CONCEPTUAL DESIGN OF A FRICTION STIR WELDING MACHINE FOR JOINING RAILS

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A research report submitted to the faculty of Engineering and the Built Environment, the University of the Witwatersrand, Johannesburg, in partial fulfilment of the requirements for the degree of Master of Science in Engineering.

Johannesburg, 2009

DECLARATION

I declare that this research report is my own, unaided work. It is being submitted to the degree of Master of Science in Engineering to the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination to any other University.

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18th day of October 2008

ABSTRACT

The main objective of the project was to conceptually design a friction stir welding machine for joining rails. The applicability of friction stir welding types and its application in rail joining was investigated. A number of machine concepts for joining rail using friction stir welding techniques were developed and a final workable concept was laid out. In addition, the existing methods and machines for joining rails were considered, including arc welding, exothermic welding, flash butt welding and manual joining (rails joined by means of splice plate). After comparing different methods of joining rails, an optimized method was selected. The capabilities of the new conceptual machine, such as its ability to accommodate various rail profiles, were demonstrated through designs and various calculations. The development cost analysis was performed and a comparison was made with the other three methods of joining rails. Consequently, it was concluded that friction stir welding concept could be applied in rail joining and the costs associated with it could be lowered.

Keywords: Friction stir welding, rail joining.

DEDICATION

To the cherished memory of my late grandparents and others:

Khavhathondwi Makhalimela Nemavhola Thonzhe

Tshinakaho Nyabele Ndou-Nemavhola

Matinyambado Jack Tshiwanammbi

Micheal Nemavhola

ACKNOWLEDGEMENTS

Initially I would like to thank my supervisor Dr. Ionel Botef for his tremendous support in this work and also for giving me the opportunity to work under his supervision. I must also thank him for his valuable support and guidance throughout the period of this research.

It is impossible to single out individuals who contributed substantially to the development and writing of this report. Many thanks go to my colleagues at work for their inspiration and advices made this project very successful. Especially: Mr Kwenzakwenkosi Thabethe, Miss Arinao Novhe and Mr Tshinanne Netshishivhe. Thanks to my brother (Thabelo Masithulela) for his spiritual support and guidance and his ability to motivate. Many thanks also go to Mr Mpho Tshidzumba for helping out in proof reading.

Above all, thanks to all my colleagues, friends and fellow students for intense discussions on my research topic.

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LIST OF SYMBOLS

T	Transmissibility, applied toque (N.m)
ω	Angular frequency of exciting force, Rotational speed (rev/min)
ω_n	Angular frequency of mounted system (rev/min)
k	Spring constant (N/mm)
x	Spring displacement (mm)
f	Frequency of exciting force, friction of power screw (Hz)
f_n	Frequency of mounted system (Hz)
K_w	Wahl factor, material and geometry factor
K_s	Stress concentration factor for static loading
C	Ratio of outer diameter to coil diameter/spring index
D	Outer diameter of a helical spring (mm)
d	Coil spring diameter (mm)
N	Number of active turns
N_t	Total number of active turns
G	Shear modulus (GPa)
L_s	Solid length of a spring (mm)
L_f	Final length (mm)
S_u	Ultimate strength (MPa)
τ_{solid}	Shear stress (MPa)
τ	Torsional stress (MPa)
$F_{applied}$	Applied force (kN)
F_{solid}	Force applied at solid state (kN)
E	Elastic modulus of elasticity (MPa)
I	Moment of inertia (mm^4)
h	Height (mm)
b	Breadth (mm)
δ	Helical spring deflection (mm)
$\delta_{\max}$	Minimum spring deflection, maximum linear displacement (mm)
σ	Normal stress (MPa)

σ_{max}	Maximum normal applied stress (MPa)
c	Distance from central axis to the surface where maximum stress occurs (mm)
f_c	Bearing friction
L	Lead, length of beam (mm)
d_c	Ball thrust bearing diameter (mm)
W	Load (weight) of the object to be lifted (kg)
A_t	Tensile stress area, Stress area (mm^2)
n	Safety factor
P	Power (Watts)
λ	Thermal conductivity, helix and lead angle
C_p	Heat capacity
ρ	Density of the material (kg/m^3)
n	Fatigue safety factor
K_f	Fatigue stress concentration factor
K_{fs}	Fatigue stress concentration factor (shear loading)
M_a	Alternating bending moment (N.m)
S_a	Endurance limit (MPa)
S_{ult}	Ultimate strength (MPa)
S_y	Yield strength of the material (MPa)
T_m	Steady or mean applied torque (N.m)
d	Shaft diameter (mm)
S_m	Mean stress (MPa)
S_a	Stress amplitude (MPa)
θ	Angular displacement of a beam (radian)
x	Position along the beam (mm)
M	Applied bending moment (N.m)
y	Distance from the central axis to the point of concern (mm)
c	Neutral axis (mm)
e	Power screw efficiency
p	Pitch (mm)
J	Polar moment of inertia (mm^4)
d_m	Mean diameter (mm)

d_c	Collar (bearing) diameter (mm)
α_n	Thread angle measured in a normal plane (mm)
MPa	Mega Pascal
GPa	Giga Pascal
mm	Millimetre
Hz	Hetz
m	meter
N	Newton
kg	Kilogram

LIST OF ACRONYMS

FSW	Friction Stir Welding
EGW	Electrogas Welding
TWI	Thompson Welding Institute
FBW	Flush-butt Welding
ETW	Exothermic Welding
FEM	Finite Element method
FEA	Finite Element Analysis
W	Wish
C	Constraints
D	Demand
PRS	Product Requirement Specification

1 INTRODUCTION

Rail welding plays an important role in railway business. A cheap and effective method for joining rails is critical for a successful railway business and performance. High quality welding of rail is essential in maintaining high standards of safety in the industry. Rail fracture due to poor quality welds costs several millions of rands per annum. Twenty percent of rail derailments are caused by poor quality welding of rails throughout the world, (Sun and Davis, 2001). Railway industries are paying enormous amounts of money buying equipment for welding which requires highly qualified technical teams of welders. The average cycle time for completing one weld varies between 3 and 10 minutes (Judge T, 1998).

Welding processes produce welds with consistent quality and good service performance and have minimum dependency on the skills of the workers (Judge T, 1998). Welding processes can introduce undesirable residual stresses and distortions in the final fabricated components as well as localised loss of mechanical properties at the weld joints (Murphy *et al*. 2007).

Two main welding methods for joining rails are used, namely: exothermic welding and flash-butt welding. These methods do not meet all the requirements for joining rails, therefore, it is critical to look at other possibilities of rail welding by applying friction welding concepts for joining rails.

Friction welding has long been recognized as a cost effective alternative method of producing highly complex shaped components for a broad variety of components in automotive, electrical, mining, transportation, or agricultural industries. Friction welding has many advantages, these include welds with a fine grain heat-affected zone structure, material and machining cost savings, 100% bond of full cross section, high production rates, automatic repeatability, welds stronger than the parent material with excellent fatigue resistance, and the advantage of joining similar and dissimilar material with no added fluxes or filler metals.

Therefore, friction welding can be applied in the railway industry.

1.1 PROBLEM STATEMENT

It is common practice world wide to join rails; these rails have by their physical nature a high carbon content. In South Africa, the majority of rails are still joined by the traditional method of bolting two rail plates (fisher plates) a method however associated with high maintenance costs. This method of joining rails produces rail discontinuity and as a result dynamic impact forces are formed. Dynamic impact forces are associated with high normal stresses on the rails, ballast and the subgrade. High stresses on both the ballast and the subgrade increase ballast settlement and produces uneven track (Judge T, 1998). As a result, the fisher plate method of joining rails is associated with high maintenance and labour cost leading to poor rail operations.

During rail joining operations, the flash-butt welding process does consume rail material, approximately 35 mm of rail and longitudinal rail movement is also needed for the flashing and forging processes (Bhadeshia, 2000). The defect rate for flash-butt welding is low as compared to exothermic welds (Sun and Davis, 2001). On the other hand, it has been deduced that the strength, ductility and the fatigue properties of exothermic weld metal have not been as high as those of the rail steel (Bhadeshia, 2000). The properties that create these differences are mainly attributed to its dendritic cast structure, porosity and inclusions.

Therefore, the friction stir welding method can be used to reduce maintenance costs considerably. Also, it is sometimes not possible to align the rails properly in order to minimise the impact forces of running trains. The expansion gap between two rails helps during the rapid change of temperature gradient. However this has been overcome by recent studies which show that the weld between two rails could also allow rails to expand without any serious damage on the rail (Bhadeshia, 2000).

Furthermore, exothermic welding, gas pressure welding and flash-butt welding processes are globally used for joining rails. These processes have their drawbacks as most of them require highly qualified personnel and high capital investment. For example the exothermic welding process requires extensive training for operators and high capital costs in order to achieve high quality weld. The flash butt welding process usually comes as a large automated truck with high mechatronics systems and it requires substantial initial capital investment. However, the FBW (Friction-butt Welding) process is environmentally friendly and requires adequate rather than extensive training in order to obtain quality welds. The gas pressure welding process is not in used in South Africa due its bad quality weld.

In South Africa the most commonly used rail joining welding process is the exothermic welding technique. The flash-butt welding process is thought to be the best at present in joining rails. However, South Africa cannot keep up with the required high initial investment costs and high maintenance costs thereafter. Consequently, there is a need to develop friction stir welding processes which could give cost benefits over and above what the traditional welding processes are offering.

Welding process should meet the following requirements (Sun and Davis, 2001):

- The total welding time should be short enough to fit available track windows.
- The welding equipment should be portable so it can be easily transported.
- Rail movement and consumption should be prevented or minimized.
- The process should be easily adaptable to all commonly-used rail sections.
- The process should be flexible enough to tolerate rails with varying wear levels.
- The average weld cost and initial equipment costs should be kept at a reasonable level.

If designed and applied correctly, the friction stir welding process can be used for joining rails in the South African railway environment.

1.2 OVERVIEW OF RAIL JOINING IN THE SOUTH AFRICAN RAILWAY INDUSTRY

1.2.1 *OBJECTIVES OF THIS STUDY*

Rail joining by means of welding plays an important role in freight performance, maintenance costs and delivery reliability in freight rail industries. Also, it has been concluded that 20% of rail infrastructure costs are directly derived from rail joining (Sun and Davis, 2001). The current methods used for joining rails have many disadvantages, including amongst other, high capital costs required, extensive training for operators required and a high level of defects attached. Therefore, the first objective of this project is to apply friction welding techniques in joining rails. Several concepts are to be generated based on friction stir welding techniques. The feasibility of the selected concepts is to be evaluated by means of performing costs analysis on the concepts. Also, engineering design methodology is to be used to validate the selected concepts.

1.3 RESEARCH CONTRIBUTION

Reproducing a higher quality weld in joining rails has remained a challenge in the railway environment. In summary the contributions made by this research to the pool of knowledge are as follows:

- The application of appropriate friction stir welding for joining rails. Various friction welding methods are to be reviewed and the best process for joining rails is to be implemented.
- The use of joining rails by means of a friction stir welding machine. At present the friction stir welding process is mainly used in joining thin (about 50 mm) aluminium alloy sheets.
- The uses of friction stir welding for joining carbon steel of irregular shape like rails.
- Tool pin profile optimization and design.

- The development and design of a friction stir welding machine for joining rails, the machine having the ability to control important parameters, such as, welding speed, pin rotational speed, welding force and the normal force.

1.4 ACADEMIC AND PRACTICAL IMPORTANCE OF THE STUDY

The friction stir welding process is mostly used for joining aluminium alloy sheets for aerospace and motor industries. This method is used in rare cases to join low carbon steel. Despite its ability to join different material of any chemical composition it has been used mostly in joining flat shaped sheets. Rail in its form can be seen as the I-beam shape. The FSW has never been used for joining complicated shapes like the I-beam ones.

This study introduces a conceptual design of joining complex shapes of high carbon content steel by means of the friction stir welding process. The FSW process is seen and has been proven to be an economical process; therefore this study will bring forth the benefits of high quality welds for joining rails. Consequently, the quality of rail welds is to be improved. At the moment studies show that the current methods for joining rails are expensive due to the high cost of training and high capital cost required.

In terms of academic requirements, this project cannot cover all the required aspects including, the microstructure of the material due to the FSW process and the design of a friction stir welding tool pin. Therefore, this leaves a wide gap for research into and an understanding of high carbon steel micro structural behaviour and the design of a tool pin. It is to be noted that this project will focus mostly on the conceptual design of the outer structure of the machine.

1.5 RESEARCH METHODOLOGY

An overview of the project approach and methodology is briefly outlined below.

1.5.1 *RESEARCH AND DESIGN CONTEXT*

The friction stir welding process has never been used for joining rails in the railway environment. In joining rails by using friction stir welding, the following aspects are required:

- The profile shape of the rail must be kept. This is to ensure that
- The material properties of the rail weld joint must be that of the parent material.
- The entering and exiting point of the rotating tool pin must not damage the rail profile

The focus of the design project will be mostly on the structural mechanical design of the machine and will therefore not be an in depth into the metallurgical processes and the macrostructure of the material. During this study the following hypothesis is to be tested:

The friction stir welding process can be used to join rails of different profiles and it can also yield a high quality weld. A critical conceptual design of a friction stir welding machine could result in adding value to safety, increasing weld joints quality and productivity and reducing costs of rail joining in railway industry in South Africa.

1.5.1.1 RESEARCH AND CONCEPTUAL DESIGN APPROACH

The conceptual design and research implemented in this report are outlined below. This approach is used mainly to address the research hypothesis. As mentioned earlier in the chapter, the successful design concept must be the one which will give costs benefits and give rise high quality welds with a limited number of machine operators. Hence, the major contribution of this research report is found in chapter 4. A brief outline of the report is given below.

Chapter 2: Literature review and background knowledge:

Process modelling and prognosis of a friction stir welding process are vital before designing the actual model of the process. Therefore, various FEM methods for envisaging friction stir welding processes have been investigated from various sources and various methods have been outlined.

Chapter 3: Design proposal

In chapter three, the design development has been presented in detail. The design development comprises six design concepts, from which the best concept has been chosen based on its performance and design requirements. Basic friction stir welding machine requirements have been explained in detail.

Chapter 4: Design specification

The overall description of the designed machine and the general assembly of the machine have been presented. These also include two subassemblies for the top structure and the bottom structure of the machine and three different views of the general assembly. A cost analysis of the friction stir welding machine was carried out and various assumptions have been made since most of the information is not available.

Chapter 5: Conclusion and recommendation

The design and research outcomes will be summarised in chapter five.

Appendix A:

The following sub-headings are clearly discussed:

- Brief history of friction stir welding
- Detailed explanation of friction stir welding terms
- General understanding of friction stir welding techniques

Appendix B:

In order to validate the concept chosen, design calculations were performed on the following components: extended shaft, rail clamp, structure supporting the electric motor, vibration isolator and the lifting mechanism. Also, the final design concept includes purchased components, such as the drive motor unit from ALSTOM; the electro-hydraulic drive from Dabeb-elram; and the coupling that connects the output shaft of the motor drive and the extended shaft from the Falk Corporation.

Appendix C:

Conditions monitoring of welded rails are discussed. These include amongst others, how welded rails are inspected, tests performed on the welds etc.

Appendix D:

Tables on how to identify rail types and rail profiles are placed in this section.

Appendix E:

Tool loading requirements for available materials like aluminium and low carbon steel are also discussed in this section.

Appendix F:

The following sub-sections are discussed:

- Selected electric motor dimensions
- Selected electro-hydraulic geometry
- Performance table for electro-hydraulic actuator
- Coupler section information from the Falk Corporation

Appendix G:

Personal/email communication

Appendix H:

Engineering graphs and tables used for calculations are placed here.

1.6 CHAPTER SUMMARY

This chapter provides a brief outline of how rails are joined in South Africa. Both the primary and secondary objectives of the study have been outlined. The research methodology for this project has also been presented.

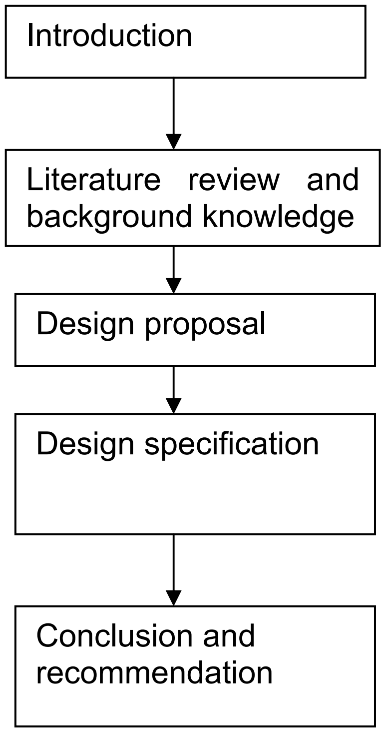


Figure 1.1: Structure of the report

2 LITERATURE REVIEW AND BACKGROUND KNOWLEDGE

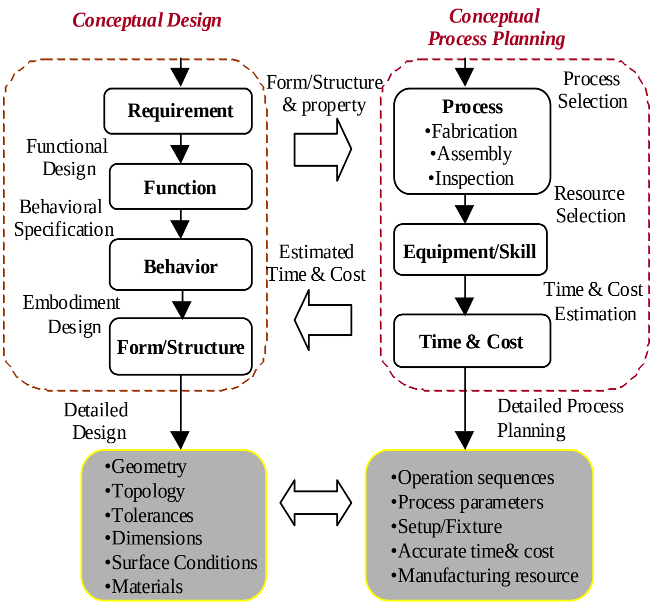


Figure 2.1: **Integration of conceptual design and process planning**

This section outlines the review and process analysis of friction stir welding as applied to joining flat sheet. The work of researchers who have contributed to mathematical modelling and FEA modelling of the friction stir welding process have been looked at. The review includes the conclusions made on what parameters are vital in friction stir welding process. Also, this section explains and introduces the reader into the world of friction welding. A brief history of welding as a general term has been discussed and introduced in Appendix B. Various types of friction welding have also been discussed. The types of friction welding discussed amongst others are: friction stir welding, linear friction welding, spin friction welding.

“Conceptual design is the creation, exploration, and presentation of ideas on how a new product will work and meet its performance requirements. Conceptual design is about what is possible for a product to be (Armour, R. et al 2007)”. To different people, conceptual design cannot mean the same things. The concept could represent e.g. the automotive future, but also represent the style and robustness of the company presenting them. “To others, conceptual design is all about making prototypes and thus limiting their idea of conceptual design”.

Stages in conceptual design (as shown in Figure 2.2) include:

- Develop Requirements
- Develop Concept
- Validate Concept and Work Statement

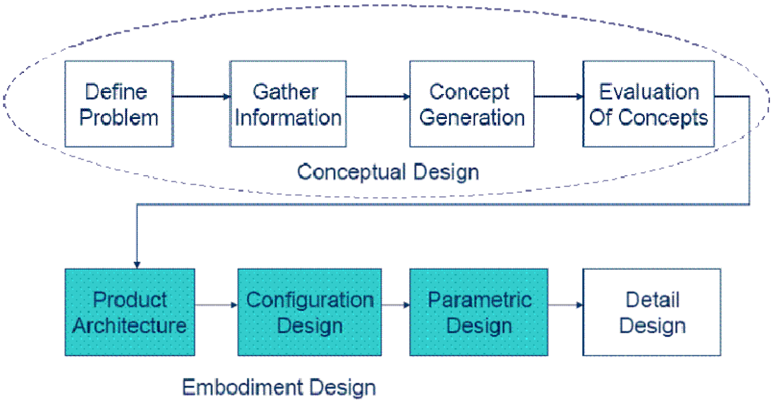


Figure 2.2: **Conceptual design**

Figure 2.1, shows how conceptual design and process planning are integrated (Shaw and Eugene, 2000). This research project focuses mainly on the conceptual design of the friction stir welding machine. Even though more work is to be done on the friction stir welding process itself, the conceptual design of the machine is to focus mainly on the structural integrity of the machine. As shown in Figure 2.1, conceptual design encompasses the generation of concepts and integration into system-level solutions, leading to a relatively detailed design.

2.1 REVIEW OF PREVIOUS WORK AND AVAILABLE LITERATURE

2.2 FRICTION STIR WELDING IN GENERAL

This process of welding is similar to angular friction welding; however it is used mostly to join two metals without fusion or filler materials and without reaching the melting temperature of the metal. Friction stir welding was invented at the Welding Institute (TWI), Cambridge, in early the 1990s. It is used mostly to join structural components made of aluminium and its alloys, copper and its alloys, lead, titanium and its alloys, magnesium alloys, zinc, plastics, and mild steel (Stockholm, 2005). This process has been proved to yield high-quality welding results as compared to the traditional way of material welding. Also, this process is mostly used and suitable for joining long and flat pieces of plates and sheets. In this process, the welds are formed by the combined action of heat due to friction and the mechanical deformation due to a rotating tool. Furthermore, the process requires the use of parameters for each application to provide optimum weld characteristics. These parameters include amongst others tool design, rotation speed, and welding force (Colligan, 2001).

The basic process of friction stir welding is shown in Figure 2.1 (FPE & Gatwick Fusion Ltd).

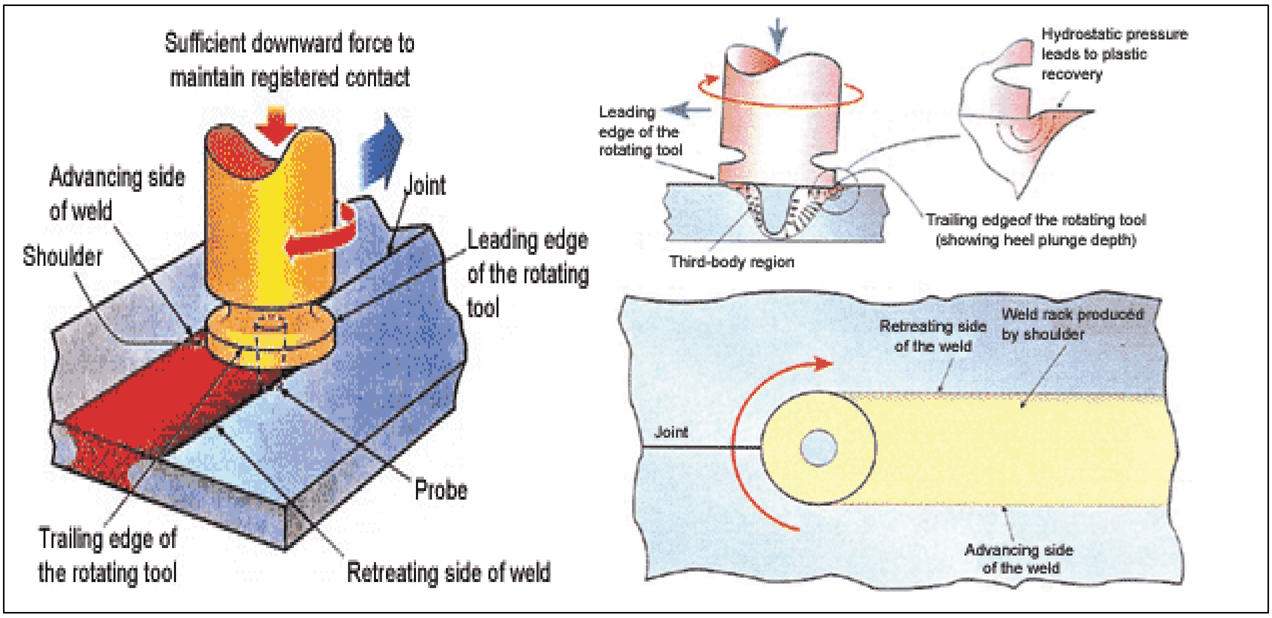


Figure 2.3: Fundamental process of friction stir welding

In the past friction stir welding used to require high capital investment; however, due to rapid technology improvement, capital investment costs have been considerably reduced (Smith *et al*, 2001). Furthermore, more industries are engaged in using the process because of its high cost benefits. At the moment products like chassis, crush horns, body enclosures, hoods, suspension links and gas tanks are being designed for the process. Also, friction stir welding does not only give the cost benefits but it also provides vast improvements in mechanical properties. Figure 2.3 shows friction stir welding parameters and terminology used to identify them.

Friction stir welding has the following benefits amongst others (Smith *et al*. 2001):

- Improved weld quality
- Reduced distortion
- Low power requirements
- No filler metal or shielding gas required
- No unsightly soot
- Adaptable to all positions
- Fatigue life 2-10 times longer than arc welding
- Able to join various non-ferrous alloys (even those considered un-weldable)
- Mechanical strength of joint equal or close to original base material
- Could weld material thickness ranging from 1 mm to 50 mm and above
- Generates no fumes or zone (that is environmentally friendly)
- Quiet
- No spatter
- No ultraviolet light
- Reduced need for cleaning of part (that is, reduced need for chemical cleaning agents)

Typical examples of friction stir welding in practice are shown in Figure 2.4 (Colligan, 2001)

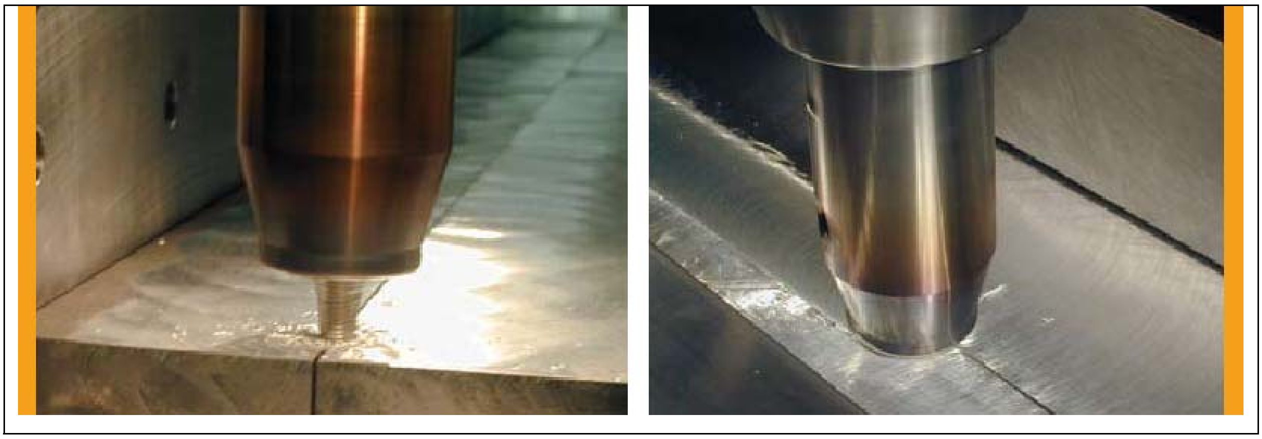


Figure 2.4: Welding tool plunge (left) and transverse (right).

Friction stir welding has a proven track record and this is shown by the increasing number of companies that are moving rapidly to adopting friction stir welding process. Colligan, (2001) summarises companies that adopted friction stir welding from 1995 to 2004 in ship construction, shown in Table 2.1 (Colligan, 2001).

Table 2.1: Companies who adopted friction stir welding from 1995 to 2004 in ship construction.

Year	Application	Company
1995	Hollow heat exchangers	Marine Aluminum, Norway
1996	Commercial shipbuilding	Marine Aluminum, Norway
1998	Delta II rockets	Boeing, US
1999	Commercial shipbuilding	SAPA, Sweden
2000	Automotive components	SAPA, Sweden
2000	Laser system housings	General Tool, US
2001	Motor housings	Hydro Aluminum (formerly Marine Aluminum), Norway
2001	Automotive components	Showa, Japan
2001	Train bodies	Hitachi, Japan
2002	Automotive components	Tower Automotive, US
2003	Aircraft structure	Eclipse, US
2003	Commercial shipbuilding	Advanced Joining Technologies, US
2004	Space shuttle external tanks	Lockheed Martin, US
2004	Food trays	RIFTEC, Germany

Smith *et al.* (2001) summarises the industry category, specific application, present process and the advantages of friction stir welding (FSW) and the information is tabulated in Table 2.2 (FPE & Gatwick Fusion Ltd website).

Table 2.2: Typical applications of friction stir welding

Industry category	Specification application	Present process	Advantages of FSW
Electrical	Heat sinks-welded laminations	GMAW	Higher density of fins-better conductivity
Electrical	cabinets, enclosures	GMAW, RSW	reduced cost, weld through corrosion coatings
Batteries	leads	Solder	Higher quality
Military	shipping pallets	GMAW	reduced costs
Extrusions	customized extrusions	Not done today	Could customized, reduces need for large press
Boats	keel, tanks	GMAW, rivets	stronger, less distortion
Golf cars, snowmobiles	chassis, Suspension	GMAW	less distortion, better fatigue life
Tanks, cylinders	Fittings, long and circum seam	GMAW	higher quality -less leaks higher uptime
Aerospace	Floors, wing spars	Rivets	higher quality -cheaper (no rivets and holes)

2.2.1 *PREVIOUS WORK ON JOINING METAL SHEETS BY FSW PROCESS*

From the literature review it can be concluded that the shape of a tool pin plays an important role in friction stir welding (Elangovan at al. 2007). They stress the influence of tool pin shape and axial force on the formation of the friction stir welding process. In their study, AA6061 aluminium alloy was used during the experiment. They summarised their findings in graphical way. Figure 2.5 (Lienert *et al*, 2003), shows how axial force and tool geometry may affect yield strength, tensile strength and joint efficiency.

Woo *et al*. (2007) also studied the influence of tool pin and shoulder on microstructure and natural aging kinetics in the FSW process. In their study, 6061-T6 aluminium alloy was used for the experiment. Natural aging behaviour after the FSW process was investigated in these studies. In this study no major conclusion which will add to this project has been made.

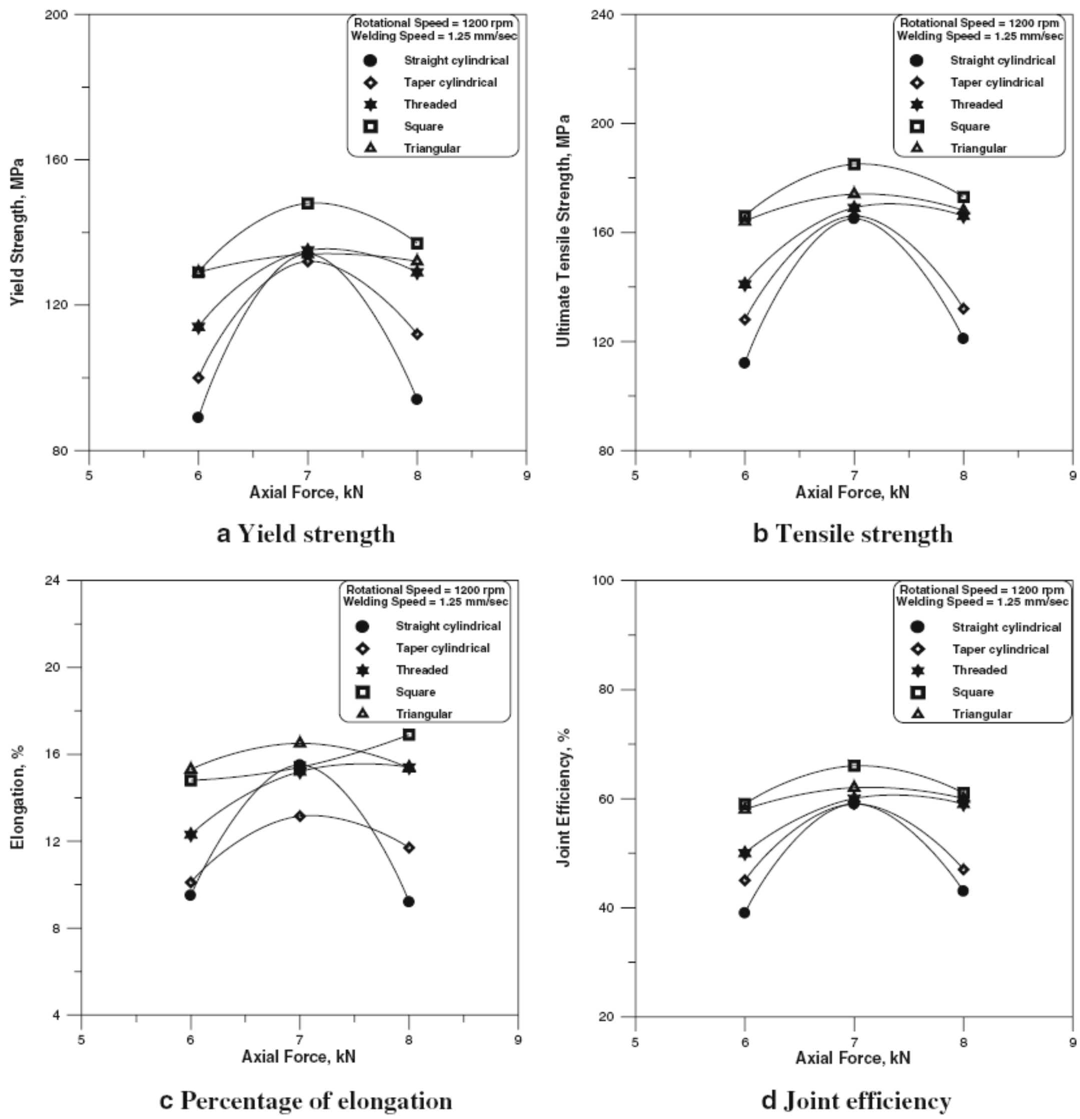


Figure 2.5: Effect of axial force and tool pin shape on tensile properties.

Kallgren, (2005) in his PhD thesis, studied the joining of copper canisters for nuclear waste by means of friction stir welding. In his studies, he focused on the characterisation of the FSW joints and modelling of the process, both analytically and numerically.

However, Zhang and Chen, (2007) have studied the material behaviour in the FSW process by means of using rate-dependent constitutive model. In both their studies, 1081 steel was used in the experiment. The following conclusions were made:

- The border of the shoulder can affect the material flow near the shoulder-plate interface.

- The mixture of the material in the lower half of the friction stir weld can benefit from the increase in the angular velocity or the decrease in the welding speed. But flaws may occur when the angular velocity is very high or the transitional velocity is very small.
- When the angular velocity applied on the pin is small or the welding speed is high, the role of the extrusion of pin on the transport of material in FSW becomes very important.
- Swirl or vortex may be easier to be observed with the increase in the angular velocity of the pin.

Zhang and Zhang (2007) present the 3D material flows and mechanical features under different process parameters by using the finite element method based on solid mechanics, which study confirms that tangent flow constitutes the major part in material flow. The increase of speeds, including the rotational speed and transitional speed can both accelerate the material flow, especially in the front of the pin on the retreating side where the fastest material occurs. Again, this study concluded that the contact pressure on the pin-plate interface is decreased with the increase of angular speed.

2.2.2 *PREVIOUS WORK ON JOINING RAILS BY FRICTION STIR WELDING PROCESS*

Literature reviews that there is no work being done on joining rails by means of friction stir welding. A rail, by its nature is composed of high carbon steel, and from literature there is little work being done on joining high carbon content steel or even low carbon steel. The friction stir welding is mostly being used for joining aluminium sheets metals and various papers have been written on joining sheets by means of the friction stir welding process.

2.2.3 *PREVIOUS WORK ON JOINING RAILS BY OTHER METHODS*

According to Ngoato, (2007), the following rail welding types are used in the South African rail industry: Flash butt welding, exothermic (exothermic) welding and arc welding (electrode welding). (See Appendix D). Arc welding is normally used where there is a skid mark in the rail. It is used only for filling up holes (skid marks) caused by the high pulling force (power) of a locomotive. Ngoato, (2007), describes flash butt welding as the fusing of two rail ends at very high pressure and temperature. The Exothermic welding is performed by igniting the mixture of aluminum and iron in a mould placed on top of two rail ends with a definite rail gap.

The greatest competition of rail welding techniques lies between exothermic welding and flash-butt welding. The exothermic weld still enjoys a larger proportion of welds as compared to that of flash-butt welds while the electric flash-butt weld has strong competition from exothermic companies (Stump, 1998).

Below are the different types of rail joining processes used to join rails in South African railways. See Appendix C on how rails are inspected after being welded and the test conducted to test the quality of rail. Each process is discussed in great detail.

2.2.3.1 RAIL JOINED BY NUTS AND BOLTS

As shown in Figures 2.6 and 2.7, this type of rail joining uses purely mechanical mechanism components which includes the fisher plate, nuts and bolt. Figure 2.6 and Figure 2.7 show how two rails are joined by means of a fisher plate. This method has been used over decades for joining rails; however, it has some limitations. In many countries, including South Africa, this type of practices is being replaced by exothermic welds which are believed to be sufficient and appropriate for the joining of rails. It has been established that the maintenance costs of rail and other components could be reduced by at least 50% if fisher-plated joint is eliminated (More, 1993). In addition, the life of rails, sleepers, ballast and subgrade components could be prolonged by reducing the number of fisher-plated rail joints in the railway line (More, 1993).

Figure 2.6: Wagon to pass jointed rails

Figure 2.7: Nut and bolt jointed rail in South Africa

2.2.3.2 RAIL JOINED BY ARC (ELECTRODE) WELDING

The Electrogas welding (EGW) was developed in 1961 and is often used in the shipbuilding industry and in the construction of storage tanks. The Arc welding process is defined as a continuous vertical position arc welding in which an arc is struck between a consumable electrode and the work piece (Howard and Helzer, 2005). Electrogas welding is unique because the electrode is not extinguished. The electrode remains stuck to the working piece until the process is completed. In railway industries applications, arc welding is used to repair the skid marks.

The heat generated by power electricity causes both electrode and the working piece to melt. Although, the electrogas welding can be used on low and medium carbon steels and stainless steel, it has some limitations. In most cases, it can not be used in high carbon content steel and other specialized materials such as copper. In order to successfully apply the electrogas welding process in quenched and tempered steels, high heat needs to be applied on the working piece. As a rule of thumb the maximum thickness which this process is able to weld is 10 mm and the maximum available diameter of the electrode is 200 mm while the height of the electrode varies from 100 mm to 20 mm (Howard and Helzer, 2005).

The electrode has positive polarity and constant voltage and direct current is supplied to the electrode. The welding current of this process usually varies from 100A to 800A and the voltage varies from 30 to 50 V (Howard and Helzer, 2005). This technique was used during the time when it was cheaper than using bolted joints. However, at present not many companies are still use this welding technique. Moreover, this welding technique is associated with long welding time and high defect rate, both of which are not desirable. As a result, most countries including South Africa, Canada and USA are not using this technique to any further extent (Stump, 1998).

2.2.3.3 FLASH BUTT WELDING AS APPLIED IN RAILWAY

The flash-butt welding technique is rated number one worldwide in terms of zero or low defect rate and high quality weld (Stump, 1998). Furthermore, welding time is much better than the above mentioned welding technique; for this technique it takes a maximum of 10 minutes to complete one weld depending on the type of rail material to be welded. The following mobile units are commercially available: Holland mobile welder and Plasser Super Stretch Flash-butt welding machine. These machines are expensive and require a large investment for maintenance. Most investors feel that the price is worth the service it delivers.

The following companies are using the Holland welder/SuperPuller combination machine: Canada National, Cionrail and CSXT. The Plasser Super Stretch unit is used by Union Pacific and Burlington Northern.

Electric flash-butt welding has high productivity levels and is able to maintain the material properties of the rail very accurately (Stump, 1998). Electric flash-butt welding depends less on human skills than the exothermic welding technique. This technique also has some limitations, for example, in an irregular welding situation such as switches. This is the place where the welding head does not have the space to hold the rails.

Zhang *et al*. (2006), conducted a study on flash butt welding of high manganese steel crossing and carbon steel. This study compares the joining of carbon steel and stainless steel and that of high manganese steel and stainless steel. The study has indicated the feasibility of flash butt welding of high manganese steel crossing and the carbon steel rail through austenite-ferrite two-phase stainless steel.

Figure 2.8 (British Standards, 2001), shows the system or mechanism used for joining rail by the process called flash butt welding.

Figure 2.8: Combined vehicle rail/road for flash butt welding

The Flash butt welding has several advantages over other methods or techniques used for joining rails.

- Reduces maintenance costs

- Faster installation
- Lowest life cycle cost
- Saves trucking time
- No weld filter material
- Smaller heat affected zone
- Smaller annealed zone
- Consistent hardness
- Highest fatigue resistance
- Average life equal to the rail
- 25% savings over exothermic

2.2.3.4 RAIL JOINED BY EXOTHERMIC WELDING PROCESS

The Exothermic welding technique had little competition until about 1989. According to Norfolk Southern, exothermic is not considered as sentimental in the railway industry but is considered as a “necessary evil” (Stump, 1998). Burlington Northern has listed the exothermic welding technique as the best when it comes to rail defects (Stump, 1998).

Mutton and Alvarez, (2003), have studied the failure mode of rail welds under high axle load conditions in the exothermic welding process. They determined that about 75% of exothermic welds broke in the Newman mainline. On the other hand, Moller *et al*. (2001) reflect that exothermic welds have the ability to support service loads. Straight break failures and horizontal split-web fractures were briefly discussed in their investigation. Horizontal split-web fractures are mentioned to be common failures in both flash-butt and exothermic welds. However, friction stir welding has the potential to overcome this.

