University of Southern Queensland

Faculty of Engineering & Surveying

Hoist System for Small Boat Stacker

A dissertation submitted by

Foster Robert Lear

In fulfilment of the requirements of

ENG4112 Research Project Part 2

Towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: 1st December 2008

Abstract

Storage space in sport and competitive rowing sheds is limited. The use of racking is common to improve the utilisation of available floor space. This project aims to develop a hoisting system suitable for the vertical storage of the small boats (tinnies) that are used for coaching and other functions in the sport.

A search of available boat hoists revealed numerous manufactures and designs mainly in the United States. It seems the American people use boat hoists in the same way as we use boat trailers in Australia. These existing hoist products are used for out of water storage of small boats and are usually mounted lake side or attached to a wharf structure. None of the hoists located are capable of storing more than one boat vertically. A new type of hoist will be required.

Due to the time available for the design work this project has focused on researching hoist requirements and design and is restricted to the hoisting system. The hoisting system is an integral part of the overall hoist structure therefore some preliminary design work of the hoist structure has also be undertaken. Additional work to be completed before prototype fabrication is the design of the lifting arm guides and bearing assemblies and the design of the hoist frame structure.

Firm commercial interest exists in this product and for other storage solutions based upon this hoist

University of Southern Queensland

Faculty of Engineering and Surveying

ENG4111 Research Project Part 1 & ENG4112 Research Project Part 2

Limitations of Use

The Council of the University of Southern Queensland, its Faculty of Engineering and Surveying, and the staff of the University of Southern Queensland, do not accept any responsibility for the truth, accuracy or completeness of material contained within or associated with this dissertation.

Persons using all or any part of this material do so at their own risk, and not at the risk of the Council of the University of Southern Queensland, its Faculty of Engineering and Surveying or the staff of the University of Southern Queensland.

This dissertation reports an educational exercise and has no purpose or validity beyond this exercise. The sole purpose of the course "Project and Dissertation" is to contribute to the overall education within the student's chosen degree programme. This document, the associated hardware, software, drawings, and other material set out in the associated appendices should not be used for any other purpose: if they are so used, it is entirely at the risk of the user.

Professor Frank Bullen
Dean
Faculty of Engineering and Surveying

Certification of Dissertation

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Foster R. Lear
Student number: 0050088715

Signature

Date

Acknowledgements

Firstly I would like to thank Doug Lumsden of Space Saver Rowing Systems for providing me with this project, and for acting as a sounding board to guide my work.

I would also like to thank Dr David Ross for his guidance and advice with the dissertation.

Finally, thankyou Debbie for your support, and telling me to stop procrastinating throughout this project.

Foster Robert Lear

University of Southern Queensland

December 2007

Contents

List of Figures

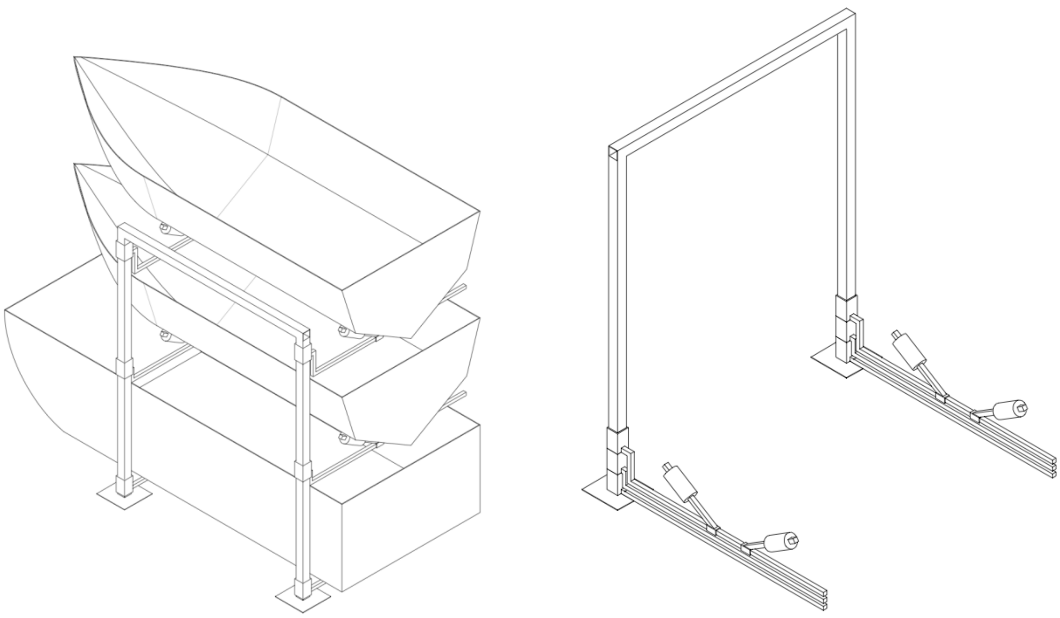
List of Tables

Chapter 1 - Introduction

1.1 Introduction

Space Saver Rowing Systems (known as SSRS) develop storage solutions for the rowing industry. They are looking to expand their product range and have identified a need for more efficient use of the floor space used for storage of small boats (tinnies) and trailers in rowing club sheds.

SSRS have provided several concept drawings of a hoist that stores 3 boats vertically, based on a fixed upright support frame and cantilevered lifting arms. These concept drawings are shown below.



(a) (b)

Figure 1.1 Hoist concept drawings. (a) Single row hoist with three boats in place. (b) Hoist concept shown with lifting arms in down position. The boats are rolled into the hoist at ground level for loading and unloading.

Single and double row hoist configurations are planned, a double row hoist can store six boats. This configuration is shown in Figure 1.3.

