside ring
DWG NO: SDVC/14/C/A4/2
Edition
Sheet
1 / 1



Drum shell should be rolled with a sheet of
3990 mm x 2132mm x 20 mm
Hardox 500 BHN

Designed by	Checked by	Approved by	Date	Date	
Terence Miller	Dorina Iounescu			2014	

Teragrader (PTY) LTD/UJ

Compacting Drum Shell

DWG NO: SDVC/14/C/A4/1	Edition	Sheet
		1 / 1

Appendix 51: Machine costing and selling price.

TABLE A: Shaft Pricing 080M40 (EN8) Bright Axle Steel

Description	Diameter	kg/m	Length needed (m)	Quantity	kg	Price per kg (max)	Price Ex VAT	VAT	Total
(Main Shaft)	110 mm	74.597	0.7	2	104.435	R 24.16	R 2 523.16	R 353.24	R 2 876.40
(Intermediate Shaft)	85 mm	44.542	0.45	2	40.088	R 25.59	R 1 025.74	R 143.60	R 1 169.35
(First Shaft)	110 mm	74.597	0.95	2	141.733	R 24.16	R 3 424.29	R 479.40	R 3 903.69
Final Price Excluding Labour									R 7 949.43

TABLE B: Plate Material needed sold standard from the supplier.

Description	Thickness mm	Area of plate sizes available from supplier	Plate Area per square m	late Price Ex VA	VAT	Total for plates
Vibratory drum shell (Hardox 500) material	20	4 x 2.4	9.6	R 32 608.27	R 4 565.16	R 37 173.43
Vibratory drum inside Rings plus outside rings(Weldox 420)	16	6 x 2.4	14.4	R 34 156.00	R 4 781.84	R 38 937.84
Vibratory drum frame , front and side plates	30	8 x 2.4	19.2	R 85 390.42	R 11 954.66	R 97 345.08
Vibratory drum frame Scraper plate + brackets	16	2.4 x 1.2	2.88	R 6 831.23	R 956.37	R 7 787.60
Final Material Price						R 181 243.95

TABLE C: Actual amount of material needed for vibratory drum and frame fabrication ex labour.

Description	Thickness mm	Area in plate form	Quantity	Fabrication Plate Area per square m	Price Ex VAT based on material area	VAT	Total
Vibratory drum shell (Hardox 500)	20	2.1 x 1.29 x π	1	8.5106	R 32 608.27	R 4 565.16	R 37 173.43
Vibratory drum outside Flange Rings (Weldox 420)	20	π x(1.24 x 1.24 - 0.545 x 0.545)/4	2	1.9487	R 5 777.79	R 808.89	R 6 586.68
Vibratory drum inside Flange Rings (Weldox 420)	20	π x(1.24 x 1.24 - 0.89 x 0.89)/4	2	1.171	R 5 777.79	R 808.89	R 6 586.68
Vibratory Drum Frame **sides** (Weldox 420)	30	2,38 x 0.4	2	1.904	R 11 290.51	R 1 580.67	R 12 871.18
Vibratory Drum Frame **front** (Weldox 420)	30	2.56 x 0.4	1	1.024	R 6 072.21	R 850.11	R 6 922.32
Botls & Nuts GR 8.8) Estimate					R 5 000.00	R 700.00	R 5 700.00
Vibratory drum frame Scraper plate + bracket (Estimate)	16	2.4 x 1.2	1	2	R 6 831.23	R 956.37	R 7 787.60
Final Price Excluding Labour							R 83 627.89