In their paper Moller *et al*. (2001), clearly stated that service performance of rails welds depends on:

- The structural behaviour of the welded joint and
- The better behaviour of the running surface.

Apart from the above characteristics, the quality of exothermic welds also depends on and are influenced by process parameters such as:

- Weld collar design
- Gap width
- Preheat conditions
- Portion chemistry

During the introduction of the electric flash-butt welding technique, manufacturers strengthen their research and development team in order to ensure high improvement in their product. The basic problem with the exothermic welding technique is that it depends on human skill and can only be used in a particular site. The manufacturers of exothermic welding tools have recently clamed that their process has involved less human interference. Very soon and not later, exothermic welding tools or manufacturers could be out of business since most projects are excluding the exothermic welding technique (Stump, 1998).

Figure 2.9 (Holland Engineering Rail Solutions website), shows how the exothermic welding technique is applied on the rail.

Figure 2.9: Example of exothermic welding applied on rail

2.2.3.5 COMPARSON BETWEEN EXOTHERMIC AND FLASH BUTT WELDING

Table 2.3: Comparison of friction stir welding and other welding processes used for joining rails

	Welding type			
	Gas (Arc) welding	Flash butt welding	Exothermic welding	Friction stir welding (proposed)
Type of rails identified	All	All	All	Selected few
Vertical and horizontal alignment of rails	Good	Excellent	excellent	Excellent
No of people required/manpower to perform the weld	3	3	6	2
Set-up time	20 minutes	3 minutes	30 minutes	5 minutes
Tools and consumables	Special tool required	No special tool	Metals	No special tool required
Recording of welding parameters	Non	Yes	Non	Non
Power required	Relatively low	High	Relatively low	Very low
Cooling control	Not known	Not known	Required	Not required
Ground smooth required	Yes	Yes	Yes	Yes
Capital (machine) costs	High	Very high	High	Low
Cutting out of defectives	Required	Required	Required	Required
Number of joints per length of rail	Not known	Not known	Not known	Not known

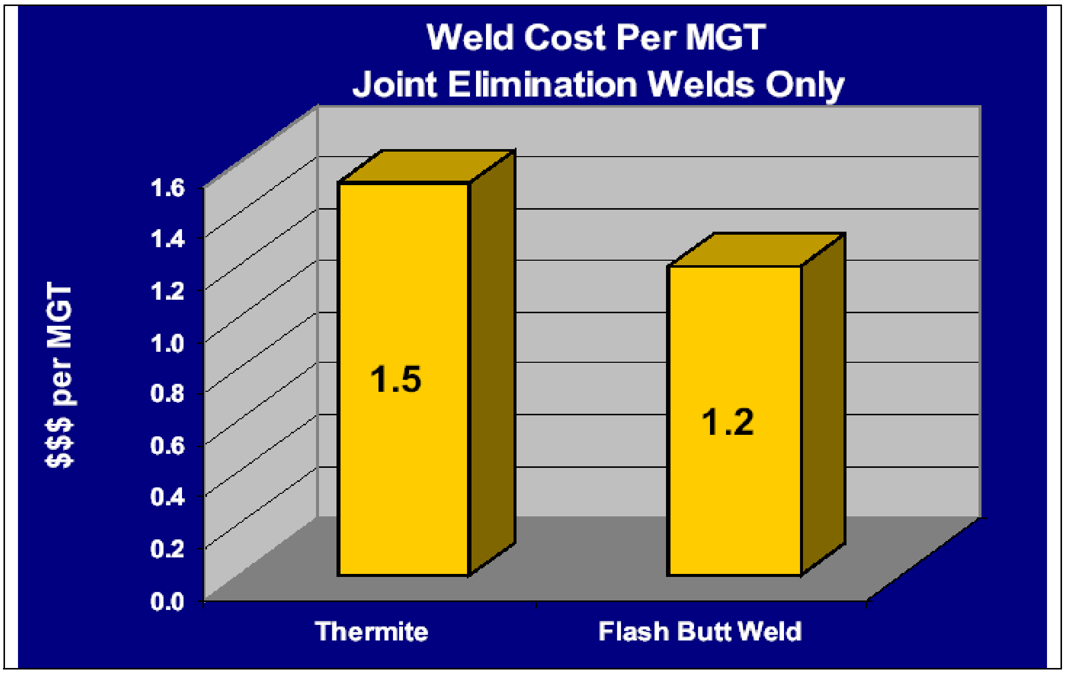


Figure 2.10: Weld cost of thermite and flash butt welding processes

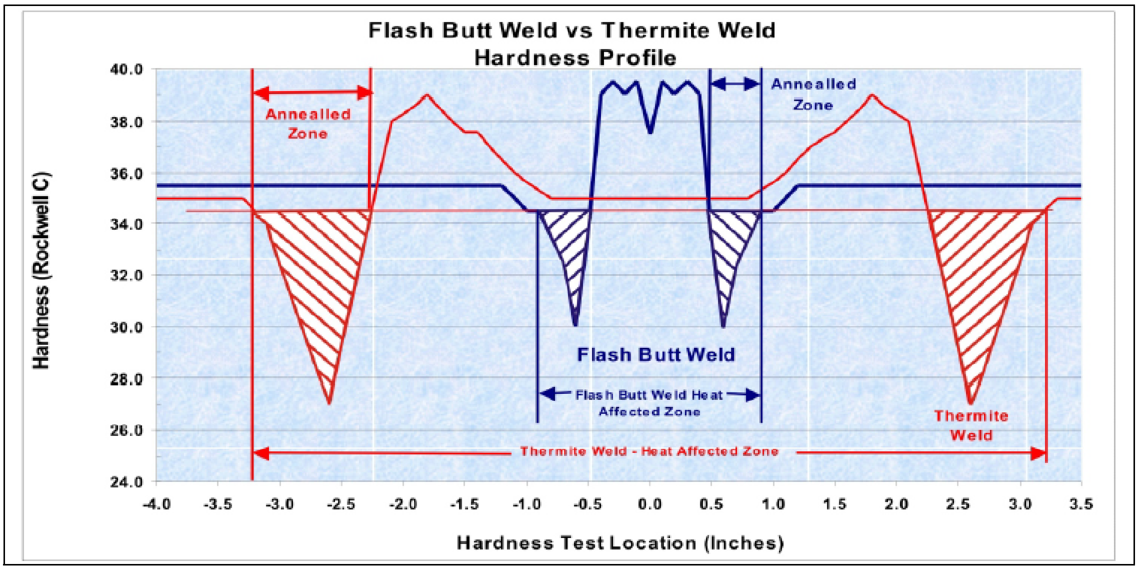


Figure 2.11: Flash butt weld vs Thermite weld hardness profile

Table 2.4: Features Comparison of welding processes.

Features	Flash Butt weld	Exothermic weld	Friction Stir Welding	Advantage of Friction Stir welding
Basic metallurgy	Forging	Casting		The process does not use any filler material and it also gives the parent material properties of rail.
Automated process	YES	NO	YES	The process does not depend much on operator skills. Average person could perform the welding
Heat affected zone	40-60 mm	145-185 mm	10-30 mm	The heat affected zone is very small as compared to other welding processes
Full Rail Profile shear	YES	NO	YES	Very good fatigue life, the weld goes through from the head to tail of the rail
Environmental pollution	LOW	High	None	No toxic gases produced or emitted
Personal Hazard	LOW	High	None	The chances of personal injury is very low as this process does not produce any molten material
Failure rate	LOW	High	None	The process has a good track record.

Figure 2.10 (Holland Engineering Rail Solutions website) shows an estimated cost per weld of both exothermic and flash butt welding techniques, and as shown in Figure 2.10, exothermic welding is more expensive that flash butt welding. Figure 2.11 (Holland Engineering Rail Solutions website) shows the weld hardness of both exothermic and flush butt welding techniques. It can also be seen from Figure 2.11 that flash butt welding has 30 Rockwell C and exothermic weld has approximately 27 Rockwell C. However, exothermic weld has a large heat affected zone as compared to the flash butt weld. Table 2.4 shows a basic feature for the three main processes, namely: exothermic, flash butt and friction stir welding techniques. In this table, various advantages of friction stir welding as compared to the other (exothermic and flash butt welding) have been given.

2.3 RAIL PROFILES USED IN SOUTH AFRICA

This section outlines the rail profile used in the South African railway environment. All profiles have disadvantages and advantages. Rail profiles are designed to carry a certain load. In the South African railway environment the following rail profiles are still in use: 30 kg/m, 40 kg/m, 43 kg/m, 48 kg/m, 57 kg/m, 60 kg/m, S-49, UIC-60 and S-60-SAR, see Appendix D. Table 2.5 shows different material types for rails mostly used in South Africa and their sizes.

The most commonly used rail profiles in South Africa are: 43 kg/m, 48 kg/m, 57 kg/m and 60 kg/m. See Appendix D for rail profile dimensions.

Table 2.5: Commonly used rail types and rail sizes

Rail type	Rail size (kg/m)
HCOB/UIC A	30 to 40
HCOB/UIC A	48
HCOB/UIC A	57
Cr Mn	48
Cr Mn	57
Cr Mn	60
HH	60

2.4 MODELLING OF FRICTION STIR WELDING PROCESS

The FSW process looks easy but its physics may become tedious and an understanding of this process is vital in achieving high quality welds. Various authors have carried out intensive research on the metal flow due to friction stir welding. Kallgren, (2005) in his PhD thesis discussed various methods used for modelling material flow due to friction stir welding. In his analysis, he used Rosenthal's analytical model, FEM solid model and FEM fluid model.

According to Mishra and Ma (2005) metal flow is complex and depends much on the type of tool pin geometry being used. Metal flow has the ability to influence the weld quality and if not understood, weld quality may be compromised. Zhang and Zhang, (2007) also provide 3D material flows and mechanical features under different process parameters by using the finite element method based on solid mechanics. In their analyses, the major conclusion made was that the tangent flow constitutes the major part in material flow. The quality of joining plates by means of friction stir welding may be increased by increasing the angular velocity of the pin. Reynolds, (2008) states that understanding material flow is critical in determining the thermo-mechanical process conditions during FSW.

Schneider and Nunes (2003) studied the characterization of plastic flow and resulting micro-textures in a friction stir weld. This study emphasizes that when the pin is inserted between the two plates, the metal becomes subjected to a thermo-mechanical processing in which the temperature, strain and strain rate is not completely understood. The optical image of the macroscopic features of a FSW transverse section is show in Figure 2.12 (Schneider and Nunes, 2003).

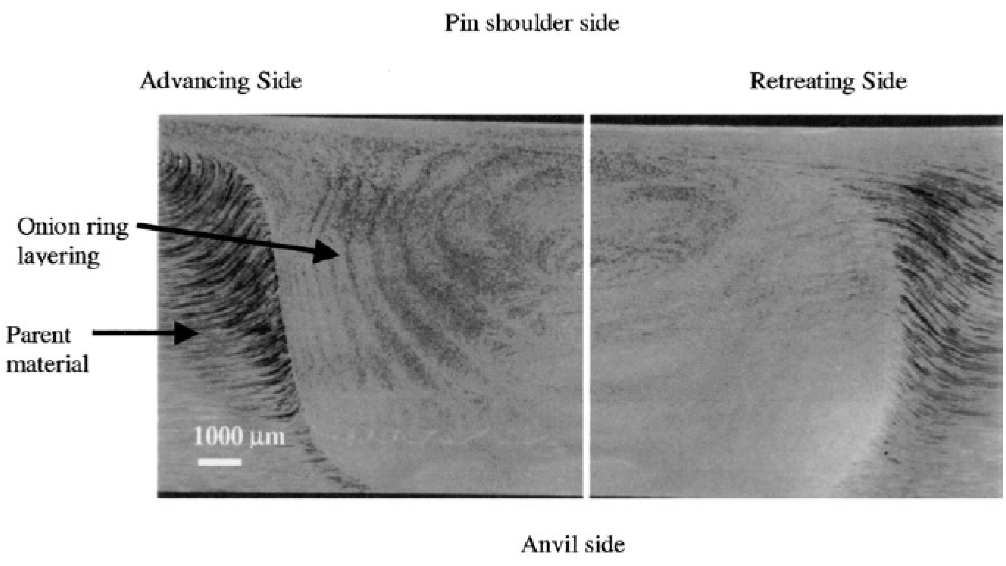


Figure 2.12: Optical image of the macroscopic features of a FSW transverse section.

Figure 2.12, clearly shows the difference between the parent material and the onion ring layer or the stir portion of the material. Figure 2.12, also suggest that if parameters such as pin speed and shoulder pressure are not controlled and understood, poor weld may be obtained. Therefore, it is important to understand clearly the microstructure of the working piece and as a direct result; high quality weld may be obtained. In addition, Murr *et al*. (1998) conducted a study on the microstructural characterisation of aluminium alloys due to the friction stir welding process. In this research article much can be learned and the approach used may also be used in studying the characterisation of high carbon steel.

Lee *et* al. (2003) studied the feasibility of joining dissimilar material. In this case aluminium alloy, cast aluminium alloy and wrought aluminium alloy were used for the purpose of this study. They clearly concluded that the friction stir welding method could be used to join dissimilar aluminium alloys which have different mechanical properties without weld zone defects under a wide range of welding conditions. Su *et al*. (2007) also came to the same conclusion that two metals of different chemical properties may be joined together by employing the friction stir welding process. They came to this conclusion by experimenting with welding two different aluminium alloys by means of FSW process. For the purpose of joining two rails, this conclusion may become useful since it is possible to find two rails of different chemical properties but of the same geometry.

2.5 DESIGN OF THE FRICTION STIR WELDING TOOL AND ITS INFLUENCE

The aim of this section is to explain how different parameters in friction stir welding affect the quality of welds carried out using friction stir welding. As compared to other welding processes, the friction stir welding process does not use a consumable working tool. Therefore, there is no filter material in the friction stir welding process. Due to the complexity of the process, several terms need to be defined for a better understanding of friction stir welding in railway. The following terms are explained in the Table 2.6 from (Mishra and Ma, 2005): namely, tool pin, tool shoulder, advancing side, retreating side, plunge depth, rotation speed and welding speed.

Figure 2.13 (Mishra and Ma, 2005) and Table 2.6 help clarify the most common terms in the process (FSW)

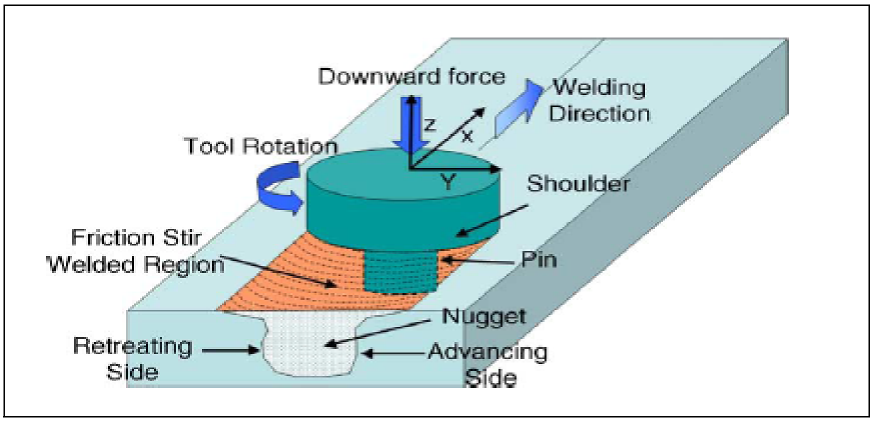


Figure 2.13: Friction Stir welding process terms.

Table 2.6: Definition of useful terms.

Terms	**Characteristics**
Tool pin	• It is extended from the shoulder of the working tool from the motor. • The shape of the tool pin is usually conical • It could be of any shape starting from a simple circular shape to a thread tool
Tool shoulder	• This is a disk shaped part of the tool pin. • It forms the weld cap in the process of friction stir welding.
Advancing side	• The side of the tool where the local direction of the tool surface due to tool rotation. • The direction of transverse is in the same direction.
Retreating side	• The side of the tool where the local direction of the tool surface due to tool rotation. • The direction of transverse is in the opposite direction.
Plunge depth	• Is the maximum length of the tool pin inside the welded piece.
Rotation speed	• Measured by rev/min and is defined as the speed of the rotating tool perpendicular to the welded plane.
Welding speed	• The speed measured in mm/sec travelling across the weld joint line.

A summary of conditions which could generate poor quality welds is shown in Table 2.7 (Mishra and Ma, 2005). The following parameters are considered: namely: thickness of the weld, tool geometry, side clamping force, rotation speed of the tool, tilt angle, rail profile, tool cooling, tool axial welding force, physical stops to prevent further movement to the welding head and the altered welding force.

Table 2.7: Summary of conditions that lead to non-desired welds quality.

Parameter change	**Non-desired welds quality**
Weld thickness	Voids on the weld root could occur due to the tool pin length
Tool geometry	Tool geometry will determine the microstructure of the weld
Side clamping force	If the side force is enough to keep the working piece (two rails) on the ground, minimum voids will develop near the surface on the advancing side
Rotating speed of the tool	Very high speed of the tool pin is not desirable as it causes overheating and grain growth. Small triangular shaped voids on the advancing side, usually concurrent with low rotation speed and low weld temperature.
Tilt angle	Tilt angle has ability to change the microstructure of the working piece (rail properties) and also the sub-surface frame.
Rail profile to be welded	Rail profile need to be checked before welding otherwise, the same rail profile need to be welded together.
Tool cooling	Surface breaking voids, grain growth (depending on effectiveness)
Tool axial welding force	This force plays a very important role in the quality of weld. If not controlled properly surface roughness and flashes at the edges of the shoulder will develop.
Physical stops to prevent further movement to the welding head	Voids on the advancing side due to large reduction in axial welding force.
Altered welding force	If the welding force is reduce a excessively, voids on the advancing side are generated.

Friction stir welding terms are explained in detail in Appendix A, section A.2.

This project does not place much detail on the design tool, but it is clear that rotating the tool plays an important role in friction stir welding. Buffa *et al*. (2005) has presented a paper on the study and design of a friction stir welding tool using the continuum based FEM model. In their studies, they argued that tool geometry plays a fundamental role in obtaining desirable microstructures in the weld and heat affected zones. Consequently, they also claimed that tool geometry may increase strength and fatigue resistance in the joint. Tool pins of cylindrical and conical geometry were used in this study. They used 3D FEM analysis to predict the process variables together with the material flow pattern and grain size in welded joints.

2.5.1 *HEAT TRANSFER ON BOTH THE TOOL PIN AND WORKPIECE*

Even though this study is only of a conceptual design heat transfer from the tool plays an important role in determining weld quality. Therefore it is warranted that literature on this subject be reviewed in detail in order to learn about and improve on the process.

Meng *et al*. (2006) optimised the stir head for FSW based on genetic algorithm. In their studies, they urged that constraints for optimization be established. A mathematical model was also developed in order to optimize the stir head for FSW. Their studies were based on the simple assumption that the function of the shaft shoulder is to provide friction contact points and sealed welding conditions so that plastic metal cannot be extruded through it. The shape of the needle has an influence on material flow and consequently tensile strength. Material in the heat affected zone of cone crew harder, and the hardness of the onward side of the cone screw weld joint is clearly larger than usual (Meng *et al*. 2006). Arora *et al*. (2008) also presented a detailed heat transfer model of friction stir welding focusing on power and torque as the main parameters in the friction welding process. Chen and Kovacevic (2003) developed an FSW thermo-mechanical model. The heat source incorporated in the model involves the friction between the material and the probe and the shoulder. A mechanical model of FSW was also briefly discussed and presented.

According to their thermo-mechanical model, the rate of heat generation caused by the friction over the entire interface of the contact will be:

$$\dot{q} = \int_{r_o}^{R_o} 2\pi\omega r^2 \mu(T) p(T) dr = \frac{2}{3}\pi\omega\mu(T) p(T)\left(R_o^3 - r_o^3\right) \quad \textbf{(2.1)}$$

Meng *et al*. (2006) also gave a mathematical model for the optimisation of a stir head dimension. Assuming that R_o is the middle diameter of the stir needle and L is the length, R_2 is the lower extreme diameter. Welding speed *v*, the cone angle of stir needle is α and the rotating speed ω.

The heat which comes from the shoulder is calculated as follows:

$$\Delta Q_J = \frac{4\pi\omega f F\left(R^3 - R^3{}_o\right)}{3R^3}\Delta t \quad \textbf{(2.2)}$$

Therefore, the heat output coefficient may be calculated by using the following relationship:

$$\frac{\Delta Q}{\Delta t} = \pi P_s f\omega\left(\frac{L^3 tg\alpha}{12\cos\alpha} + \frac{R_o^2 L}{\cos\alpha}\right) + \frac{4\pi\omega F f\left(R^3 - R_o^3\right)}{3R^3} \quad \textbf{(2.3)}$$

R and R_o are decisive parameters and must be optimized, (Meng *et al*. 2006). The angle of the stir needle can also influence the heat output coefficient. According to Meng at al. (2006), it has been reiterated than the value of R must be between 17 and 27 mm and the value of R_0 must be between 3 and 5 mm. A mathematical model which allows the value of R and R_o to be larger than 27 mm and 5 mm respectively must be investigated since there is a great possibility that the FSW of rail may require larger and strong working tool. If not developed this model may be used but with care.

Meng *et al*. (2006), proposed the operation process of algorithm to optimize the FSW process. The operation process logarithm is shown in Figure 2.14 (Song and Kovacevis, 2003).

The role of heat generated by means of friction on both the workpiece and the tool pin needs to be taken into account as it plays an important role in determining the quality of a weld. The understanding of the heat transfer process in the workpiece is helpful in predicting thermal cycles in the welding workpiece and hardness in the welding zone (Song and Kovacevis, 2003). Tool pin wear-out may be prevented by understanding the temperature profile in the tool pin. In light of the above Song and Kovacevis (2003) addressed this issue by studying heat transfer modelling for both workpiece and tool: as the understanding of temperature profile in the tool pin is vital in the welding of high melting temperature like carbon steel. Their mathematical and numerical model solutions were validated by a friction stir welding experiments. In their paper three-dimensional transient-heat transfer models for both the tool and the workpiece in friction stir welding.

In developing their mathematical model for heat transfer for both workpiece and tool in the FSW, the following were introduced into the model.

- The heat generated at the tool shoulder/workpiece interface is the frictional heat.
- The radiation heat can be neglected.
- The tool pin is a cylinder; the thread of the pin can be neglected.
- The temperature cannot exceed the material melting temperature in the welding.
- The heat transfer at the workpiece/backing plate can be simplified to a conversion heat transfer with an effective coefficient.

According to Song and Kovacevis (2003), heat generated at the tool shoulder/workpice interface may be calculated by using the following relationship:

$$q_{fi} = 2\pi F_n R_i \omega$$

And the heat generated at the tool pin/workpiece interface is represented by the following relationship:

$$Q_{in} = 2\pi r_p hk \bar{Y} \frac{V_m}{\sqrt{3}} + \frac{2\mu k \bar{Y} \pi r_p h V_{rp}}{\sqrt{3\left(1+\mu^2\right)}} + \frac{4F_p \mu V_m \cos\theta}{\pi} \qquad \textbf{(2.4)}$$

Where:

$$\begin{aligned} \theta &= 90^o \lambda \tan^{-1}(\mu) \\ V_m &= \frac{\sin\lambda}{\sin\left(180^o.\theta.\lambda\right)} v_p \\ V_{rp} &= \frac{\sin\theta}{\sin\left(180^o.\theta.\lambda\right)} v_p \\ v_p &= r_p \omega \end{aligned} \qquad \textbf{(2.5)}$$

It is important to note that in this model the tool pin is assumed to be a plain cylinder and not a threaded cylinder. Song and Kovacevis (2003), have summarised a flowchart for calculating the heat or temperature distribution on both the tool pin and the workpiece. The flowchart is reproduced in Figure 2.14.

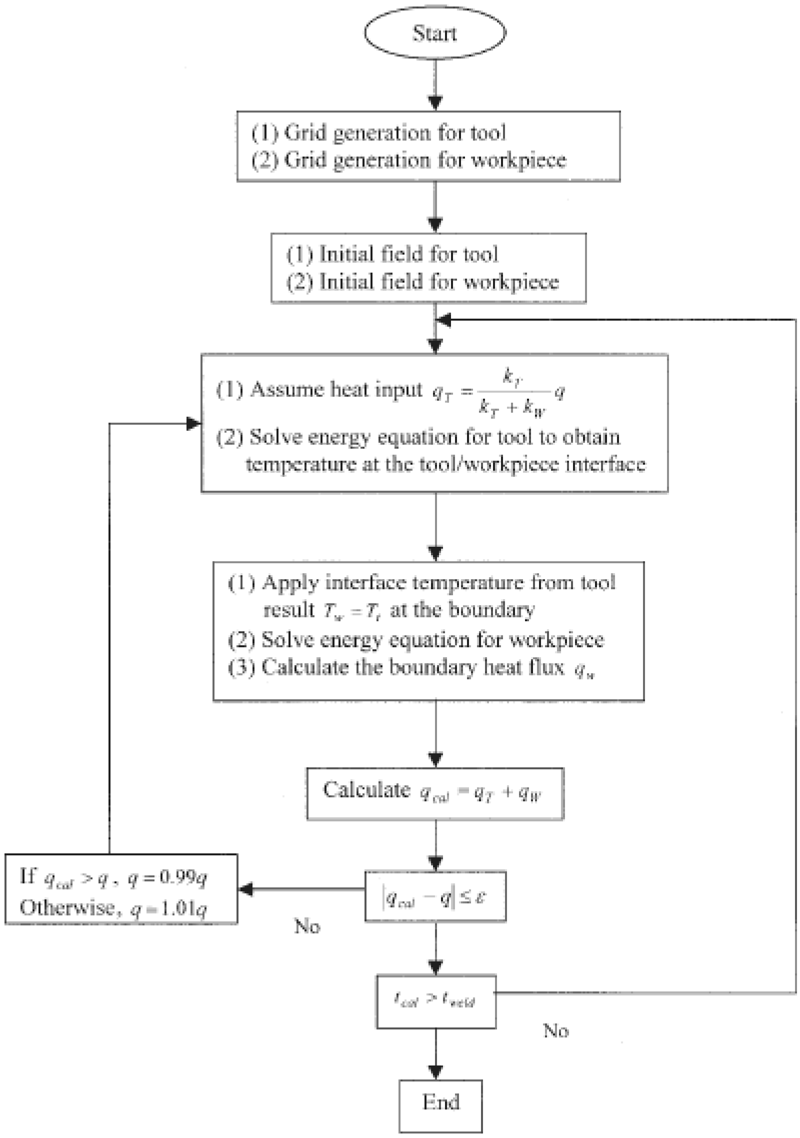


Figure 2.14: Flow-chart of a calculation.

By using this heat transfer model, the temperature contour of both the tool and the work piece may be calculated. This will yield the benefit of predicting the weld quality and also the size of the rotating tool. This model depends mostly on the material properties of the workpiece (in this case the rail) and the tool geometry and material properties of the tool pin. It is important to predict and understand the material flow during FSW. An understanding of the FSW process may result in the optimisation of the process associated with the mechanical properties of welded joint (Heurtier *et al*. 2005). Zhang and Zhang (2008) concluded that when the welding speed becomes higher, the rotating speed must be increased simultaneously to avoid any possible welding defects such as void. They also added that the simultaneous increase of the rotating and translating speeds of the welding tool can lead to an increase of residual stress.

Elongovan *et al*. (2008) also studied the influence of the tool pin profile and axial force on the formation of friction stir welding on AA6061 aluminium alloy. Welding parameters such as tool rotating speed, welding speed, axial force and the tool pin profile plays a major role in deciding the weld quality. In this study, five different tool pin profiles were used to join aluminium alloys, namely: straight cylindrical, tapered cylindrical, threaded cylindrical, triangular and square. In addition, three different axial forces were used in order to evaluate the weld quality. It was then concluded that the square tool pin profile produces mechanically sound and metallurgical defect free welds compared to other tool pin profiles. The tool pin profiles used are shown in Figure 2.15 (Elangovan at al. 2008).

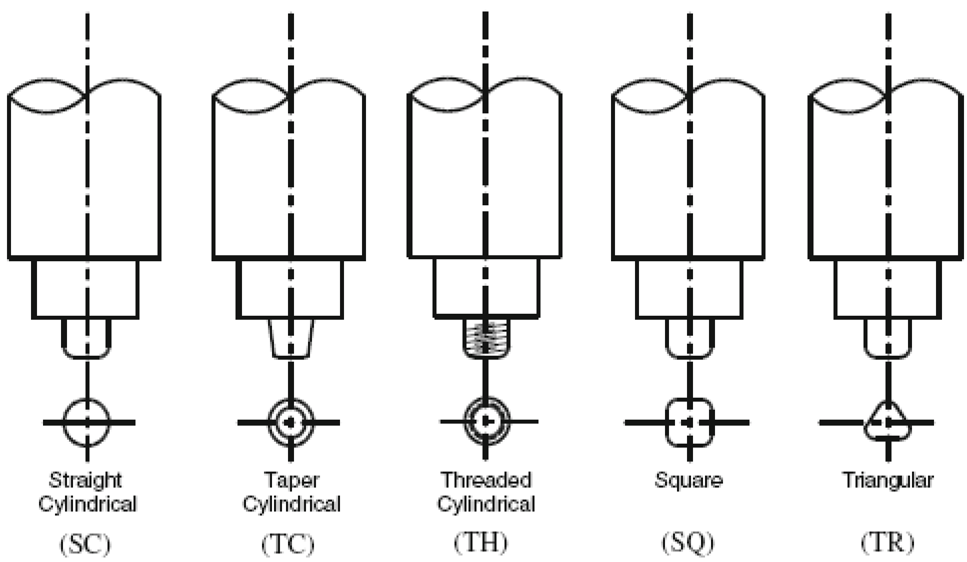
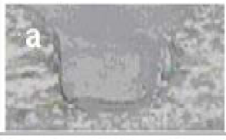
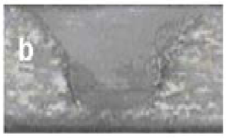


Figure 2.15: FSW tool pin profiles.

Although all the microstructure based on different pin tool profile are tabulated in Table 2.8 (Elangovan *et al*. 2008), it is in one's best interest to show the microstructural results of a square shaped tool profile. Following on Elongovan *et al*. (2008), the results are shown in Figure 2.16 (Meng *et al*. 2006).

Table 2.8: Microstructure of the joints fabricated by square pin profiled tool.

Axial Force (kN)	Macrostructure RS / AS	Size of FSP zone (mm) W	Size of FSP zone (mm) H	Shape of FSP zone	Name of the defect and location	Quality of weld metal consolidation	Probable reason
6		11.3 6.7 4.1	5.7	Inverted Trapezoidal	No defect	Good	Sufficient flow of the metal by pulsing action of the pin profile
7		12.1 6.7 4.6	5.6	-do-	-do-	-do-	Eccentricity of the pin profile causes dynamic orbit with associated pulsating action resulted in good weld.
8		11.8 7.1 5.3	5.9	Inverted Trapezoidal with shear lips	-do-	-do-	Excess heat input due to additional axial force reduces thickness of the plate in the weld zone

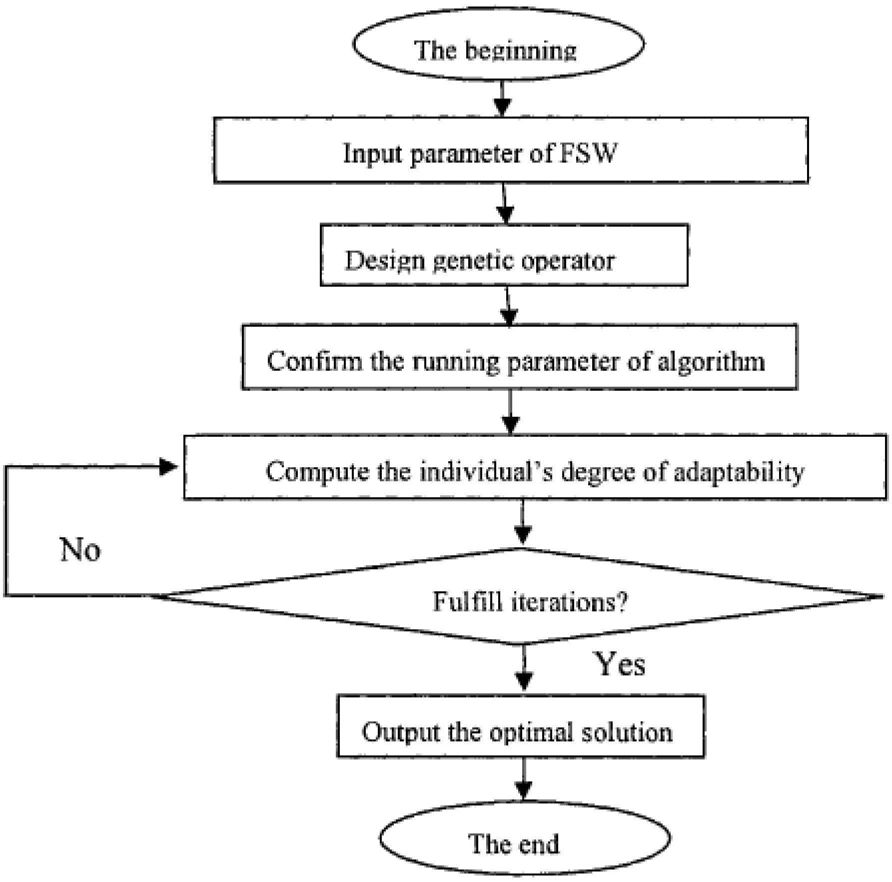


Figure 2.16: The operation process of algorithm for the optimization for stir head dimension.

In order to produce defect free welds, the following must be taken into consideration: tool pin profile, axial force, welding speed and rotational speed. Since the FSW bond depends critically on the plastic deformation of the material caused by the tool pin that plunges into the material and travels the joining line. Woo e*t al*. (2007) concluded that tool geometry, (i.e the shapes, and sizes of the pin and shoulder) is essential in FSW processing because this parameter has the ability to determine the quality of a joint and micro structural characteristics.

2.5.2 *LOAD DISTRIBUTION ON FRICTION STIR WELDING TOOL PIN*

The criticality of predicting the load distribution action of the tool pin on the friction welding process was clearly refined by Soresen and Stahl (2007). Their model is mostly beneficial to the material with limited toughness such as polycrystalline cubic boron nitride. Force distribution is mainly dependent on the length and diameter of the rotating tool pin. Their major contribution made by their studies reveals that the force due to the pin increases with pin length, but appears not to vary significantly with pin diameter. Unexpectedly, large pin length resulted in large force variation.

The total load on a pin does not give the appropriate tool pin design, therefore it is important that the total force distribution is modelled in order to identify the net effect on tool pin stress and thereby increase the probability of failure of the tool pin. Ulysee (2002) came to the same conclusion as that of Soresen and Stahl (2007), which clearly state that pin forces increase with the welding speed but when rotational speed increases the pin forces decreases.

Soresen and Stahl (2007) represented the force distribution of long pins by the following relationship:

$$F_p(L) = \frac{\beta(L - L_o)^2}{2} \qquad \textbf{(2.6)}$$

$$\beta = \frac{2F_p(L)}{(L - L_o)^2} \quad \textbf{(2.7)}$$

2.6 FRICTION STIR WELDING MACHINES

Friction stir welding commercial machines in various configurations are in existence. The following configurations of friction stir welding are: multi axis, moving gantry, portable and robotic, as shown in Figure 2.17 (FPE & Gatwick Fusion Ltd website).

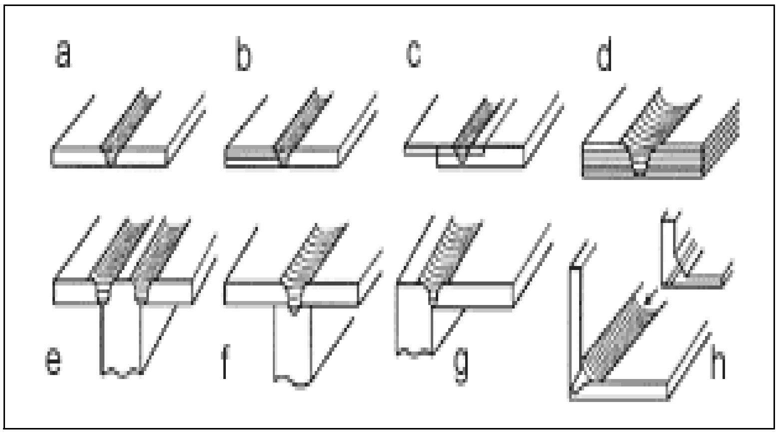


Figure 2.17: Joint geometry's suited to Friction Stir Welding.

Description of symbols in Figure 2.15

a. Square butt
b. Combined butt and lap
c. Single lap
d. Multiple lap
e. 3 piece T butt
f. 2 piece T butt
g. Edge butt
h. Possible corner fillet weld

Figure 2.18 (Colligan, 2001) shows a typical example of a vertical milling machine converted to a friction stir welding machine. With the right tool and simple modifications, the milling machine could be used for friction stir welding. The machine basically consists of a motor which helps to spin the friction welding tool and provide the heat on the specimen due to friction.

Figure 2.18: Friction stir welding machine.

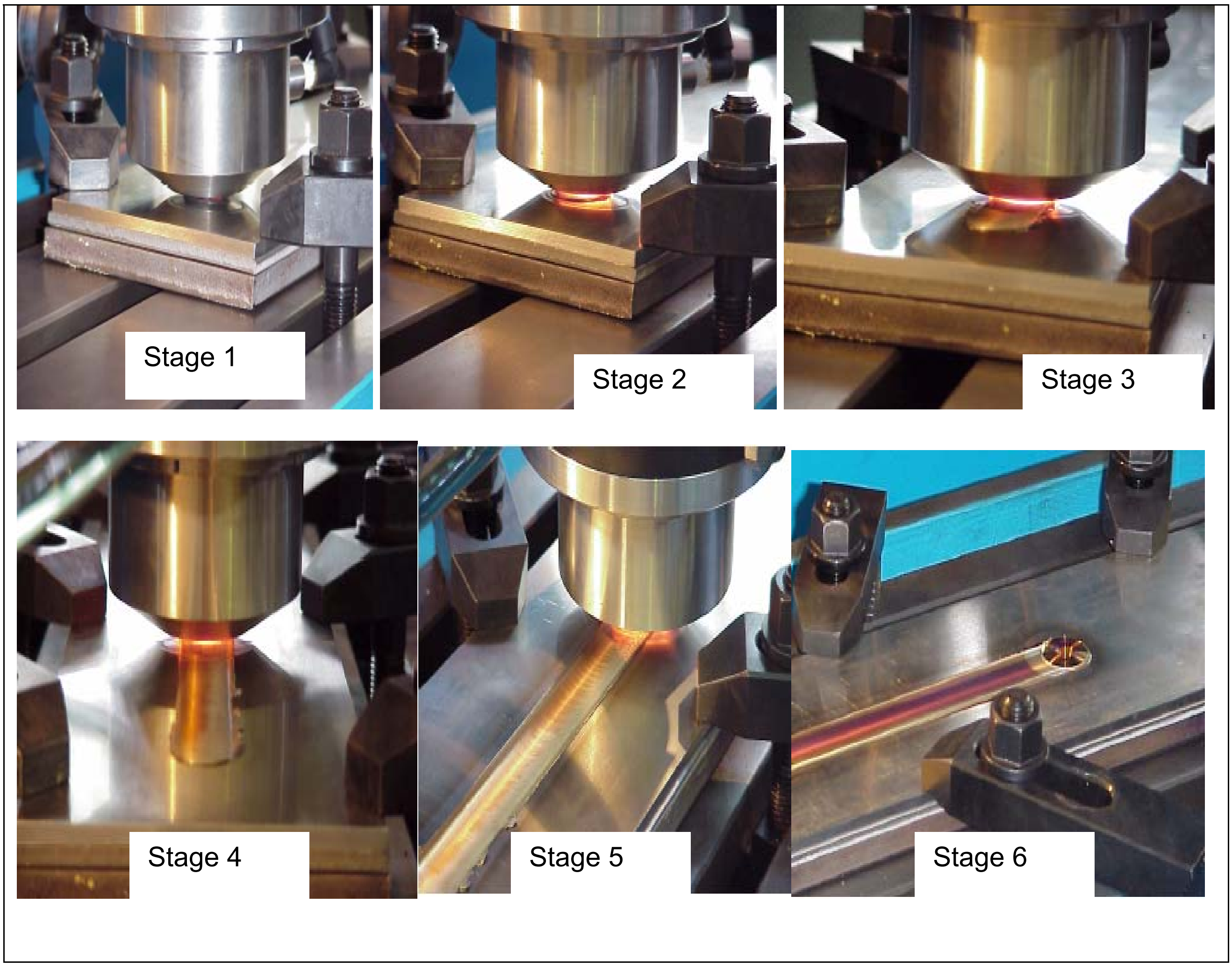


Figure 2.19: Process of friction stir welding for steel plate.

Figure 2.19 (American Welding Society website), shows the process of friction stir welding when joining 12 mm thick steel plate. The machine which has the ability to rotate at a maximum speed of 1000 rpm was used for this application. Two steel plates are fixed on the machine by means of four clamps. The function of clamps is to make sure the plate does not move apart when the rotating pin is forced on to it. Figure 2.19 (Stage 1) shows two plates about to be welded together by means of friction stir welding. Stage 2 shows the initial point of the rotating pin. Similarly stage 3 shows the rotating pin after the initial movement. Stage 6 shows the finished product of steel welded plate. This is practical example of the friction stir welding technique used in joining two thick steel plates.

As seen from the above available machine research, in most cases lazer machines can be converted to become friction stir welding machines.

3 DESIGN PROPOSAL

3.1 DESIGN REQUIREMENT SPECIFICATION

This section outlines the basic friction stir welding machine requirements. This helps the designer in focusing on the intended features. In order to develop various concepts of a friction stir welding machine, the following product requirement specification is tabulated.