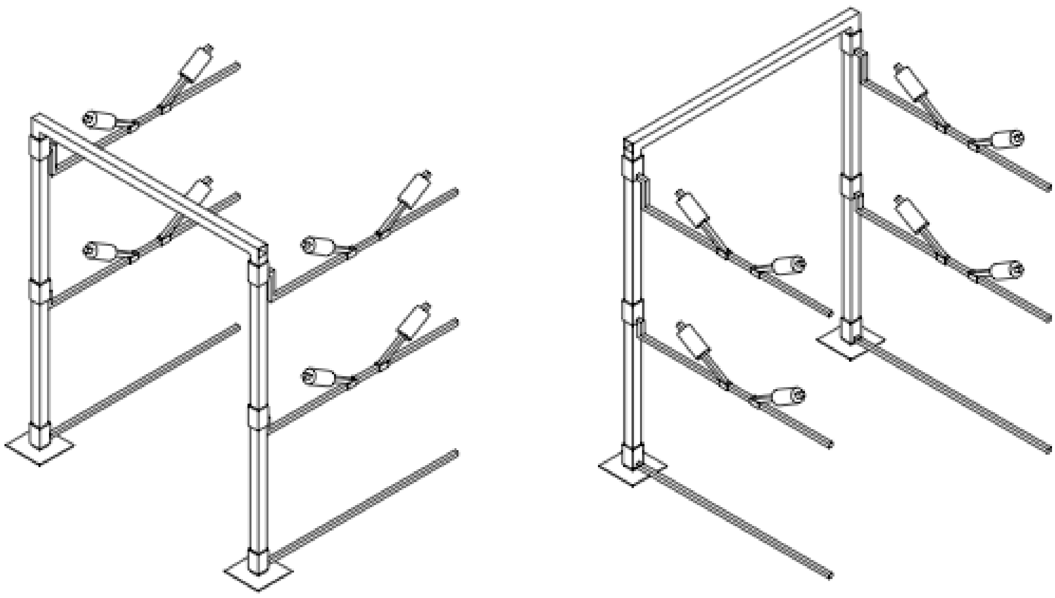


Figure 1.2 Two Views of hoist structure and lifting arms.

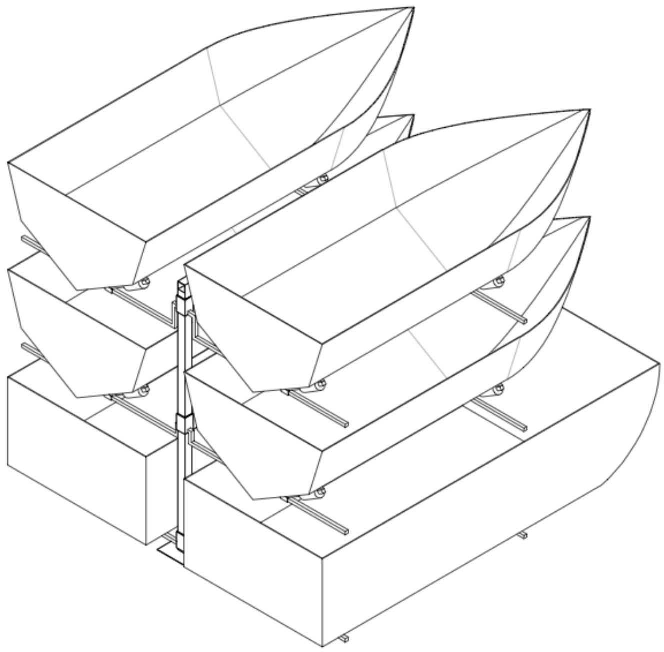


Figure 1.3 Concept drawing showing double row configuration.

1.2 Problem Statement

Boat hoists are widely used for out of water storage and maintenance boats. An internet search for 'boat hoists' revealed numerous boat hoist suppliers, designs and related products. Most of the products come from the US.

All of the ‘boat hoists’ found were designed to lift a single water craft, none are capable of stacking multiple boats vertically. Therefore to achieve the outcomes SSRS is looking for a new design to be developed.

It is anticipated this design work will be applied to other storage solutions in the future.

1.3 Objectives

This project aims to research hoisting system design and requirements. From this research, design concepts will be developed for critical evaluation. One system will then be selecting for development into a prototype design.

The bulk of the work in this project will be focused on the mechanical lifting system, because this system is part of the overall structure some consideration of the hoist structure design will be required.

1.4 Methodology

The steps to achieve the objectives of this project are as follows:

- Research legal requirements for hoists.
- Research relevant technical standards.
- Research existing boat hoists and related products.
- Create a list of potential solutions.
- Establish estimated costs for selected viable hoisting options.
- Discuss and evaluate viable options for the hoisting system with SSRS, before deciding which combination is best suited for the prototype.
- Define variables such as safe working load, critical dimensions etc.
- Select and detail the hoisting system components.
- Develop the prototype design.
- Fabricate prototype for testing and design refinement.

1.5 Overview

The work of this project has been separated into the following sections or chapters:

- Chapter 1 Introduction
- Chapter 2 Background research
- Chapter 3 Hoist structure concepts
- Chapter 4 Hoisting system options
- Chapter 5 Final design
- Chapter 6 Conclusions and further work

1.6 Conclusion

The introduction chapter has outlined the process that will be applied to developing a prototype hoisting system design.

Chapter 2 - Background research

2.1 Introduction

This chapter will review literature and research information for the purpose of developing the design criteria, the concept designs and the prototype design .

Specific areas to be reviewed include the following:

- Concept drawings and notes taken during meetings with SSRS, discussing their requirements.
- State based Occupational Health and Safety legislation and regulations pertaining to hoists.
- Relevant Australian Standards relating to hoists and specific components selected for the hoisting system.
- Existing products designs and patents.
- Boat and trailer manufacturers for typical parameters.
- Component and product manufacturer's information.

2.2 SSRS requirements

The requirements from SSRS cover the entire boat hoist, some items are more relevant to the structural design than the hoisting system, all points have been included.

The list has been produced from notes taken during meetings with SSRS and concept drawings they have provided:

- Design a boat hoist to maximise use of floor space where boats and trailers are stored. Hoist to store three boats vertically.
- The design should be flexible to allowing single and double row configurations and storage of a combination of boats and trailers as required. The design is likely to be adapted for the storage of other items.
- Design to a relevant technical standard to ensure compliance with Government Occupational Health and Safety Legislation.
- Safety systems are required to prevent hoist from collapsing in the event of component failure.

- Solution to be low cost, simple, safe and easy to use.
- The product is intended to be marketed to boat sellers, the hoist should be aesthetically pleasing and allow room for branding.
- Hoist to be suitable for installation inside and outside in the coastal environment. Subject to wind, rain, salt, etc.
- Each lifting arm will be fitted with adjustable supports for various boat hull and trailer configurations. A securing mechanism will hold the boats and trailers in place on the hoist.
- The hoist is to be loaded at ground level, and therefore requires the lifting arms to lay flat and as close to floor as possible. The top level is hoisted first and then the next boat to be lifted is rolled into place. Once the second level is lifted, an additional boat can be rolled underneath giving the 3 high storage solution required by SSRS.
- Hoist will be prefabricated for assembly onsite. The hoist is to be easily packed for transport. Design for transport of the hoists will be undertaken after the prototype has been tested.