***TABLE D*: Power Transmission Prices with NSK Bearings**

Description	Part Number	Quantity	Price per unit Ex VAT	Price Ex VAT after QTY is added	VAT	Total
Vibration Dampers (Seal n Devices)	Z160-16F-FT/FT	24	R 225.68	R 5 416.37	R 758.29	R 6 174.66
Drive Shaft (Spicer & Hardy) (Estimate)	10 Series	1	R 20 000.00	R 20 000.00	R 2 800.00	R 22 800.00
HD 100 2 Gearbox (Bonfiglioli)	HDO 100 2 LP 10.9 LD VP B3 FAN	1	R 117 237.00	R 117 237.00	R 16 413.18	R 133 650.18
F 200 - F Fenner Flex Tyre Coupling (BMG)	F 200 - F	2	R 27 931.27	R 55 862.54	R 7 820.76	R 63 683.30
Fenner Triplex Chain Drive (BMG)	As per Spec.	2	R 43 800.00	R 87 600.00	R 12 264.00	R 99 864.00
Fenner Friction Wedge Belt Drive (BMG) (Belt prices are estimated	As per Spec.	2	R 26 101.81	R 52 203.62	R 7 308.51	R 59 512.13
Main Shaft Eccentric Weight Bearings. (BMG)	22324 CAMKE4C3/H2324	4	R 9 350.58	R 37 402.32	R 5 236.32	R 42 638.64
Main Shaft Frame Bearings. (BMG)	23024 CCK	2	R 3 751.40	R 7 502.80	R 1 050.39	R 8 553.19
Intermediate shaft Bearings (BMG)	22219 E	2	R 2 197.89	R 4 395.78	R 615.41	R 5 011.19
First Shaft Bearings (BMG)	23024 CCK	2	R 3 751.40	R 7 502.80	R 1 050.39	R 8 553.19
Final Price Excluding Labour						R 450 440.48

***TABLE E*: Power Transmission Prices with SKF Bearings**

Description	Part Number	Quantity	Price per unit Ex VAT	Price Ex VAT after QTY is added	VAT	Total
Vibration Dampers (Seal n Devices)	Z160-16F-FT/FT	24	R 225.68	R 5 416.37	R 758.29	R 6 174.66
Drive Shaft (Spicer & Hardy) (Estimate)	10 Series	1	R 20 000.00	R 20 000.00	R 2 800.00	R 22 800.00
HD 100 2 Gearbox (Bonfiglioli)	HDO 100 2 LP 10.9 LD VP B3 FAN	1	R 117 237.00	R 117 237.00	R 16 413.18	R 133 650.18
F 200 - F Fenner Flex Tyre Coupling (BMG)	F 200 - F	2	R 27 931.27	R 55 862.54	R 7 820.76	R 63 683.30
Fenner Triplex Chain Drive (BMG)	As per Spec.	2	R 43 800.00	R 87 600.00	R 12 264.00	R 99 864.00
Fenner Friction Wedge Belt Drive (BMG)	As per Spec.	2	R 26 101.81	R 52 203.62	R 7 308.51	R 59 512.13
Main Shaft Eccentric Weight Bearings. (SKF)	22324 CAMKE4C3/H2324	4	R 11 303.09	R 45 212.36	R 6 329.73	R 51 542.09
Main Shaft Frame Bearings. (SKF)	23024 CCK	2	R 4 438.25	R 8 876.50	R 1 242.71	R 10 119.21
Intermediate shaft Bearings (SKF)	22219 E	2	R 2 980.37	R 5 960.74	R 834.50	R 6 795.24
First Shaft Bearings (SKF)	23024 CCK	2	R 4 438.25	R 8 876.50	R 1 242.71	R 10 119.21
Final Price Excluding Labour						R 464 260.02

NB: There are two bearing suppliers (BMG and SKF) stipulated in the above tables and both their costs are summarised below.

Cost pertaining to the use of BMG NSK bearings.

Final Cost = Cost Table A (Shafting) + Cost Table C (Fabricating) + Cost Table D (BMG Power transmission)

Final Cost = R 7 949.43 + R 83 627.89 + R 450 440.48 = R 542 019.8

Cost pertaining to the use of SKF bearings.

Final Cost = Cost Table A (Shafting) + Cost Table C (Fabricating) + Cost Table D (BMG Power transmission)

Final Cost = R 7 949.43 + R 83 627.89 + R 464 260.02 = R 555 837.34

NB: There is no major cost difference between BMG and SKF final costs. SKF in general, is the better supplier with better quality bearings when compared with BMG, thus, the SKF route will be used for further costing.

Total fabrication cost

To get the total fabrication cost, labour has to be included. Labour will be estimated as 50% of the total cost for shafting and fabrication.

Total Fabrication cost = 1.5 x (R 83 627.89 + R 7 949.43) = R 137 365.98

Selling Price

To get the final selling price of the machine, an estimated mark-up of 35% will be added to all relevant costs.

Selling price = 1.35 x (R 137 365.98 + R 464 260.02) = R 812 195.1