The following symbols are used:

W – Wish
C – Constraints
D – Demand

Table 3.1: Product Requirement Specification

Factor	***W***	***C***	***D***
General form of design			
The product design concept must be operated by one or two persons. The person operating the machine needs not wear any safety clothes and the machine must be environmentally friendly. The machine set-up time on site must be very low with minimum separate components. The weight of the machine components must be capable of being carried by one or two persons from the store room to the welding site.		**C**	
Geometry			
The diameter and shape of the pin tool must in agreement with the tool pin design criteria.		**C**	
The motor must have an output diameter of between 50 mm and 96 mm. the output diameter of the motor is the shaft that is to be connected to the working pin. The optimal diameter of the working tool based on the geometry of the rail must be between 60 mm and 100 mm diameter, therefore, it is desired that the		**C**	

motor output diameter has a diameter which is equal to or less than that of the working tool.			
Side block aligners (rail clamps) must be of adjustable height to accommodate different rail types. The block must be adjusted from 127 mm to 165 mm as an absolute maximum. In practice rails are found not to be of the same size mainly due to the operating conditions that they are subjected to. Consequently, rails have different sizes; therefore, the rail clamp must have the ability to adjust height. Rails have different profiles; therefore, there is a need to have a rail clamp which can be used for several rail profiles.		**C**	
Kinematics			
The welding speed of the tool must be controllable. The motor must be installed with a speed controller to assist in experimentation of weld quality.		**C**	
The working tool pin must be changeable easily. A wide range of tool diameter must be accommodated. The tool diameter must vary from 60 mm to 100 mm diameter. This must be accommodated by using the coupler which has the variable diameter hole.		**C**	
The tool pin must be able to move in a transverse direction. The force applied in the traverse direction must also be controllable. Therefore, force control is required.			**D**
Actuator which can be maintained cheaply is, user friendly and small geometry must be used to apply the required welding force on the working piece (rail)			
Vibration due to motor rotation must have a minimum impact on the machine structure. The noise due to tool pin rotation must also be minimized.		**W**	
A maximum rotational speed of 2940 rpm. The rotating speed is based on the type of material to be welded.			**D**
Cost			
The friction stir welding machine must be able to be produced or	**W**		

manufactured locally using low cost manufacturing techniques.			
Energy			
The final concept of a friction stir welding machine must use maximum power of 35 kW. The power needed is the combination of power required by the main motor and the power required by the actuator. Based on the rotational speed required, the power and torque required by motor can be easily calculated.	**W**		
A normal 220 V source must be required for operation and the frequency of 50 Hz	**W**		
Life span			
The concept to be considered must be able to succeed the duration of a test and trial test.		**C**	
Ideally the friction stir welding machine must attain the maximum number of years of actual use in the field without any major service.	**W**		
Maintenance			
In order to reduce the machine set-up time, the friction stir welding machine should be assembled and dissembled easily. If possible non-power and simple mechanical tools must be used when assembling or disassembling.	**W**		
The friction stir welding machine should be easily maintained during field operation	**W**		
Safety			
General rules of safety are applied. The product must pose no danger to the operator.	**W**		
Materials			
Movable parts must be small and weight light so that only one person is able to carry the parts. The operator must not feel exhausted due to the heaviness of the machine after use.	**W**		
All material to be used in building the designed product must be of normal low, medium and high carbon steel except the tool pin which requires strong and special material.	**W**		
Low and high carbon steel and aluminium may be used to build	**W**		

machine components. Low and high carbon steel are very affordable and have the desired mechanical properties. In addition aluminium has the advantage of being light weight, which is also a concern in the new product.			
Operations			
Must be operated by a maximum of four operators for set-up and two operators or a single operator for the actual welding.	**W**		
Takes minimum possible time to complete one single weld	**W**		
Manufacturing and Assembly			
It must be assembled easily and locally.			**D**
It must be able to be assembled in a very short time to reduce the overall time for performing the actual welding process.	**W**		
Installation			
It must be easily installed in the railway environment. No special tool is to be used for installation purposes.	**W**		
Ergonomics			
User friendly	**W**		
Appearance			
The whole structure should be well structured.	**W**		
Entire unit should be aesthetically pleasing	**W**		
Performance			
The tool pin must be able to rotate at the desired speed all the time			**D**
The maximum speed of the machine must be marked accurately and precisely in a visible area to operators	**W**		
Place or area of operation			
The designed product must be flexible enough to be used in straight rail lines and open places. The unit must be able to friction stir weld the following rail types: (i) 43 kg/m; (ii) 48 kg/m; (iii) 51 kg/m and (iv) 60 kg/m.		**C**	

3.2 PROPOSED MECHANICAL TESTING OF WELDED RAIL

In order to validate the process of friction stir welding, tests must be performed. These tests are as follows:

3.2.1 *TENSILE TESTING*

This test is performed by trying to pull two rails apart from each other by a known force magnitude. The magnitude of the force to be applied should be according to the manufacturing specification. The main aim of this test is to make sure that the rails do not pull apart when subjected to a high tensile force.

3.2.2 *FATIGUE AND FRACTURE MECHANICS TESTING*

When the train is in motion, the rail is subjected to a different load at any given time depending on several factors, such as operating condition, loading condition, weather condition and so on. Therefore, it becomes apparent that most rails joined by means of friction stir welding are subjected to fatigue testing. The load to be applied on the rail should depend on the manufacturing specification and the operating conditions.

3.2.3 *BENDING TESTING*

When the train is on the rail, the rail has a tendency to bend by certain millimeters. The bending movement imposed by the load on the rail has a allowable limit. Therefore, it is crucial that the welded rail be subjected to a bending test. This is to ensure that the weld does not deflect beyond the maximum prescribed limit.

3.2.4 *IMPACT TESTING*

When the train is in dynamic motion it has a tendency to pump due to its suspension system and the condition of the rail. If this aspect is not controlled, derailment due to rail breakage may occur. This will cost the rail company a fortune to repair. Consequently, it is vital to make sure that rails welded by the friction stir technique are tested. This will also aid in monitoring the overall weld quality.

3.3 CONCEPT DEVELOPMENT

This section introduces various concepts developed during the design process. The objective of this section is to provide the reader with an overview of how the concepts were developed. Figure 3.1 shows the hand operated tool across the rail.

3.3.1 *VERTICAL OPERATED STIR TOOL*

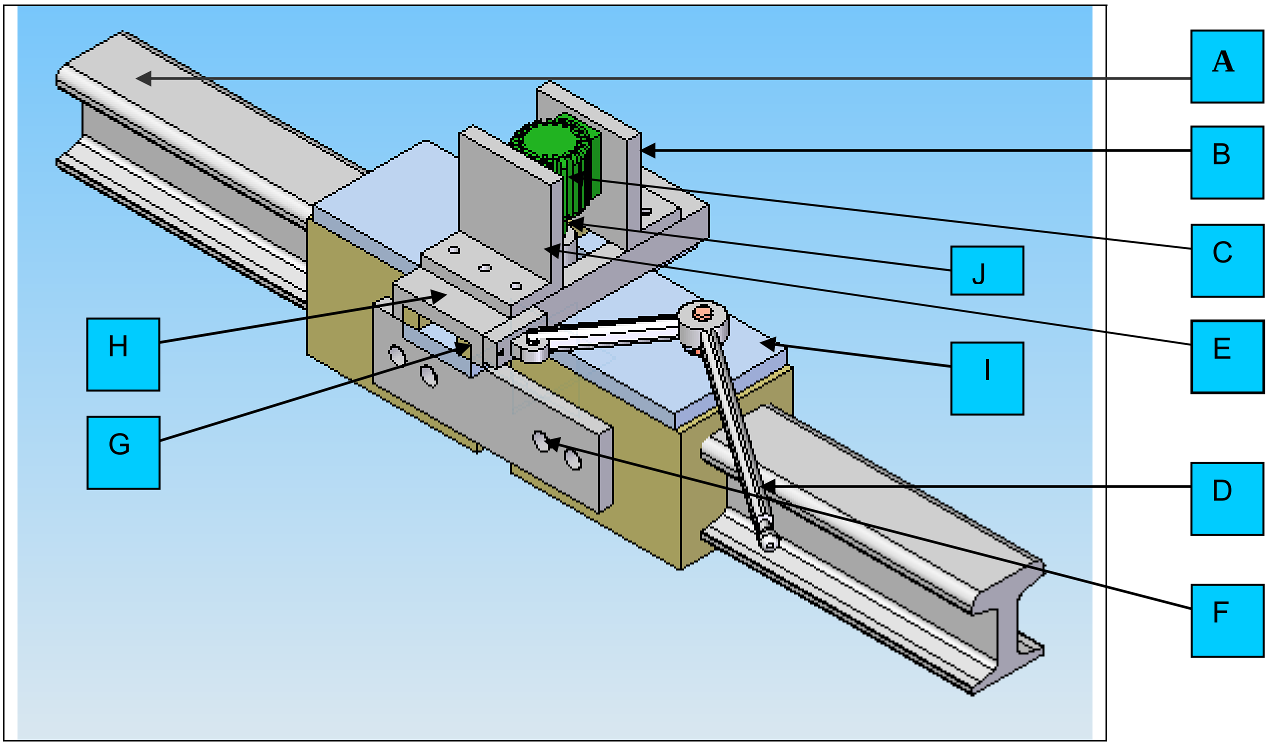


Figure 3.1: Hand powered tool across the rail

3.3.1.1 COMPONENTS NAME AND DESCRIPTIONS

A – Rail type used in one of the company rail lines (48 kg/m)

B – Motor side supports (left)

C – Electric motor

D – Hand lever

E - Motor side supports (right)

F – Side supports

G – Sliding bar

H – Motor base

I – Structure support

J – Tool pin

3.3.1.2 CONCEPT ASSESSMENT AND DECISION

Figure 3.1 shows the first concept developed. It can be seen from the figure that the friction stir welding machine for joining rail is to be hand operated. The force required to get the acceptable weld on the rail depends on the types of rail and the metallurgical properties of the rail. In order to apply the acceptable required force two or three persons is needed to operate or apply the force by means of an arm lever. This will cause many problems in the operation of the machine, including human error and also the irregularities of the weld. It is natural that people of different ages and ethic groups have different strengths. Consequently, people will tend to apply a different welding force. Welding force is a major factor in friction welding and as a result, welding force has a major impact on the quality of weld. As a result this concept will not be able to yield the required welding results and does not meet a basic friction stir welding machine's specification requirement.

The side support blocks which are used to align and clamp the rail are only made for a special rail type. This will definitely limit the flexibility of the design. One of the main requirements of this design (machine – friction stir welding machine) is to make sure that the side supports have an adjustable height of 127 mm to 165 mm in order to accommodate various rail heights. Four steel blocks are also used to make sure that the side supports do not fall apart during welding operations. Four steel blocks labelled F in Figure 3.1 have the capacity to fall off in operation. This method is not very reliable and it could cause injury to the operator. Based on these reasons, this concept was abandoned.

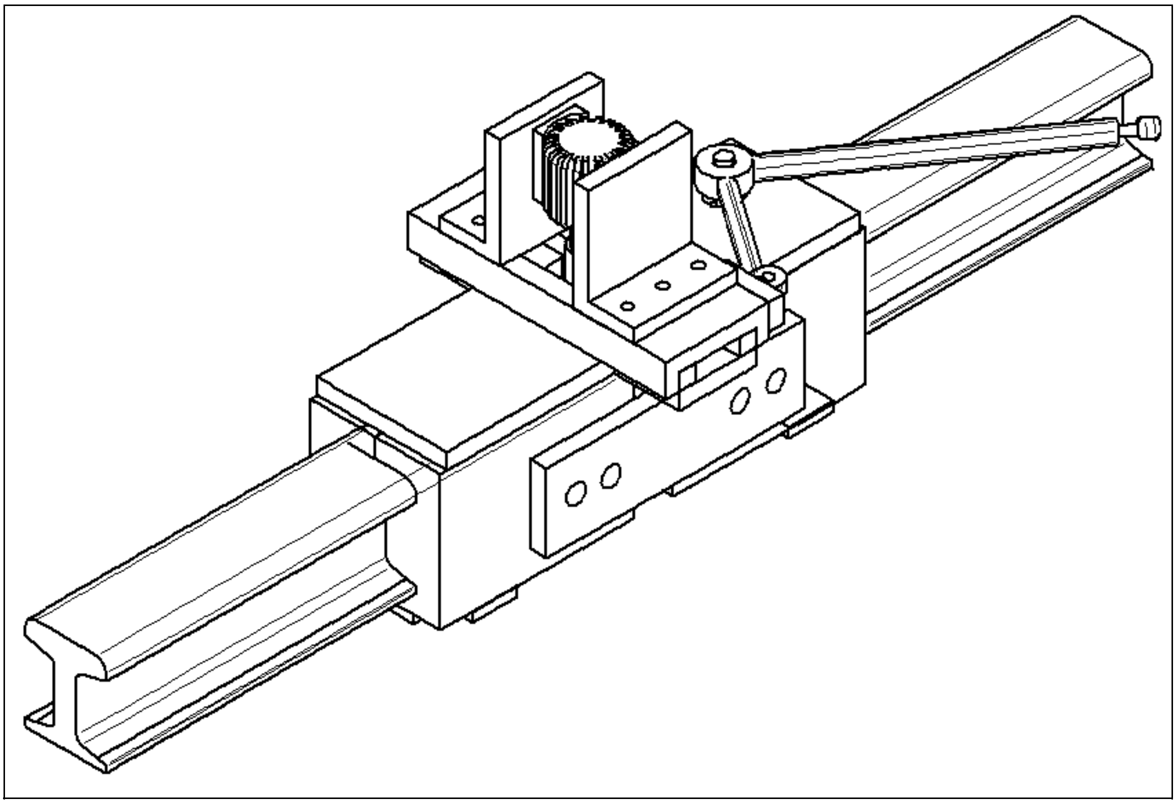


Figure 3.2: Friction stir welding machine applied in rail joining-concept 1

3.3.2 *HORIZONTAL OPERATED CUTTING TOOL*

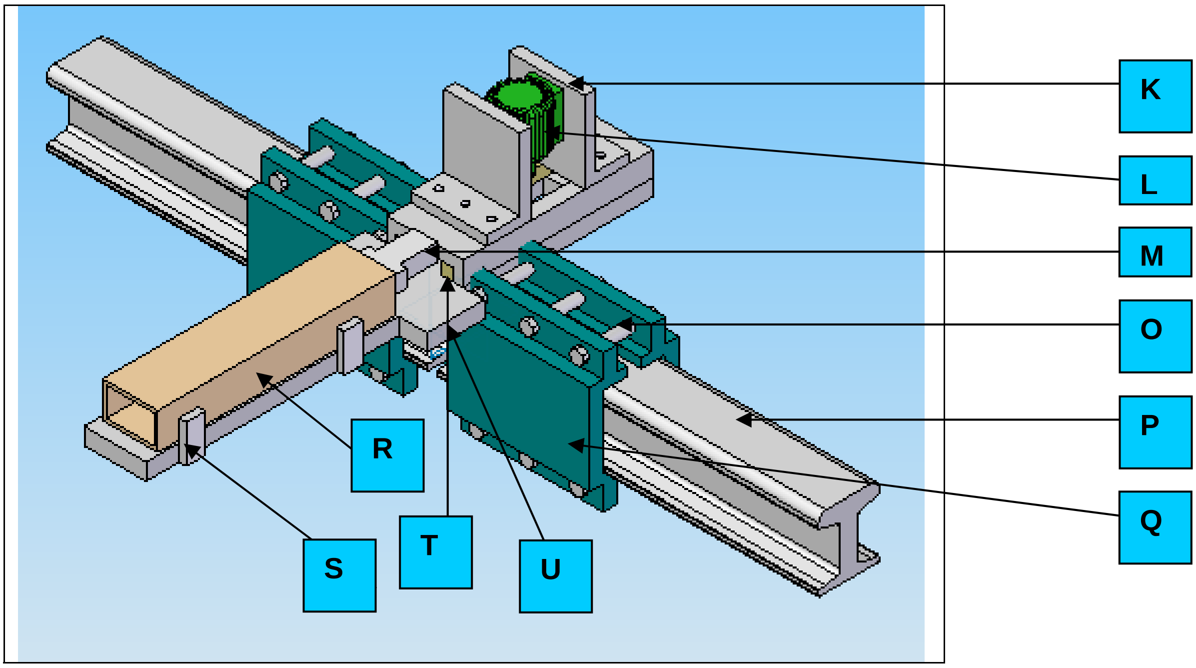


Figure 3.3: Torsional spring powered tool across the rail

K - Motor side supports (left)

L – Electric motor

M – Sliding bar (rectangular shape)

O – Tightening nuts

P – Rail type

Q – Rail clamp

R – Sliding bar holder (with torsional spring inside)

S – Sliding bar holder supports

T – Transitional bar

U – Structure support

Figure 3.3 shows improvements in terms of operational requirements and the number of machine components required. The rail clamp is used to align the rail and any rail profile could be used in the Railway Company. The rail clamp (labelled Q in Figure 3.3) does not depend on the profile of a particular rail in the line but only on the height size of the rail. The rail clamp can easily be clamped onto the rail using bolts and nuts. In fact only one operator could perform the task without any difficulty.

In this design concept, the welding force exerted on the rail is applied by means of torsional spring. The spiral spring is inserted inside the steel rectangular bar whereas the rectangular bar is inserted inside the steel bar. Depending on the properties or the design of torsional spring, enough force could be obtained to push against the motor base. The steel bar side support is used to fix the steel bar and also to ensure that transitional movement is constrained. The motor support is still the same as in the previous concept. It is assumed that the horizontal movement of the rail due to the force experienced by the movement of the spinning tool is constrained by making sure that the side block supports are fixed and tightened to a certain specific torque.

Although the concept meets the majority of product requirements specification factor, the side supports need to be changed so that they are able to accommodate various rail profiles. Another factor which hinders this concept is that torsional springs are not widely used, expensive and have limited operational life as compared to other spring types. This could increase the product cost by a significant amount. Therefore, this design concept was abandoned. In order for the product to meet the product requirement specification, the following modifications were done. The modification is shown in Figure 3.4.

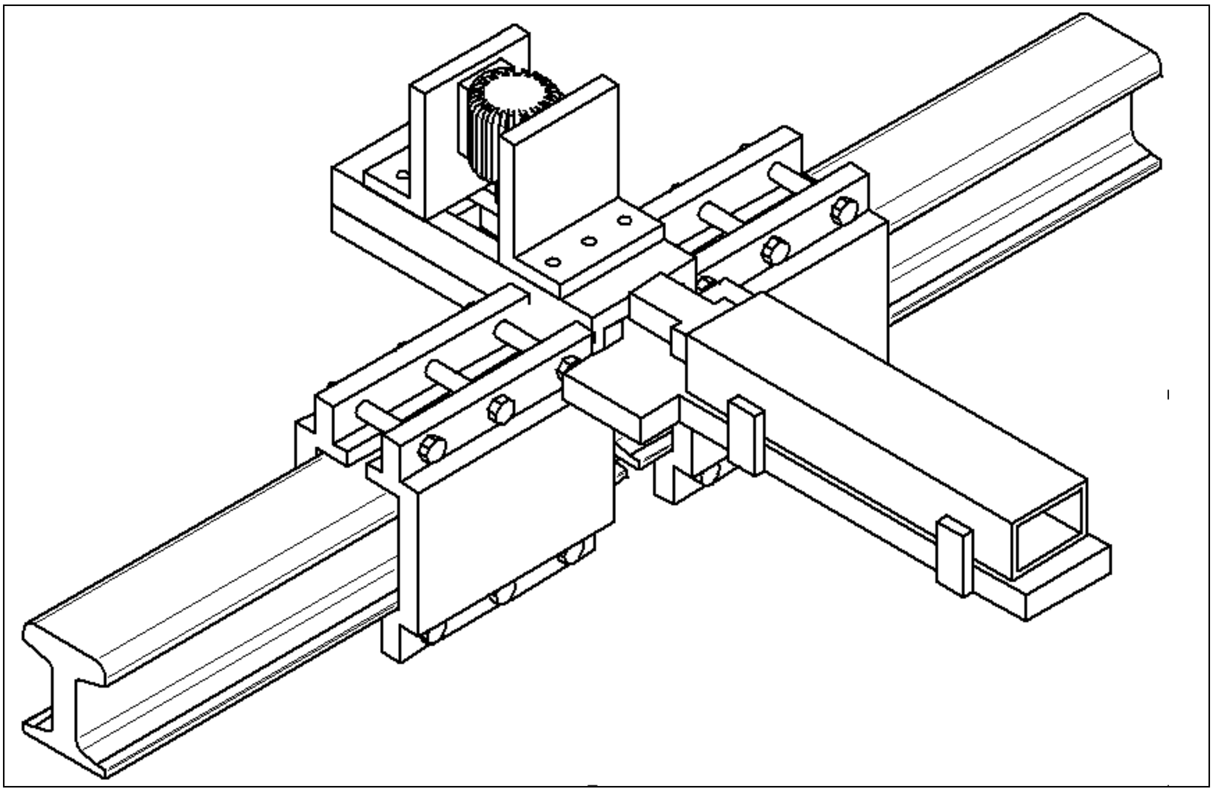


Figure 3.4: Friction stir welding machine applied in rail joining-concept 2

3.3.3 *HYDRAULLIC POWERED TOOL ACROSS THE RAIL*

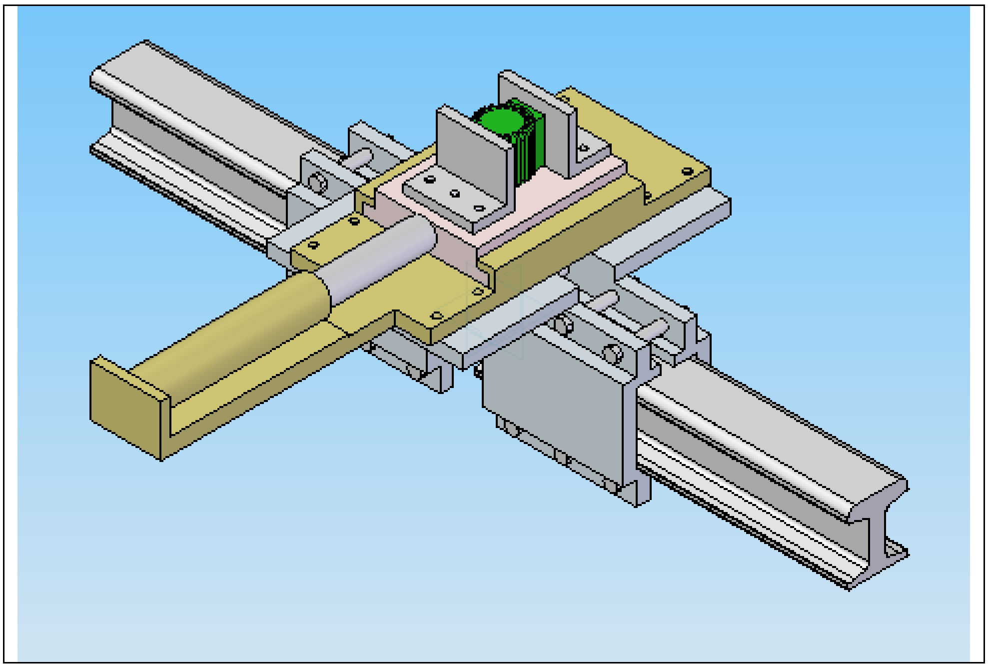


Figure 3.5: Hydraulic powered tool across the rail

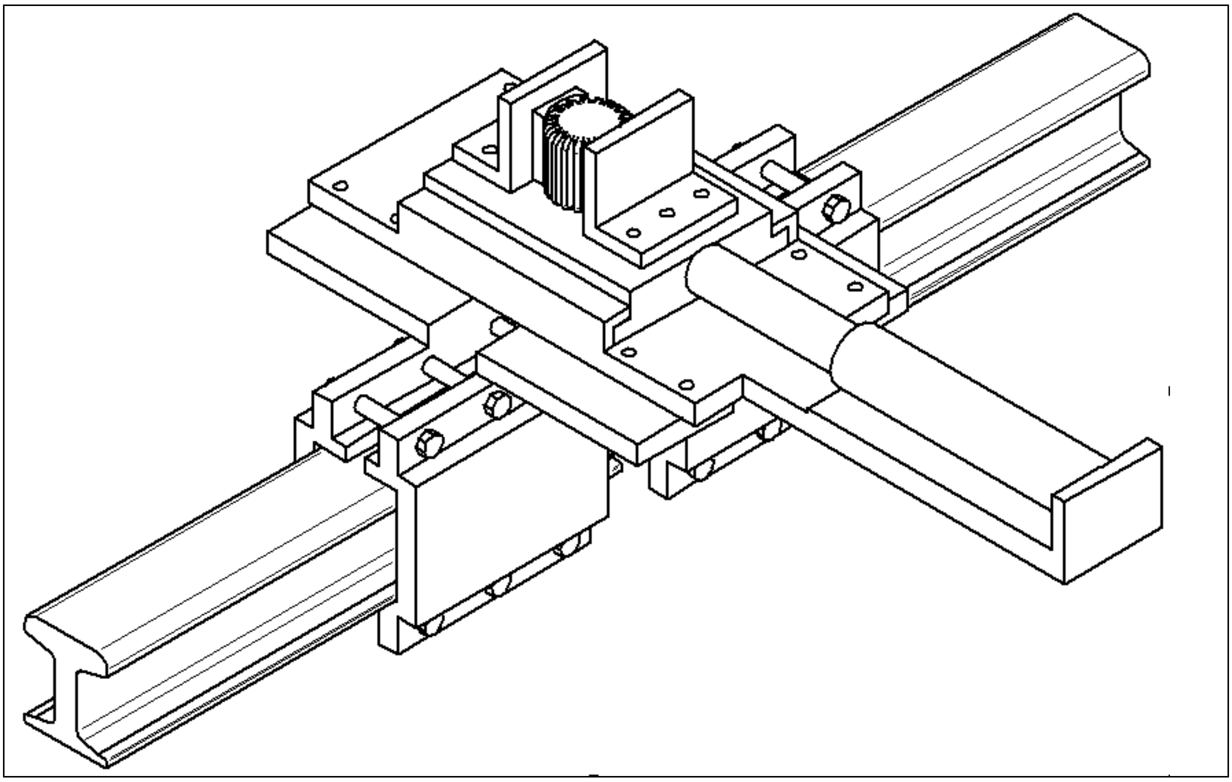


Figure 3.6: Hydraulic powered tool across the rail (sketch)

Figure 3.5 and 3.6 show the third concept which is not exactly the same as the second concept shown in Figure 3.4. The major difference between the second and third concepts is that the force to be applied on the working piece is being applied by the helical spring. Note that torsional springs are used in the second design concept.

Another major difference between the second and third concepts is that the third concept was designed in such a way that the force actuator is fixed in all direction. In other words the sliding chunk (which holds the motor bar together) is constrained in a radial direction. This is to make sure that the sliding chunk does not move in any unwanted direction during the operation.

3.3.4 *ADJUSTABLE RAIL CLAMP*

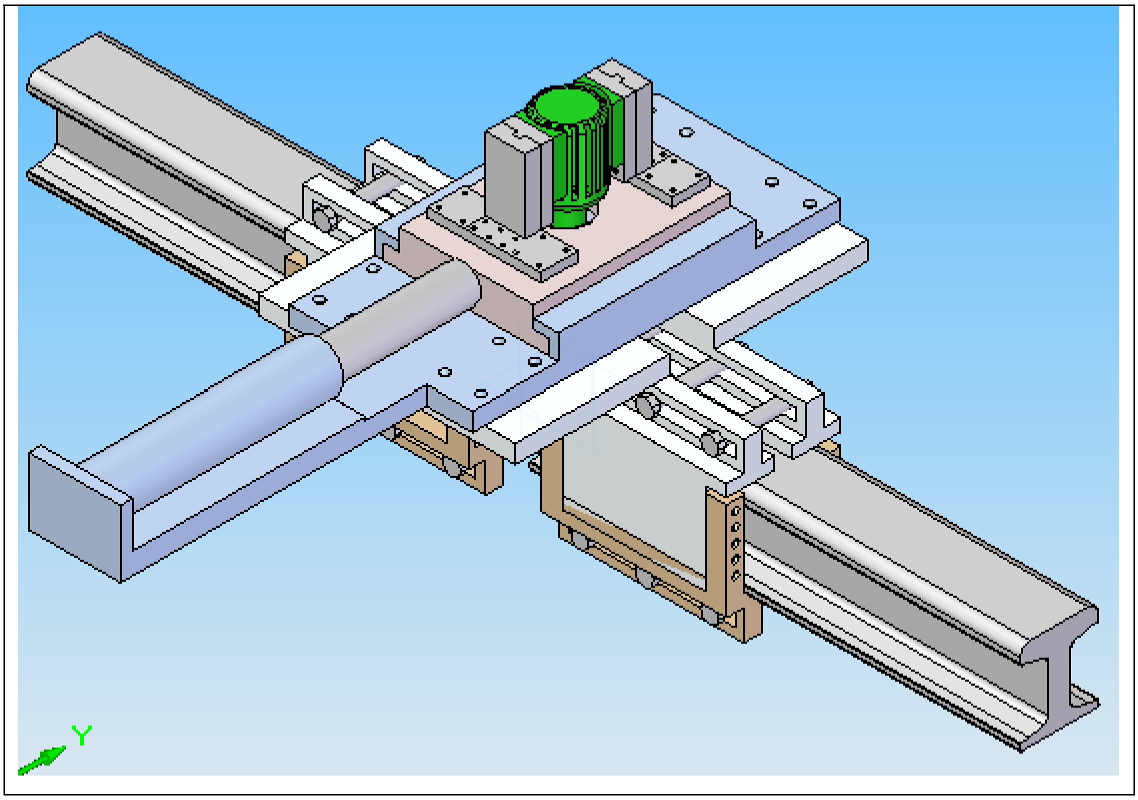


Figure 3.7: Adjustable rail clamp

The design concept model shown in Figure 3.7 shows the adjustable rail clamp. This has a huge advantage on the design model because it is able to hold various rail profiles countered in service. Therefore, if the rail clamp has the adjustable height all or most of rail profile or rail types could be friction stir welded by using a single rail clamp. In this concept, the motor has the ability to be adjusted vertically. The machine operator can change the height of the motor, depending on the height of tool pin to be used.

As shown in Figure 3.7, the rail camp has a cut-out through out. The operator does not have to align holes in order to clamp or grab the rail. The disadvantage of the mentioned above feature is that the friction stir welding of rails requires a “perfect” alignment of rails and as a result of the above, misalignment could occur. In order to maximise the height adjustment of the rail clamp, instead of a hole cut-out, simple flat holes are used on the outer or top rail clamp piece.

Another important feature which is required in the conceptual design of friction stir welding machine is the vibration isolator. The concept shown in Figure 3.7 does not have the vibration isolator and as a result it also was abandoned. The advantage of a vibration isolator is that the vibrations due to motor rotation on the top structure do not affect the bottom structure of the machine.

3.3.5 *ADJUSTABLE RAIL CLAMP WITH POWER SCREW*

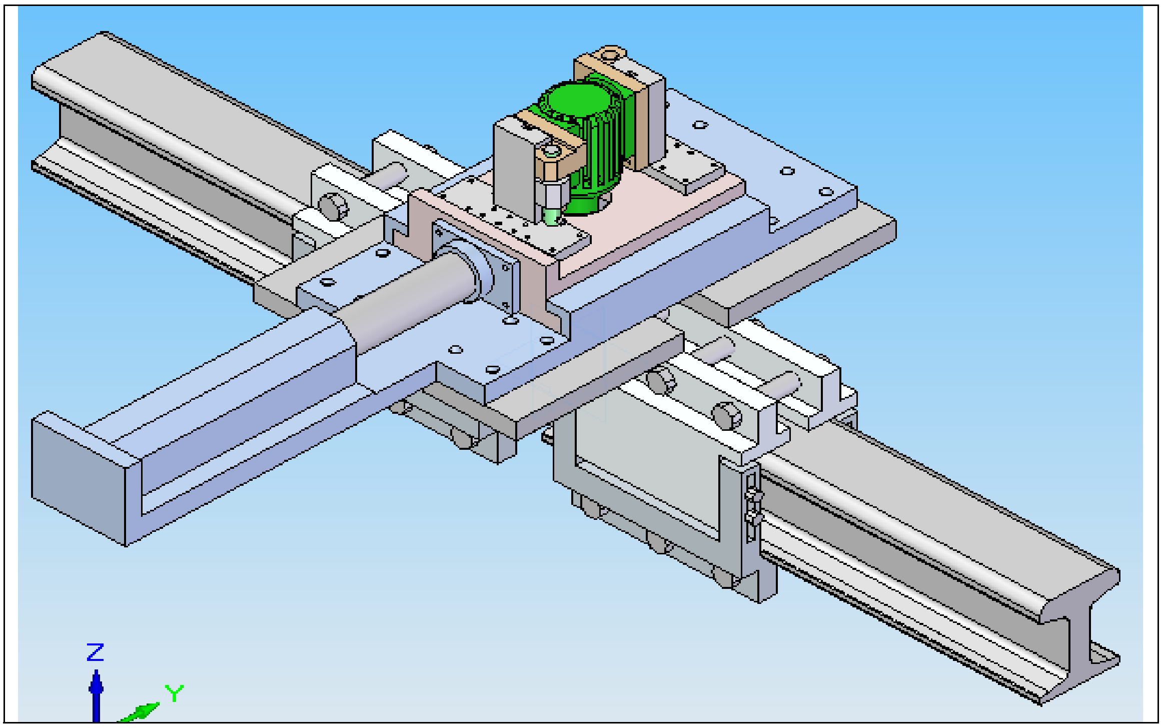


Figure 3.8: Adjustable rail clamp with power screw

The adjustable rail clamp model in Figure 3.8 shows a significant improvement because the bottom part of the rail clamp can be easily adjusted without any restriction. This means that there are no specific holes on the bottom part of the clamp. It can be seen from Figure 3.8 that the bottom part can move vertically up and down without any restriction. An adjustable height of between 127 mm and 155 mm can be achieved easily by this feature.

3.3.6 *ADJUSTABLE RAIL CLAMP WITH VIBRATION SEPERATOR*

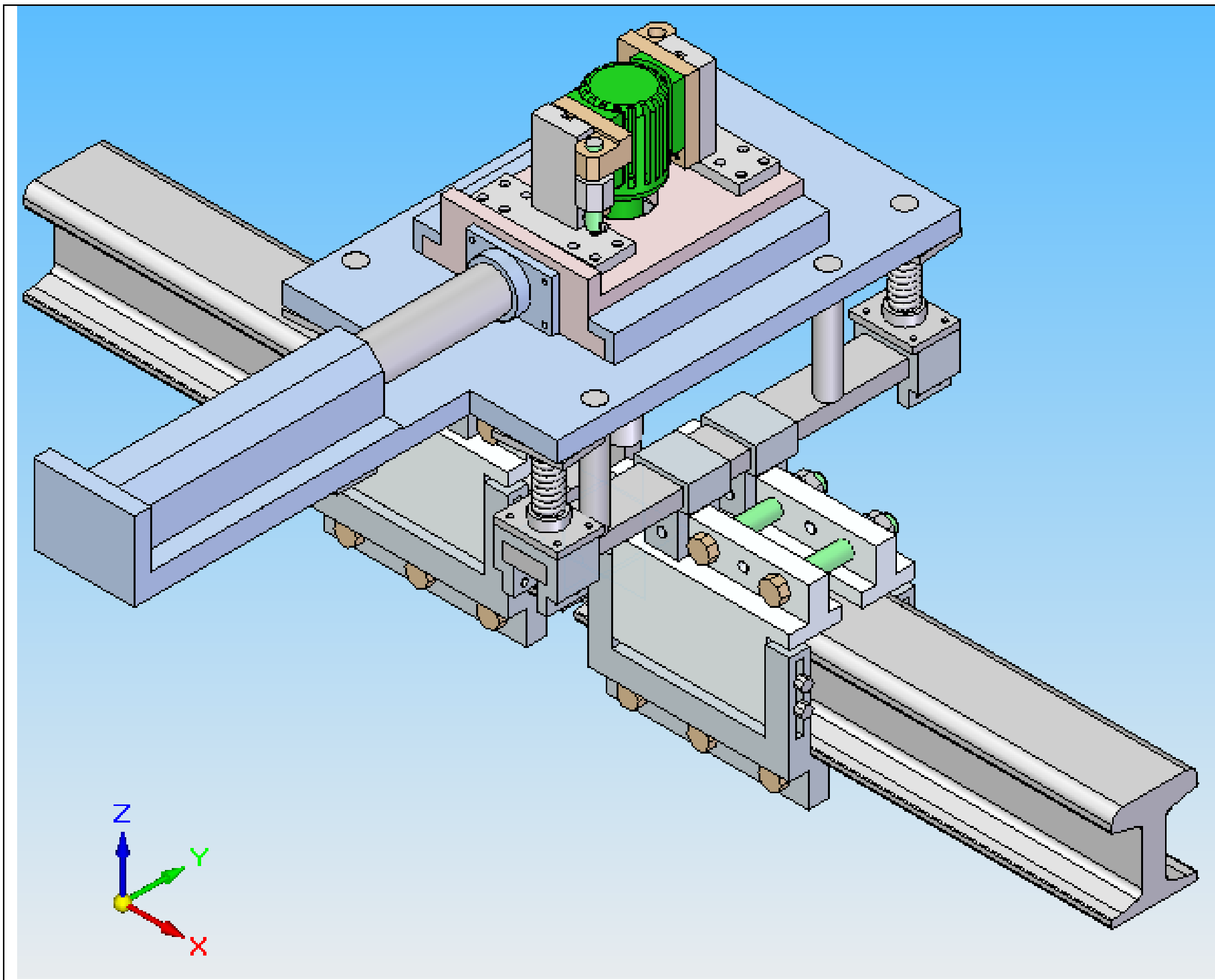


Figure 3.9: Adjustable rail clamp with vibration isolator

Figure 3.9 shows the final design concept of a friction stir welding machine for joining rails. The design concept shown in this figure meets most of the conceptual design requirements tabulated in Table 3.1. The design concept has been updated to accommodate the vibration isolator with a sliding guide. The primary function of the sliding guide is to ensure that the top structure does not off-set relative to the bottom structure. Moreover, its primary function is to align the top and bottom structure of the machine. Therefore, the design concept shown in Figure 3.9 was chosen as the final design concept. However, this design concept is modified to reflect the required design concept.

4 DESIGN SPECIFICATION

This section depicts the overall description of the proposed design concept, , performance capabilities, cost analysis and the general arrangement of the device. The device has been designed in such a way that it could accommodate various types of rails used in South African railway lines. At the moment in South Africa more than twelve rail types are being used, therefore, rail clamps must be designed in such a way that at least three rails are accommodated by one rail clamp. The strength and size of the rotating pin to be used depend on a number of factors such as thickness, material, working conditions of the working piece etc.

4.1 OVERALL DESCRIPTION OF THE PROPOSED DEVICE

The structure of the friction stir welding machine for joining rails is composed of two main subassemblies, namely; the bottom and upper structures. The upper structure consists of the driving unit which comprises of the motor and a gear. The motor selected has a built in gear which can be removed in order to change the speed of the rotating pin. In this arrangement different gear sets can be used in order to achieve the desired speed. Another unit on the top structure of the machine is the electro-hydraulic device. The electro-hydraulic device is capable of applying a force of 220kN at a maximum speed of 2000 mm/sec. The top structure main electric motor is selected and it has the maximum angular speed of 2950 rpm and an applied torque of approximately 60 N.m.

In Figure 4.1, there are four helical springs used to reduce vibration from the top structure to the bottom structure. The bottom structure as shown in Figure 4.2 requires a minimum vibration in order to maximise weld quality. Also, helical springs are designed in such a way that they have high lateral stiffness. The higher the lateral stiffness the lower the chances of lateral movement of the structure due to the weld force applied by the electric-hydraulic actuator. The extension shaft joining the main electric motor and the rotating pin has been designed to accommodate both the torsional stress and the bending stress. Figure 4.3 shows different views of the top structure. The extension has the supported bar across the plate length that aims to make sure that the weld force (pressure) is applied on the pin or shaft. The shaft does not attain any significant movement. Significant movement on the extension shaft could have an impact on the weld quality; consequently the movement must be minimal.

The pin has the ability to rotate in a specific direction and it can also translate in both directions. The main electric motor is on top of the small plate which has the ability to slide along the direction of the electro-hydraulic actuator and the electric motor is fixed onto the sliding plate. The sliding plate is connected to an electro-hydraulic actuator by means of a joint as shown in Figure 4.3. For the purpose of alignment and other factors required to obtain a quality weld, a special bracket has been designed to accommodate any misalignment of the rotating pin. The bracket has the ability to rotate up to 30 degrees without any obstruction from any other machine component.