2.3 Government legislation for hoists

Each Australian state and territory has enacted Occupational Health and Safety Legislation to provide a framework for Occupational Health and Safety Regulations for identifying, eliminating or controlling work related risks. These bodies consider cranes and hoists to be high risk plant.

Current government legislation and regulations reviewed:

- Queensland Workplace Health and Safety Act 1995.
- Queensland Workplace Health and Safety Regulation 1997, Schedules 3 and 4.
- NSW Occupational Health and Safety Act 2000 No. 40 Part 1.
- NSW Occupational Health and Safety Regulation 2001 Chapter 5.

The Queensland Department of Employment and Industrial Relations, Chief Safety Engineer, was contacted seeking advice on how this small boat hoist would be viewed in regard to Workplace Safety Regulations.

His advice was that under the current regulations (Queensland Workplace Health and Safety Regulation 1997), this type of hoist would not be classified as Registrable Plant or a Registrable Plant Design. However there is still a need for the design to comply with safety requirements. Designing to a technical standard is one way of fulfilling the designers requirements. This was confirmed by email on 6th August 2008.

The implications of this opinion are:

- The design does not require registration with Occupational Health & Safety in Queensland.
- Individual plant do not require inspection initially, or on an ongoing basis.
- There is a requirement to design to a technical standard, i.e. Australian Standards.

NSW Occupational Health and Safety were also contacted for their opinion on the requirements for this hoist under current legislation. They advised relevant sections of the Act and the Regulations applicable to hoists.

The Act and Regulations place obligations on designers, manufacturers, suppliers and operators of cranes, hoists and winches. These obligations are to ensure the plant is safe and doesn’t pose a risk to persons who are exposed to its use. Chapter 5 of the Regulations specify the provisions that apply to high risk plant.

The Regulations specify the following relevant to hoists:

- All cranes hoists and winches must now be designed to AS 1418 Cranes, hoists and winches.
- The ‘Regulation’ also places obligation of importers of hoist equipment and mechanical components used for cranes, hoists and winches, to ensure products comply with local Australian standards. Other standards maybe used providing they are at least equivalent to the applicable Australian Standard.

2.4 Australian Standards

2.4.1 Introduction to Australian Standards for cranes, hoists and winches

From the previous section Australian Occupational Health and Safety legislation, requires hoists to be designed and built to Australian Standards.

The following standards and specific sections have been identified as relevant to the design of this hoist:

- AS 1418.1-2002 Cranes Hoists and Winches, Part 1, General requirements.
- AS 2549-1996 Cranes (including hoists and winches) – Glossary of terms.
- AS 2089-2008 Sheave blocks for lifting purposes.
- AS 2759–2004 Steel wire rope—use, operation and maintenance.
- AS 1403-2004 Design of rotating steel shafts.
- Handbook 48-1999 Steel Structures Design Handbook.

AS 1418.1-2002 Cranes Hoists and Winches, Part 1, General Requirements. This standard is the basis for design of all cranes hoists and winches. Additional standards are required, depending on the classification of the hoist and for specific components used within the hoist.

2.4.2 Classification of cranes hoists and winches

There is a system of classification for cranes, hoist and winches. Its purpose is to rationalise the design of structures and machinery, to create standards by which users and manufacturers can match a particular crane to a service for which it is required.

Classifications only consider the conditions of operation and the intended service life of the crane.

Classifications are determined with the following information:

- Maximum number of operating cycles for intended service life. A single operating cycle is defined as commencing from an unloaded state through a loaded state and back to an unloaded state.

- Load Spectrum factor. The number of times a load of particular magnitude in relation to the crane capacity is lifted.

There are three classes of classification for cranes hoist and winches:

- Group classification (C) - applied to the overall crane.
- Structural classification (S) - applied to each part of the cranes structure.
- Mechanical classification (M) - applied to each mechanical component of the crane.

This project is only concerned with mechanical components of the hoisting system. Therefore only the mechanical classification process will be detailed.

2.4.2.1 Mechanical classification

The mechanical classification is determined with the use of the following tables and information contained in section 7.4 of AS 1418.1 (2002).

- Class of Utilisation of Mechanism, AS 1418.1 (2002) Table 7.3.4.2
- Nominal Load Spectrum Factor and State of Loading for Crane Mechanisms, AS 1418.1 (2002) Table 7.3.4.3
- Group Classification of Crane Mechanisms AS 1418.1 (2002) Table 7.3.4.4

2.4.3 Hoist loads

Loads are categorised and must be combined when assessing maximum stresses the mechanisms will experience. This is done as per AS 1418.1 (2002) section 7.9. Loads to be considered include the following:

- Loads from intended crane operation.
- Load caused by environment.
- Loads incurred in and out of service.
- Loads during erection.
- Loads during testing and fault finding.

Loads are categorised into principle loads, additional loads, and special loads, as per table 7.4.3 and listed below:

Principle loads:

- R1 Dead load of the mechanism or components.
- R2 Dead loads acting on the components i.e. crane hooks.
- R3 Loads due to live load acting on the crane hook.
- R4 Loads due to the dynamic affect of maximum acceleration of loads on the crane hook.
- R5 Loads due to maximum acceleration of the crane mechanism, including those due to the inertia of the mechanism itself.
- R6 Friction loads.
- V1 Wind forces acting horizontally in any direction.
- V2 Wind forces acting when hoist is out of service in any direction.

Additional Loads:

- Wind, snow and ice, temperature extremes, oblique travel.

Special loads:

- B1 load due to collision with buffers.
- B2 emergency conditions.

AS 1418.1 2002, section 7.9 describes how to use Table 7.9 ‘Load combinations for crane mechanisms’. For vertical motion loads R1,R3,R4,R5,R6 are to be considered. Other loads should be included if special conditions of operation require their inclusion.

2.4.4 Fail to safety (fail-safe systems)

AS 1418.1 2002 Appendix D discusses the following principles of fail safe systems.

The principle of fail safe is for the failure of a component to not cease service or reduce the safety of the plant. However, any failed component should be readily detectable, if not the probability of failure should be eliminated by design, quality control or by an inspection and maintenance routine. This is because any subsequent second failure may result in the plant becoming inoperable or unsafe.

Emergency stop system - locks out all power to the hoist and must be manually reset. The contactor that is operated by the emergency stop system must be correctly rated to limit the risk of the contacts becoming welded as a result of some fault condition This however is one of the key functions of a contactor and is not considered to cause any design issues.