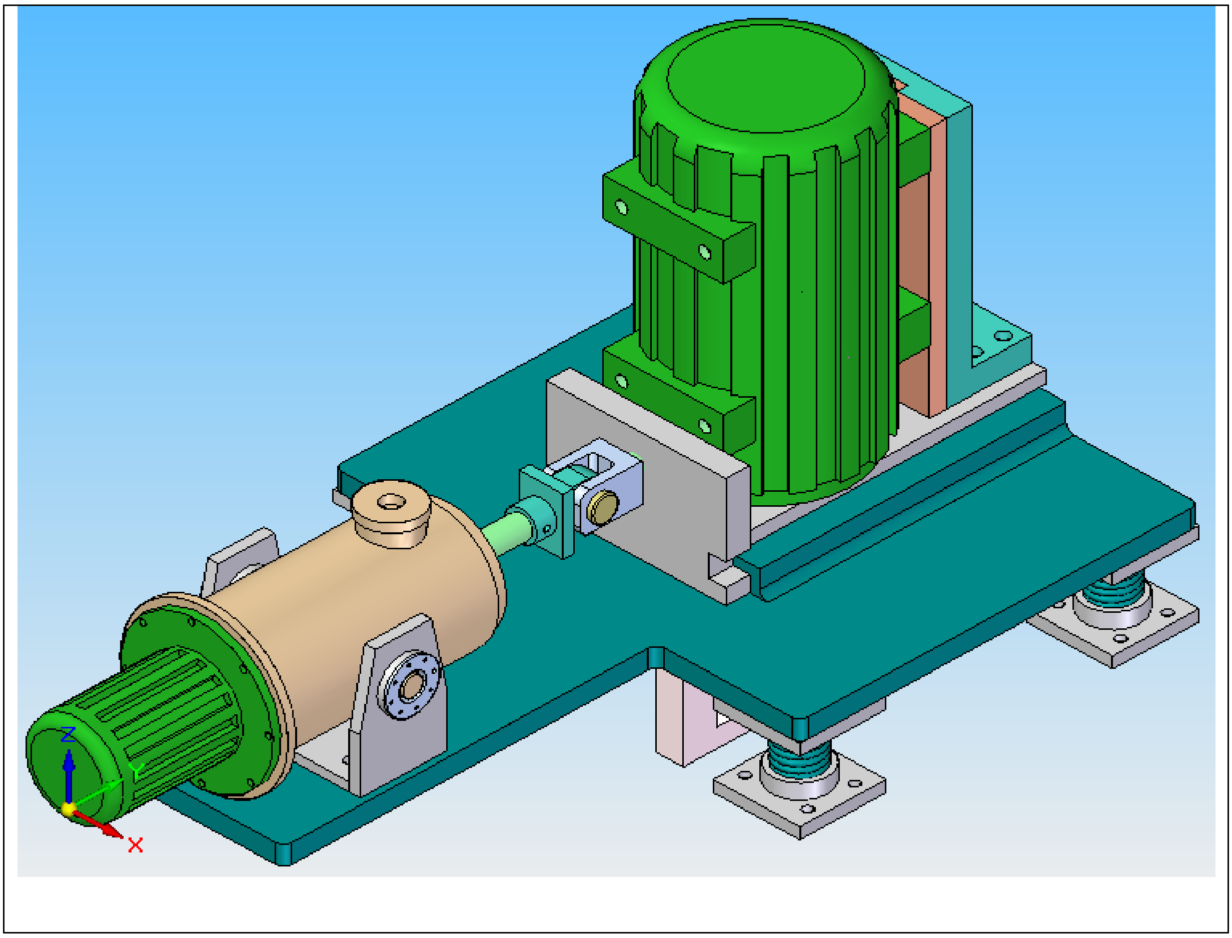


Figure 4.1: Top structure of friction stir welding machine for joining rails

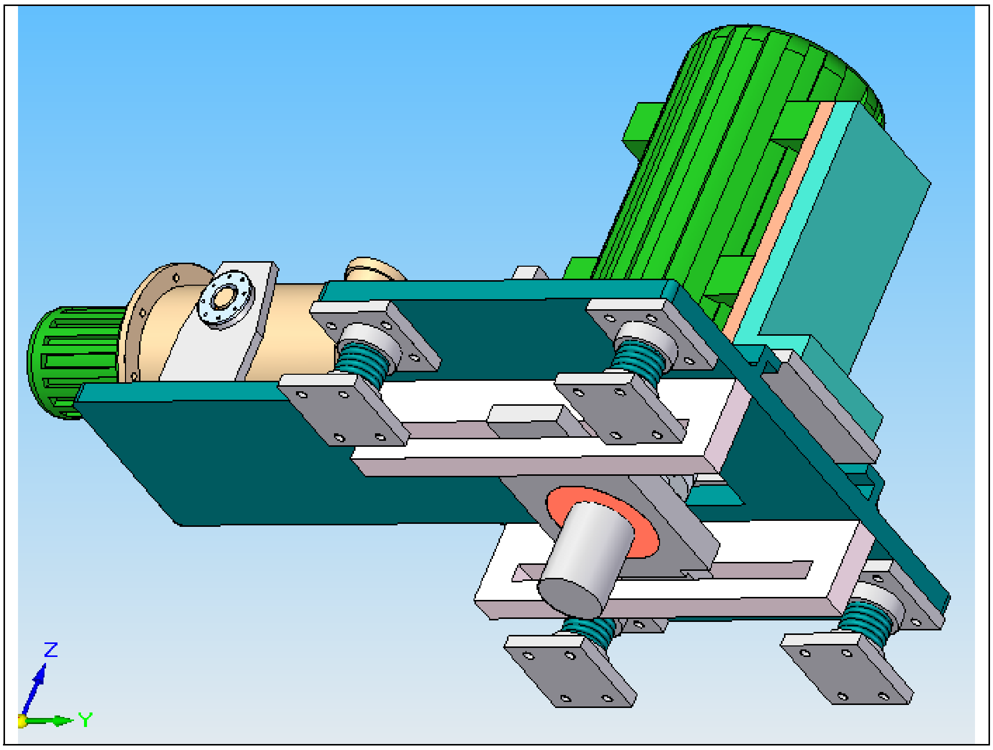


Figure 4.2: Bottom (different) view of top structure

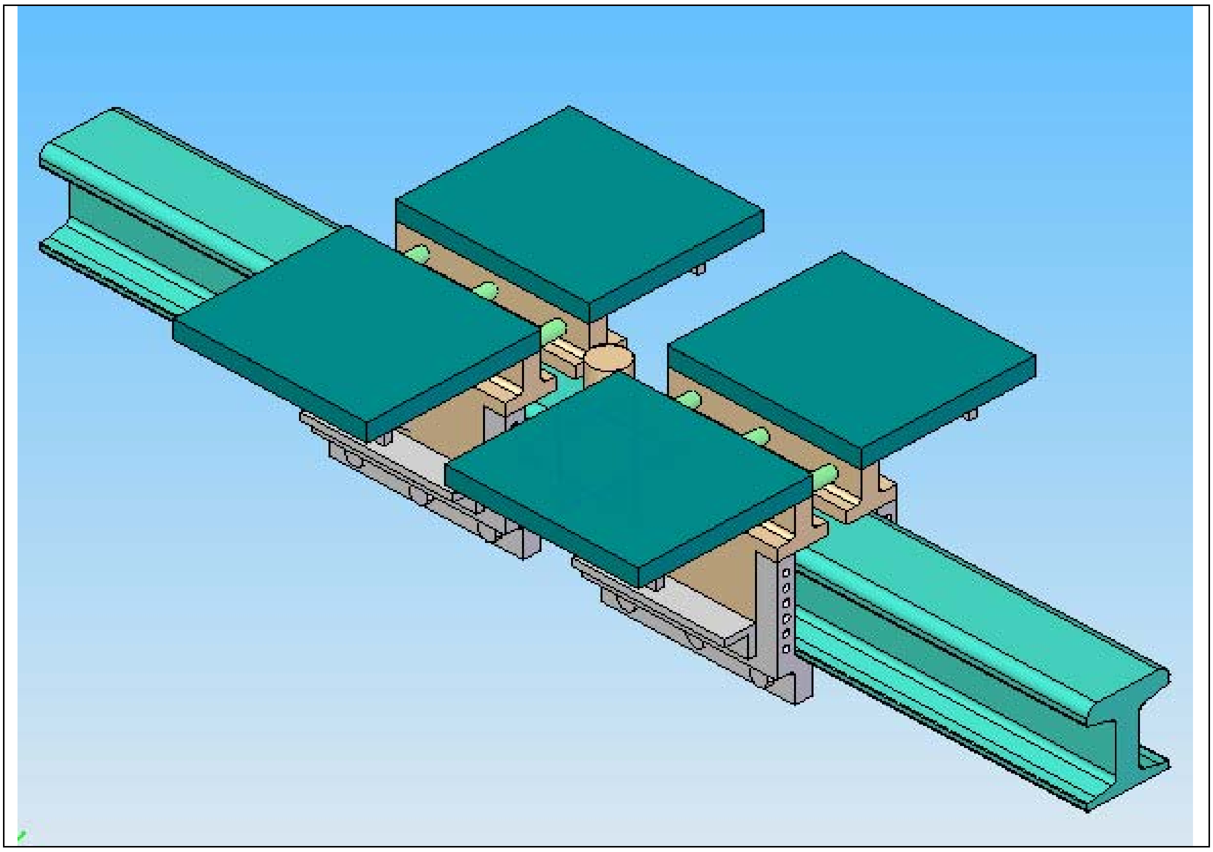


Figure 4.3: Bottom structure of a friction stir welding machine for joining rails

Figure 4.3 shows the bottom structure of a friction stir welding machine that aims to hold tight the two rails to be welded together and also to support the upper structure. The two rails to be welded must be held tight by two pieces of specially designed steel plates (referred to as “rail clamp”). Three bolts and nuts are used to keep the two steel plates together. Based on engineering calculations the number of bolts required to keep the two rails together for each rail clamp set is six, that is three at the bottom and three on top. Figure 4.2 shows a different view of upper structure.

In practice, the rails to be welded are of different heights and sizes, therefore the rail clamps have been designed in such a way that the height could be adjustable. The bottom part of the rail clamp has seven holes used for height adjustment. The rail clamp must be as rigid as possible in order to increase its strength in holding the two rails together.

The rail clamps were designed in such a way that their height (rail clamp height) is adjustable. The rail clamp height is adjusted by means of a bolt and nut. The rail clamps have a minimum height of 127 mm and the maximum height of 157 mm. To ensure that one rail clamp is capable of friction welding two to three rails of different profiles or dimensions.

Figure 4.4 and 4.8 show the general arrangement of a friction stir welding machine for joining rails as described above. Figure 4.5, 4.6 and 4.7 show the side, front and right views respectively. Figure 4.8 is the same as Figure 4.4 except that it does not have the edge lines.

In order to validate that the concept of a friction stir welding machine for joining rails has the ability to work in real life, various engineering calculations have been done. Appendix B shows detailed calculations how friction stir welding machine will work in practice. In most cases, due to the unavailability of data various assumptions have been made in order to make engineering calculations and these assumptions have been clearly stated in each section of Appendix B. Appendix H shows all the tables and figures used in the engineering calculations found in Appendix B.

4.2 GENERAL ARRANGEMENT OF THE DEVICE

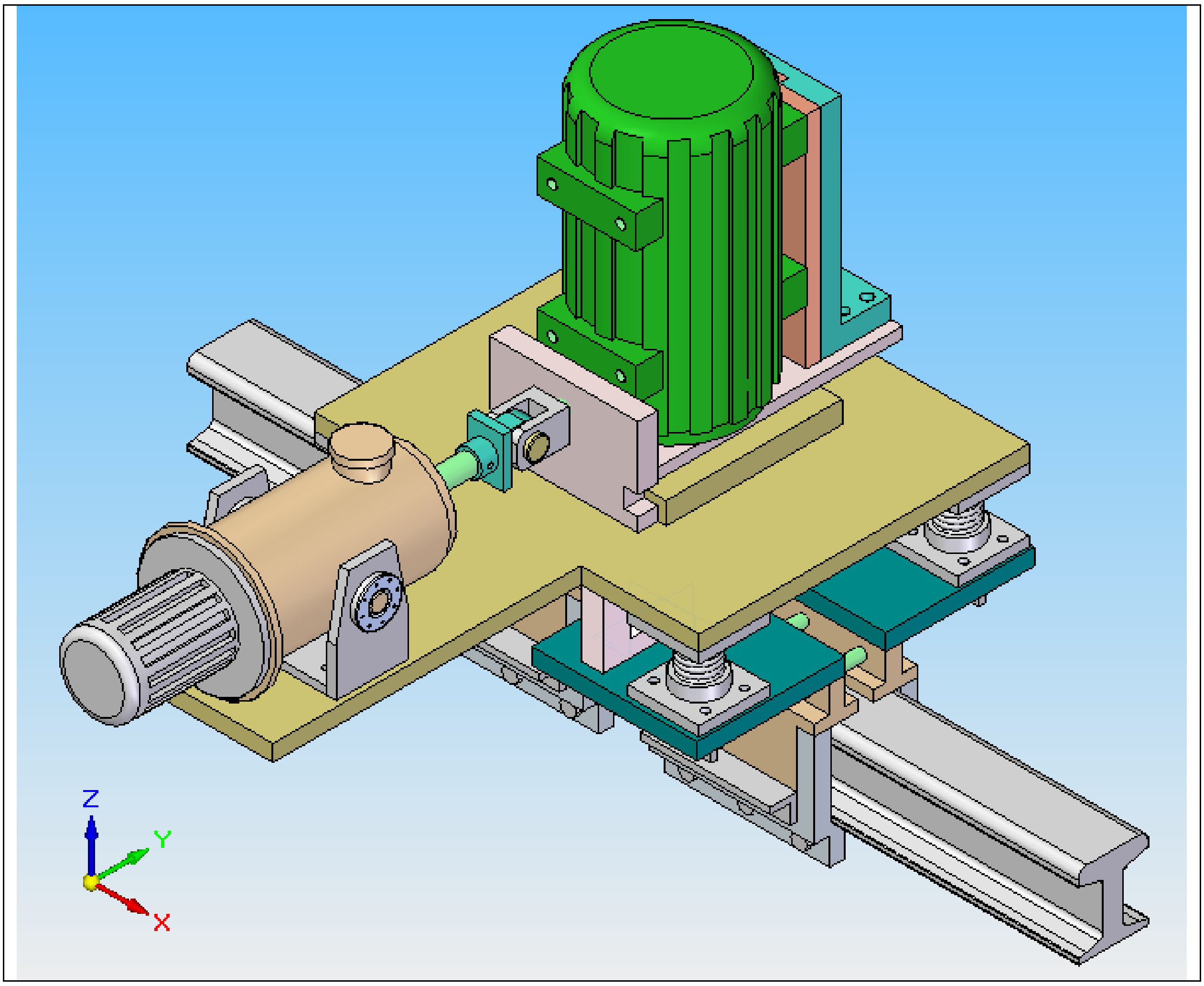


Figure 4.4: Friction stir welding machine used for joining rails

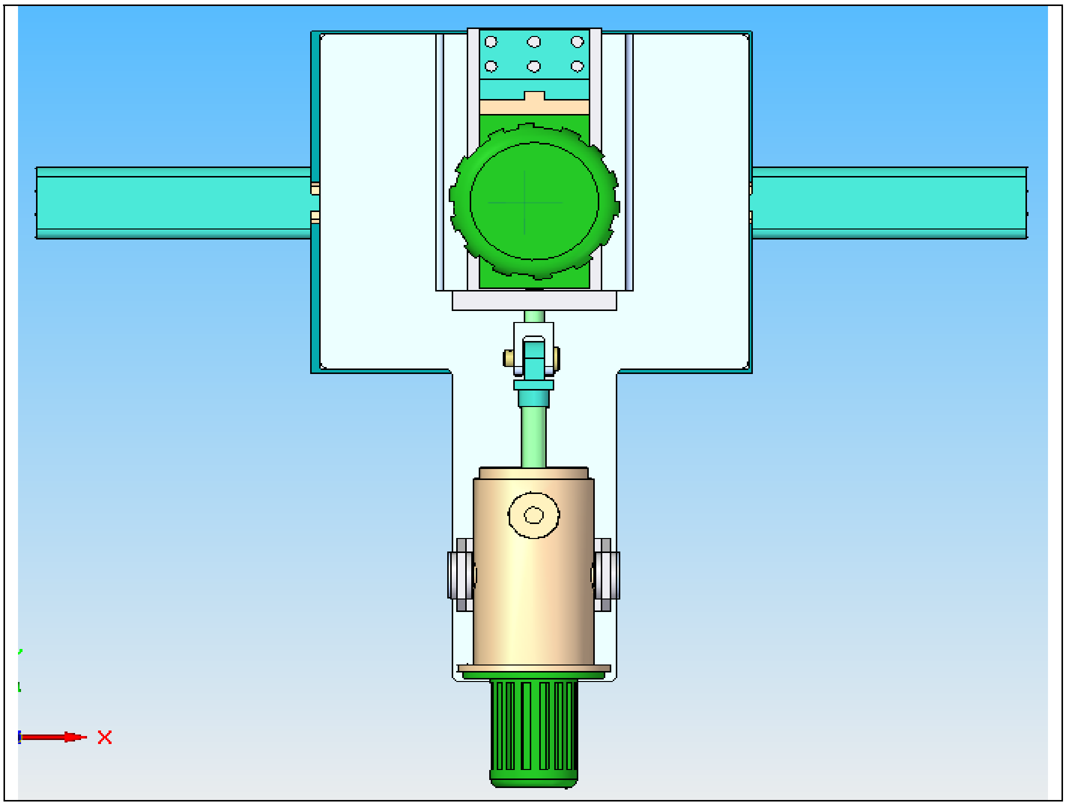


Figure 4.5: Side view of proposed friction stir welding machine

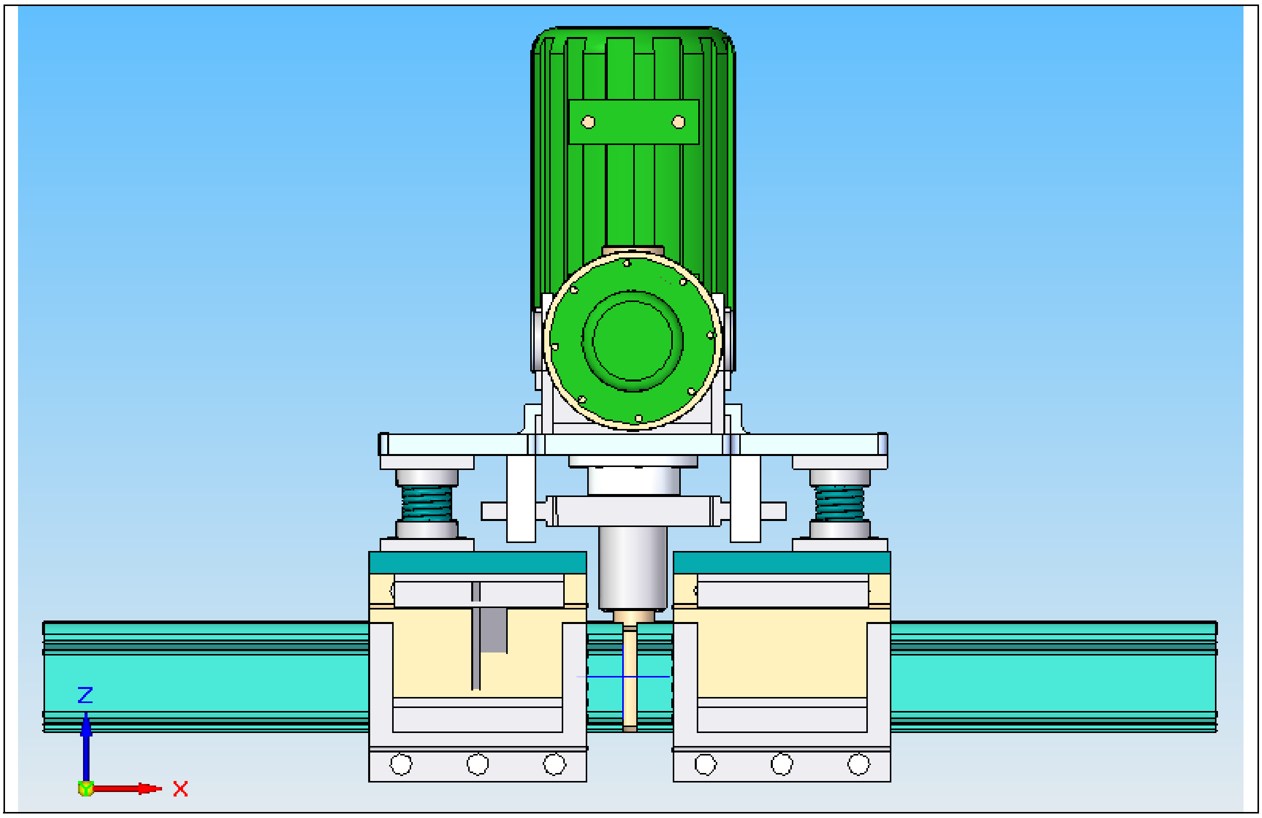


Figure 4.6: Front view of proposed friction stir welding machine

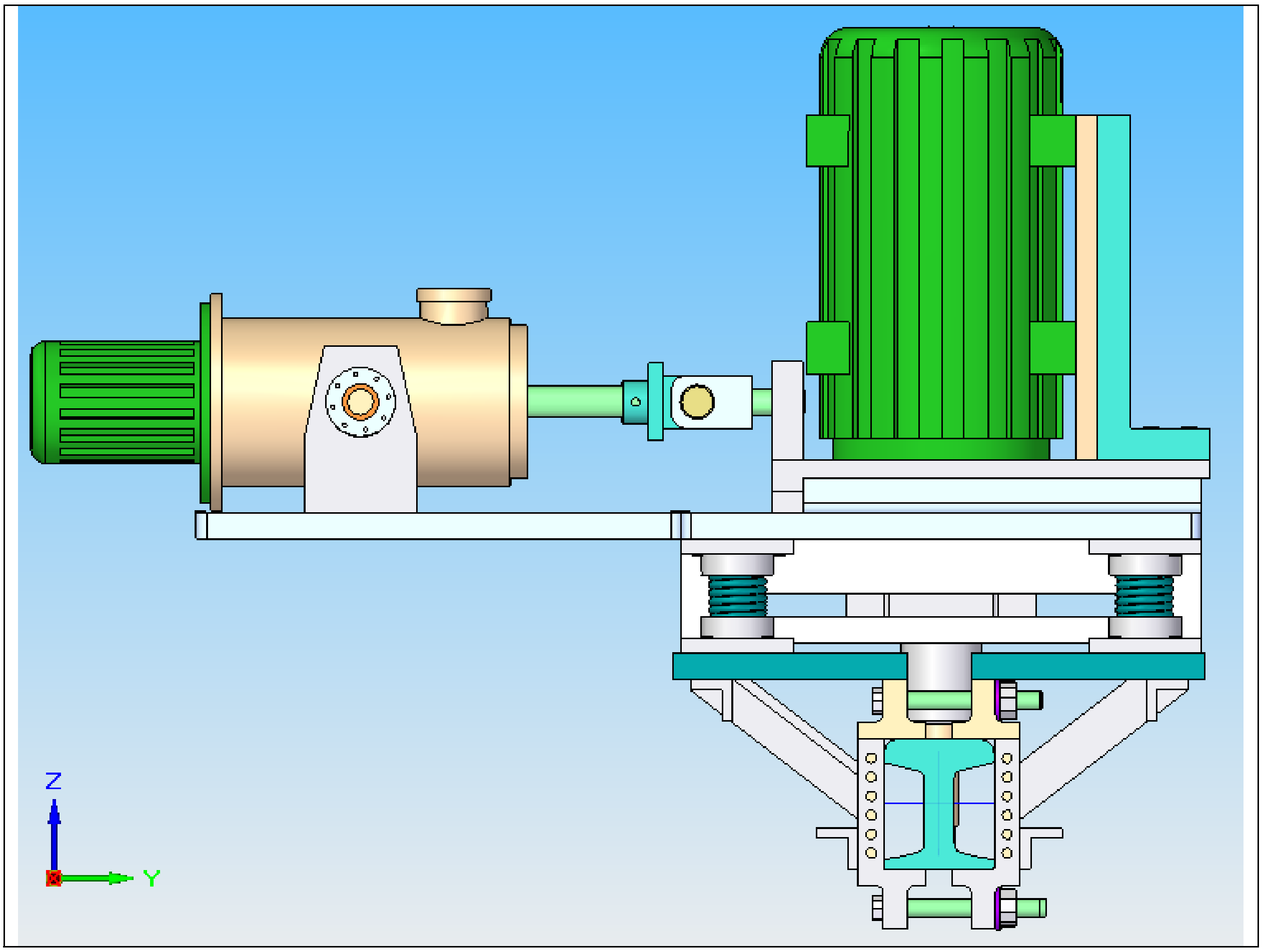


Figure 4.7: Right view of proposed friction stir welding machine

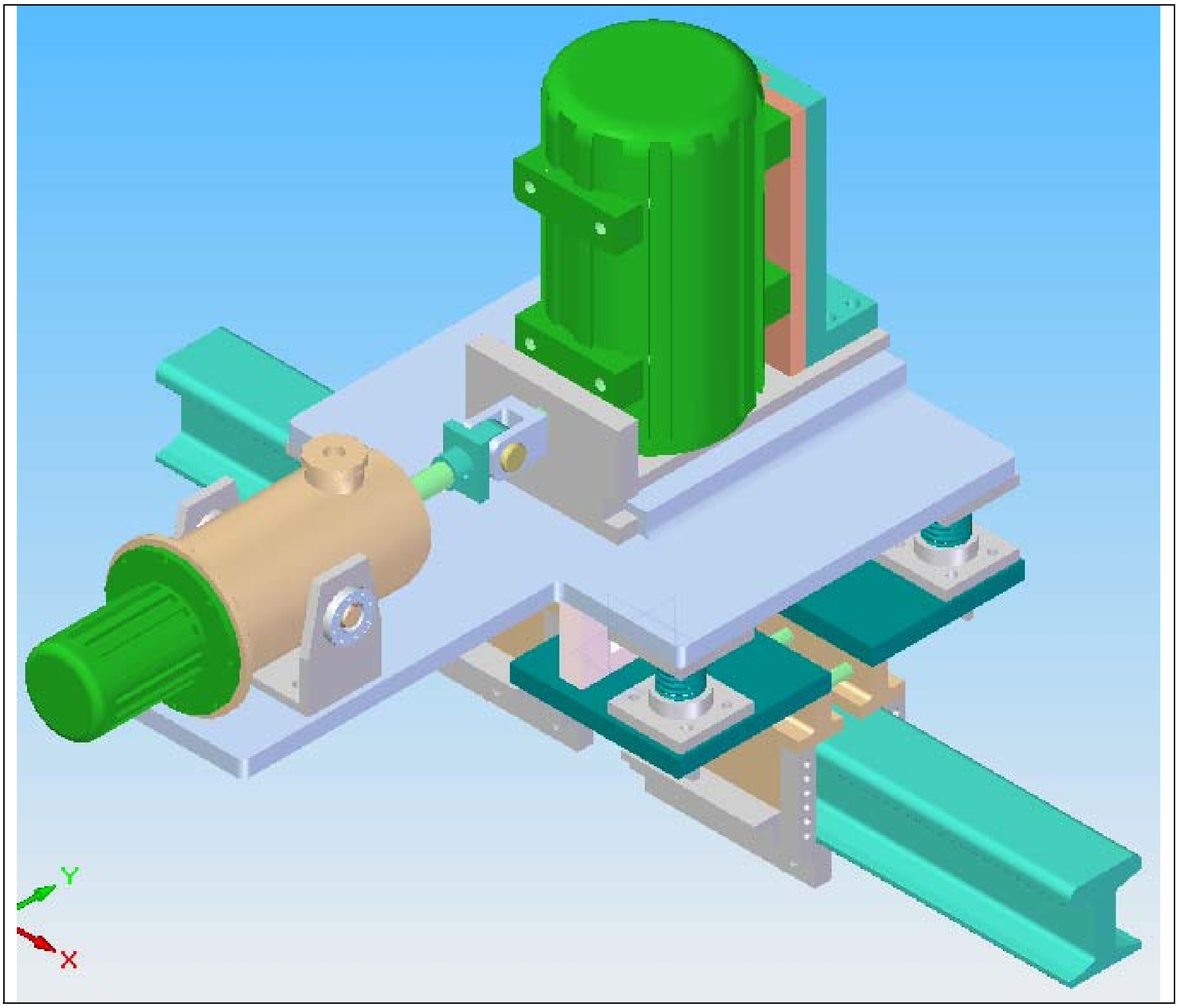


Figure 4.8: Final design of a friction stir welding machine for joining rail

4.3 PERFORMANCE CAPABILITY OF THE DEVICE

The designed friction stir welding machine for joining rails was designed. Table 4.1 compares the product requirement specification (PRS) and the actual designed device. YES means that the feature in the machine has been designed successfully.

Table 4.1: Comparison of PRS and the actual designed device

Factor	***Comments on actual design***
General form of design	
The product design concept must be operated by one or two persons. The person operating the machine needs not wear any safety clothes and the machine must be environmentally friendly. The machine set-up time on site must be very low with minimum separate components. The weight of the machine components must be capable of being carried by one or two persons from the store room to the welding site.	YES
Geometry	
The diameter and shape of the pin tool must in agreement with the tool pin design criteria.	YES
The motor must have an output diameter of between 50 mm and 96 mm. the output diameter of the motor is the shaft that is to be connected to the working pin. The optimal diameter of the working tool based on the geometry of the rail must be between 60 mm and 100 mm diameter, therefore, it is desired that the motor output diameter has a diameter which is equal to or less than that of the working tool.	YES
Side block aligners (rail clamps) must be of adjustable height to accommodate different rail types. The block must be adjusted from 127 mm to 165 mm as an absolute maximum. In practice rails are found not to be of the same size mainly due to the operating conditions that they are subjected to. Consequently, rails have different sizes; therefore, the rail clamp must have the ability to adjust height. Rails have different profiles; therefore, there is a need to have a rail clamp which can be used for several rail profiles.	YES
Kinematics	
The welding speed of the tool must be controllable. The motor must be installed with a speed controller to assist in	YES

experimentation of weld quality.	
The working tool pin must be changeable easily. A wide range of tool diameter must be accommodated. The tool diameter must vary from 60 mm to 100 mm diameter. This must be accommodated by using the coupler which has the variable diameter hole.	YES
The tool pin must be able to move in a transverse direction. The force applied in the traverse direction must also be controllable. Therefore, force control is required.	YES
Actuator which can be maintained cheaply is, user friendly and small geometry must be used to apply the required welding force on the working piece (rail)	YES
Vibration due to motor rotation must have a minimum impact on the machine structure. The noise due to tool pin rotation must also be minimized.	YES
A maximum rotational speed of 2940 rpm. The rotating speed is based on the type of material to be welded.	YES
Cost	
The friction stir welding machine must be able to be produced or manufactured locally using low cost manufacturing techniques.	YES
Energy	
The final concept of a friction stir welding machine must use maximum power of 35 kW. The power needed is the combination of power required by the main motor and the power required by the actuator. Based on the rotational speed required, the power and torque required by motor can be easily calculated.	YES
A normal 220 V source must be required for operation and the frequency of 50 Hz	YES
Life span	
The concept to be considered must be able to succeed the duration of a test and trial test.	
Ideally the friction stir welding machine must attain the maximum number of years of actual use in the field without any major	YES

service.	
Maintenance	
In order to reduce the machine set-up time, the friction stir welding machine should be assembled and dissembled easily. If possible non-power and simple mechanical tools must be used when assembling or disassembling.	YES
The friction stir welding machine should be easily maintained during field operation	YES
Safety	
General rules of safety are applied. The product must pose no danger to the operator.	YES
Materials	
Movable parts must be small and weight light so that only one person is able to carry the parts. The operator must not feel exhausted due to the heaviness of the machine after use.	YES
All material to be used in building the designed product must be of normal low, medium and high carbon steel except the tool pin which requires strong and special material.	YES
Low and high carbon steel and aluminium may be used to build machine components. Low and high carbon steel are very affordable and have the desired mechanical properties. In addition aluminium has the advantage of being light weight, which is also a concern in the new product.	YES
Operations	
Must be operated by a maximum of four operators for set-up and two operators or a single operator for the actual welding.	YES
Takes minimum possible time to complete one single weld	YES
Manufacturing and Assembly	
It must be assembled easily and locally.	YES
It must be able to be assembled in a very short time to reduce the overall time for performing the actual welding process.	YES
Installation	
It must be easily installed in the railway environment. No special	YES

tool is to be used for installation purposes.	
Ergonomics	
User friendly	YES
Appearance	
The whole structure should be well structured.	YES
Entire unit should be aesthetically pleasing	YES
Performance	
The tool pin must be able to rotate at the desired speed all the time	
The maximum speed of the machine must be marked accurately and precisely in a visible area to operators	YES
Place or area of operation	
The designed product must be flexible enough to be used in straight rail lines and open places. The unit must be able to friction stir weld the following rail types: (i) 43 kg/m; (ii) 48 kg/m; (iii) 51 kg/m and (iv) 60 kg/m.	YES, only 43, 48 and 50 kg/m

4.4 COST ANALYSIS

A cost analysis of friction stir welding versus traditional arc welding and other processes used for rail joining has been concluded. Although it has been proven that friction stir welding has a greater cost benefit than traditional arc welding, sometimes it may not be beneficial to use friction stir welding. For example, in the automotive industry for medium to high production friction stir welding (FSW) costs 20% high than the arc welds (Colligan, 2001). Therefore, it is essential to conduct a cost analysis on a friction stir welding machine for joining rails as compared to the other methods currently being used for joining rails.

Rail welding costs are not available from the users and the manufacturers. However, some information could lead to a fair conclusion on cost comparison. According to More (1993) Charles A. Totten, has compiled number of reports on rail maintenance and the results are summarised below for further discussion on rail welding costs.

Bethlehem's Burns Harbor Div. which has worked for many years in the field of rail maintenance has observed for two years the costs of rail welding for different methods used. The processes that Bethlehem's Burns Harbor Div. observed are: flash-butt welding, arc welding, bolted joint and exothermic welding.

Burns Harbor has used the flash butt weld joint since the beginning of 1987 in rerailing 39 runways. More than 14 of the 20 miles of crane rail in the plant have been replaced. It was noted that 909 flash butt weld joints has been made successfully without any failures. The wheel of diameter of between 24 and 27 inches is used in these special cranes, most of which have at least 120 000 lb wheel loads. Table 4.2 (Charles, 1995) summarises the founding by Bethlehem's Burns Harbor Div.

Table 4.2: Statistics of rail joint

Joint Type	**Assembly time/Joint**	**Cost/Joint, $**	**Failures in 2-year period**	**Maintenance Cost/Joint (1st year), $**	**Rerail man-hr/linear ft**
Bolted	2 hr	430	95	1000	1.5
Manual Welded (Arc weld)	10-12 hr	2000	16	750	2.0
Exothermic	4 hr	620	8	240	1.7
Flash Butt	3 min	633	0	0	0.5

Table 4.2 shows that rail joining by means of bolts and a fisher plate takes a longer time than all the other three methods because there is no formal or standardised clamp used for aligning the rails to be joined and some hand tools have to be used. Highly skilled personnel are also needed to perform the job.

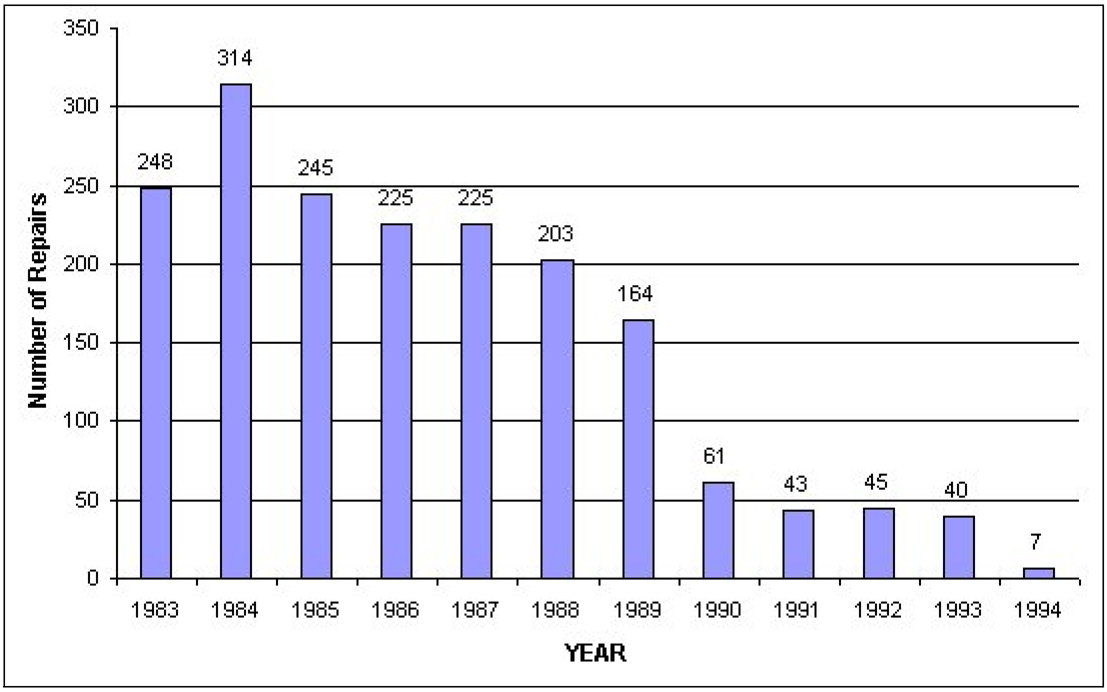


Figure 4.9: Rail joint repairs from 1983 to 1994

Figure 4.9 Charles, (1995) shows the number of repairs performed over the years 1983 to 1994. From 1990, the number of repairs decreased from 64 to 7. The variation in the number of repairs over the years could have been due to different factors. Flash butt welding was introduced at the beginning of 1990; as a result, the number of repairs was reduced significantly. In 1994, the number of repairs done was only 7. This shows the efficiency and reliability of flash butt welding in rail welding. It can also be predicted that the number of weld repairs will go down once the friction stir welding for joining rails is fully understood and fully operational.

Table 4.3: Statistics of rail joint with estimated friction stir welding Figures

Joint Type	Assembly time/Joint (minutes)	Cost/Joint, (Rand)	Failures in 2-year period	Maintenance Cost/Joint (1st year), $	Rerail man-hr/linear ft
Bolted	120	3010	95	7000	1.5
Manual Welded (Arc weld)	660	14000	16	5250	2
Exothermic	240	4340	8	1680	1.7
Flash Butt	3	4431	0	0	0.5
Friction stir welding	4	1352.979	0	0	

In order to understand the costs for friction stir welding used for joining rails, the following assumptions were made in the costs analysis;

- The Electricity charge of 24.64 cents per kWh
- Labour charge of R200 per day
- The material cost of the machine itself together with the components and the rotating pin is not included in the analysis
- Transportation costs of the machine have been omitted
- The friction stir weld joint will last up to four years without major or any repair.
- The maintenance cost of the designed machine was assumed to be minimal as compared to other major costs such as electricity and labour

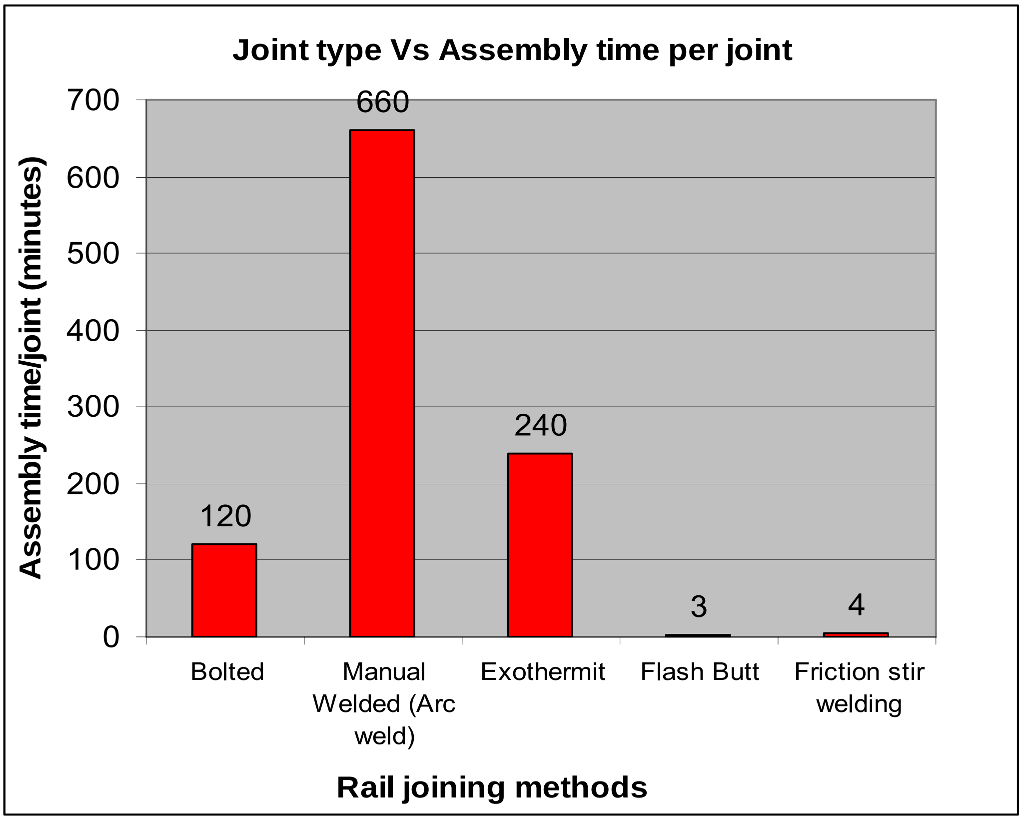


Figure 4.10: Assembly time per joint for all methods used

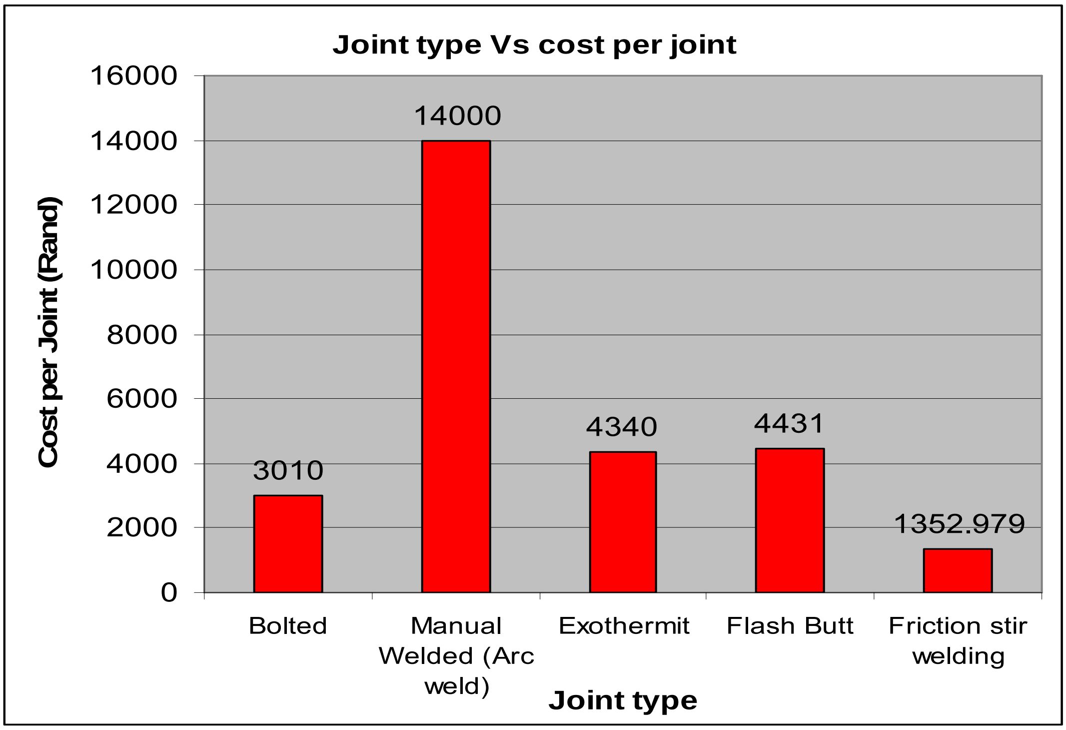


Figure 4.11: Cost per joint for all methods used

The cost per joint for the designed friction stir welding machine seems to be very low as compared to all other processes used for joining rails. The friction stir weld joint is assumed to be the most durable and reliable joint due to the fact that it has the ability to withstand high impact forces. It has been proven that friction stir welding methods yields the material properties which are similar to the parent material as shown in Figure 4.10 and 4.11. As a result, the friction stir welding technique has the ability to withstand more fatigue circles. Costs for buying and manufacturing the friction stir welding machine are very low as it is made from the low, medium and high carbon steel content which is relatively cheap in the South African market. Costs associated with friction stir welding are very low merely because of its high ability to align rail and short welding time. The friction stir weld joint has shown In the literature review that it can eradicate most of the maintenance problems in a rail line.

5 CONCLUSION AND RECOMMENDATIONS

5.1 INTRODUCTION

This design report has been organised into five chapters (described briefly below) which were structured, unified and focussed on solving one conceptual design problem:

- The first chapter set the scene by defining the project objectives and the reasons for selecting friction stir welding for joining rails.
- Chapter 2 gives a summary of the friction stir welding technique and its applications, and a literature review on the friction stir welding process.
- Chapter 3 provides the design proposal of friction stir welding machine for joining rails including the product requirement specification. Chapter 3 also, is concerned with the design development of a friction stir welding machine. Here, all concepts are presented and the best combinations of those concepts were identified in order to meet the production requirement specification. In addition, the chapter gives an overview of the detailed components design and the selection of standard components;
- Chapter 4 provides a design specification of the designed friction stir welding machine; cost analysis of friction stir welding concept was considered as compared to currently used methods for joining rails and, finally,
- Chapter 5 briefly summaries the previous chapters, and prior to making a conclusion about the design, this chapter explains how the new design has an impact in rail welding.

5.2 CONCLUSION ABOUT THE CONCEPTUAL DESIGN

A friction stir welding machine suitable for joining rails has been designed based on the product requirement specification. It was shown that the costs of welding two rails by means of the friction stir welding technique were lower as compared to the other rails joining processes. Finally, it has been established that the introduction of friction stir welding machine for joining rails has major benefits as compared to the traditional methods for joining rails

5.3 CONCEPTUAL DESIGN IMPLICATIONS

This section provides the theoretical implications of the conceptual design of a friction stir welding machine. Its also supports that this design report has not only made a significant contribution to knowledge in its immediate discipline, but also it has had implications for a wider body of knowledge where other disciplines could benefit from its findings.

5.3.1 *IMPLICATIONS FOR THEORY*

Four methods; namely; arc welding, exothermic welding, flash butt welding and manual joining (rails joined by means of splice plate) for joining rails have for many years received tremendous support from railway business giants all over the world. However, the support for arc welding is slowly deteriorating mainly because many companies are moving away from these methods because of the poor quality of weld and the high costs of training personnel associated with it. In the railway environment, the challenge remains - quality of weld is critical irrespective of the method used.

5.3.2 *IMPLICATIONS FOR PRACTICE*

The research literature has shown that the introduction of friction welding brings benefits, which improves not only the productivity, consistency, clarity, accuracy and the quality of the welding activity itself, but also improve various related activities such as production lead-time, production scheduling and capacity utilization.

The research findings have shown that the application of friction stir welding for joining rails could facilitate robust rail joining welds. In essence, this could assist in reducing rail derailment due to the poor quality of rail welds. As a result of this, friction stir welding’s scope of application becomes wider and in line with the new demands in product development practices that require a direct link with activities and systems, including parameter optimisation and practical tests. Consequently, friction welding could facilitate and improve the quality of welds in rail joining, promote safer working environment and contribute towards the implementation of friction stir welding for joining rails.

5.4 RESEARCH LIMITATIONS

This design report has aimed to:

- Conceptually design the friction stir welding machine for joining rails.
- Perform engineering calculations on the selected components to validate the conceptual design on important (complex) components of the machine.

This design project has focused on the conceptual design of a friction stir welding machine for joining rails. Several components were chosen and detailed engineering calculations were performed on them. The friction stir welding machine was designed using Solid Edge software.

Some of the report limitations acknowledged by the author that do not detract from the significance of the report's findings refer to the need to practically build the machine and conduct various tests in order to validate the machine performance in reality.

5.5 FURTHER RESEARCH

This section has been written to help students and other researchers in the selection and design of future research directions.

Finally, this conceptual design report has shown that it is both theoretically and practically possible to find new solutions for rail joining by using the friction stir welding technique. Also, the report has laid a foundation for further research. The following recommendations are made for further research:

- There are some areas of concern which need to be looked at in great detail.

- Very little work has been done on pin selection, pin material and its metallurgical properties. The life of the pin used for stirring the material of two working pieces is of great importance. Therefore, there is a need for further research and design to the pin selection and its life associated with friction stir welding for joining rails.
- Due to time constraints on the project, the metallurgical process was not covered in great length. Therefore, it is of utmost importance to look at the metallurgical process of rail welding in detail. This will help in predicting the quality of weld by the friction stir welding process. This work could cover the following areas: heat flow of material, heat affected zone, failure mode, hardness, etc
- In order to observe the capability of the machine in practice, it is crucial to build the machine in such a way that so that practical tests can be conducted. These tests should assist in finding the optimum parameters of friction stir welding for joining rails.
- The modeling of the proposed machine design is essential in successfully building the machine and in knowing exactly the stress acting on the structure due to structural loading. A FEA model using either Nastran or Pastran or other available FEA software must be done.

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APPENDIX A: UNDERSTANDING OF FSW TECHNIQUE

A.1 BACKGROUND HISTORY OF WELDING

Welding started back in the ancient times. The simplest example would be the Bronze Age where gold circular boxes were made by pressure welding lap joints together. These boxes have been estimated to have been made 2000 years ago. In the time of the Iron Age people in Egypt and in the Mediterranean area were able to weld pieces of iron together. Using the welding techniques these people were able to produce many tools dating back to 1000 BC (American Welding Society website).

It has been found that during the Middle Ages most iron pieces were welded together by means of hammering. In other words two pieces were hammered together in order to form one piece. The technique of hammering to weld was used until welding as it is known to day was invented. In about 1836 Edmund Davy of England discovered acetylene. In 1800 Sir Humpry Davy was credited with the production of an arc between two carbon electrodes using a battery. Sources mention that gas welding and cutting were developed in the late 1800s (American Welding Society website).

According to the American Welding Society, friction welding originates from the late 1800s. The first patent was issued around 1891 on the process of friction welding. More work was done in Europe between 1920 and 1944 and this is the time were more patents has been issued. Rockwell International, AMF and Caterpillar have together contributed much in the field of friction welding in the 1960s (NCT Incorporated website). As a result of their contribution Rockwell International built its own machines to weld spindles to truck differential housings where AMF designed and built the machines to weld steering worm shafts. The Caterpillar Company concentrated more on welding turbochargers and hydraulic cylinders (NCT Incorporated website).

A.2 DETAILED EXPLANATION OF FSW TERMS

TOOL AXIAL WELDING FORCE

- The tool axial welding force need to be controlled in an adequate way.
- The axial welding force must not be too low or too high as this has influence on weld quality.
- For example if the axial welding force is too low the heat generated is too low and if the force is very high the tool shoulder will penetrate deep into the working piece thereby causing excess flash at the edges of the shoulder which is not desirable.
- In addition, the axial welding force could also affect the surface texture of the working piece (Stockholm, 2005).

SIDE CLAMPING FORCE

- Side clamping force must be very high as compared to the force generated by the stir tool during the friction stir welding process.
- The spinning tool pin will try to push the two rails apart; therefore, the clamping force must be greater than the force generated.
- The clamping force depends very heavily on the size of the working piece (the thickness of the rail profile).
- When the thickness of the working tool increases, the tool pin also increases. If the clamping force is not high enough to prevent the working piece from separating, voids are normally formed just below the surface on the advancing side of the weld (Stockholm, 2005).

LENGTH OF TOOL PIN

- The length of the tool pin must be controlled to make sure that it is not too short or too long.
- If the length of the tool pin is too short, the two working pieces to be welded together do not join completely.
- This is normally seen during the bending test because the two pieces will fracture early during tensile or root bend testing.

- If the tool pin is too short stress raiser will occur which could initiate a crack on the working piece.
- The tool pin is not allowed to be too long as it can be damaged by contacting other materials their support the welding process (Stockholm, 2005).

THICKNESS

- It is very important to be consistent with the wall thickness of the working piece because when thickness changes the tool geometry also changes.
- The tool geometry changes in order to accommodate the heating, stir and height of the weld.
- If the thickness of the piece increases and the tool geometry remains the same or the tool geometry is small, there is an area near the bottom of the weld which will not be welded. This gives poor quality welds.

TOOL GEOMETRY

- In friction stir welding tool geometry is the principal parameter.
- It has been found that tool pin geometry/shape has great influence in heat generation, plastic flow and the stirring in the weld.
- The width of the weld represents the size and shape of the tool pin. When the tool geometry is modified or redesigned the micro-structural properties of the weld changes.
- In the early stages of friction stir welding only the tool pin with a constant cross section was available, however nowadays, pin tools with complicated shapes are available.
- The tool pin and shoulder are made from different materials; this is because the tool pin is usually required to have certain material properties as compared to the shoulder. It is obvious that the tool pin must be stronger than the shoulder and that is why the tool pin is usually made from tungsten, carbides, cermets, super alloys and refractory metals.

The main function of the shoulder is to generate heat to assist the tool pin to soften material around it. Usually the shoulder has a flat underside, however a spiral underside is found in some shoulders for other applications. During a research for reduced size, Triflute^{TM} geometry was developed better. The Triflute^{TM} provide the user with a better material flow and adequate stir action Stockholm, (2005). It is important to note that different tool geometry is under development by other companies. See Figures 7.1, 7.2 and 7.3 (Stockholm, 2005).

Figure 7.1: Different tool profile geometry from (a) Whorl ^{TM} pin (b) (c) Triflute ^{TM} pins.

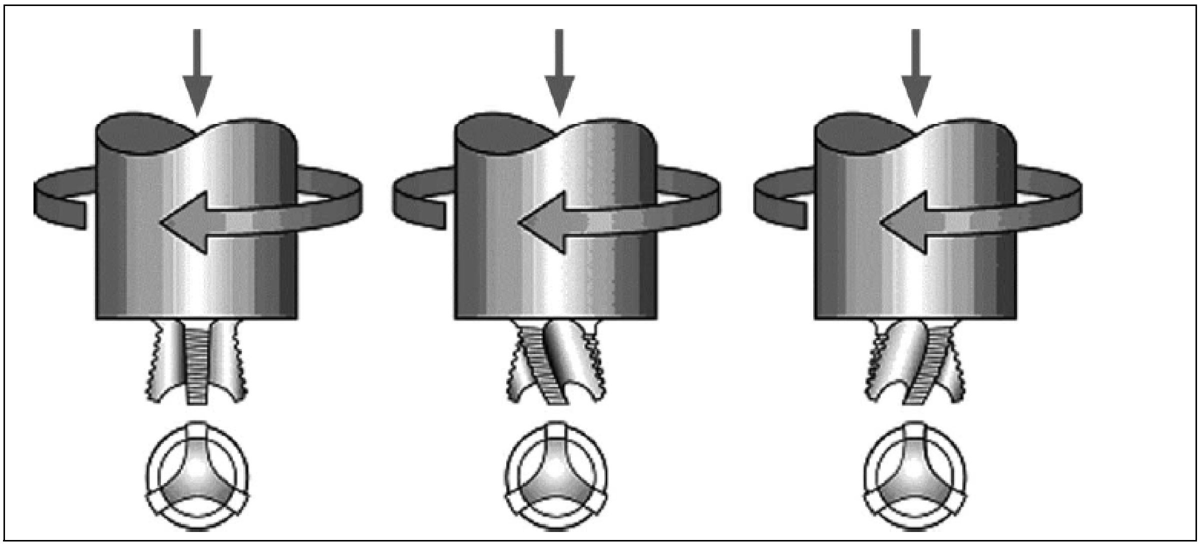


Figure 7.2: Flared- Triflute ^{TM} tool pin geometries developed by the welding institute, (a) neutral flutes, (b) left flutes and (c) right hand flutes.

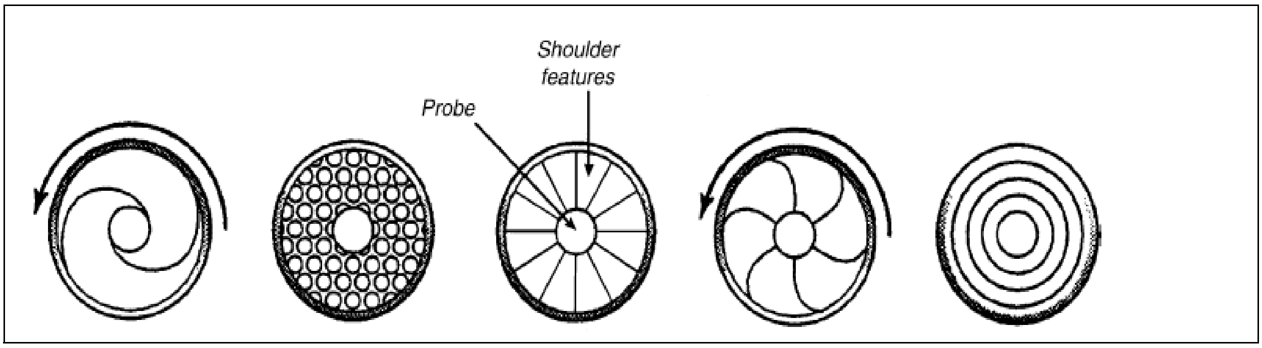


Figure 7.3: Different shoulder profiles viewed from underneath the shoulder (Copy right © TWI).

WELDING SPEED

- It is an important friction stir welding parameter and must be controlled as accurately as possible.
- In rail friction stir welding the welding speed could be controlled by an actuator or hydraulic speed controller.
- The hydraulic speed controller needs to be selected carefully so that it can be used in friction stir welding.
- This parameter is obtained mainly by trial and error as there is no readily available information or analytical method to determine the speed required.
- The machine must be designed in such a way that speed is variable in order to get the optimal speed for a particular application.

TOOL ROTATION SPEED

- It is not fully known at which speed the pin tool should run in order to get a high quality weld on rails.
- Additional experiments need to be done so that the rotation speed of the tool pin is optimized.
- The more carbon content in steel the higher the rotation of tool pin is required.
- The rotational speed is one of the most friction stir welding process variables (Stockholm, 2005).

TILT ANGLE

- In order to prevent side sub-surface voids, various attempts have been made to increase the tilt angle on the tool pin.
- When the tilt angle is increased, the forging on the back edge of the shoulder is increased.
- A large tilt angle gives a uniform material flow.
- More research has to be done on the tilt angle of high carbon steel material (Stockholm, 2005)

PILOT HOLE AT WELD START

- When performing friction stir welding on plates, a pilot hole is drilled in order to minimize the start torque of the tool pin.
- The pilot hole helps prolong the life of the tool pin; in most cases if the hole is not drilled the pin tool has a tendency to fracture at the beginning stage of friction stir welding.
- Due to the high development rate of the tool pin, stronger pin tools are available which are able to work well with a simple drilled hole at the beginning of the friction stir welding process (Stockholm, 2005).

TOOL COOLING

- A tool cooling system is required in order to successfully friction stir weld the two high carbon content steel.
- The temperature of the tool pin and the shoulder are to be kept constant.
- If the temperature of both the tool pin and the shoulder is not controlled a fracture will occur between the tool pin and the shoulder.

TOOL PARKING

- When one complete circle of the friction stir welding of rail, the initial pilot hole drilled for process stimulation is over welded by a tool pin.

- Rail is usually subjected to high stresses and therefore a pilot hole or tool parking hole must never be allowed to occur as this will create a source of crack propagation.
- Therefore, it could be concluded that the tool pin must start to rotate 10 mm from the leading edge surface.
- The tool pin must also be allowed to move right across the rail.
- The entire problem caused by tool parking and the pilot hole is solved.

A.3 GENERAL UNDERSTANDING OF FRICTION WELDING TECHNIQUES

BRIEF DESCRIPTION

The definition of friction welding according the American Welding Society abstract is as follows:

"In the direct drive variation of friction welding, one of the workpieces is attached to a motor driven unit, while the other is restrained from rotation. The motor driven workpiece is rotated at a predetermined constant speed. The workpieces to be welded are moved together, and then a friction welding force is applied. Heat is generated as the faying surfaces (weld interface) rub together. This continues for a predetermined time, or until a preset amount of upset takes place. The rotational driving force is discontinued, and the rotating workpiece is stopped by the application of a braking force. The friction welding force is maintained or increased for a predetermined time after rotation ceases (forge force)" (American Welding Society website)

Several parameters are vital in friction welding, and these are: speed of the moving or rotating component, forge pressure, displacement and the duration of the spinning component. The parameters are interdependent and in most cases several trials are done before the mass production. The trials are performed in order to maximize the properties of welds on the welding piece. It has been found that almost any thermoplastic material can be friction welded [American Welding Society website].

The process of friction welding is illustrated in Figure 7.4. The process is mainly composed of three phases as shown.

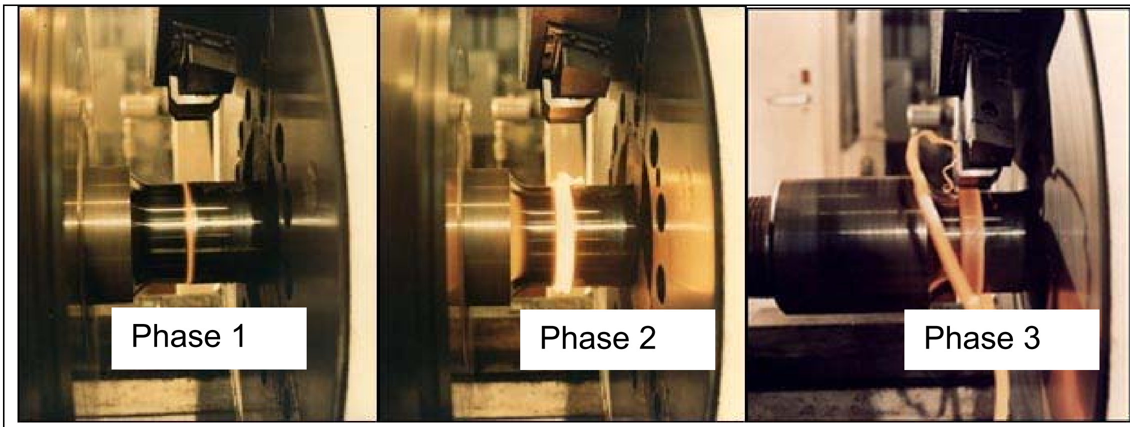


Figure 7.4: Three main phases of friction welding process

Phase 1 in Figure 7.4 (NCT Incorporated website), is the initial phase with a low temperature interface where one stationary component is placed against the spinning component to produce friction or heat. Phase 2 shows the state where enough heat is generated and all components are in a plastic state. A predetermined Axial forging force is applied in order to join the two components. In phase 3, plastic state flashing is easily removed (NCT Incorporated website).

Friction welding yields a very high strength because of full cross-sectional forging. It also provides low stress porosity and in most cases there is no need for very expensive pre-machining. This process also allows the joining of two different materials and still yields good material property. An example is the joining of steel to stainless steel and copper to aluminium or vice versa (NCT Incorporated website)

Friction welding technique is a low temperature welding technique. This means that a small area is affected by heat. By using this technique most material properties are not greatly affected as compared to the traditional liquefied metal. In this technique no third metal is added to the process. Also, it is possible to remove the flash (the plastic state material displaced during forging) while the welding piece is still soft and hot. In addition, costly grinding is avoided by removing flash while hot.

BENEFITS AND APPLICATIONS OF FRICTION WELDING

Benefits:

The friction welding process is an environmental friendly process as it uses minimum energy consumption and it does not generate any smoke or gases which are harmful to society or the environment. Furthermore, friction welding gives the benefits of free design, where designers are free to combine various factors in one piece of design. These factors are: conductivity, reluctance, hardness, strength, weight, tubular and non magnetic. The friction welding process requires less labour as compared to the traditional way of welding and it could also give the designer the benefit of simple component design (NCT Incorporated website).

This process is ideal for a prototype or small component because it gives flexibility of initial low costs. Furthermore, the friction welding process is a low cycle time process, it takes less time to produce or weld one component. This leads to low costs and high production business environment. The process does not require much of surface preparation and saw cuts are most commonly used (NCT Incorporated website).

Applications:

The friction welding process is used in many industries in different applications. The first industry which friction welding is commonly used is Automotive. In the Automotive Industry the following components are welded: air bag inflators, transmission gears, axles, axle housings (NCT Incorporated website).

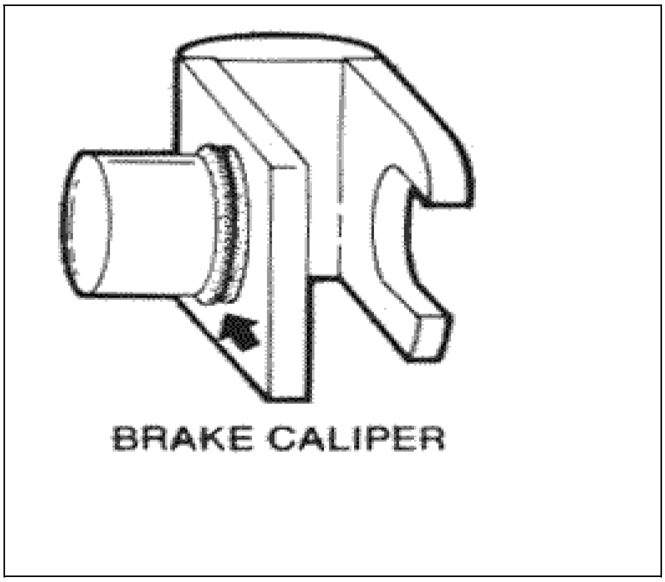


Figure 7.5: Friction welding on brake calliper

Figure 7.5 shows the application of friction welding in the automotive industry. The brake calliper as shown in Figure 7.5 is welded by using friction welding techniques. Other industries which use the friction welding process are food, military, medical, marine, mining/drilling, Bi-metal and Hydraulic. The following are examples of components manufactured by friction stir welding in different industries, namely: hollow heat exchangers, delta II rockets, laser system housings, aircraft structure, space shuttle external tanks and food trays. See Appendix G on how components are welded together.

TYPES OF FRICTION WELDING

LINEAR VIBRATION WELDING

In this type of friction welding, the components to be welded are brought into contact and all components are moved relative to each other to generate enough friction. The components are rubbed against each other and at the same time forged towards each other. Furthermore, parts are vibrated at amplitudes of between 1.0 mm and 1.8 mm at a frequency of 200Hz or the amplitude of between 2.0 mm and 4.0 mm at a frequency of 100Hz. Examples of linear friction welding applications are shown in Figure 7.6 (The Welding Institute website).

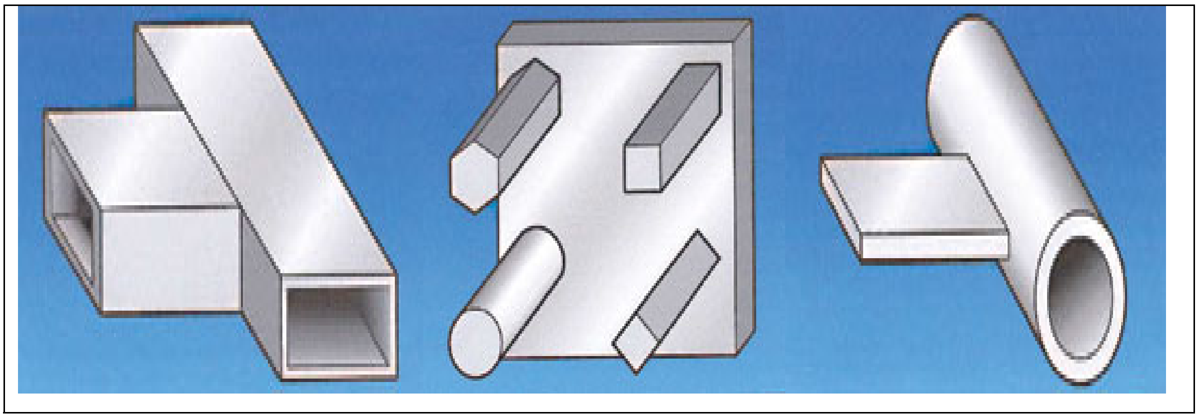


Figure 7.6: Example of linear friction welding

ORBITAL FRICTION WELDING

"In orbital welding each point on the surface of one part orbits a different point on the face of the other part. The orbit is of constant rotational speed and is identical for all points on the joint surface. This motion is stopped after sufficient material is melted and the thermoplastic then solidifies to form a weld" (The Welding Institute website)

"Orbital welding is a relatively new technique, and tends to fill the size gap between benchtop ultrasonic units and linear vibration welders, and most applications tend to be for automotive components" (Thomson Friction Welding website). Figure 7.7 (The Welding Institute website) shows an example of linear friction welding (Thomson Friction Welding website])

SPIN FRICTION WELDING

Spin friction welding works well when both parts have a circular cross section. Spin friction welding is used in a variety of applications such as the manufacture of polyethylene floats, aerosol bottles, transmission shafts and PVC pipes and fittings. The great advantage of spin friction is that it can be performed below the surface of a liquid. An example of spin friction welding is shown in the Figure 7.7.

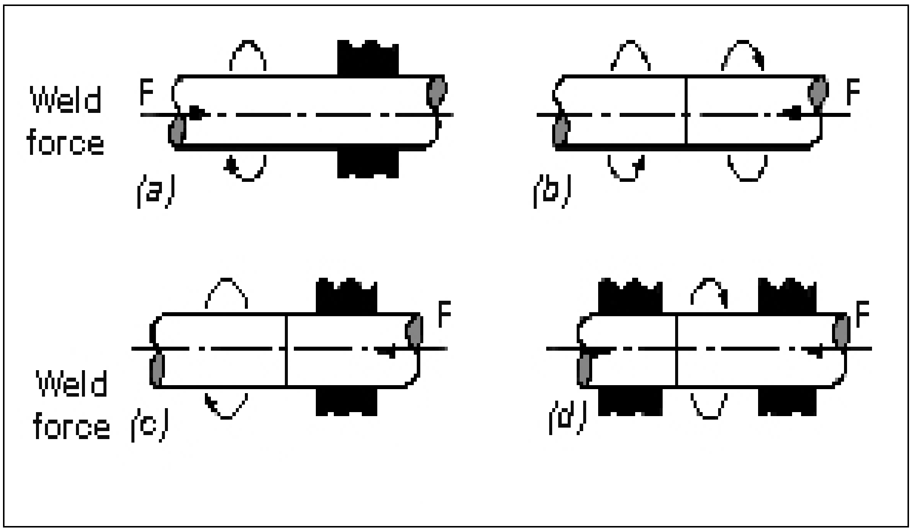


Figure 7.7: Possible configurations for spin welding thermoplastics

ANGULAR FRICTION WELDING

Angular friction welding is comparable to linear vibration welding. The distinction is that angular friction welding requires angular motion whereas linear motion is needed for linear vibration welding. The vibration on angular friction is rotates the workpiece through fewer degrees.

APPENDIX B: DESIGN CALACULATIONS

B.1 LOADS ACTING ON THE STRUCTURE: ASSESSMENT AND DECISIONS

The friction stir welding machine for joining rails has never been built before; therefore, all loads acting on the structure are obtained by means of detailed research for similar materials or processes. Most work has been done on the friction stir welding of aluminium and less work has been done on steel material. As a result, most of the information contained in here has been for aluminium material.

In order to fully understand the loading specification of the designed structure, the following must be known:

- the type of material to be welded
- the thickness of the material to be welded
- type of welding tool to be used

The machine is designed to friction stir weld a number of rails used in South African railway lines. Table 0.1A in Appendix A defines the rails used in South Africa and their material properties or composition.

In Appendix C, Tables of different materials, overall sizes and loading specification are shown. From Table 0.3C, in particular reference to steels, shows that for a mild steel plate of about 12 mm the average rotation speed of 600 rpm, and the transverse speed of about 100 mm/min were recorded. The torque which was acting on the rotating tool was about 55 N.m and a normal force of about 18 kN was recorded (Lienert *et al*, 2003). From the above information, it can be deduced that for a plate thickness of 120 mm, the torque acting on the tool could be around 550 N.m with the required rotational speed of 4000 rpm and the minimum welding speed or transverse speed of 1000 mm/min (16.67 mm/sec). The required force to push the rotating pin in the direction of the weld depends very much on the welding speed and the material to be welded on. Therefore, the maximum allowable applied force in the direction of the weld is estimated to be between 44 kN and 55 kN.

Based on the above information the following information in Table 7.1 has been drawn up in order to summarise the design requirements parameters. These parameters are required in a detailed design analysis of the machine components to be used. In addition, these estimated parameters are only the estimates since there is no source of information. The unavailability of information is due to the fact that friction stir welding is very new and not much study has been done on it.

7.1: Estimated welding required parameters for design

Properties	Minimum Value	Maximum Value
Rotational speed (rpm)	2500	4000
Torque	55 N.m	65 N.m
Transverse or welding speed (mm/sec)	16.67	30
Estimated Pushing force (kN)	44	55

B.2 SELECTION OF STANDARD COMPONENTS

DRIVE MOTOR SELECTION

In order to select motor drive successfully various issues need to be investigated and fully understood for example, power required to drive the load or tool pin. Shigley et al, (2004) states that power can be obtained by utilising the following relationship:

$$P = \frac{2\pi T\omega}{60} \qquad \textbf{(7.1)}$$

Where P is the power required to drive the load
T is the torque and ω is the rotational speed.

Equation 7.1 is used together with Table 7.1 so that the power required is obtained.

From equation 7.1, the power required to drive the load is found to be:

$$P = \frac{2\pi \times 2500 \times 65}{60}$$
$$= 17.02kW$$

ELECTRO-HYDRAULIC DRIVE SELECTION

The electro-hydraulic actuator was selected on its overall dimensions and the required maximum force. The overall dimension included the ease with which it can be mounted on the designed machine and the portability of the device. Another factor which has contributed to the selection of the electro-hydraulic actuator was the technical performance of the device. Also, the actuator must be able to translate (Translational speed) by the required speed and must be able to apply the maximum and minimum force required. Table 7.1 specify the welding speed of 30 mm/sec and model 139 is capable of providing the speed of 100 mm/sec and was selected. See Appendix F for the technical specification of the selected electro-hydraulic actuator.

COUPLER SELECTION

The coupler is used to join the extended shaft to the electric motor and the string tool pin. The selection of couplers is based on the SteelFlex selection guide (The Falk Corporation, 2004) that requires the following information:

- (i) kilowatt (kW) or torque
- (ii) running rotational speed (rpm)
- (iii) Type of equipment to be connected (motor to pump, gear drive to conveyor etc)
- (iv) Shaft diameters (for both driver and driven shafts)
- (v) Shaft gaps
- (vi) Physical space limitations
- (vii) Special bore or finish information and type of fit required.

The coupler between the motor and the extended shaft is to be selected based on the information above. The selected coupler is used between the gearbox output shaft and the extended shaft. The standard selection procedure used is adopted from (The

Falk Corporation, 2004). See Appendix F for the detailed coupler selection procedure. Therefore, the gearbox output shaft has a rotating speed of 2500 rpm and an output power of 17.02 kW. The driver diameter is measured at 65 mm and the driven extended shaft has a diameter of 92 mm.

To determine the required rating, the following relationship is used:

$$Systemtorque = \frac{kW \times 9549}{rpm}$$
$$= \frac{17.02 \times 9549}{2500}$$
$$= 65.01 Nm$$

Looking at The Falk Corporation, (2004 Appendix F shows the service factor for machine tools is equal to 1.75. Therefore, the required minimum coupling rating is given by:

Required coupling rate = 1.75 (65.01)
= 113.77 N.m

Looking at The Falk Corporation (2004), Appendix F, the suitable coupler is 1090T. This type of coupler is capable of absorbing a rotational speed of 3600 rpm and the torque rating of 3730 Nm. Both the rotational speed and the rated torque do not (1090T) exceed the specified speed of 2500 rpm and the rated torque of 65 N.m. Therefore, the selected coupler meets the minimum parameters specified. The maximum and minimum bore diameters are specified to be 95 mm and 27 mm and the shaft diameters to be coupled 65 mm and 91 mm. Therefore, the selected coupler meets the basic geometric requirements. Appendix F shows detailed information needed for selecting the above mentioned coupler.

The coupler is also suitable for joining the rotating tool pin and the extended shaft since a rigid coupler is required to make sure that the pin and the shaft and the pin are as rigid as possible. Therefore, a reference to The Falk Corporation (2004) was made to select the perfect rigid coupler suitable for coupling the rotating tool pin and the extended shaft. From The Falk Corporation, (2004), Appendix F; size 1020G type of coupler has been selected. It is able to accommodate the maximum rotational speed of 5600 rpm and the torque rating of 4270 Nm. The maximum and minimum bore diameters are 98 mm and 26 mm respectively. Therefore, the above selected coupler is able to accommodate the required parameter on both sides.

B.3 DESIGN OF SELECTED MACHINE COMPONENTS

Forces which are applied on the friction stir welding technique are significantly higher than those applied in arc welding. As a result, the structure to be used for friction stir welding must be rigid.

The following components were chosen for the detailed design.

(i) Extended shaft
(ii) Rail clamp
(iii) Structure supporting or fixing the electric motor (motor support bracket)
(iv) Vibration isolator between the rail clamp structure and top (motor supporting structure)
(v) Lifting mechanism

DESIGN OF AN EXTENDED SHAFT

The extended shaft is used as an extension rod to connect the motor and the pin (working tool). The shaft is designed in such a way that is zero defection will occur when the maximum load is applied to the shaft. This force is applied by means of an electro-hydraulic cylinder connected to the shaft. The shaft is assumed to be subjected to pure bending load due to the electro-hydraulic cylinder and pure torsonal load due to the rotating motor output shaft. The following assumptions are made in order to design the extended shaft successfully:

- The shaft is manufactured accurately based on the requirements of surface finish and material composition
- The shaft diameter is of a specified size

- The load acting on the shaft acts in a manner similar to that of a Cantilever beam.
- The beam is to be designed for static and fatigue loadings

Shaft design may be achieved by using different fatigue-failure criteria, namely:

(i) MSS-Seoderberg
(ii) DE-Goodman
(iii) ASME-elliptic
(iv) DE-Gerber

The failure criteria stated are used to select the suitable shaft diameter. Therefore, the following equations are used. All equations used are taken from Shigley *et al*, (2004).

- **For MSS-Seoderberg fatigue-failure criteria**

$$d = \left\{ \frac{32n}{\pi} \left[\left(K_f \frac{M_a}{S_e} \right)^2 + \left(K_{fs} \frac{T_m}{S_y} \right)^2 \right]^{1/2} \right\}^{1/3} \quad \textbf{(7.2)}$$

- **For DE-Goodman**

$$d = \left\{ \frac{16n}{\pi} \left(2 \frac{K_f M_a}{S_e} + \sqrt{3} \frac{K_{fs} T_m}{S_{ut}} \right) \right\}^{1/3} \quad \textbf{(7.3)}$$

- **For ASME-elliptic**

$$d = \left\{ \frac{16n}{\pi} \left[4 \left(K_f \frac{M_a}{S_e} \right)^2 + 3 \left(K_{fs} \frac{T_m}{S_y} \right)^2 \right]^{1/2} \right\}^{1/3} \quad \textbf{(7.4)}$$

- **For DE-Gerber**

$$d = \left(\frac{16nK_f M_a}{\pi S_e} \left\{ 1 + \left[1 + 3\left(\frac{K_{fs} T_m S_e}{K_f M_a S_{ult}} \right)^2 \right]^{1/2} \right\} \right)^{1/3} \quad \textbf{(7.5)}$$

Where:

n – Fatigue safety factor

K_f – Fatigue stress concentration factor

K_{fs} – Fatigue stress concentration factor (shear loading)

M_a – Alternating bending moment

S_e – Endurance limit

S_{ult} – Ultimate strength

S_y – Yield strength of the material

T_m – Steady or mean applied torque

Given information for shaft design:

n = 2

M_a = 40 kN*0.15m

= 6.0 kN.m

= 6000 N.m

T_a = 100 N.m

S_y = 1000 MPa

S_{ult} = 1200 Mpa

The following data is assumed;

D/d = 1.1

r/d = 0.05

Analysis:

The first step is to determine the endurance limit of the shaft. The endurance limit is calculated by the following relationship for most steel materials:

$$
\begin{aligned}
S_e' &= 0.504 S_{ult} \\
&= 0.504 \times 1200 MPa \\
&= 604 MPa
\end{aligned}
\qquad \textbf{(7.6)}
$$

The endurance limit of the shaft itself is given by the following relationship:

$$
\begin{aligned}
S_e &= k_a k_b k_c k_d k_e k_f S_e' \\
&= 0.963(1)(0.85)(1)(0.814)(1)(604) MPa \\
&= 401.2 MPa
\end{aligned}
\qquad \textbf{(7.7)}
$$

The stress concentration factor (bending load) is calculated by the following relationship:

$$K_f = \frac{K_t}{1 + \frac{2(K_t - 1)}{K_t} \frac{\sqrt{a}}{\sqrt{r}}} \qquad \textbf{(7.8)}$$

$$K_f = 1 + (K_t - 1)q \qquad \textbf{(7.9)}$$

Referring to Juvinal and Marshek (2006), the stress concentration factor found in Figure 0.1N, Appendix H, corresponding to r/d of 0.05. In Figure 0.1 N in Appendix H, the value of K_t corresponding to r/d of 0.05 is found to be 1.87.

The notch sensitivity factor is obtained by assuming that the notch radius r is equal to 2mm and an ultimate strength of 1.2 GPa. The notch sensitivity factor Figure 0.2N Appendix H Juvinal and Marshek (2006) based on the above mentioned is equal to 0.95.

Therefore,

$$K_f = 1 + (1.87 - 1) \times 0.95$$
$$= 1.8265 \qquad \textbf{(7.10)}$$

The stress concentration factor (torsional load) is calculated by the following relationship

$$K_{fs} = \frac{K_{ts}}{1 + \frac{2(K_{ts} - 1)}{K_{ts}} \frac{\sqrt{a}}{\sqrt{r}}} \qquad \textbf{(7.11)}$$

$$K_{fs} = 1 + (K_{ts} - 1)q \qquad \textbf{(7.12)}$$

The stress concentration factor (torsional loading) is found from Figure 0.1N Appendix H corresponds to r/d of 0.05. According to Figure 0.1N the value of K_{ts} is found to be 1.55.

Notch sensitivity factor is obtained by assuming that the notch radius r is equal to 2mm and ultimate strength of 1.2 GPa. The notch sensitivity factor is found from Figure 0.4N Appendix H Juvinal and Marshek (2006) based on the above mentioned is equal to 0.95.

Therefore,

$$K_{fs} = 1 + (1.55 - 1) \times 0.95$$
$$= 1.5225 \qquad \textbf{(7.13)}$$

The constants found above are used for determining the required diameter of the shaft by using various criteria. The calculated diameter is defined as the minimum diameter required to withstand loadings on the shaft.