Fail safe brake - brake requires power to keep the brake off, any interruption in power supply to the brake causes the brake to operate most commonly by spring pressure. Electric motors are available with inbuilt brakes that operate when the power supply to the motor is interrupted.

Structural elements - there are two basic methods of controlling structural failures first is to fit secondary structures to support the loading in case of failure of the primary structure. The second method is to design a primary structure to minimise the probability of failure.

Ratchet locks, redundancy in ratchet locks require a primary and secondary ratchet pawl. The secondary ratchet pawl must either engage with separate ratchet teeth or on a separate part of the primary ratchet teeth.

2.4.5 Wire ropes

AS 1418.1 2002 sections 7.14-7.16 discusses the requirements of wire ropes in cranes, hoists and winches and is summarised below:

- Each rope and component in rope systems shall be suitable for its particular application.
- Guys, fixed and stationary ropes have slightly lower coefficient of utilisation (Z_P) than hoisting ropes see AS 1418.1 2002 table 7.15.
- Maximum design load shall be determined by dynamic analysis.
- Where a reeved system has more than 10 parts an allowance must be made for friction as per AS 1418.1 (2002) Appendix G.

Note: Reeved systems increase the wire rope tension, due to bearing and journal friction and internal friction within the wire rope as it passes over the sheaves. An example of a reeved system can be seen in Figure 2.5

Wire rope selection is based upon the following information:

- Classification of the mechanism.
- Coefficient of utilisation (Z_P) for reeved systems from AS 1418.1 (2002) Table 7.16.2.1
- Calculated dynamic loading is multiplied by (Z_P) to determine the maximum load allowable for the wire rope.
- Reeved systems with more than 10 parts require an allowance for friction affects in the sheave bearings and internal friction within the wire rope induced by flexing and un-flexing as the rope passes over each sheave.

Minimum wire rope breaking load (F_O) is given by AS 1418.1 (2002) formula number 7.16.2.4 as below:

$$F_O = S_R \times Z_P \tag{2-1}$$

Where S_R = maximum wire rope tension in newtons, and

Z_P = minimum coefficient of utilization.

2.4.6 Sheaves for wire rope

Sheave design information is detailed in AS 1418.1 (2002) section 7.17-18 and AS 2089 (2008) Sheave blocks for lifting purposes.

2.4.6.1 General requirements for sheaves

- Sheave material must comply with the relevant Australian Standard, or equivalent as per AS 1418.1 clause 7.17.1. This clause specifies which standard is applicable for different sheave materials.
- Sheave guards should be fitted where there is a possibility of the rope being dislodged if there is slack in the rope at any time.
- If enclosure of the sheave is required, to protect personnel against injury or

protect the sheave from falling debris, it should still be visible to the operator.

- Fleet angle shall not exceed 5 degrees for direction of groove for sheaves.

Additional information on sheaves for lifting purposes is found in AS 2089 (2008) Sheave blocks for lifting purposes. A summary of this information is below:

- Clause 5.2.2 Bending stress in sheave axle is not to exceed 0.5 times the yield stress of the axle material.
- Clause 5.2.3 Shear stress in sheave axles for steel with strength grade of < 4.6 not to exceed 62 MPa. For higher tensile steel, not to exceed UTS_{STEEL} / 6.9 MPa.
- Clause 5.2.4 States maximum allowable bearing pressures for non-metal materials should not exceed the manufacturers recommended maximum.
- Clause 5.2.5 Discusses the bearing pressure between axles and axle support members. Where the steel has a UTS < 430 MPa then maximum bearing pressure is 125 MPa. If the steel has a UTS > 430 MPa then the maximum bearing pressure is calculated using UTS_{STEEL} / 3.4 MPa.
- Section 5.9 Axle surfaces shall be appropriate to bearings used. Axles shall be secured by means other than welding to prevent rotation and lateral movement of axle. Nuts shall be positively secured.
- Section 5.3 Discusses side and partition plates for sheaves for rope retention. With the following requirements:
    - Side plates shall not interfere with smooth rotation of the sheave.
    - Smooth edges.
    - Plates shall project beyond the sheave to protect the rope.
    - Measures are to be taken to prevent wire from jamming between the sheave and side plates. Either by clearance between sheave and plates less than 1 mm + 0.4% sheave diameter or by means of constraint.
    - Where the rope is not adequately constrained the sheave shall be shaped so that the rope cannot run on the outer perimeter of the sheave.

2.4.6.2 Sheave dimensions

Minimum sheave diameter is determined using the following information:

- Classification of the Mechanism

- Minimum ratio of sheave pitch diameter to wire diameter (h_S) from AS 1418.1 (2002) Table 7.18
- Minimum wire diameter (d_{min})

Minimum sheave pitch diameter (D_S) is given by AS 1418.1 (2002) formula 7.18(2) as follows:

$$\text{Minimum sheave pitch diameter } (D_S) \geq h_S \text{ x } d_{min} \qquad (2\text{-}2)$$

According to AS 2089 (2008) the minimum required groove depth for sheaves for cranes is the 1.0 * rope diameter. However (A.Nobles&Son 2006) recommend 1.5 * rope diameter to be the minimum.

2.4.6.3 Sheave bearings and bushes

This section will discuss the types of bearings and bushes suitable for sheaves and the design criteria applicable for the chosen type of bearing.

Types of bushing available for sheaves:

- Bronze Bushes
- Taper roller bearings
- Double row cylindrical roller bearings
- PTFE lined bushes

PTFE lined bushes are a composite structure of epoxy encapsulated glass fibre impregnated with PTFE. Properties of Reinforced Teflon bushes from the ‘Nobles’ datasheet and the ‘Machinery’s Handbook’ as follows:

- High load carrying capacity (140Mpa)
- Excellent self lubricating properties i.e. low friction
- Wide temperature range
- Chemical resistant

The above properties make Reinforced PTFE bearings the logical choice for the sheave bearings.

In the Machinery's Handbook (Erik Oberg 2008) operating clearance for plain bearing should be 0.001-0.0025" per inch of journal diameter. This clearance applies to reinforced PTFE bearings, however some plastic bearings can swell as they absorb water and therefore the initial clearance is increased to 0.004-0.006" per inch. The potential for swelling can be checked by boiling a bearing in water, measuring the diameter before and after.

To ensure good bearing life the sheave axle bearing surface finish must be adequate. According to (A.D. Deutschmann 1975) surface finish requirements for a journal bearing is 32 μ in., rms. This level of surface finish can be achieved by turning or surface grinding.

Sheave loading is calculated using the following formula (Erik Oberg 2008). Note: the original formula is in psi and has been converted to metric.

Bearing pressure $P_B = W / 2R_J L$ N/mm2 (or MPa) (2-3)

Where W = force in newtons,

R_J = journal radius in mm,

L = bearing length.

From Engineering Mechanics (J.L. Meriam 1987) typical coefficient of friction (static and dynamic) for teflon (PTFE) on steel is 0.04. The following equation calculates the moment required to overcome friction in a plain bearing:

$M_{friction} = \mu L r$ (N) (2-4)

Where μ = coefficient of friction,

L = bearing load (N),

r = bearing radius (m).

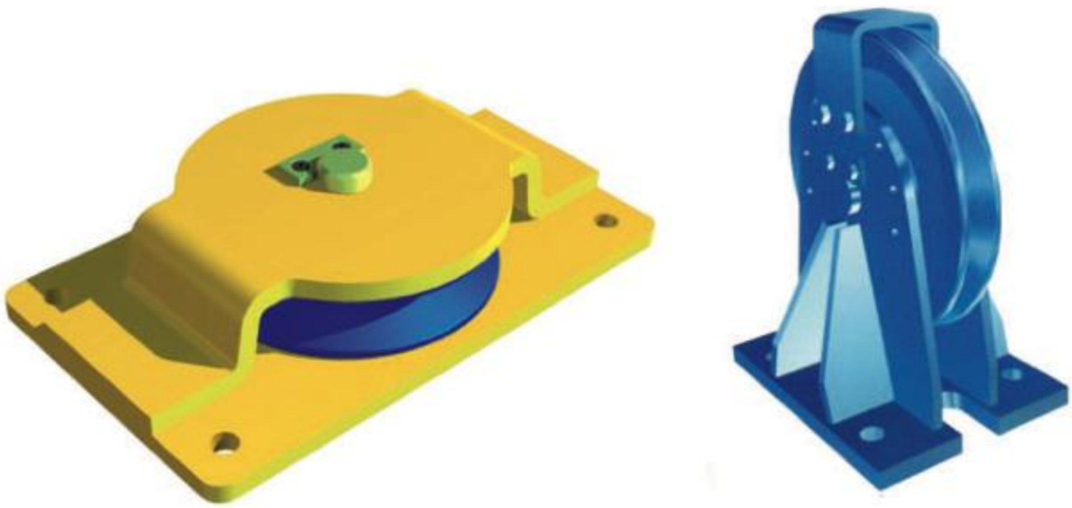


Figure 2.1 Sheave mounting options.

Figure 2.1 shows two mounting arrangements for wire rope sheaves. Both arrangements have bearings either side of the sheave to limit bending stress within the axle. If possible the sheave mounting in this hoist will use the same approach.

2.4.7 Wire rope drums

2.4.7.1 Wire rope drum general requirements

AS 1418.1 (2002) section 7.18-7.19 and Appendix I, K and L deal with wire rope drum design.

A summary of relevant information from these sections is given below:

- Drum material grade standards are detailed in AS 1418.1 (2002) section 7.19.1.
- Grooved drums shall have at least 2 grooves full for each rope when each rope is payed out to maximum.
- The drum should be sized, allowing the rope to be stored in a single layer with at least one complete groove empty for each rope. If rope is to be stored in more than one layer additional requirements relating to drum flanges apply.
- The drum groove pitch diameter shall be a minimum of 1.06 times wire rope diameter. Groove profiles are detailed in AS 1418.1 (2002) Appendix K.
- Fleet angle shall not exceed 5 degrees for direction of groove for grooved drum.

Additional information on wire rope anchorages is discussed in AS 2759 (2004) Steel wire rope—use, operation and maintenance.

- Rope anchorage must be able to withstand 2 times rated capacity.
- Anchorage shall not decrease rope strength by more than 20 per cent.
- Any sharp contacting edges should be removed.
- Where anchorage relies on clamping, at least 2 clamps be used.
- Clamping plates should support as much of the rope circumference as possible.
- AS 2759 (2004) Figures 9.2(a), (b) show various types of rope anchorages suitable for reverse and non-reverse winding rope drums these figures are attached in Appendix G.

2.4.7.2 Drum dimensions

Minimum drum pitch diameter is determined in a similar method as sheave diameters, using the following information:

- Classification of the Mechanism.
- Minimum ratio of drum pitch diameter to wire diameter (h_d) from AS 1418.1 (2002) Table 7.18.
- Minimum wire diameter (d_{min})

Minimum drum pitch diameter (D_d) is given by AS 1418.1 (2002) formula 7.18(1) as follows:

$$\text{Minimum drum pitch diameter } (D_d) \geq h_d \text{ x } d_{min} \tag{2-5}$$

Minimum drum thickness calculations are given by AS 1418.1 (2002) formula 7.19.5. This is an involved calculation and will be undertaken in the final design. Thickness cannot be less than 5mm for grey iron and not less than 3 mm for other materials.

$$T_D = (T_{DB}{}^2 + T_{DB*}\, T_{DC} + T_{DC}{}^2)^{1/2} \tag{2-6}$$

Where T_{DB} = minimum thickness allowing for bending stress in mm.

$$T_{DB} = \frac{1250 \times M}{D_{DM}{}^2 \times F_B} \tag{2-7}$$

M = maximum bending moment in Nm,

D_{DM} = mean diameter of drum shell in mm,

F_B = permissible bending stress in MPa

= 0.67 times yield stress.

And T_{DC} = minimum thickness allowing for compressive stress in mm.

$$T_{DC} = \frac{1000 \times K_{RL} \times P_{RS}}{p \times F_C} \tag{2-8}$$

K_{RL} = rope lay factor = 1.0 for single layer,

P_{RS} = maximum unfactored static rope load in kN,

p = pitch of rope coil in mm,

F_C = permissible compressive stress from AS 1418.1 table 7.19.5

= 125 MPa,

d = nominal rope diameter in mm.

Drum width for this hoist will be calculated to store all the wire rope required in a single layer. Two turns are required to remain on the drum when the hoist is fully lowered. One turn should be empty once the hoist is fully raised therefore minimum drum width can be calculated by the following formula:

Drum width = (hoist travel / drum pitch circumference +3)*Wire diameter. (2-9)

2.4.8 Rotating steel shafts and bearings

Steel shafts for cranes shall comply with AS 1403 (2004) Design of rotating steel shafts with load factors in AS 1418.1 (2002) Table 4.8 to be considered.