- **For MSS-Seoderberg fatigue-failure criteria**

$$d = \left\{ \frac{32n}{\pi} \left[\left(K_f \frac{M_a}{S_e} \right)^2 + \left(K_{fs} \frac{T_m}{S_y} \right)^2 \right]^{1/2} \right\}^{1/3} \quad \textbf{(7.14)}$$

Inserting values on the equation above:

$$d = \left\{ \frac{32(2)}{\pi} \left[\left(1.8265 \frac{600}{604 \times 10^6} \right)^2 + \left(1.5225 \quad \frac{100}{1000 \times 10^6} \right)^2 \right]^{1/2} \right\}^{1/3} \quad \textbf{(7.15)}$$

$d = 71.76mm$

- **For DE-Goodman**

$$d = \left\{ \frac{16n}{\pi} \left(2 \frac{K_f M_a}{S_e} + \sqrt{3} \frac{K_{fs} T_m}{S_{ut}} \right) \right\}^{1/3} \quad \textbf{(7.16)}$$

$$d = \left\{ \frac{16(2)}{\pi} \left(2 \frac{1.8265 \times 6000}{604 \times 10^6} + \sqrt{3} \frac{1.5225 \times 100}{1200 \times 10^6} \right) \right\}^{1/3} \quad \textbf{(7.17)}$$

$d = 71.91mm$

- **For ASME-elliptic**

$$d = \left\{ \frac{16n}{\pi} \left[4 \left(K_f \frac{M_a}{S_e} \right)^2 + 3 \left(K_{fs} \frac{T_m}{S_y} \right)^2 \right]^{1/2} \right\}^{1/3} \quad \textbf{(7.18)}$$

$$d = \left\{ \frac{16(2)}{\pi} \left[4 \left(1.8265 \frac{6000}{604 \times 10^6} \right)^2 + 3 \left(1.5225 \frac{100}{1000 \times 10^6} \right)^2 \right]^{1/2} \right\}^{1/3} \quad \textbf{(7.19)}$$

$d = 71.76mm$

- **For DE-Gerber**

$$d = \left(\frac{16nK_f M_a}{\pi S_e} \left\{ 1 + \left[1 + 3 \left(\frac{K_{fs} T_m S_e}{K_f M_a S_{ult}} \right)^2 \right]^{1/2} \right\} \right)^{1/3} \quad \textbf{(7.20)}$$

$$d = \left(\frac{16(2)(1.8265)(6000)}{\pi(604 \times 10^6)} \left\{ 1 + \left[1 + 3 \left(\frac{1.5225(100)(604 \times 10^6)}{1.8265(6000)(1200 \times 10^6)} \right)^2 \right]^{1/2} \right\} \right)^{1/3} \quad \textbf{(7.21)}$$

$d = 71.76mm$

Table 7.2: Summary of shaft design based on different criterion

Criterion	Diameter (d) (mm)
MSS-Soderberg	71.76
DE-Goodman	71.91
ASME-elliptic	71.76
DE-Gerber	71.76

Based on the summary of results in Table 7.2, the minimum diameter required based on the load assumption applied is 72 mm.

Therefore, the minimum shaft diameter is said to be

d = 76 mm

D/d = 1.2 **(7.22)**

Therefore,

D = 92 mm

After obtaining all relevant data, important parameters need to be determined to draw and plot the designer's fatigue diagram. The designer's fatigue diagram is only constructed in the DE-elliptic criterion case.

VIBRATION ISOLATOR

The main purpose of the vibration isolator is to reduce the magnitude of force transmitted from a vibrating machine to its supporting structure. Various components are used as vibration isolators, for example: coil spring (steel), resilient material or rubbers, cork, felt and dense fibreglass. Consequently, there is a need to design the vibration isolator. In addition, the motor mounted on the top structure has the ability to transmit vibration to the bottom structure (rail clamp and other components). As a result, there is a need to design a vibration isolator to help in reducing force magnitude and increasing weld quality in the system.

The purpose of rail clamps is to ensure that the rail is rigidly clamped together. If vibration is transmitted in large amounts by the motor, the rail clamp could suffer early damage and undesired misalignment could occur.

The following assumptions are made:

- The foundation (the support rail clamp) is massive and rigid and the mounted motor vibrates at constant amplitude.
- If the motor vibrates at constant force, the reduction in force is on the ratio of the exciting frequency to the natural frequency of the system (Kutz, 1999).

The factor which specifies the effectiveness of a vibration isolator is defined as transmissibility. "Transmissibility is defined as the ratio of the force applied to the isolator by means of machine, to the force transmitted by the isolator to the foundation" (Kutz, 1999).

Therefore,

$$Transmissibility = \frac{transmittedforce}{impressedforce} \qquad \textbf{(7.23)}$$

Ideally transmissibility must be equal to zero and in a real situation the objective is to make it very close to zero. This is usually achieved by designing the system to have a lower natural frequency of the mounted machine as compared to that of the exciting force.

By assuming that there is no damping in the system, transmissibility could be expressed as follows:

$$T = \frac{1}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \qquad \textbf{(7.24)}$$

Where T = the transmissibility, expressed as a fraction

ω = the circular frequency of the exciting force, in radians per second

ω_n = the circular frequency of the mounted system, in radians per second

If the ratio of frequency of the exciting force and the frequency of the mounted system is zero, this implies that the transmissibility ratio is one. In this condition there is no need to insert the insulator as there is no benefit to fitting one. If the ratio of frequency of the exciting force and the frequency of the mounted system is greater than zero but less than 1.41, the designed isolator tends to increase the magnitude of transmitted force. This region is called, “the region of implication” (Kutz, 1999). From equation 7.24, if the ratio ω/ω_n is equal to unity, the theoretical amplitude of the transmitted force goes to infinity, since it is the point where the natural frequency of the system is equal to the frequency of the disturbing force.

If the ratio ω/ω_n becomes greater than 1.41, the transmissibility becomes negative. The negative sign is simply due to the phase relation and the motion and could be disregarded if only the force transmitted is a concern. Therefore, the transmissibility could be written as:

$$T = \frac{1}{\left(f/f_n\right)^2 - 1} \tag{7.25}$$

Where $\omega = 2\pi f$ **(7.26)**

The natural frequency is closely related to the static deflection of the spring; therefore, the natural frequency is expressed as follows:

$$\omega_n = \sqrt{k/m} = \sqrt{g/\delta_{static}} \tag{7.27}$$

The above equation 7.27, may be converted to equation 7.28 as follows:

$$f_n = 3.14\sqrt{1/d} \tag{7.28}$$

Where f_n = natural frequency, Hertz

d = the deflection, in inches

According to Kutz (1999), the natural frequency of the isolator should be about one-tenth to about one-sixth of the driving frequency for critical conditions, and the corresponding transmissibility factor should be between 1% and 3%. For non-critical or less critical condition, the natural frequency of the isolator should be between one-sixth and one-third of the driving frequency and the corresponding transmissibility should be about 3% and 12% (Kutz, 1999).

Design calculation

The frequency transmitted to the structure is found by assuming that the motor is rotating at a maximum speed of 4000 revolutions per minute. Therefore, ω=4000 rpm, when converted to Hertz, ω=66.67 Hz. The load applied to a bottom or supporting structure is estimated by summing the masses of all components from the top structure. The total weight is found to be approximately 150 kg, which can be easily converted to magnitude force of 1471.5 N. ($F_{applied}$ = 1471.5 N).

From the design point of view, the above specified force applied to the isolator need to have a safety factor and the safety factor is assumed to be 1.8. This clearly implies that the force applied to the isolator is now equal to 2648.7 N.

It is assumed that the spring has the ability to deflect 2.5 mm when maximum force is applied. The natural frequency of the isolation system could be found by equation 4-6 above. Since *d* is expressed in inches, 2.5 mm equals 0.1 inches. Therefore, the natural frequency of the isolation in the system is found to be:

$$\begin{aligned} f_n &= 3.14\sqrt{1/0.1} \\ &= 9.93Hz \end{aligned} \qquad \textbf{(7.29)}$$

By using equation 7.25, the transmissibility is found to be;

$$\begin{aligned} T &= \frac{1}{\left(66.67/9.93\right)^2 - 1} \\ &= 0.02269 \\ &= 2.269\% \end{aligned} \qquad \textbf{(7.30)}$$

The condition is critical since the ratio ω/ω_n is approximately seven times the natural frequency. In other words the natural frequency of the isolating system is seven times that of the rotating machine. The transmissibility is also between 1% and 3%.

Design scenario

The helical spring design process was adopted from Juvinal and Marshek (2006). In order to design the helical spring geometry which will assist in reducing the force transmitted to the bottom structure, the following scenario is assumed.

- The helical spring must deflect 30 mm when the maximum 2.7 kN force is applied and the loading is assumed to be static.
- The helical spring must be able to fit accurately into a hole of 40 mm diameter.
- The helical spring is assumed to have been made from oil tempered ASTM 229 wire.
- "There is no unfavourable residual stress""
- "Both end plates are in contact with nearly a full turn of wire"
- The end plate loads coincide with the spring axis"

Considering the above, the spring constant is calculated as follows:

The following equations are to be used in the design of the helical spring.

$$K_w = \frac{4C-1}{4C-4} + \frac{0.615}{C} \tag{7.31}$$

Where C is D/d

$$K_s = 1 + \frac{0.5}{C} \tag{7.32}$$

$$\tau = \frac{8FD}{\pi d^3} K_s = \frac{8F}{\pi d^2} C K_s \tag{7.33}$$

$$k = \frac{d^4 G}{8D^3 N} = \frac{dG}{8NC^3} \tag{7.34}$$

$$N_t = N + 2 \tag{7.35}$$

Where N = number of active turns

N_t = total number of turns

$$L_s = N_t d \tag{7.36}$$

Design analysis

- **Spring constant**

$$F = kx \qquad \textbf{(7.37)}$$

Where *k* is the spring constant
x is the deflection
F is the applied force

Therefore, k = 2648.7/30
= 88.26 N/mm

Therefore, the required spring rate is 88.26 N/mm.

- **Clash allowance**

The clash allowance must be 10% of the maximum force applied.
Therefore,

Clash allowance = $10\% \frac{2648.7N}{88.26N/mm}$

= 3 mm

It is also important to calculate the solid force of the helical spring. The solid force can be defined as the maximum force in N that can be applied before the spring becomes solid. Therefore, the solid force is found to be:

F_{solid} = 2648.7 + 88.26 (3) N
= 2913.48 N

The spring diameter must be able to fit a hole of 40 mm and there must be clearance on the spring. Therefore, It was determined that *D* + *d* must not exceed 40 mm. The reason why clearance is needed is that the spring outside diameter has a tendency to increase due to compressive force on the spring. If D is assumed to be 34 mm, clearance is obtained. D and d parameters must be solved in order to make sure that all stress requirements are satisfied.

In order to calculate the value of d, K_s and τ_{solid} need to be evaluated first. Juvinal and Marshel (2006) K_s varies very slowly for values of C between 6 and 12. Therefore, K_s is assumed to be 1.05. The value of τ_{solid} can be obtained by using Juvinal and Marshel (2006) and by first assuming the diameter of the wire to be 2.7 mm. The corresponding ultimate stress for ASTN A229 is 1562.5 MPa. Therefore, the corresponding shear stress τ_{solid} is calculated as follows:

τ_{solid} = 0.45S_u

= 703.125 MPa

Therefore, d is calculated by using equation 7.34 as follows:

$$\tau_{solid} = \frac{8F_{solid}D}{\pi d^3}K_s$$

$$703.125 \times 10^6 = \frac{8 \times 2913.48 \times 0.034}{\pi \times d^3}1.05$$

$$d^3 = 0.000000376m$$

$$d = 0.007222m$$

$$d = 7.22mm$$

To obtain greater spring clearance, d is chosen to be equal to 7.2 mm. To avoid estimating K_s equation 7.34 is used.

Using Figure 0.6N Appendix H, the corresponding value of ultimate stress for d = 7.2 mm is found to be 1300 MPa, therefore, the corresponding τ_{solid} is found to be 585 MPa.

$$\tau_{solid} = \frac{8F_{solid}}{\pi d^2} CK_s$$

$$585 \times 10^6 = \frac{8 \times 2913.48}{\pi \times 0.00722^2} CK_s \quad \textbf{(7.38)}$$

$$CK_s = 4.11$$

In Juvinal and Marshek (2006), the value of C is estimated to be 1.11. Therefore, D can be calculated as follows:

$$C = \frac{D}{d} \quad \textbf{(7.39)}$$

$$D = 4 \times 7.22$$

$$= 29mm$$

The second trial is more accurate and has given more clearance since the initial estimate of the outer diameter was 34 mm and the required outside diameter was found to be 29 mm.

The number of turns required could be calculated by using equation 7.39. Juvinal and Marshek (2006) the tensile modulus of material to be used (ASTM 229) is found to be 79 GPa. Therefore:

$$G = 79GPa$$

$$k = \frac{d^4 G}{8D^3 N} \quad \textbf{(7.40)}$$

$$88.26 \times 10^3 = \frac{0.00722^4 \times 79 \times 10^9}{8 \times 0.029^3 N}$$

$$N = 12.47$$

Using equation 7.40, the total number of turns is calculated as follows:

N_t = 12.47 +2

= 14.47

Using equation 7.40, L_s is calculated as follows:

$L_s = N_t d$

$= 14.47(7.22)$

$= 104.47$ mm

Therefore, the free length of the spring is 105 mm. this is merely because the spring will deflect 30 mm when 2.7 kN is applied.

To verify the buckling criteria of the spring Figure 0.5N Appendix H is used because the coil springs loaded under compression act like a columns. This happens particularly for large ratios of free length to mean diameter. The following criterion is used for checking buckling on the spring.

$$\frac{\delta_{static}}{L_f} = \frac{30mm}{104.47mm} = 0.29$$

$$\frac{L_f}{D} = \frac{104.47mm}{29mm} = 3.6 \qquad \textbf{(7.41)}$$

Looking at Figure 0.6N Appendix H Juvinal and Marshek (2006), it can be seen that even though the spring end plates are constrained parallel, the spring is far from the buckling region. In other words the spring parameters are within the stable region to avoid buckling.

Finally based on the design process followed above, the spring parameters which satisfy the stress and the spring rate requirements are as follows:

$$d = 7.2mm$$
$$D = 29mm$$
$$N = 12.47$$
$$L_f = 104.5mm$$

Important note:

The spring must be manufactured within the specified limit of dimensions otherwise early failure could be caused or the applied force reduction limited.

RAIL CLAMP

The rail clamp is supposed to keep both rails together and in line. This means that rail clamps must be designed in such a way that they are very rigid. The component must be able to withstand all forces subjected to it by the tool pin. The tool pin has a tendency to try and pull apart the two rails which are to be welded. Therefore, the rail clamp must be designed in such a way that it can withstand the force exerted on it by the rotating pin. The rail clamp must be strong enough and it must not by any chance bend due to the forces applied to it. In other words the rail clamp must have minimum (zero) deflection at all times during the friction stir welding operation.

The rail clamp must also be desisigned in such a way that it is adjustable in terms of height. The rail clamp must be adjusted from 127 mm to 150 mm in height. This is mainly due to the fact that rails are of different profiles (sizes) and they tend to deform or wear during operation. Therefore, if the rail clamp has adjustable height it may be able to accommodate several rails to be welded.

Design criterion

The following assumptions are made:

- The transitional force is equal to the axial force (400kN)
- The clamping force must be at least 1.5 times the axial force to avoid slipping on the rail.
- For simplicity the rail clamp is modelled on the supported beam shown below.
- Normal steel properties are to be used in the analysis

Analysis:

The following symbols and dimensions are used for rail clamp design analysis:

$d_4 = d_2 = 50$ mm

$d_1 = d_3 = 20$ mm

$d_7 = d_8 = 28$ mm

$d_5 = d_6 = 28$ mm

$d_1 = d_3 = d$

$d_2 = d_4 = e$

Summing the forces along the x- direction (refer to Figure 33):

$$\sum F = 0$$
$$300 + 300 = F_1 + F_2 \quad \textbf{(7.42)}$$

Summing moment (clock wise direction assumed to be positive) (refer to Figure 7.8 and Figure 7.9):

$$\sum M_{F_2} = (d_1 + d_2)300 + 0.127(F_1) - (d_4 + d_3 + 127)300 = 0 \quad \textbf{(7.43)}$$

Therefore, solving for F_1

$$F_1 = \frac{(d_1 + d_2)300 - (d_4 + d_3 + 0.127)300}{0.127}$$
$$F_1 = \frac{300}{0.15}[0.07 - (0.07 + 0.15)] \quad \textbf{(7.44)}$$
$$F_1 = 300kN$$

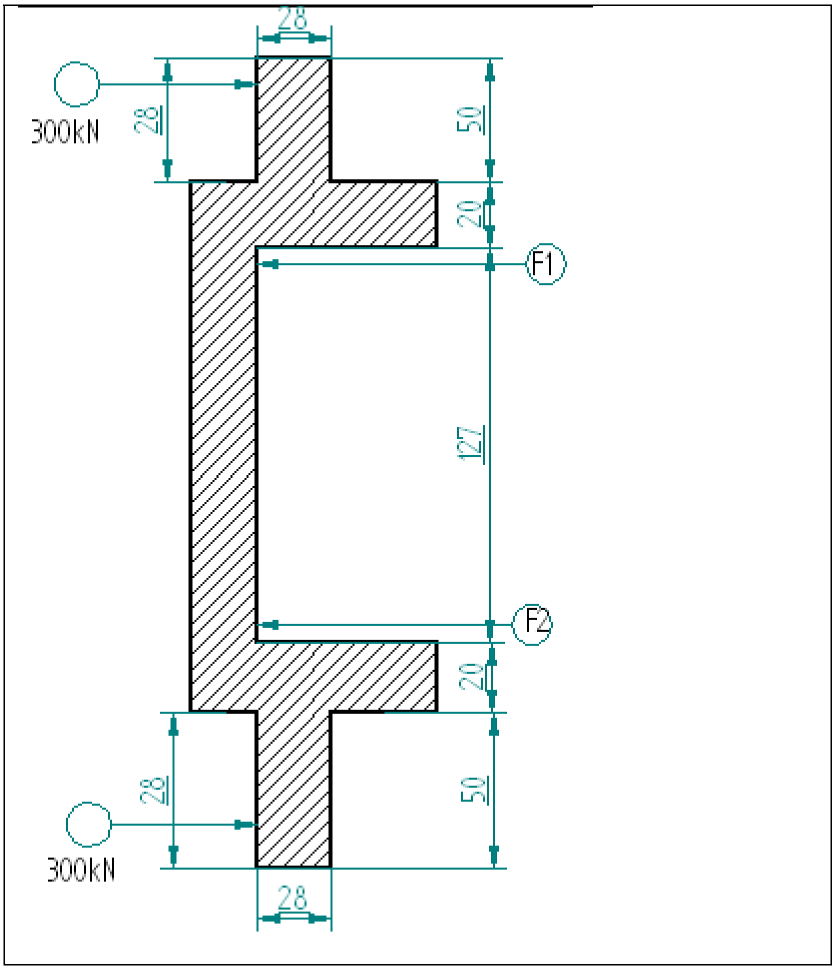


Figure 7.8: Forces assumed to be applied on the rail clamp

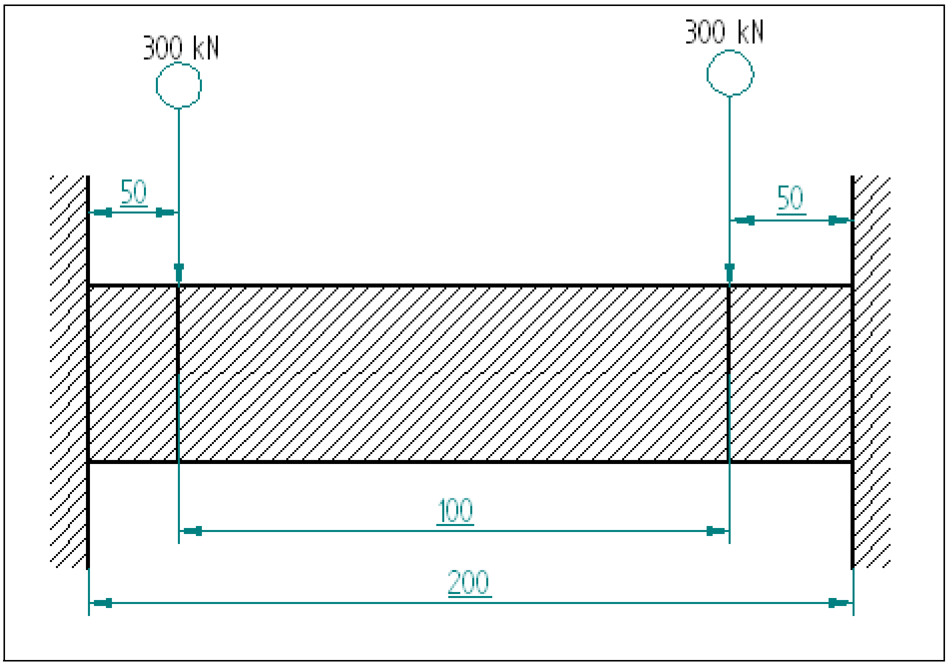


Figure 7.9: Simplified rail model

Figure 7.10 shows the rail clamp used to tighten two rails. It is assumed that the force F1 and F2 are applied due to the reaction forces of the rail flange. Figure 7.11 is a simplified representation of Figure 7.10. It has been assumed that the rail clamp is fixed at both ends by means of bolting. Figure 7.10 shows the worse case scenario of loading application.

In order to obtain the minimum thickness required on the plate, mild steel is assumed to be the selected material to be used. The scenario in Figure 7.10 shows the two equal forces applied 50 mm away from the ends of the plate. For simplicity, it is assumed that the sum of the two forces (600 kN) is applied to the centre of the beam. This is obviously the worse case scenario. This is to ensure that the well known standard for the beam bending theory is used.

Figure 7.12, shows the worse case scenario for Figure 7.11. The worse case scenario was only assumed to simplify the analysis. It is the worse case scenario because the load is applied at the centre of the beam instead of the two loads applied at the far end. When two forces are applied to the far end of the beam, they do not bend the beam much as compared to the forces applied at the centre. Greater deflection is expected when a higher force is applied to the beam at the centre. This will ensure that standard tables from reliable sources are used with confidence.

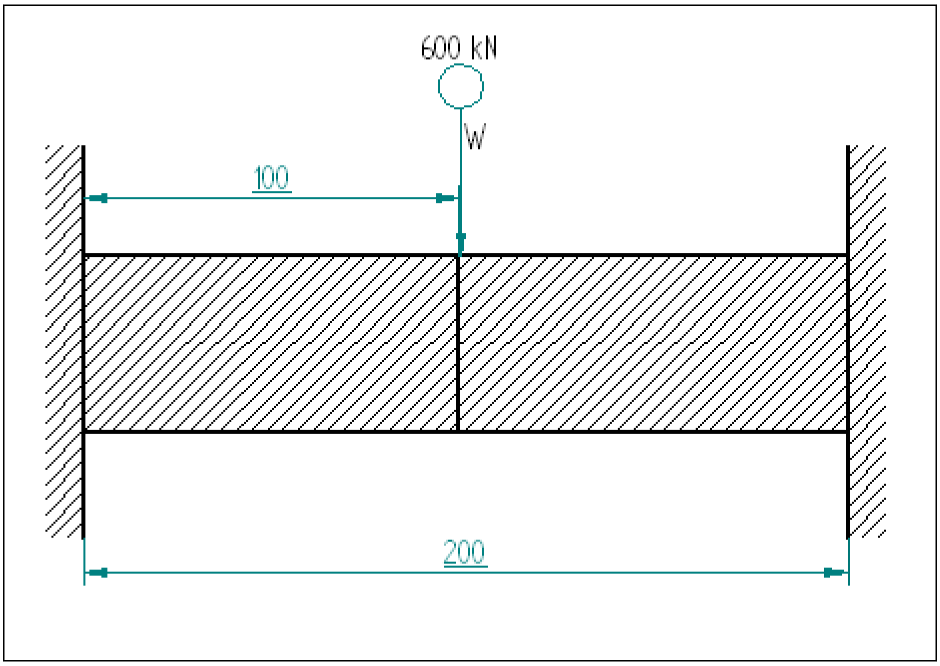


Figure 7.10: Worse case scenario

In the worse case scenario,

Maximum deflection is found by using the following relationship:

$$f = \frac{W}{EI}\frac{l^3}{192}(\max) \quad \textbf{(7.45)}$$

$$M_x = \frac{Wl}{8}(\max) \quad \textbf{(7.46)}$$

The beam must have zero or nearly zero deflection. Therefore, it is fair to assume that the maximum deflection allowed on the beam is 0.0001 m (nearly zero)

By using equation 7.45, the following results are obtained:

$$0.0001 = \frac{600 \times 10^3 (0.2)^3}{200 \times 10^9 I (192)} \quad \textbf{(7.47)}$$
$$I = 1.25 \times 10^{-6} m^4$$

I is the moment of inertia and is simply calculated as follows:

$$I = \frac{1}{12}bh^3 \quad \textbf{(7.48)}$$

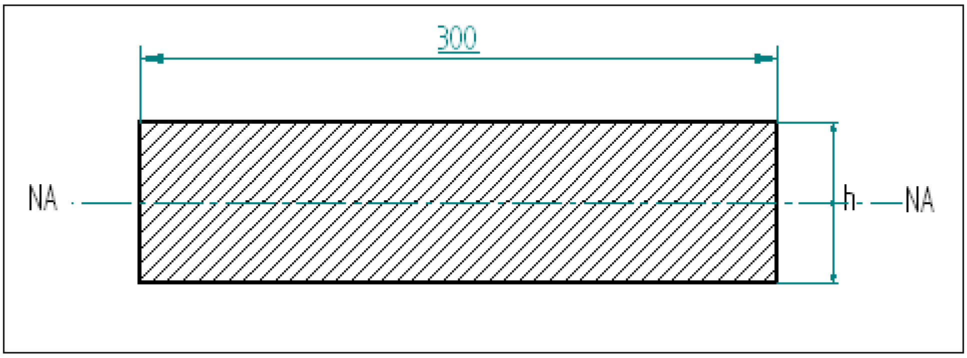


Figure 7.11: Front cross-section of the rail clamp

By equating equation 7.47 and 7.48 and assuming that the value of *b* is equal to 300 mm, the following is obtained. (Refer to Figure 4.13):

$$h^3 = 0.00005$$

Therefore,

$$h = 0.03684m$$
$$h = 36.84mm$$

Therefore, the minimum thickness required on the rail clamp is 38 mm.

Bolt selection:

The rail clamp is subjected to heavy forces and is expected to hold two rails together. Therefore, the number of bolts required to hold the two rails must be selected. Initially it was assumed that the number of bolts required on the top and bottom of rail ramp was three on each side. Therefore, three bolts at the top and three are the bottom. The side view of the rail clamp is shown on the Figure 7.12:

Using Figure 7.12, the tensile force acting on the holes is calculated as follows;

$$F_A = \frac{150}{3}$$
$$= 50kN$$

$$F_B = \frac{150}{3}$$
$$= 50kN$$

For design overload:

$$6(300) = 300kN$$

Therefore,

$$A_t = \frac{F}{S_p} \tag{7.49}$$

Consequently,

$$A_t = \frac{300}{970}$$
$$A_t = 309.27mm^2 \tag{7.50}$$

Therefore, M24x3 was chosen from Shigley et al, (2004) Table 0.2N, and Appendix H. The proof strength of 970 MPa was chosen from Table 0.2N fromin Appendix H. In addition, M24 bolt is the minimum diameter which can be used to clamp rails together. For safety reason M30x3.5 was chosen.

Specifications for bolt to be used is as follows:

Table 7.3: Bolt material and geometry specification

Material	**Diameter**	**Proof load stress**	**Yield strength**	**Tensile strength**	**Core Hardness Rockwell**	**Pitch**	**Minor diameter**
SAE Class 12.9	1.6 through 36	970 MPa	1100 MPa	1220 MPa	C38 (max) and C44 (min)	3.5 mm	28.7 mm

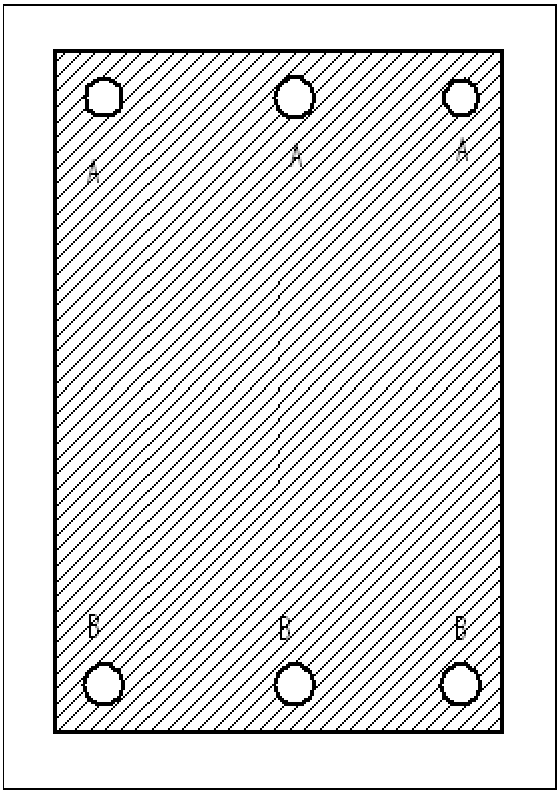


Figure 7.12: Side view of rail clamp

MOTOR SUPPORT BRACKET

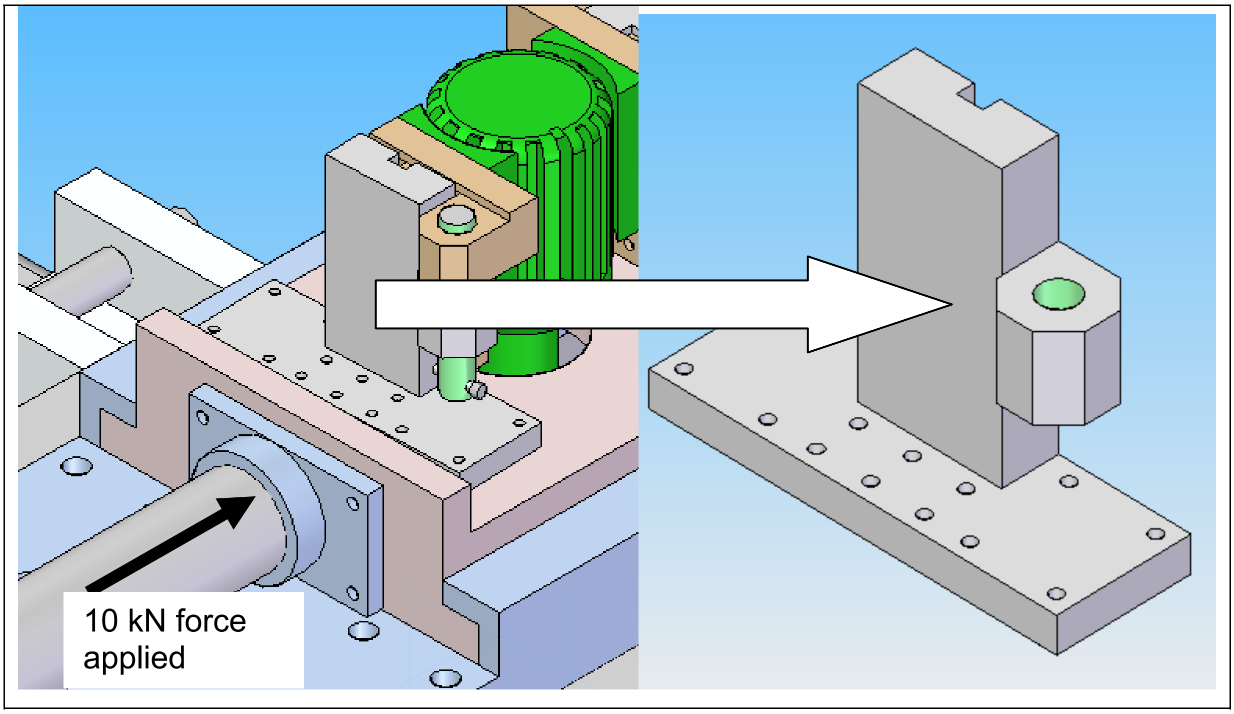


Figure 7.13: Assembly of motor support bracket to be designed

The motor support bracket must be designed in such a way that there is zero deflection because a force of reasonable magnitude is to be applied by the actuator or motor coupled with gears to make sure that enough friction is generated by the tool pin coupled to the top motor. Therefore, the top motor structure must be very rigid, otherwise the principle of friction stir welding will not succeed.

Initially it was assumed that the rail material was at room temperature and more force is needed to generate enough friction or heat to begin the welding process. For the worst case scenario it is assumed that the force to be applied to the motor support is equivalent to that applied by the tool pin axial. Mishra and Ma (2005) it have shown that for a plate thickness of 70 mm the force needed to be applied is about 10 kN but the force applied depends in various factors, for example: material properties of plate or piece to be friction welded, quality of weld required etc.

Based on the findings of Mishra and Ma (2005) it is safe to assume that the force to be applied to the motor support is 18 kN. The minimum thickness of rail to be welded is 127 mm and the maximum thickness is 150 mm. It can also be assumed that the safety factor to be used is 1.5. As shown in Figure 7.13, the force F is applied by the actuator and the reaction forces are due to the motor support, therefore, the motor support bracket is modelled as a Cantilever beam fully fixed at one end and free at the other. Figure 7.13 is simplified and represented in Figure 7.14.

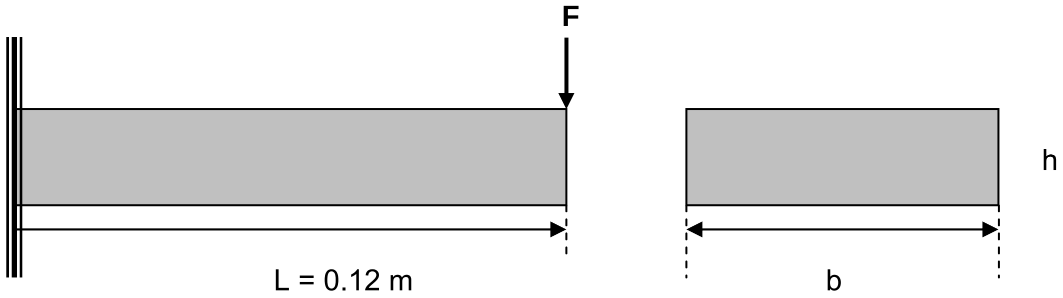


Figure 7.14: Model of motor support bracket as a Cantilever beam

Based on the requirements of the motor support bracket discussed above, the bracket must be designed in such a way that there is no deflection on it as the load is applied. Therefore, the deflection of beam formula must be utilised in order to optimise the geometry of the motor support bracket.

Refering to Juvinal and Marshek (2006), the following formulas are to be utilised.

$$\theta = \frac{FL^2}{2EI} \qquad \textbf{(7.51)}$$

$$\delta_{\max} = \frac{FL^3}{3EI} \qquad \textbf{(7.52)}$$

$$\delta = \frac{Fx^2}{6EI}(3L - x) \qquad \textbf{(7.53)}$$

Where F is the force applied on the beam, E is the elastic modulus of beam material and I is the moment of inertia.

Two motor support brackets are used to support and fix the motor in position. Therefore, the reaction forces are to be shared between the two brackets. Due to symmetry, the half model must be used that is: the design calculation will only be done on one motor support bracket. A force of 40 kN is assumed to be applied on both the support brackets, therefore, each one will carry 20 kN force.

For simplicity, it is assumed that the maximum deflection allowed when force F is applied to the beam is 0.0000001 m. Therefore, equation 7.53 is used to calculate the required geometry such that the maximum deflection is 1 mm. The length of the beam is assumed to be 120 mm. By using equation 7.53, the following is found.

$$0.0000001 = \frac{30\times10^3\times0.12}{3\times200\times10^9\times\frac{1}{12}\times bh^3} \quad \textbf{(7.54)}$$

By assuming ***b*** to vary from 26 mm to 126 mm and solving ***h*** from equation 7.54 above, the following results tabulated below yield.

Table 7.4: Optimum shape for 1 mm deflection of a bracket

***b* (m)**	***h***	***h* (m)**	***h* (mm)**
0.026	0.001246154	0.107611	107.6111
0.036	0.0009	0.096549	96.54894
0.046	0.000704348	0.088974	88.97385
0.056	0.000578571	0.083327	83.32698
0.066	0.000490909	0.078886	78.88608
0.076	0.000426316	0.075262	75.26224
0.086	0.000376744	0.072224	72.22411
0.096	0.0003375	0.069624	69.62383
0.106	0.00030566	0.067362	67.3617
0.116	0.00027931	0.065368	65.36757
0.126	0.000257143	0.06359	63.59039

From Table 7.4, the optimum cross-section for a motor support bracket is a 126×64 mm plate. This is based on the assumption that the beam is allowed to deflect 1 mm.

Also, there is a need to perform bending moment calculations to show the maximum stress on the plate. For a simple stress analysis, the following assumptions are made:

- The beam is initially straight

- The beam is loaded in a plane
- The shear stress in the beam is uniform across the beam width at each location from the neutral.

Refering to Juvinal and Marshek (2006), the maximum stress can be calculated from the following formula.

$$\sigma = \frac{My}{I} \qquad \textbf{(7.55)}$$

$$\sigma_{\max} = \frac{Mc}{I} \qquad \textbf{(7.56)}$$

Where *M* represents the moment applied on the beam and ***c*** is the distance from the neutral axis to the surface of the beam.

From Figure 7.15, M is calculated to be 3.24 kN.m and c is equal to 0.032 m.
Therefore, from equation 7.56, the maximum stress on the beam is calculated as follows:

$$\sigma_{\max} = \frac{3.24 \times 10^3 \times 0.032}{\frac{1}{12} \times 0.126 \times 0.064^3}$$

= 37.67 MPa

The maximum stress is way below the yield stress of carbon steel 1020 HR of 290 MPa (Juvinal and Marshek, 2006).

Furthermore, a fatigue analysis needs to be done in order to verify how variable loads can affect the structure. The applied stress is about eight times the yield stress, therefore from Boldea and Nasar (2002) suggests that if that is the case it is not necessary to perform a fatigue analysis.

Bolts and nuts analysis

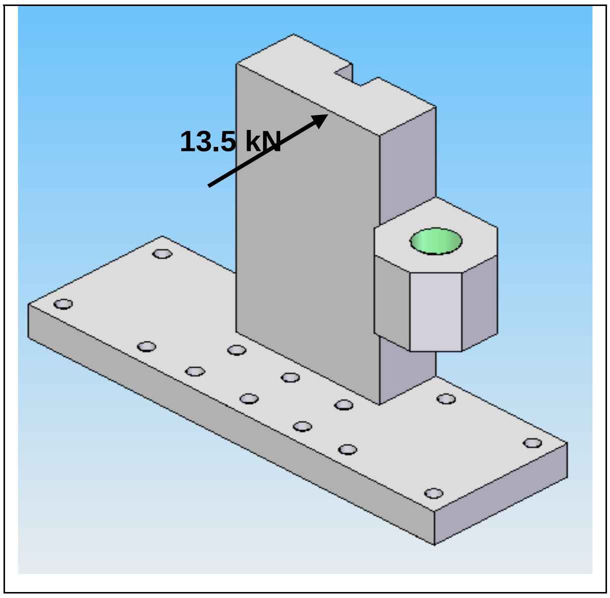


Figure 7.15: selecting bolts for motor support bracket

For mounting the motor support bracket, bolts and nuts are to be used. Therefore, there is a need to perform basic design calculations to determine the sizes as well as the number of bolts needed in order to withstand the force applied. In order to fully understand the situation of design, the following assumptions are made:

(i) There is no deflection on clamped members
(ii) The shear load is carried by friction
(iii) There are eight bolts to be used to clamp the bracket
(iv) A safety factor or design overload of six (6)

Summation of moments about point G (the fixed point on the bracket as shown in Figure 7.15), gives the following results:

$$81(200) = 2F_D(20) + 2F_E(80) + 2F_B(266) + 2F_A(326)$$

Assuming that $F_A = F_B = F_C = F_D$

The above equation yields the following results,

F_A = 11.70 kN

As shown in Table 0.3N, Appendix H, class SAE 4.6 has a proof strength of 225 MPa. Hence the required tensile stress area is

$$A_t = \frac{11.70}{225} \qquad \textbf{(7.57)}$$
$$= 52mm^2$$

By referring to Table 0.2N, Appendix H, the required thread size is to be M10×1.5

In summary eight bolts of diameter 10mm and pitch 1.5 mm must be used to hold the motor support bracket. Bolts must be manufactured using class SAE 4.6 steel material. See Appendix H, Table 0.1N for Table of materials that must be used.

LIFTING MECHANISM

The lifting mechanism is used for lifting and adjusting the height of the motor which is coupled to a tool pin. The lifting mechanism must not be moved and it must be self locking and require a lower force to lift the motor. In addition, the motor must be able to be operated by a single person. Therefore, a power screw is necessary for lifting and adjusting the motor height.

The following assumptions are made in the design of the power screw for lifting or adjusting the motor height.

(i) The starting and running friction remain steady
(ii) Starting friction is about one-third higher than running friction
(iii) The force or torque to be applied on the power screw by a person is known

The following formulas are to be used in the design process of a power screw:

$$T = \frac{Wd_m}{2}\frac{f\pi d_m + L\cos\alpha_n}{\pi d_m - fL} + \frac{Wf_c d_c}{2} \qquad \textbf{(7.58)}$$

It is vital to design a efficient power screw which requires a low force to lift the load (weight of the motor). This is done to make sure that the force applied on the screw is properly utilized. Efficiency is simply defined as the ratio of work input to that of work output. The work input for a power screw is defined as $2\pi T$ and the work output is defined as WL (force times distance). Therefore, the efficiency is defined as follows, ignoring the friction.

$$e = \frac{L}{\pi d_m} \frac{\pi d_m \cos\alpha_n - fL}{\pi f d_m + L\cos\alpha_n} \qquad \textbf{(7.59)}$$

$$T = Fa \qquad \textbf{(7.60)}$$

$$\tan\lambda = \frac{L}{\pi d_m} \qquad \textbf{(7.61)}$$

Where:

L – Lead
d_n – Mean diameter of thread contact
T – Torque
W – Load (weight) of the object to be lifted
d_c – Ball thrust bearing diameter
f_c – Bearing friction
f – Friction of power screw

Design calculations

Basic power screw dimension

The aim of this section is to design a basic power screw geometry such as lead (L), pitch (p), thread depth, mean pitch diameter, helix angle and the efficiency of jack during the lowering and raising of the load.

Assume that the force to be applied to the mechanism to lift the motor is 20 N and the arm is approximately 0.15 m. Using equation 7.60, the torque to be applied to the lift arm is calculated to be 3 N.m. The weight of the motor is assumed to be 20 kg. The load to be lifted is approximately 196.2 N (the value was calculated by assuming the gravity acceleration of 9.81 m/s^2).

In design calculation the following parameters are made based on the following assumptions:

- The running friction does not change with time
- The initial friction does not change with time
- Initial friction is one third greater than the running friction

  - f_c = 0.07
  - f = 0.11
  - d_c = 21 mm

In Table 0.4N Appendix H, Juvinal and Marshek (2006) for the diameter of 16 mm, the corresponding *p* is 2. Therefore:

p =2 and

L = 2*p*
  = 4 mm

Thread depth = *p*/2

= 1 mm

D_m = *d*-*p*/2

= 16-1

= 15 mm

$$\lambda = \tan^{-1}\left(\frac{4}{\pi \times 15}\right)$$
$$= 4.851^{\circ}$$

$$\alpha_n = \tan^{-1}(\tan\alpha\cos\lambda)$$
$$= \tan^{-1}(\tan 14.5^{\circ}\cos 4.851^{\circ})$$
$$= 14.45^{\circ}$$

To calculate the torque needed to apply the motor of 20 kg weight, equation 7.60 is used:

$$T = \frac{20\times 9.81\times 0.015}{2}\frac{0.11\times\pi\times 0.015 + 0.004\cos 14.45^{\circ}}{\pi\times 0.015 - 0.11\times 0.004} + \frac{20\times 9.81\times 0.021\times 0.07}{2}$$
$$= 0.42969 N.m$$

If the arm length is assumed to be 70 mm, then the force required to lift the motor up is calculated to be;

T = Fxa

Therefore, F =0.42969/0.07

= 6.1384 N

Consequently, the force required to lift the motor upwards is approximately 6.14 N. The force is very low and only one operator is able to perform. This is one of the most important requirements in the design of the lifting mechanism of rail friction stir welding machine.

Power screw subjected stresses

Torsional stresses

The power screw and threaded fasteners are normally subjected to torsional stress during operating conditions. Torsional stress in a power screw is represented by the following relationship:

$$\tau = \frac{Tc}{J} = \frac{16T}{\pi d^3} \tag{7.62}$$

Where d is the thread root diameter and T is the torque applied on the power screw. Using the values obtained above and equation 7.62, the torsional stress is found to be:

$$\tau = \frac{16 \times 0.4297}{\pi \times 0.015^3}$$
$$= 0.684 MPa$$

From the above calculation, the torsional stress acting on steel material could be converted to a normal stress by a factor of 0.577 Boldea and Nasar (2002). Therefore, the ultimate stress is found to be 0.3741 MPa. The stress applied on the power screw is very low and as a result, normal mild steel may be used in building the power screw.

Axial load

The axial load applied to the power screw is the weight of the motor times the acceleration due to gravity, therefore, the load applied to it is 196.2 N. By using the stress formula; the following normal stress is applied to the power screw.

$$\sigma_{applied} = \frac{F}{A}$$

$$\sigma_{applied} = \frac{196.2}{\pi \times 0.015}$$
$$= 0.2776 MPa$$

Again, the applied normal stress is very low and as a result steel carbon 1002 A[b] of yield stress 131 MPa Juvinal and Marshek (2006) may be used to build the power screw.

If the above mentioned material is used, the safety factor based on yield strength is calculated to be 471. So, this material is safer to use in building a power screw.

APPENDIX C: RAIL WELDING CONDITIONS MONITORING

C.1 CONDITIONS MONITORING FOR WELDING RAILS

INSPECTION AND DECISION MAKING

The exothermic welding process is a unique process whereby two rails are welded together. It must be noted that most exothermic weld portions are found in rails in the South African environment. When there is a need for exothermic welding the following inspections must be done in order to maximise the quality of exothermic welds. This type of welding can be used for maintenance purposes or for joining rails after derailment.

The following steps are to be performed by qualified personnel before welding the rail:

- The type of rails must be checked very carefully as some of the rails cannot be joined by this type of welding. In the South African railway environment all chrome manganese blocks are normally marked with aluminium or silver paint. The aluminium paint is used only to maximise or to facilitate identification.
- All joints to be exothermic welded are normally checked for alignment to see to it that both rails are correctly aligned before welding. Otherwise poor alignment could cause train derailment.
- A minimum of 2 meters must be maintained between an obtuse weld and an exothermic weld.
- The permanent closure rail must fit as close to the existing rail as possible not differing by more than 3 mm in rail level. If the height exceeds the minimum 3mm level, then a step mould must be used. The rail must preferably have no fisher-bolt holes and must be at least 4.2 meters long and no casting may take place if more than 6 mm difference occurs.
- When it is necessary to insert a permanent closure rail curve where side wear prevails, the total length of the defect in the curve concerned must be cut out and the rails pulled up so that the permanent closure rail may be welded onto the straight portion of the track where no side wear prevails.

- Where exothermic welds are welded onto existing long-welded rails, the fasteners or sleepers, 80 on both sides of the joint, must be loosened for distressing.

TESTS FOR ACCEPTANCE/ LABORATORY TESTS

In order to make sure that exothermic welds are in line with the requirements, a number of tests are performed. British standard BS EN 14730-1:2006 has outlined all tests required for quality assurance purposes or for going on research purposes. Therefore, the following takes are conducted:

- Visual surface examination

On the cast welded surface the following should be checked in order to maximize the quality of exothermic welds. This test is used to make sure that there is no crack which is longer than or equal to 2 mm. The joints between weld collars and rail as well as flashing and rail must not have cracks. Pores with the dimensions of more than 3 mm should not be allowed. The ground weld surface must also be checked to make sure that there are no visible cracks or defects like pores, slag, sand inclusions, metal beads etc. Visible heat affected zones must also be checked as their widths are not allowed to be more than 20 mm.

- Running surface hardness test

The aim of this test is to make sure that weld hardness is within the required tolerance or value. It must be noted that the hardness value range depends mostly on the type of rail welded. The table below shows how hardness varies with rail grade. Table 0.1M shows the hardness range (HBW) of different rail grades:

Table 0.1M: Hardness value for various rail grades.

Rail grade	Weld range (HBW)	
	Rail running on the unaffected parent rail	Weld center-line
R200 R220 R260 R260 R260Mn R320Cr R350HT R350LHT	200 to 240 220 to 260 260 to 300 260 to 300 260 to 300 320 to 360 360 to 390 360 to 390	230 ± 20 250 ± 20 580 ± 20 300 ± 20 280 ± 20 330 ± 20 350 ± 20 350 ± 20
NOTE: 0.5 mm should be ground from the running surface before a hardness impression is made		

- Slow blend tests

The main aim of this test is to make sure that the rails joined by the exothermic welding process can withstand the lateral forces imposed by a moving train along the rail. The minimum fracture load in kN to be applied on the rail when testing is 5 kN. The force is simply defined as F = 0.00032.S where S is the section modulus for the base of the rail.

- Internal examination

In this type of test weld soundness of the head, web and foot of the rail containing weld is examined ultrasonically. This is usually done by sectioning the rail in all directions. The Figure 0.2M (British Standard, BS EN 14730-1:2006) shows how the rail containing the weld should be sectioned. The maximum dimension of any pores, slag inclusions, sand inclusions or metal beads are recorded. Fusion zones and heat affected zone are examined and recorded in the Table 0.1M:

Table 0.2M: Ranges of heat affected zone

Less than or equal to	Heat treated rail	Non-heat treated rail
20 mm	x	x
30 mm	x	x
40 mm	x	x
50 mm	x	-
60 mm	x	-

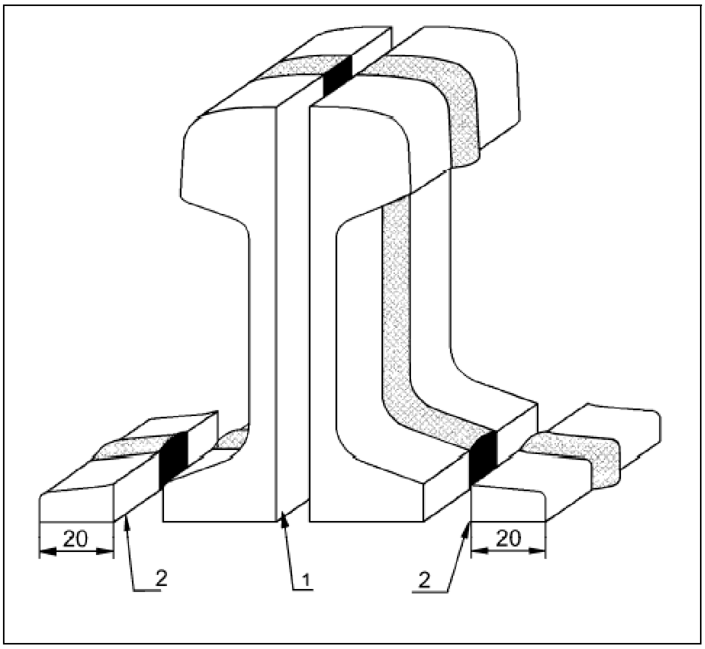


Figure 0.1M: Sectioning of rail containing weld

- Fatigue test

The loading variation is due to train movement bumps or due to normal variation forces. Therefore, all rails are subjected to fatigue forces in operation. It is very important also to check whether rail can survive the fatigue stresses due to force variation. Fatigue strength and standard deviation of welds on rails is normally specified by railway authorities. In this test all the unbroken rails are broken by slow bending test in order to examine the fracture faces.

- Chemical analysis

Chemical analysis is performed in order to make sure that the chemical composition of the weld does not deviate very much from the parent rail chemical composition. The following chemical contents are usually found in rails: namely: Carbon, Silicon, Manganese, Phosphorus, Sulphur, Chromium, Nickel, Aluminium, Copper, Tin, Antimony, Titanium, Niobium and Vanadium.

Inspection and decision making

Inspection is done according to prescribed specifications.

Approval tests

2. Visual inspection

The aim of this test is to make sure that all welding parameters are visually inspected before any test is done on the welds. The parameters to be visually inspected include: welding, trimming, pressing, clamping or profile finishing imperfections, such as tears, cavities, cracks, damage, geometrical non-conformities, and thermal damage in particular in the electrode contact areas.

3. Weld trimming

This test is aims to prevent poor trimming processes which could cause cracks along the rail length. Attention is mostly given to the quality of trimming on the undesirable quality of the rail foot.

The following tests are also carried out to make sure that the quality of weld is of a high standard.

- Weld straightness and flatness
- magnetic particles or dye penetrant inspection
- bend testing
- macro examination
- micro examination
- hardness testing
- fatigue testing

EXOTHERMIC WELDING AS APPLIED IN RAILWAY

The exothermic welding technique had little competition until about 1989. According to Norfolk Southern exothermic is not considered as sentimental in the railway industry but is considered as “necessary evil” (Stump, 1998). According to Stump (1998), Burlington Northern has listed exothermic welding technique as the best when it comes to rail defects. As stated, exothermic welds are considered to be extremely difficult to produce good quality weld.

During the introduction of the electric flash-butt welding technique, manufacturers strengthen their research and development teams in order to ensure a high improvement in their product. The basic problem with the exothermic welding technique is that it involves human skills and it can be applied only in a particular site. The manufacturers of exothermic welding tools have recently claimed that their process involves less human interference. Very soon and not later, exothermic welding tools or manufacturers could be out of business since most projects are excluding them (Stump, 1998).

Figure 0.2M: Example of exothermic welding applied on rail

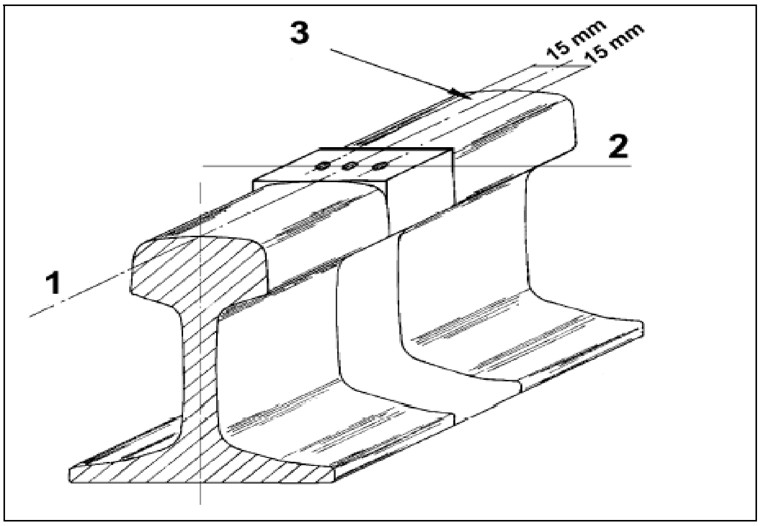


Figure 0.3M: Example of exothermic welding of rails

Figure 0.3M (Holland Engineering rail Solutions website) shows the jig used to perform the exothermic welding. Similarly, Figure 0.4M (British Standard, BS EN 14730-1:2006) shows two rails joined by means of exothermic welding. It can be seen from the latter figure that when two rails are welded together by means of exothermic welding, more work is still required to cut out the edges. This is to make sure that the rail has the required profile.

ADVANTAGES AND DISADVANTAGES OF ARC (ELECTRODE) WELDING AS APPLIED IN RAILWAY

As mentioned above, the electrode welding as applied in the railway industry yields very good material properties. It is very easy to produce good fatigue material properties by using this technique. However this technique is not very good when it comes to bending and static load properties. The main advantage of this technique is that it yields high quality welds. It is very expensive by nature as it requires a highly specialized operator.

COMPARISON BETWEEN FRICTION STIR WELDING AND OTHER WELDING METHODS USED TO JOIN RAILS

In this section three types of welding used for joining rails and the proposed friction stir welding is outlined. The aim of this section is to give an overall picture of the processes used for joining rails.

Table 0.3M: Experimental Values for Standard Rail Welds.

Test			Welding Method		
			JNR Gas Pressure Welding	**Flash-Butt Welding**	**Thermite Welding**
Fatigue strength (kgf/mm)	One type rotation bending		29-31	23-28	27-31
	Bending (actual rail)		34	29.5-34	18-22
Static bending test	Maximum bending load (t)	Head up Head Down	121-137 118-131	116-139 99-118	99-110 88-99
	Deflection (mm)	Head up Head Down	25-84 23-90	30-97 13-63	17-23 11-18
Drop Weight test	Height (m)	Head up Head Down	2-3.5	1.5-5 3-8(5)	2.5-4
	Deflection (mm)	Head up Head Down	15-53	7-69 7-57	4-9.4

Stump (1998), was able to determine the weld strength of the rail welding processes used in the field, namely: Exothermic, Flash-butt and Arc (gas) welding for three different test. The following tests were performed on a standard rail in order to compare the weld strength for different tests, namely; Fatigue test, static bending test and drop weight (impact tests). Table 0.1M was found from Stump, (1998).

It can be seen from Table 0.1M, the JNR Gas pressure welding technique has the highest fatigue properties followed by the exothermic weld properties. The flash-butt welding technique yields the highest bending resistance followed by the JNR Gas pressure welding. Flash-butt welding also yields the highest impact resistance followed by the JNR Gas pressure welding. Even though the exothermic welding process is thought to be the best from a structural point of view the results yielded are worse as compared to the other two techniques. Based on the information in Table 0.2M, it cab be concluded that flash-butt welding yields the best material properties as compared to the other techniques (Stump, 1998).

Figure 0.4M: Gap under straight and jointed rails.

APPENDIX D: RAIL SPECIFICATIONS

D. 1 IDENTIFICATION OF RAIL TYPES FOR WELDING PURPOSES

Tabel 3.1: Roll marks

IDENTIFICATION OF RAIL TYPES FOR WELDING PURPOSES								
No.	ROLL MARKS							
	H.C.O.B							
1	←—	NOM MASS		▽			19—	SAS
2	←—	NOM MASS	2MCC	▽		9	19—	SAR
3	←—	NOM MASS	⊔	▽	±	9	19—	SAS
4	←—	NOM MASS	⊟	▽		9	19—	SAR
5	←—	NOM MASS	2MCC	▽	±	9	19—	SAS
6	←—	NOM MASS	2MCC	▽	Z	9	19—	SAS
7	←—	NOM MASS	⊏⊐	▽	Z	9	19—	SAR
8	←—	NOM MASS		▽	Z	9	19—	SAS
	UIC A							
9	←—	NOM MASS	═	▽		9	19—	SAR
10	←—	NOM MASS	═	▽	±	9	19—	SAS
	UIC B							
11	←—	NOM MASS	═	▽		9	19—	SAR
	UIC C							
12	←—	NOM MASS	2MCC	▽		9	19—	SAS
13	←—	NOM MASS	2MCC	▽		9	19—	SAR
14	←—	NOM MASS	═	▽		9	19—	SAS
	Cr - Mn							
15	←—	NOM MASS	═	▽	9	19—	SAR	
16	←—	NOM MASS	≡	KRUPP		19—	SAR	
	*HEAD HARDENED /KROON VERHARD (Post-2000 supply)							
17	DO	═ —	04	SAR57				
	*HEAD HARDENED /KROON VERHARD (Pre-2000 supply onCoal line only)							
18	DO	96	1X	UIC 60	═ —			
19	THYSSEN			UIC 60	HH	350		
20	THYSSEN			UIC 60	HHLA	350		
21	HY	96	X	UIC 60	□═ I			
22	HY	96	X	UIC 60	□═ I			
23	UIC 60	LDVT	IIIIIIII					
24	UIC 60	LD	NKK	THH	TH34**			
25	UIC 60	LD	NKK	THH	TH37**			

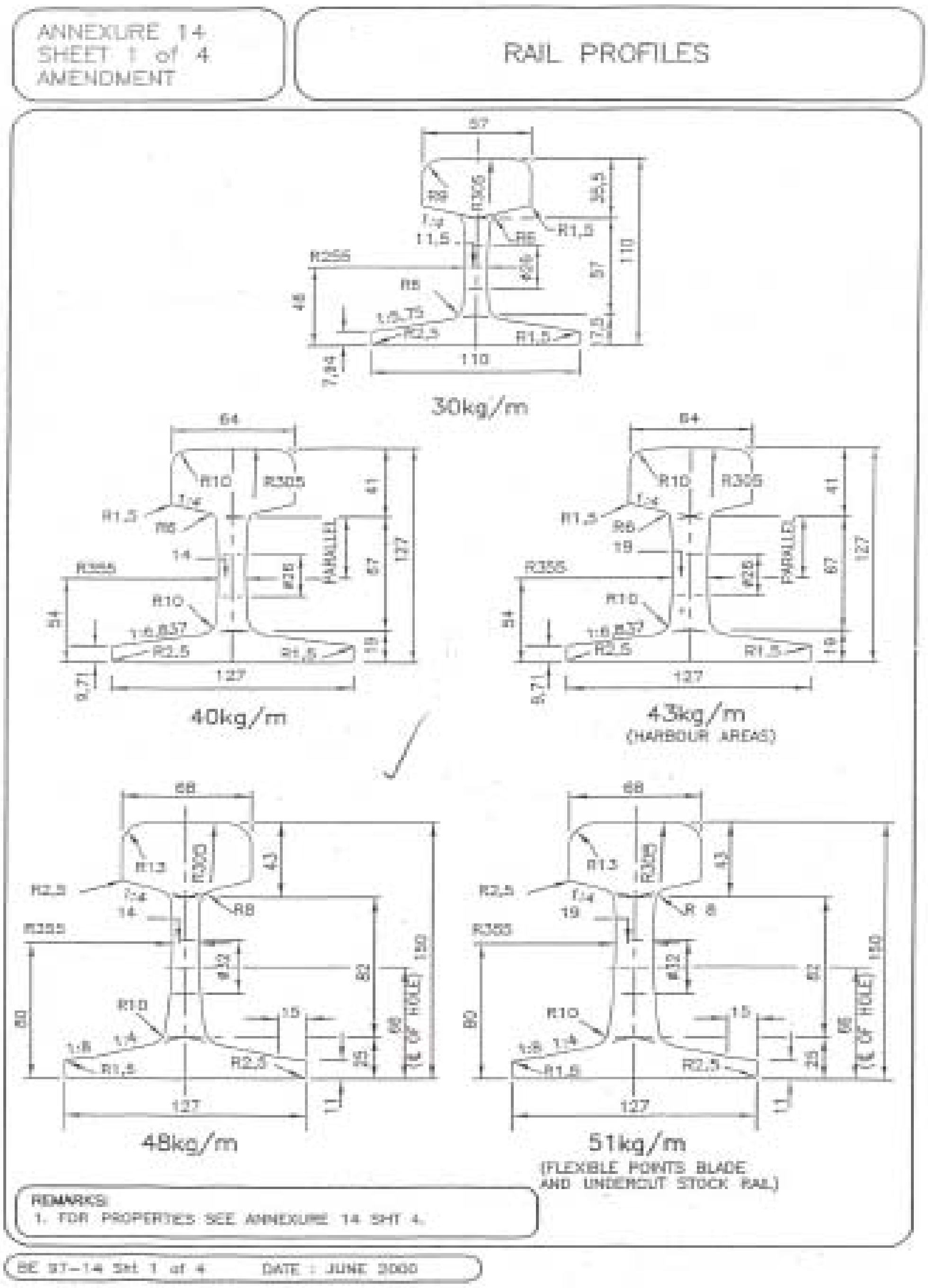
* Spoornet identification may not be present on the head hardened rails/
Spoornet identifikasie mag dalk nie verskyn op die kroon verharde stawe nie.
** Hot stamped on other side/Warm gestempel aan teenoorgestelde kant

□ Siemens-Martin (basic) process

□ Electric process

□ Steel refined by Oxygen-bloom process

D.2 RAIL PROFILES



APPENDIX E: FSW TECHNICAL PERFORMANCE

E.1 LOADING REQUIREMENTS FOR AVAILABLE MATERIALS

Table 0.1C: A summary of grain size in nugget zone of FSW aluminium alloys.

Material	Plate thickness (mm)	Tool geometry	Rotation rate (rpm)	Transverse speed	Grain size (micro mm)
7075Al-T6	6.35			127	2-4
6061Al-T6	6.3	Cylindrical	300-1000	90-150	10
Al-Li-Cu	7.6				9
7075Al-T651	6.35	Threaded, cylindrical	350,400	102, 152	3.8, 7.5
6063Al-T4,T5	4		360	800-2450	5.9-17.8
6013Al-T4,T6	4		1400	400-450	10-15
1100Al	6	Cylindrical	400	60	4
5054Al	6				6
1080Al-O	4				20
5083Al-O	6				4
2017Al-T6	3	Threaded, cylindrical	1250	60	9-10
2095Al	1.6		1000	126-252	1.6
Al-Cu-Mg-Ag-T6	4		850	75	5
2024Al-T351	6			80	2-3
7010Al-T7651	6.35		180, 450	95	1.7, 6
7050Al-T651	6.35		350	15	1-4
Al-4Mg-1Zr	10	Threaded, cylindrical	350	102	1.5
2024Al	6.35	Threaded, cylindrical	200-300	25.4	2.0-3.9
7475Al	6.35				2.2
5083Al	6.35	Threaded, cylindrical	400	25.4	6.0
2519Al-T87	25.4		275	101.6	2-12

Table 0.2C: A summary of FSW of copper alloys.

Materials	Plate thickness (mm)	Tool materials	Rotation rate (rpm)	Transverse speed (mm/min)
Pure copper	3.0	Tool steel		
Pure copper	10-25	Sintered tungsten-based alloy		
Pure copper	10-50	High-temperature materials with specific geometry design		
Oxygen-free copper	1.5-5.0	Sintered carbide ISO K40UF (WC-Co), Ni-based superalloy (Inconel 718), Cr-Mo-V type hot worl tool steel	375-1250	250-400
Pure copper	4		1250	61
60/40 brass	2		250-1500	500-2000

Table 0.3C: FSW parameters and tool materials for FSW of steels.

Materials to be welded	Plate thickness (mm)	Tool rotation rate (rpm)	Tool transverse speed (mm/min)	Tool materials
12% Cr steel	12		240	-
Low carbon steel	12, 15		102	-
AISI 1010	6.4	450-650	25-102	Mo and W-based alloys
304L	3.2, 6.4	300, 500	102	W alloy
304	6.0	550	78	Polycrystalline cubic boron
304L, 316L	5, 10	300-700	150, 180	
Al 6XN	6.4, 12.7		102	W alloy
HSLA-65	6.4, 12.7	400-450	99-120	W
DH-36	6.4		102-457	W alloy
C-Mn	6.4			Polycrystalline cubic boron nitride

Table 0.4C: FSW parameters and tool geometries for FSW of magnesium alloys and resultant grain sizes in stirred zone.

Materials/plate thickness (mm)	Tool geometry	Tool rotation rate (rpm)	Tool traverse speed (mm/min)	Grain size in stirred zone (micro meters)
Cast AM50, AM50, AZ91/6	Plain threaded pin or MX Triflute	250-500	160-450	
Wrought AZ31/6.4	Plain threaded pin or MX Triflute	250-500	160-450	
Thixomolded AZ91D/2	Screw pin	880-1750	50-500	2-5
Wrought AZ31B-H24/4	Simple tool	1250-2500	87-507	90
Cast AZ91D/4		1098-3600	32-187	7-19
Thixomolded AM60/2		800-2450	90-750	0.9-5.4
Wrought AZ31B/6.4	Screw pin	800-1000	60	25
Thixomolded AM60/2	Screw pin	2000	120	10-15
Cast AZ91D/5			55	
Wrought AZ61/6.3		1220	90	Less than 14

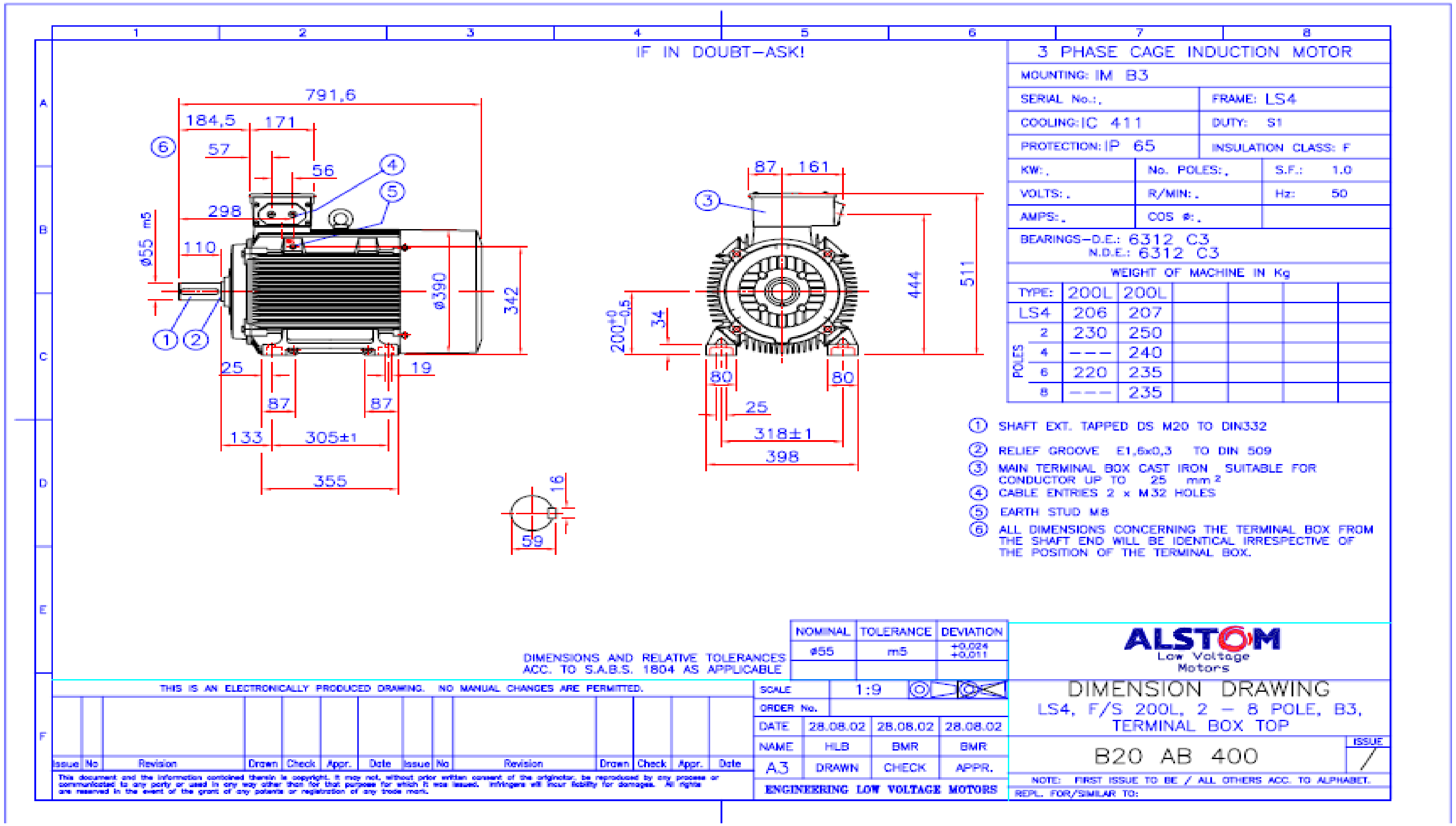
Table 0.5C: FSW parameters and tool materials for FSW of aluminium matrix composites.

Materials	Plate thickness (mm)	Plate thickness (mm)	Tool rotation rate (rpm)	Tool transverse speed (mm/min)	Tool materials
6092-SiC	17			102	
6061-BC	15-30			114-138	H13 tool steel
A339-SiC	10		670	60	20-carbon steel
6061-Al2O3	20		650	60	20-Carbon steel
6061-Al2O3	10, 20	2, 4	650	100-2500	
7093-SiC	25		500-3000		
7093-SiC	25			150	

All figures in this section are adopted from The Falk Corporation, (2004)

APPENDIX F: SELECTED COMPONENTS TECHNICAL SPECIFICATION

E.1 SELECTED MOTOR DIMENSIONS



E.2 INDUCTION MOTOR TECHNICAL SPECIFICATION

ALSTOM **CAST IRON STANDARD THREE PHASE T.E.F.C. SQUIRREL CAGE INDUCTION MOTORS**

PERFORMANCE DATA AT 400 VOLTS

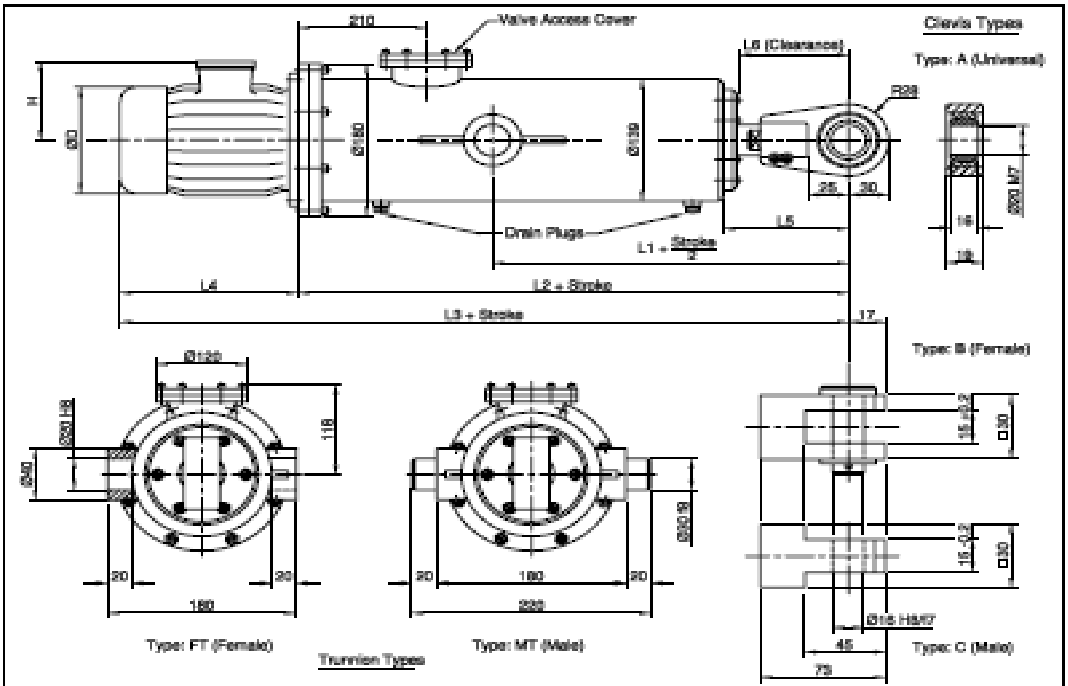
OUTPUT	FRAME	TYPE	SPEED	POLE	CURRENT	FLT	EFFICIENCY			POWER FACTOR (Pu)					LOCKED ROTOR RATIOS				NLC	BDT	% SLIP AT BDT	LRT (s)		ROTOR INERTIA	MOTOR MASS
							(%)	(%)	(%)						Star		D.O.L.								
kW		LS4	r/min		A	Nm	4/4	3/4	2/4	4/4	3/4	2/4	NL	LR	Torque	Current	Torque	Current	A	p.u.	BDT	Cold	Hot	kg.m²	kg
15	200L	207	725	8	32,0	197	89,5	90,5	89,4	0,76	0,73	0,63	0,09	0,58	0,44	1,28	2,05	4,75	16,6	2,20	10	21	10	0,41	235
18,5	200L	206	975	6	36,5	181	90,9	91,5	89,5	0,83	0,81	0,72	0,09	0,58	0,57	1,69	2,65	6,30	15,0	2,40	8,7	15	7	0,32	220
18,5	225S	220	730	8	37,0	242	89,9	91,0	89,5	0,80	0,78	0,68	0,10	0,55	0,42	1,33	1,95	4,95	17,6	2,15	8,9	19	9	0,63	300
22	200L	207	975	6	40,5	215	91,7	92,2	90,0	0,85	0,84	0,76	0,10	0,57	0,50	1,56	2,30	5,80	15,2	2,10	8,2	16	7	0,37	235
22	225M	223	730	8	43,5	288	90,4	91,4	90,0	0,81	0,76	0,69	0,09	0,53	0,41	1,30	1,90	4,85	19,5	2,10	9	22	10	0,74	325
30	200L	206	2940	2	53,5	97	91,8	91,4	89,0	0,88	0,86	0,82	0,21	0,46	0,52	1,94	2,20	6,95	16,3	2,75	8,8	19	9	0,13	230
30	200L	207	1465	4	54,5	195	92,0	92,4	91,0	0,86	0,83	0,75	0,10	0,55	0,60	1,96	2,55	6,90	20,9	2,60	9,3	12	6	0,26	240
30	225M	223	980	6	54,5	292	92,5	92,8	91,0	0,86	0,84	0,75	0,10	0,55	0,58	1,79	2,70	6,65	20,9	2,25	6,8	15	7	0,63	305
30	250S	253	730	8	59,5	392	91,3	92,4	90,5	0,80	0,78	0,71	0,07	0,52	0,45	1,33	2,10	4,95	23,8	2,05	7,2	26	12	1,2	460
37	200L	207	2940	2	65,0	120	92,5	92,5	90,5	0,89	0,87	0,82	0,18	0,46	0,56	2,05	2,35	6,90	19,5	2,95	8,5	15	7	0,17	250
37	225S	220	1475	4	66,0	239	92,8	92,8	90,5	0,87	0,84	0,74	0,09	0,52	0,66	1,98	2,80	6,90	27,6	2,80	7,5	13	6	0,47	300
37	250S	253	980	6	68,5	360	93,0	93,3	93,0	0,84	0,81	0,76	0,08	0,53	0,52	1,68	2,40	6,25	22,8	2,05	5,9	16	7	0,97	435
37	250M	255	730	8	71,0	484	92,0	93,3	93,0	0,82	0,80	0,71	0,07	0,50	0,48	1,41	2,20	5,25	28,0	2,30	7,1	29	13	1,4	490
45	225M	223	2950	2	78,0	146	92,8	93,0	91,0	0,90	0,89	0,85	0,19	0,43	0,54	1,96	2,30	6,90	20,0	2,65	6,9	25	11	0,24	310
45	225M	223	1475	4	79,0	291	93,6	93,8	92,5	0,88	0,85	0,78	0,08	0,50	0,65	2,04	2,75	6,90	28,5	2,65	7,1	16	7	0,57	330
45	250M	255	980	6	83,0	438	93,4	93,6	93,0	0,84	0,81	0,75	0,08	0,52	0,56	1,80	2,60	6,70	29,0	2,15	5,7	15	7	1,2	460
45	280S	283	735	8	85,5	584	92,5	93,6	91,5	0,82	0,79	0,70	0,07	0,50	0,49	1,54	2,25	5,75	31,8	2,10	6,3	24	11	1,8	630
55	250S	253	2965	2	96,0	177	92,9	93,1	91,0	0,89	0,87	0,80	0,17	0,45	0,57	1,98	2,40	6,90	28,0	2,60	4,8	19	9	0,45	435
55	250S	253	1475	4	97,5	356	93,8	94,0	92,5	0,87	0,84	0,80	0,09	0,48	0,58	1,79	2,45	6,40	29,2	2,20	5,5	21	10	0,89	460
55	280S	283	985	6	100,0	533	93,5	93,8	92,5	0,85	0,83	0,79	0,08	0,52	0,58	1,81	2,70	6,75	31,4	2,40	6,2	14	6	1,7	610
55	280M	285	735	8	103,0	714	93,0	93,9	92,5	0,83	0,81	0,75	0,07	0,49	0,51	1,62	2,35	6,05	35,6	2,10	6,1	24	11	2,2	675
75	250M	255	2970	2	130,0	241	93,4	92,8	90,5	0,89	0,88	0,82	0,17	0,45	0,61	2,01	2,60	6,90	30,9	2,55	4,3	22	10	0,63	490
75	250M	255	1475	4	132,0	485	94,1	94,3	93,5	0,87	0,85	0,79	0,08	0,47	0,66	1,98	2,80	6,90	39,9	2,40	5,2	22	10	1,2	510
75	280M	285	985	6	134,0	727	93,8	94,2	93,8	0,86	0,83	0,79	0,07	0,52	0,61	1,88	2,80	6,90	40,9	2,40	5,9	14	7	2,3	650
75	315S	310	740	8	139,0	967	93,7	94,4	93,0	0,83	0,80	0,72	0,06	0,47	0,51	1,77	2,35	6,60	57,5	2,55	5,5	21	9	3,5	880
90	280S	283	2970	2	152,5	289	94,5	94,5	93,5	0,90	0,88	0,83	0,15	0,36	0,47	2,05	2,00	6,90	38,5	2,70	4	29	13	0,94	640
90	280S	283	1480	4	158,0	580	94,6	94,7	93,5	0,87	0,84	0,81	0,08	0,40	0,59	2,10	2,50	6,90	52,3	2,60	4,6	26	12	1,8	690
90	315S	310	985	6	160,0	872	94,4	94,6	93,5	0,86	0,80	0,79	0,06	0,46	0,51	1,72	2,35	6,40	49,4	2,30	5,2	25	11	3,2	860
90	315M	311	740	8	166,5	1161	94,1	94,5	93,5	0,83	0,80	0,72	0,04	0,46	0,51	1,75	2,35	6,50	67,0	2,70	5,4	21	10	4,2	960
110	280M	285	2970	2	184,0	354	94,8	94,8	94,0	0,91	0,90	0,85	0,14	0,36	0,50	2,08	2,10	6,90	41,3	2,70	3,8	31	14	1,2	725
110	280M	285	1480	4	189,5	709	95,3	95,2	94,5	0,88	0,84	0,78	0,08	0,39	0,60	2,12	2,55	7,00	60,3	2,60	4,4	30	14	2,2	765
110	315M	311	985	6	194,5	1066	94,8	95,0	94,8	0,86	0,81	0,78	0,06	0,46	0,53	1,79	2,45	6,65	58,9	2,35	5	24	11	3,9	940
110	315L	312	740	8	203,5	1419	94,1	94,5	94,0	0,83	0,81	0,75	0,04	0,45	0,52	1,76	2,40	6,55	80,8	2,75	5,3	22	10	5,2	1070
132	315S	310	2980	2	226,0	423	95,1	94,6	93,0	0,89	0,88	0,82	0,14	0,32	0,45	2,12	1,90	7,00	63,7	2,90	3,1	30	14	1,6	870
132	315S	310	1485	4	229,5	848	95,5	95,5	94,5	0,87	0,85	0,80	0,08	0,39	0,58	2,04	2,45	6,90	71,3	2,45	3,5	29	13	3,0	940
132	315L	312	985	6	233,5	1279	94,8	95,0	93,5	0,86	0,84	0,81	0,06	0,46	0,55	1,83	2,55	6,80	70,3	2,35	4,9	22	10	4,9	1070
132	315LX	314	740	8	243,0	1703	94,5	94,9	94,0	0,83	0,78	0,68	0,04	0,45	0,52	1,80	2,50	6,50	99,8	2,50	5,1	23	10	7,2	1454
160	315M	311	2980	2	268,0	513	95,7	95,3	94,5	0,90	0,89	0,85	0,13	0,31	0,43	2,01	1,80	6,90	64,1	2,70	2,9	28	13	2,0	950
160	315M	311	1485	4	274,0	1028	95,8	95,7	95,0	0,88	0,86	0,82	0,08	0,40	0,57	1,96	2,40	6,90	72,2	2,25	3,4	31	14	3,7	1030
160	315L	313	985	6	286,5	1551	94,8	95,1	94,0	0,85	0,84	0,76	0,05	0,45	0,54	1,72	2,50	6,80	90,3	2,35	5,5	21	9	5,7	1255
185	315L	312	2980	2	306,5	593	95,7	95,7	94,0	0,91	0,90	0,86	0,12	0,31	0,45	2,01	1,90	6,90	64,6	2,65	2,8	27	12	2,3	1070
185	315L	312	1485	4	316,5	1189	95,9	95,9	94,0	0,88	0,86	0,81	0,07	0,41	0,63	2,14	2,65	7,00	89,3	2,40	3,4	27	12	4,2	1170
185	315LX	314	985	6	331,5	1793	94,8	95,2	94,0	0,85	0,83	0,76	0,05	0,45	0,56	1,80	2,60	6,70	114,0	2,40	5,5	24	11	7,2	1454
200	315L	313	2980	2	331,5	641	95,7	95,7	94,0	0,91	0,89	0,87	0,12	0,33	0,47	2,04	2,00	6,90	66,5	2,65	2,7	27	12	2,6	1130
200	315L	313	1485	4	345,5	1286	96,0	96,0	95,0	0,87	0,85	0,80	0,07	0,40	0,66	2,20	2,80	7,00	99,8	2,55	3,4	28	13	4,5	1255
200	315LX	315	983	6	358,0	1942	94,8	95,3	95,0	0,85	0,82	0,76	0,05	0,40	0,56	1,75	2,60	6,55	125,4	2,40	5,5	21	10	7,2	1460
225	315LX	314	2975	2	372,5	722	95,8	95,9	94,5	0,91	0,90	0,88	0,11	0,33	0,47	1,96	2,00	6,90	69,4	2,55	2,6	27	12	3,2	1370
225	315LX	314	1485	4	389,0	1446	96,0	96,1	95,5	0,87	0,86	0,81	0,06	0,39	0,61	2,10	2,60	6,90	115,9	2,40	3,4	25	11	5,5	1425
250	315LX	315	2975	2	414,0	802	95,8	95,9	94,5	0,91	0,90	0,88	0,10	0,32	0,48	2,12	2,05	7,00	86,5	2,75	2,8	27	12	3,2	1370
250	315LX	315	1485	4	432,0	1607	96,0	96,2	94,5	0,87	0,86	0,81	0,06	0,40	0,61	2,04	2,60	6,90	128,3	2,35	3,5	29	13	5,5	1425

Output - Rated output (kW)
Frame - IEC frame size
Type - Range type
Speed - Motor rated speed(r/min)
FLT - Full load torque (Nm)
Efficiency - efficiency under different conditions
Locked rotor ratios - Pu value for torque and current
NLC - No load current @ nominal voltage
BDT - Break down torque
Power factor - Power factor under different conditions
LRT - Locked rotor time (seconds)

PQ01

E.3 SELECTED ELECTRO-HYDRAULIC GEOMETRY

dabeb-elram ELECTRO - HYDRAULIC ACTUATORS	MODEL DEA 139 GENERAL ARRANGEMENT AND OVERALL DIMENSIONS	DATA SHEET REF: 139/1 Date 02/01



Variable Dimensions		Motor Rating (kW)									
		0.25	0.37								
L1		315	315								
L2		420	420								
L3*		627	627								
L4*		207	207								
L5		75	75								
L6		55	55								
D *		140	140								
H *		126	126								
Approx Mass (Kg)	Basic	40	40								
	Per 100 Stroke	3	3								

For strokes less than 250, dimensions L1, L2 & L3 to be based on 250 Stroke
Maximum stroke length: 650 (For longer strokes consult Dabeb Elram)

* These dimensions are nominal, and will vary slightly according to electric motor manufacture.
For actuators fitted with piston rod protection bellows, please refer to Data Sheet No OF 24
Please consult the optional features section of the catalogue (blue pages) for details of the many options available.