This standard advises high strength steel shaft, fatigue strength is generally affected more by stress raising characteristics than shaft made from low strength steels. Therefore shaft and fittings design will endeavor to reduce these stress raising factors.

The formula for calculating minimum shaft diameter is from AS 1403 (2004) Table 2 as follows:

For a shaft starting less than 600 times a year and more than 900 revolutions per year with no torque reversals (the torque on this hoist wire drum is always in the same direction). Minimum shaft diameter is given by Formula 4, As follows

$$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{4}{T_q}^2} \qquad (2\text{-}10)$$

Where D = minimum shaft diameter (mm)

F_S = safety factor = 1.2 (as per AS 1403 (2004), Table 2, note 5)

F_R = endurance limit of shaft material, reversed bending during rotation or 0.45*tensile strength.

K_S = Size factor as per AS 1403, clause 8.1

K = Stress raising factor as per AS 1403, clause 8.2

M_q = bending moment at shaft cross section in (Nm)

P_q = maximum axial tensile force at shaft cross section (N)

T_q = Maximum torque at shaft cross section (Nm)

For solid shafts with a maximum of 2 keyways according to AS 1403 (2004) clause 6.2(a) the outside diameter shall not be less than D.

For hollow shafts from AS 1403 (2004) Section 6.3:

- The hole shall have a constant diameter and coaxial with outside.
- The wall thickness shall be greater than 0.15*D.
- The outside diameter (D_O) shall be greater than the value calculated by the formula,

$$D_O^3 = D^3 + 1.7D_I^3 \qquad (2\text{-}11)$$

Where D = calculated minimum diameter,

D_I = shaft internal diameter.

A worked example for a calculating the shaft diameters for a crane hoist drive is included in AS 1403 (2004) Appendix E.

According to AS 1418.1 (2002) section 7.10.2 and 7.10.3.5, bearings design shall be based upon load factor K_m (see section 2.4.4) with the bearing load following limits:

- Ball bearings maximum load applied < $K_m^{1/3}$ *full load.
- Roller bearings maximum load applied < $K_m^{1/3}$ *full load.

The support structure of a bearing shall be designed so that no major part of the crane or load will be dropped as a result of bearing failure.

2.5 Existing product and design review

2.5.1 Introduction

This chapter is a review of existing boat hoist, hoist safety latch designs and products. Key features and operating principles will be discussed for each item.

Review of existing designs is important in the design process to avoid the need to reinvent the wheel. A design or product maybe found that can be used by SSRS as is, or with minimal modification. At this time none of the designs reviewed are capable or designed to stack multiple boats vertically. Some of the features of the reviewed designs are potential solutions for the purpose of this project, in particular the hoisting system.

One thing to keep in mind is the actual performance of a patent is unknown, whereas an existing product should have overcome teething problems. From my research into existing products it appears at least some of the patents reviewed are being marketed now.

2.5.2 Patent - boat dock and lift

This patent (Layton J. Reprogle 1987) is a cantilevered boat hoist .

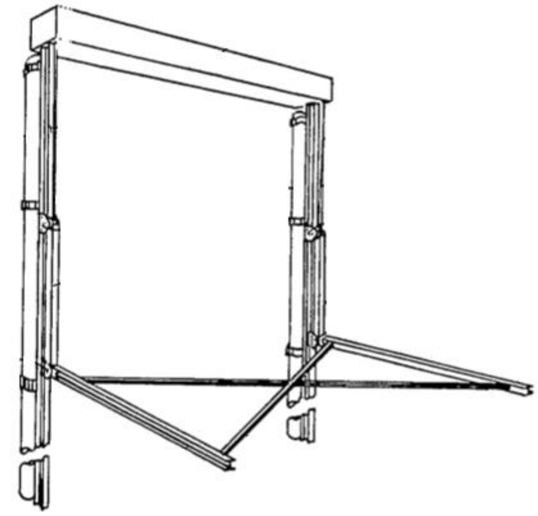


Figure 2.2 Cantilevered boat hoist.

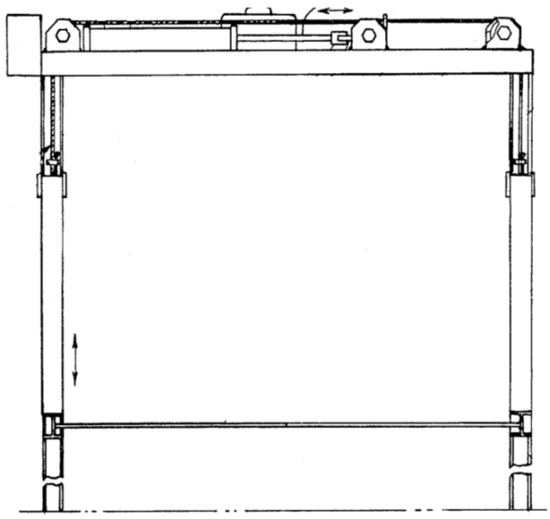


Figure 2.3 Plan view of cantilevered boat hoist showing actuator location.

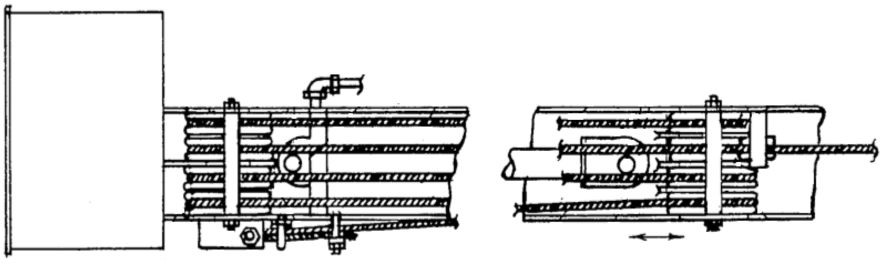


Figure 2.4 Multi sheaved hoisting system used in cantilever boat hoist.

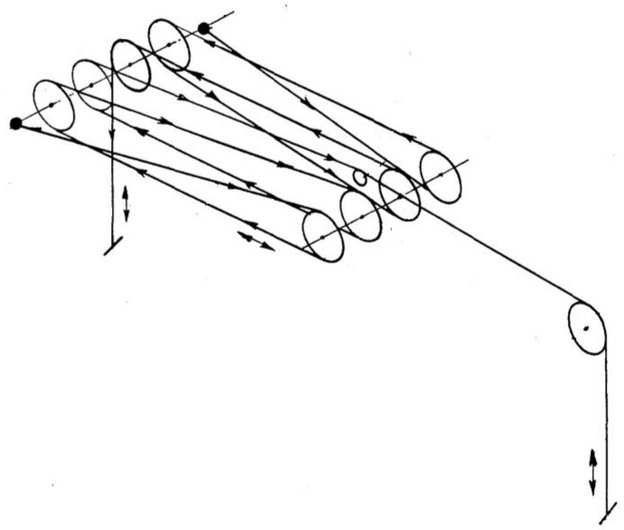


Figure 2.5 Schematic drawing of cable and sheave arrangement.

Key Features:

- Lifting hardware located on top horizontal member clear of water and easily protected from environmental elements.
- Powered by single hydraulic cylinder acting on reeved wire rope system achieving mechanical disadvantage of 1:4. This ratio means the hoist moves four times the distance the cylinder moves.
- Two wire ropes, one attached to each lifting arm, are simultaneously operated by one cylinder resulting in even lifting of arms.

2.5.3 Patent - floating dock and boat lift

This patent (Holmgren 1997) describes a floating dock arrangement. The hoisting system is based upon a motor driven screw thread providing linear motion for the hoisting wires.

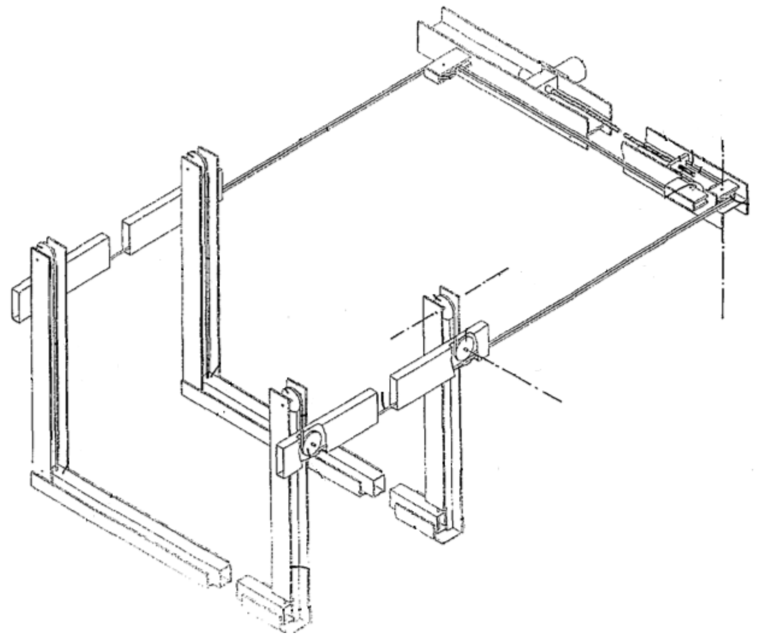


Figure 2.6 Boat hoist.

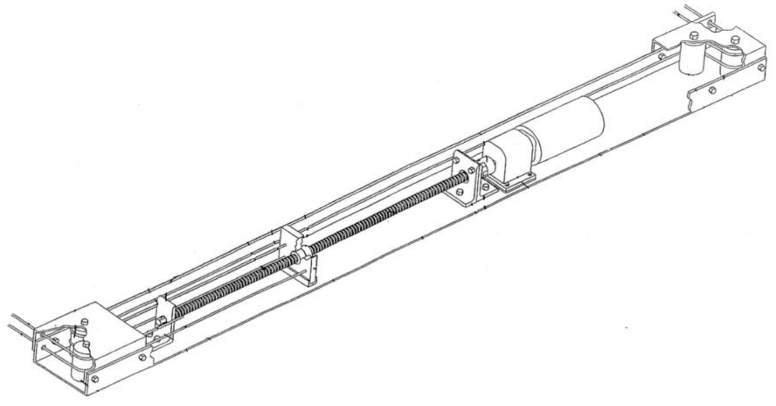


Figure 2.7 Screw lifting mechanism.

- Mechanical advantage achieved by screw thread.
- Use of multiple wires possible.
- Even movement of each wire due to single lifting point negates need for levelling mechanisms.
- Option of low voltage motor powered and independent power supply via batteries.
- Option of solar charger for batteries.

2.5.4 Patent - boat hoist

This patent (Byron L. Godbersen 1983) is for a hand powered hoist and its main design feature is the system for maintaining a level hoisting platform, free of twisting during operation.

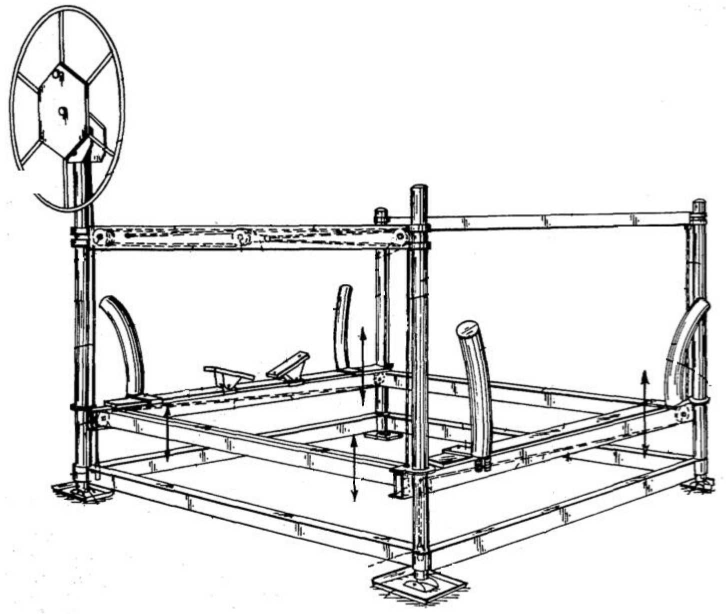


Figure 2.8 Free standing boat hoist.