For full specifications and ordering information,see the white pages.
Certified drawings are available on request.

In the interests of continued development, we reserve the right to make changes without notice.

E.4 PERFORMANCE TABLE FOR ELECTRO-HYDRAULIC ACTUATORS

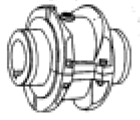
Table 0.1F: Performance for electro-hydraulic

DEA RANGE - PERFORMANCE TABLE					
Model	Max. Extend Thrust kgF	Max. Retract Thrust kgF	Max. Stroke mm	Max Speed mm/sec	Motor Range kW
139	300	250	650	250	0.37
159	500	400	800	250	0.37 - 1.5
193	1500	1000	1200	250	0.37 - 2.2
219	2500	2000	1600	200	0.37 - 3.0
245	4000	3000	2500	200	0.55 - 5.5
273	5500	4000	3500	160	0.75 - 7.5
298	7000	5500	4000	140	0.75 - 7.5
368	10000	9000	5000	120	0.75 - 15
419	22500	16000	5000	120	2.2 - 18.5
508	40000	30000	5000	100	2.2 - 18.5
610	80000	60000	5000	50	2.2 - 30

E.6 COUPLER SELECTION INFORMATION FROM THE FALK CORPORATION

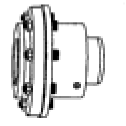
Falk Steelflex Grid Couplings

A general purpose, lubricated design that combines the economy and high torque capacity of a gear coupling with the torsional flexibility of an elastomer coupling. Backed by a 5–year lubrication warranty, Falk Steelflex couplings require no periodic maintenance when lubricated with Falk LTG (Long Term Grease) at installation. Featuring 25 sizes, Steelflex couplings can accommodate torque loads of 932 000 (Nm) and shaft diameters of 508 millimeters.



A double flexing, close–coupled design for use in four bearing systems. Features a horizontally split cover which allows for grid replacement without the movement of the connected equipment. (See Page 14.)

Type T10 Close Coupled

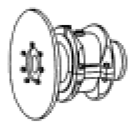


For use on line shaft applications. Can be used in place of single engagement gear couplings to provide torsional resiliency and lower overall operating cost. (See Pages 28 & 29.)

Type T50 Piloted

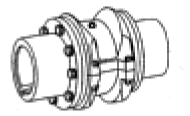
A double flexing design featuring a vertically split steel cover. Ideal for higher running speeds. (See Page 15.)

Type T20 Close Coupled



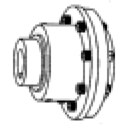
Proven to be far superior to drum-type brakes in cost, construction and performance. (See Pages 30 thru 32.)

Type T63 Disc Brake



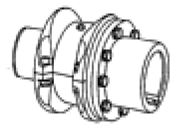
Complete center section drops out for easy service of connected equipment bearings and seals. Ideal for pump applications. (See Pages 16 & 17.)

Type T31 Full Spacer



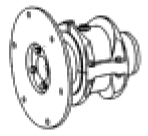
Designed for operating speeds beyond those of the T10 and T20 designs. Features a one-piece cover and balanced components. (See Page 33.)

Type T70 High Speed



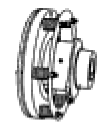
An economical spacer design for easy service of connected equipment bearings and seals. Ideal for pump applications. (See Pages 18 & 19.)

Type T35 Half Spacer



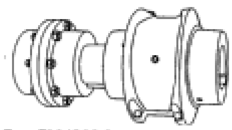
Used primarily to connect the flywheel of an engine to the driven machinery. It provides for higher torque ratings with resulting smaller sizes and lower costs than elastomer couplings. (See Page 34.)

Type T90 Flywheel



Provides adjustable slipping action to protect connected equipment from shock, jams, or temporary overloads. (See Pages 20 thru 27.)

Type T41, T42, T44 & T45 Controlled Torque

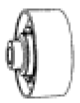


A combination of two standard Falk couplings. Utilizes readily available components for an economical price and shorter lead time than T31/T35 couplings. (See Page 35.)

Type T10/G82 Spacer

Double piloted design for connecting equipment where the distance between shafts is too large for a spacer type coupling. (See Pages 28 & 29.)

Type T50 Floating Shaft



Provides a built-in braking surface right at or near the centerline of the coupling . . . saves space and dollars. (See Selection Guide 431-310.)

Type BW Brakewheel

WARNING! Mixing grid coupling components from different manufacturers may cause premature failure and possible personal injury or property damage from flying debris.

6 (M421-110)

Standard Selection Method (except T41/T44 &T63)

The standard selection method can be used for most motor, turbine, or engine driven applications. The following information is required to select a flexible coupling:

- Kilowatt (kW) or torque
- Running rpm
- Application or type of equipment to be connected (motor to pump, gear drive to conveyor, etc.)
- Shaft diameters
- Shaft gaps
- Physical space limitations
- Special bore or finish information and type of fit

Exceptions are High Peak Loads and Brake Applications. For these conditions use the Formula Selection Method in the next column, or consult your local Falk Representative for assistance.

1. **RATING:** Determine system torque. If torque is not given, calculate as shown below:

$$\text{System Torque (Nm)} = \frac{\text{kW} \times 9549}{\text{rpm}}$$

Where kilowatt (kW) is the actual or transmitted power required by the application (if unknown, use the motor or turbine nameplate rating) and rpm is the actual speed the coupling is rotating. Applications that require rapid changes in direction or torque reversals should be referred to Falk Engineering.

2. **SERVICE FACTOR:** Determine appropriate service factor from Table 4, Page 12.
3. **REQUIRED MINIMUM COUPLING RATING:** Determine the required minimum coupling rating as shown below:

   Minimum Coupling Rating = S.F. (Service Factor) x Torque (Nm)
4. **TYPE:** Refer to Page 6 and select the appropriate coupling type.
5. **SIZE:** Turn to appropriate pages for the coupling type chosen and trace down the torque column to a value that is equal or greater than that determined in Step 3 above. The coupling size is shown in the first column.
6. **CHECK:** Check speed (rpm), bore, gap, and dimensions.

STANDARD SELECTION EXAMPLE:

Select a coupling to connect a 55 kW, 1500 rpm electric motor driving a lobe type blower. Motor shaft diameter is 60 mm, blower shaft diameter is 45 mm. Shaft extensions are 140 mm and 110 mm. Selection is replacing a gear type coupling with a 3 mm gap.

1. **DETERMINE REQUIRED RATING:**

$$\text{System Torque (Nm)} = \frac{55 \text{ kW} \times 9549}{1500 \text{ rpm}} = 350 \text{ Nm}$$

2. **SERVICE FACTOR:** From Table 4 = 1.25
3. **REQUIRED MINIMUM COUPLING RATING:**

   1.25 x 350 Nm = 438 Nm
4. **SIZE:** From Page 14 a Size 1070T is the proper selection based on a torque rating of 904 Nm exceeding the required minimum coupling rating of 438 Nm.
5. **CHECK:** Allowable speed capacity of 4125 (1070T10) exceeds the required speed of 1500 rpm. Maximum bore capacity of 67 mm exceeds the actual shaft diameters.

Type T63 Static (holding) Brake Applications

1. **SIZE:** The brake rating must equal or exceed the application requirements. Determine the required coupling size by comparing the application loads (from Steps A and B below) to the coupling brake rating listed on Page 31. Use the highest torque value calculated to determine the coupling size.

A. For normal service applications, use the application torque in Nm.

$$\text{System Torque (Nm)} = \frac{\text{Transmitted kW} \times 9549}{\text{rpm}}$$

B. For repetitive high peak load applications, use the system peak torque in Nm. (Repetitive is defined as more than 1000 times during the expected coupling life.)

2. **CALIPER TORQUE BRAKE RATING:** For the coupling size selected, compare the caliper brake torque rating on Page 31 to the holding torque requirement of the application. Falk recommends that the caliper torque rating (min.) be at least two times the holding torque requirement for static applications to compensate for the possibility of foreign matter on the disc surfaces, loss of condition of the brake pad surfaces, or other conditions that may affect the holding ability of the caliper brake.

   Caliper brakes and brake discs listed are designed primarily for static and/or emergency brake applications. **NOTE:** Check brake system and lining wear after emergency stops. They can, however, also be used for dynamic stopping if only used occasionally, such as shutting down the equipment for the day or between shift changes. For stopping high inertia systems or for applications that require more frequent stopping, consult your local Falk Representative.
3. **CHECK:** Check maximum bores, speeds, and dimensions.

Type T63 Stopping Or Service Brake Applications

1. **SIZE:** The coupling brake rating must equal or exceed the application requirements. Determine the required coupling size by comparing the application loads (from Steps A, B and C below) to the coupling brake rating listed on Page 28. Use the highest torque value calculated to determine the coupling size.

A. For the selected caliper brake and disc diameter ,use the maximum brake torque in Nm.

B. For normal service applications, use the application torque in Nm.

$$\text{System Torque (Nm)} = \frac{\text{Transmitted kW} \times 9549}{\text{rpm}}$$

C. For repetitive high peak load applications, use the system peak torque in Nm (Repetitive is defined as more than 1000 times during the expected coupling life.)

2. **CHECK:** Check maximum bores, speeds, and dimensions.

Service Factors

TABLE 4 — Flexible Coupling Service Factors for Motor ♦ and Turbine Drives

Service factors listed are typical values based on normal operation of the drive systems.

Alphabetical listing of applications

Application	Service Factor
AERATOR	2.0
AGITATORS	
Vertical and Horizontal Screw, Propeller, Paddle	1.0
BARGE HAUL PULLER	1.5
BLOWERS	
Centrifugal	1.0
Lobe or Vane	1.25
CAR DUMPERS	2.5
CAR PULLERS	1.5
CLARIFIER OR CLASSIFIER	1.0
COMPRESSORS	
Centrifugal	1.0
Rotary, Lobe or Vane	1.25
Rotary, Screw	1.0
Reciprocating	
Direct Connected	Refer to Falk
Without Flywheel	Refer to Falk
■With Flywheel and Gear between Compressor and Prime Mover	
1 cylinder, single acting	3.0
1 cylinder, double acting	3.0
2 cylinders, single acting	3.0
2 cylinders, double acting	3.0
3 cylinders, single acting	3.0
3 cylinders, double acting	2.0
4 or more cly., single act.	1.75
4 or more cyl., double act.	1.75
▲CONVEYORS	
Apron, Assembly, Belt, Chain, Flight, Screw	1.0
Bucket	1.25
Live Roll, Shaker and Reciprocating	3.0
▲CRANES AND HOIST	
Main Hoist	1.75▲
Skip Hoist	1.75▲
Slope	1.5
Bridge, Travel or Trolley	1.75
DYNAMOMETER	1.0
ELEVATORS	
Bucket, Centrifugal Discharge	1.25
Freight or Passenger	**Not Approved**
Gravity Discharge	1.25
ESCALATORS	**Not Approved**
EXCITER, GENERATOR	1.0
EXTRUDER, PLASTIC	1.5
FANS	
Centrifugal	1.0
Cooling Tower	2.0
Forced Draft — Across the Line start	1.5
Forced Draft Motor Driven thru fluid or electric slip clutch	1.0
Gas Recirculating	1.5
Induced Draft with damper control or blade cleaner	1.25
Induced Draft without controls	2.0
FEEDERS	
Apron, Belt, Disc, Screw	1.0
Reciprocating	2.5
GENERATORS	
Even Load	1.0
Hoist or Railway Service	1.5
Welder Load	2.0
HAMMERMILL	1.75
LAUNDRY WASHER OR TUMBLER	2.0
LINE SHAFTS	
Any Processing Machinery	1.5
MACHINE TOOLS	
Auxiliary and Traverse Drive	1.0
Bending Roll, Notching Press, Punch Press, Planer, Plate Reversing	1.75
Main Drive	1.5
MAN LIFTS	**Not Approved**
METAL FORMING MACHINES	
Continuous Caster	1.75
Draw Bench Carriage and Main Drive	2.0
Extruder	2.0
Forming Machine and Forming Mills	2.0
Slitters	1.0
Wire Drawing or Flattening	1.75
Wire Winder	1.5
Coilers and Uncoilers	1.5
MIXERS (see Agitators)	
Concrete	1.75
Muller	1.5
PRESS, PRINTING	1.5
PUG MILL	1.75
PULVERIZERS	
Hammermill and Hog	1.75
Roller	1.5
PUMPS	
Boiler Feed	1.5
Centrifugal —	
Constant Speed	1.0
Frequent Speed Changes under Load	1.25
Descaling, with accumulators	1.25
Gear, Rotary, or Vane	1.25
Reciprocating, Plunger Piston	
1 cyl., single or double act.	3.0
2 cyl., single acting	2.0
2 cyl., double acting	1.75
3 or more cylinders	1.5
Screw Pump, Progressing Cavity	1.25
Vacuum Pump	1.25
SCREENS	
Air Washing	1.0
Grizzly	2.0
Rotary Coal or Sand	1.5
Vibrating	2.5
Water	1.0
SKI TOWS & LIFTS	**Not Approved**
STEERING GEAR	1.0
STOKER	1.0
TIRE SHREDDER	1.50
TUMBLING BARREL	1.75
WINCH, MANEUVERING	
Dredge, Marine	1.5
WINDLASS	1.5
WOODWORKING MACHINERY	1.0
WORK LIFT PLATFORMS	**Not Approved**

Alphabetical listing of industries

Industry	Service Factor
AGGREGATE PROCESSING, CEMENT, MINING KILNS; TUBE, ROD AND BALL MILLS	
Direct or on L.S. shaft of Reducer, with final drive Machined Spur Gears	2.0
Single Helical or Herringbone Gears	1.75
Conveyors, Feeders, Screens, Elevators	See General Listing
Crushers, Ore or Stone	2.5
Dryer, Rotary	1.75
Grizzly	2.0
Hammermill or Hog	1.75
Tumbling Mill or Barrel	1.75
BREWING AND DISTILLING	
Bottle and Can Filling Machines	1.0
Brew Kettle	1.0
Cookers, Continuous Duty	1.25
Lauter Tub	1.5
Mash Tub	1.25
Scale Hopper, Frequent Peaks	1.75
CLAY WORKING INDUSTRY	
Brick Press, Briquette Machine, Clay Working Machine, Pug Mill	1.75
DREDGES	
Cable Reel	1.75
Conveyors	1.25
Cutter head, Jig Drive	2.0
Maneuvering Winch	1.5
Pumps (uniform load)	1.5
Screen Drive, Stacker	1.75
Utility Winch	1.5
FOOD INDUSTRY	
Beet Slicer	1.75
Bottling, Can Filling Machine	1.0
Cereal Cooker	1.25
Dough Mixer, Meat Grinder	1.75
LUMBER	
Band Resaw	1.5
Circular Resaw, Cut-off	1.75
Edger, Head Rig, Hog	2.0
Gang Saw (Reciprocating)	Refer to Falk
Log Haul	2.0
Planer	1.75
Rolls, Non-Reversing	1.25
Rolls, Reversing	2.0
Sawdust Conveyor	1.25
Slab Conveyor	1.75
Sorting Table	1.5
Trimmer	1.75
✱METAL ROLLING MILLS	
Coilers (Up or Down) Cold Mills only	1.5
Coilers (Up or Down) Hot Mills only	2.0
Coke Plants	
Pusher Ram Drive	2.5
Door Opener	2.0
Pusher or Larry Car Traction Drive	3.0
Continuous Caster	1.75
Cold Mills —	
Strip Mills	Refer to Falk
Temper Mills	Refer to Falk
Cooling Beds	1.5
Drawbench	2.0
Feed Rolls - Blooming Mills	3.0
Furnace Pushers	2.0
Hot and Cold Saws	2.0
Hot Mills —	
Strip or Sheet Mills	Refer to Falk
Reversing Blooming	Refer to Falk
or Slabbing Mills	Refer to Falk
Edger Drives	Refer to Falk
Ingot Cars	2.0
Manipulators	3.0
Merchant Mills	Refer to Falk
Mill Tables	
Roughing Breakdown Mills	3.0
Hot Bed or Transfer, non-reversing	1.5
Runout, reversing	3.0
Runout, non-reversing, non-plugging	2.0
Reel Drives	1.75
Rod Mills	Refer to Falk
Screwdown	2.0
Seamless Tube Mills	
Piercer	3.0
Thrust Block	2.0
Tube Conveyor Rolls	2.0
Reeler	2.0
Kick Out	2.0
Shear, Croppers	Refer to Falk
Sideguards	3.0
Skelp Mills	Refer to Falk
Slitters, Steel Mill only	1.75
Soaking Pit Cover Drives —	
Lift	1.0
Travel	2.0
Straighteners	2.0
Unscramblers (Billet Bundle Busters)	2.0
Wire Drawing Machinery	1.75
OIL INDUSTRY	
Chiller	1.25
Oil well Pumping (not over 150% peak torque)	2.0
Paraffin Filter Press	1.5
Rotary Kiln	2.0
PAPER MILLS	
Barker Auxiliary, Hydraulic	2.0
Barker, Mechanical	2.0
Barking Drum	
L.S. shaft of reducer with final drive - Helical or Herringbone Gear	2.0
Machined Spur Gear	2.5
Cast Tooth Spur Gear	3.0
Beater & Pulper	1.75
Bleachers, Coaters	1.0
Calender & Super Calender	1.75
Chipper	2.5
Converting Machine	1.25
Couch	1.75
Cutter, Felt Whipper	2.0
Cylinder	1.75
Dryer	1.75
Felt Stretcher	1.25
Fourdrinier	1.75
Jordan	2.0
Log Haul	2.0
Line Shaft	1.5
Press	1.75
Pulp Grinder	1.75
Reel, Rewinder, Winder	1.5
Stock Chest, Washer, Thickener	1.5
Stock Pumps, Centrifugal	
Constant Speed	1.0
Frequent Speed Changes Under Load	1.25
Suction Roll	1.75
Vacuum Pumps	1.25
RUBBER INDUSTRY	
Calender	2.0
Cracker, Plasticator	2.5
Extruder	1.75
Intensive or Banbury Mixer	2.5
Mixing Mill, Refiner or Sheeter	
One or two in line	2.5
Three or four in line	2.0
Five or more in line	1.75
Tire Building Machine	2.5
Tire & Tube Press Opener (Peak Torque)	1.0
Tuber, Strainer, Pelletizer	1.75
Warming Mill	
One or two Mills in line	2.0
Three or more Mills in line	1.75
Washer	2.5
SEWAGE DISPOSAL EQUIPMENT	
Bar Screen, Chemical Feeders, Collectors, Dewatering Screen, Grit Collector	1.0
SUGAR INDUSTRY	
Cane Carrier & Leveler	1.75
Cane Knife & Crusher	2.0
Mill Stands, Turbine Driver With all helical or Herringbone gears	1.5
Electric Drive or Steam Engine Drive with Helical, Herringbone, or Spur Gears with any Prime Mover	1.75
TEXTILE INDUSTRY	
Batcher	1.25
Calender, Card Machine	1.5
Cloth Finishing Machine	1.5
Dry Can, Loom	1.5
Dyeing Machinery	1.25
Knitting Machine	Refer to Falk
Mangle, Napper, Soaper	1.25
Spinner, Tenter Frame, Winder	1.5

♦ For engine drives, refer to Table 5. Electric motors, generators, engines, compressors and other machines fitted with sleeves or straight roller bearings usually require limited end float couplings. If in doubt, provide axial clearances and centering forces to Falk for a recommendation.

■ For balanced opposed design, refer to Falk.

▲ If people are occasionally transported, refer to Falk for the selection of the proper size coupling.

✱ For high peak load applications (such as Metal Rolling Mills) refer to Falk.

TABLE 5 — Engine Drive Service Factors ▼

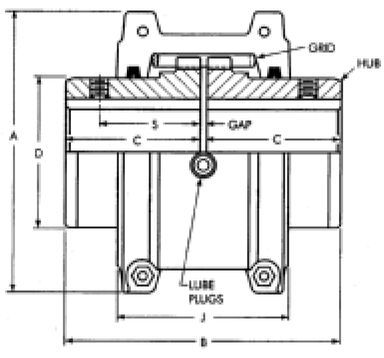
Service Factors for engine drives are those required for applications where good flywheel regulation prevents torque fluctuations greater than ±20%. For drives where torque fluctuations are greater or where the operation is near a serious critical or torsional vibration, a mass elastic study is necessary.

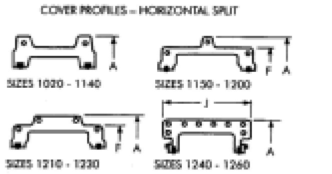
No. of Cylinders	4 or 5 ▼					6 or more ▼				
Table 2 S.F.	1.0	1.25	1.5	1.75	2.0	1.0	1.25	1.5	1.75	2.0
Engine S.F.	2.0	2.25	2.5	2.75	3.0	1.5	1.75	2.0	2.25	2.5

▼ To use Table 5, first determine application service factor from Table 4. Use that factor to determine ENGINE Service Factor from Table 5. When service factor from Table 4 is greater than 2.0, or where 1, 2, or 3 cylinder engines are involved, refer complete application details to Falk Engineering.

Type T10

Close Coupled/Dimensions — Millimeters





Sizes 1020 thru 1230T10 covers are cast aluminum alloy;
Sizes 1240 thru 1260T10 are fabricated steel.

SIZE ★	Torque Rating Nm †	Allow Speed rpm ‡	Max Bore mm ■	Min Bore mm ■	Cplg Wt With No Bore-kg	Lube Wt kg	DIMENSIONS — MILLIMETERS A	B	C	D	F	J	S	Gap
1020T	52	4500	28	13	1,93	0,0272	97,0	98,2	47,6	39,7	....	66,7	39,1	3
1030T	149	4500	35	13	2,58	0,0408	105,7	98,2	47,6	49,2	....	68,3	39,1	3
1040T	249	4500	43	13	3,34	0,0544	114,3	104,6	50,8	57,2	....	69,9	40,1	3
1050T	435	4500	50	13	5,44	0,0680	135,1	123,6	60,3	66,7	....	80,9	44,7	3
1060T	684	4350	56	20	7,44	0,0862	147,8	130,0	63,5	76,2	....	93,5	52,3	3
1070T	994	4125	67	20	10,4	0,113	158,8	155,4	76,2	87,3	....	96,8	53,8	3
1080T	2 050	3600	80	27	17,9	0,172	190,5	180,8	88,9	104,8	....	115,6	64,5	3
1090T	3 730	3600	95	27	25,6	0,254	211,1	199,8	98,4	123,8	....	122,2	71,6	3
1100T	6 280	2440	110	42	42,0	0,426	251,0	246,2	120,6	142,1	....	155,4	....	5
1110T	9 320	2250	120	42	54,3	0,508	269,7	259,0	127,0	160,3	....	161,5	....	5
1120T	13 700	2025	140	61	81,2	0,735	307,8	304,4	149,2	179,4	....	191,5	....	6
1130T	19 900	1800	170	67	121	0,907	345,9	329,8	161,9	217,5	....	195,1	....	6
1140T	28 600	1650	200	67	178	1,13	384,0	374,4	184,2	254,0	....	200,2	....	6
1150T	39 800	1500	215	108	234	1,95	453,1	371,8	182,9	269,2	391,2	271,5	....	6
1160T	55 900	1350	240	121	317	2,81	501,9	402,2	198,1	304,8	436,9	278,4	....	6
1170T	74 600	1225	280	134	448	3,49	566,9	437,8	215,9	355,6	487,2	307,3	....	6
1180T	103 000	1100	300	153	619	3,76	629,9	483,6	238,8	393,7	554,7	321,1	....	6
1190T	137 000	1050	335	153	776	4,40	675,6	524,2	259,1	436,9	607,8	325,1	....	6
1200T	186 000	900	360	178	1 058	5,62	756,9	564,8	279,4	497,8	660,4	355,6	....	6
1210T	249 000	820	390	178	1 424	10,5	844,6	622,6	304,8	530,4	750,8	431,8	....	13
1220T	336 000	730	420	203	1 785	16,1	920,8	663,2	325,1	571,5	822,2	490,2	....	13
1230T	435 000	680	450	203	2 367	24,0	1 003,3	703,8	345,4	609,6	904,7	546,1	....	13
1240T	559 000	630	480	254	2 950	33,8	1 087,1	749,6	368,3	647,7	....	647,7	....	13
1250T	746 000	580	♦	254	3 833	50,1	1 181,1	815,6	401,3	711,2	....	698,5	....	13
1260T	932 000	540	♦	254	4 682	67,2	1 260,9	876,6	431,8	762,0	....	752,0	....	13

★ Refer to Page 5 for General Information and Reference Notes.
♦ Refer to Falk.

APPENDIX G: PERSONAL/EMAIL COMMUNICATION

Fulufhelo Masithulela **Transnet Freight Rail PTA**

From: Justice Ngoato Transnet Freight Rail PTA
Sent: 02 October 2007 13:21
To: Fulufhelo Masithulela Transnet Freight Rail PTA
Subject: Track welding

Hi Fulu,

There's basically two major types of welding done on the track. i.e Flash butt welding - which is, in simplicity, the fusing of two rail ends at very high pressure and temperature. And the second one is exothermic (aluminothermic) welding - (a mixture of basically aluminium and iron is ignited in a mold placed on top of two rail ends with a definite rail gap) - reaction is exothermic thus melting temperatures are met.

We also do skidmark repairs using the arc welding (electrode welding) method. The procedure followed depends on the type of rail i.e Cr-Mn or HH rail. The latest development required the use of HH rails throughout our tracks so the procedure on Cr-Mn rails is only valid where these type of rails still exist. We have in the past conducted audits for approval of these welding electrodes so that the composition of the electrodes matches that of the rail.

Attached pls recieve our flash butt welding specification and a british standard regarding exothermic welding. Pls note that the british standard is intended to give you the basic idea on exothermic welding. It is not to be regarded as the technical procedure of exothermic welding within Transnet Freight Rail. Thermitrx, a company that supplies most of our exothermic consumables as well as technical skills on the subiect holds the specification according to my knowledge - writing under correction. The flash butt welding specification is also under review currently, therefore certain things may change. Pls treat it as only a guideline.

Should you need clarity on anything else, pls do not hesitate to contact me.

Kind regards,

Justice Ngoato
Eng. Technician (Metallurgy)
Rail Technology Cluster
Transnet Freight Rail
Koedoespoort
PRETORIA, RSA

Office. 012 842 6246
Fax. 012 842 6235

(.....Life is precious, let every year of your career be beautified with great achievements......)

Spec S170 part0.doc (713 KB) Spec S170 part2.doc (135 KB) Spec S170 part1.doc (134 KB) Spec S170 part3.doc (116 KB) Spec S170 part4.doc (109 KB) Spec S170 part5.doc (108 KB) BS EN 14730-1.pdf (566 KB)

APPENDIX H: ENGINEERING GRAPHS FOR DESIGN CALCULATIONS

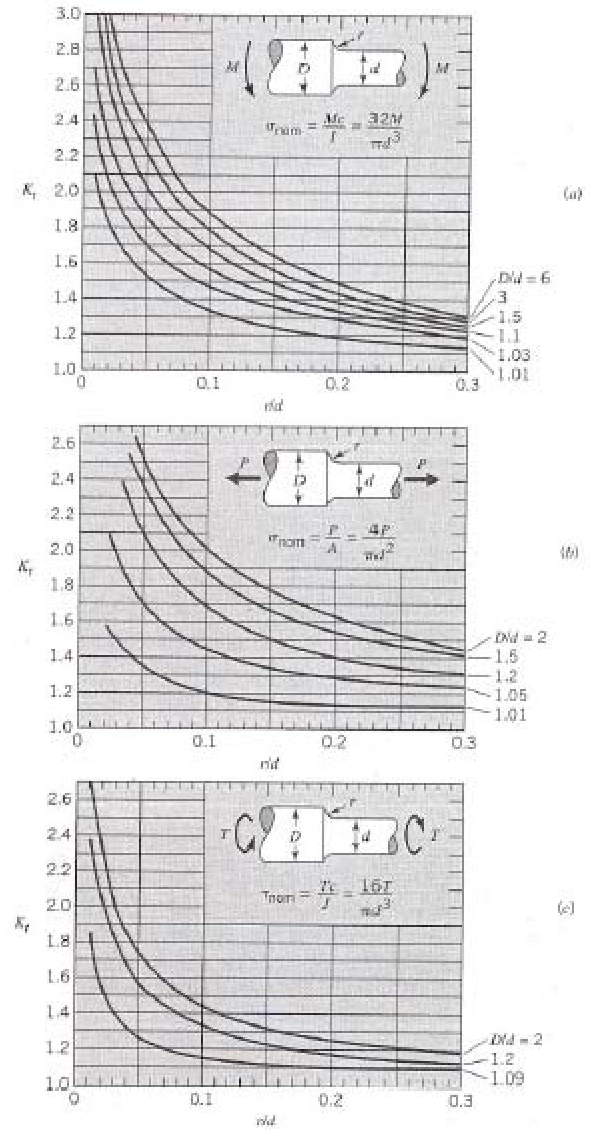


Figure 0.1N: Shaft with fillet (a) bending; (b) axial load; (c) torsion

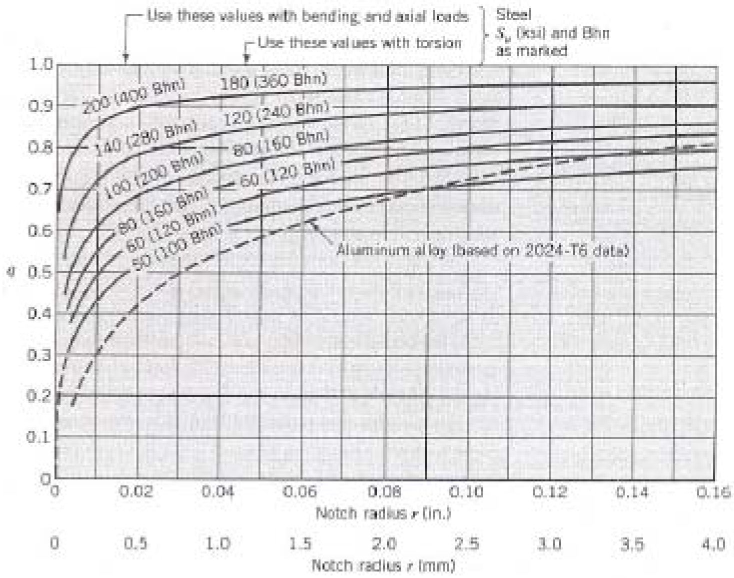


Figure 0.2N: Notch sensitivity curves.

Table 0.2N: Basic dimensions of united screw thread.

Size	Major Diameter d (in.)	Coarse Threads—UNC: Threads per Inch	Coarse Threads—UNC: Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC: Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF: Threads per Inch	Fine Threads—UNF: Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF: Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599
$\frac{9}{16}$	0.5625	12	0.4603	0.182	18	0.4943	0.203
$\frac{5}{8}$	0.6250	11	0.5135	0.226	18	0.5568	0.256
$\frac{3}{4}$	0.7500	10	0.6273	0.334	16	0.6733	0.373
$\frac{7}{8}$	0.8750	9	0.7387	0.462	14	0.7874	0.509
1	1.0000	8	0.8466	0.606	12	0.8978	0.663
$1\frac{1}{8}$	1.1250	7	0.9497	0.763	12	1.0228	0.856
$1\frac{1}{4}$	1.2500	7	1.0747	0.969	12	1.1478	1.073
$1\frac{3}{8}$	1.3750	6	1.1705	1.155	12	1.2728	1.315
$1\frac{1}{2}$	1.5000	6	1.2955	1.405	12	1.3978	1.581
$1\frac{3}{4}$	1.7500	5	1.5046	1.90			
2	2.0000	$4\frac{1}{2}$	1.7274	2.50			
$2\frac{1}{4}$	2.2500	$4\frac{1}{2}$	1.9774	3.25			
$2\frac{1}{2}$	2.5000	4	2.1933	4.00			
$2\frac{3}{4}$	2.7500	4	2.4433	4.93			
3	3.0000	4	2.6933	5.97			
$3\frac{1}{4}$	3.2500	4	2.9433	7.10			
$3\frac{1}{2}$	3.5000	4	3.1933	8.33			
$3\frac{3}{4}$	3.7500	4	3.4433	9.66			
4	4.0000	4	3.6933	11.08			

Note: See ANSI standard B1.1-1974 for full details. Unified threads are specified as "$\frac{1}{2}$ in.–13UNC," "1 in.–12UNF."

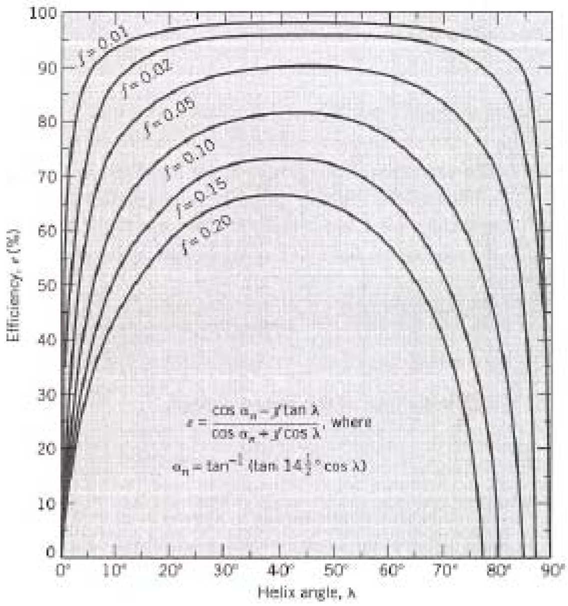


Figure 0.3N: Efficiency of acme screw threads when collar friction I negligible.

Table 0.3N: Specifications for steel used in inch series screws and bolts.

SAE Grade	Diameter d (in.)	Proof Load (Strength)[a] S_p (ksi)	Yield Strength[b] S_y (ksi)	Tensile Strength S_u (ksi)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness Rockwell Min	M
1	$\frac{1}{4}$ thru $1\frac{1}{2}$	33	36	60	18	35	B70	B
2	$\frac{1}{4}$ thru $\frac{3}{4}$	55	57	74	18	35	B80	B
2	Over $\frac{3}{4}$ to $1\frac{1}{2}$	33	36	60	18	35	B70	B
5	$\frac{1}{4}$ thru 1	85	92	120	14	35	C25	C
5	Over 1 to $1\frac{1}{2}$	74	81	105	14	35	C19	C
5.2	$\frac{1}{4}$ thru 1	85	92	120	14	35	C26	C
7	$\frac{1}{4}$ thru $1\frac{1}{2}$	105	115	133	12	35	C28	C
8	$\frac{1}{4}$ thru $1\frac{1}{2}$	120	130	150	12	35	C33	C

Table 0.4N: Specifications for steel used in milimeter series screws and bolts.

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

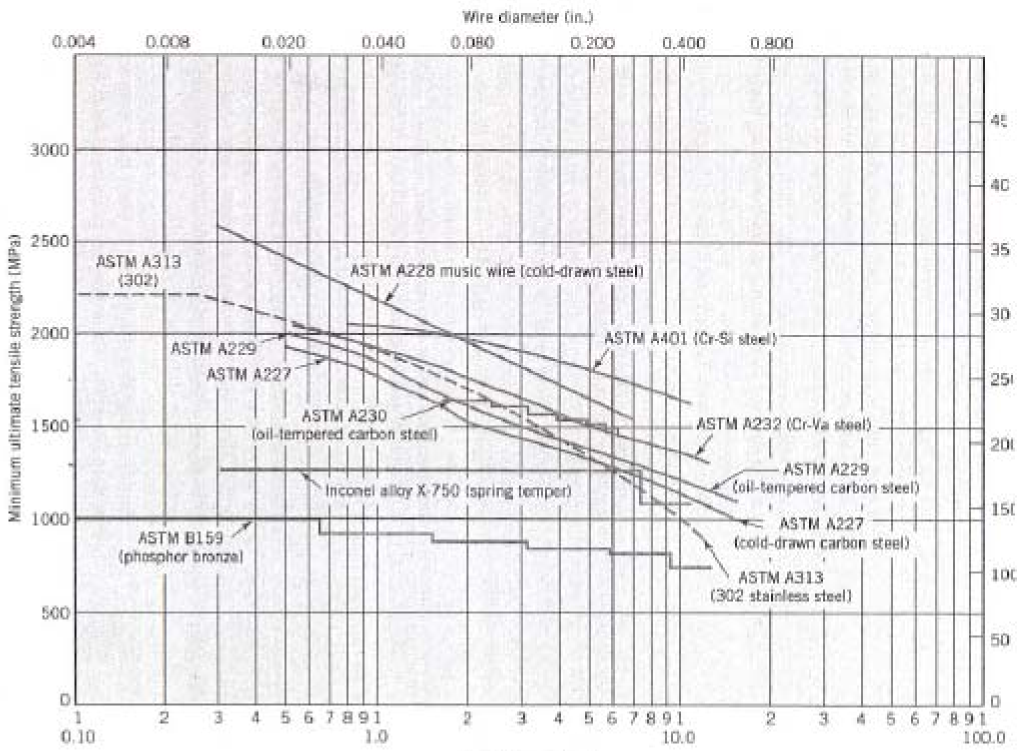


Figure 0.4N: Tensile strengths of various spring wire materials and diameters, minimum.

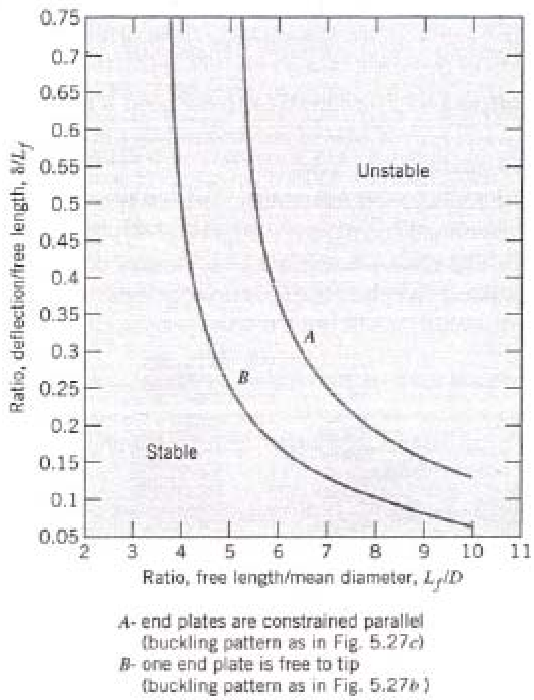


Figure 0.5N: Buckling conditions for helical compression springs.

All figures and tables in this section are adopted from Juvinal and Mershak, (2006).