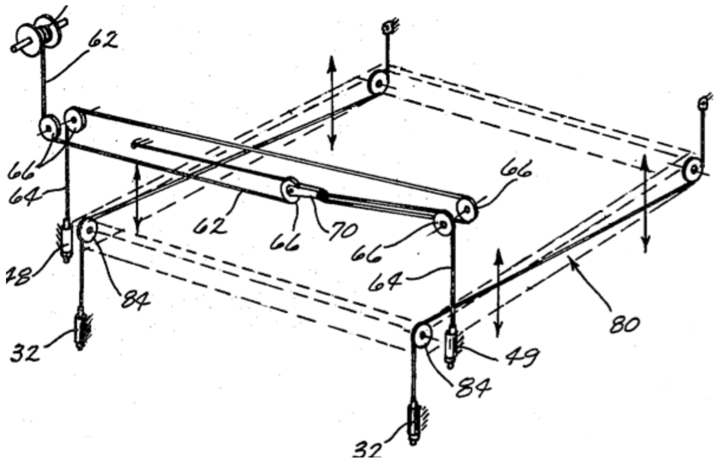


Figure 2.9 Wire system for free standing hoist.

- Single lift wire acting via pulley to two lifting points. Ensures even lifting of hoisted side.
- Self levelling wires across ends of hoisting platform.
- Significant mechanical advantage achieved with large diameter hand-wheel driving geared winch drum.

- Additional 2:1 mechanical advantage achieved by pulley (item 66) and single hoisting wire (item 62).

2.5.5 Patent - powered boatlift with electronic controls

This patent (Wayne G. Floe 2006) describes a boat hoist with a single hoisting point at one of the 4 corners of the lifting platform. The lifting platform is kept level by the application of 3 self levelling wires in a similar fashion to previous patents. The key additional features in this design are related to the electronic control of the hoisting system.

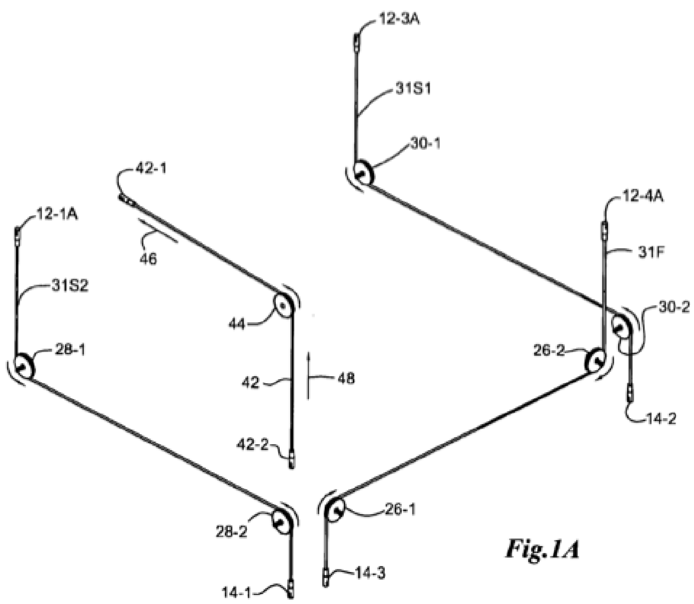


Figure 2.10 Wire drawing of hoisting and levelling system.

Wire 42 is the single hoisting wire, wires 31F1, 31S1 and 31S2 are levelling wires to maintain a level hoisting platform.

- Hoisting system is a powered screw thread housed within a horizontal member acting on a single hoisting wire.
- Screw thread is driven by a reversible electric motor via first a belt drive, and secondly by a chain drive. Both sets of drive are geared to achieve mechanical advantage.
- Adjustable feet for levelling hoist frame on uneven surface.
- Electrically released mechanical brake. Brake engages automatically when

stopped.

- Regenerative electric braking system to limit speed during lowering.
- Mechanical limit switches driven from the gearbox output shaft stop hoist on upper and lower limits.
- Reverse delay, to prevent shock loads resulting from sudden reversal of control signal.
- Load sensing circuit (motor current monitor) locks out hoist if overloaded, requires manual reset.
- Control by either manual push-buttons or by remote control.

2.5.6 Patent - programmable lift control

This patent (James A. Endres 1997) discusses a control system for automatic hoisting of the platform. However, the hoisting system itself is different from previously examined designs.

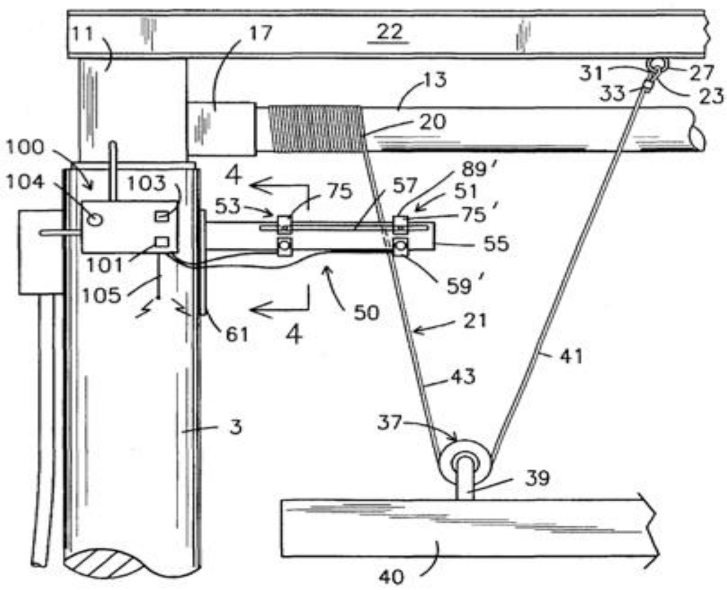


Figure 2.11 Boat hoist winch and limit switch arrangement.

- The wire drum in this hoist system is a length of pipe instead of the usual drum end.
- Because the wire is layed only a single wrap there is never any problem with a

difference in drum diameter causing uneven hoisting.

- Magnetic proximity switches are mounted so they are operated by movement of beam (item 57) when contact is made with the wire (item 21) due to the changing angle as more or less wire is wound onto the pipe.
- A possible improvement to this design would be to use a grooved drum which tends to locate the wire more positively, keeping the wraps tight.
- 2:1 mechanical advantage is achieved with pulley (item 37).

2.5.7 Patent application - electromagnetic release.

This latching system patent application (Wolfgang Raffler 2006) is for a vehicle hoist. The latch is based on a spring loaded latching pawl that engages with a perforated steel section. The latch pawl engages with each step in the perforation. When lowering the hoist the weight is taken off the pawl and then a solenoid retracts the pawl against the spring pressure. The solenoid is connected to the end of the chain shown in figure 2.12.

This particular latch is of interest because of the mechanism simplicity and the perforated plate is easily incorporated into this hoist.

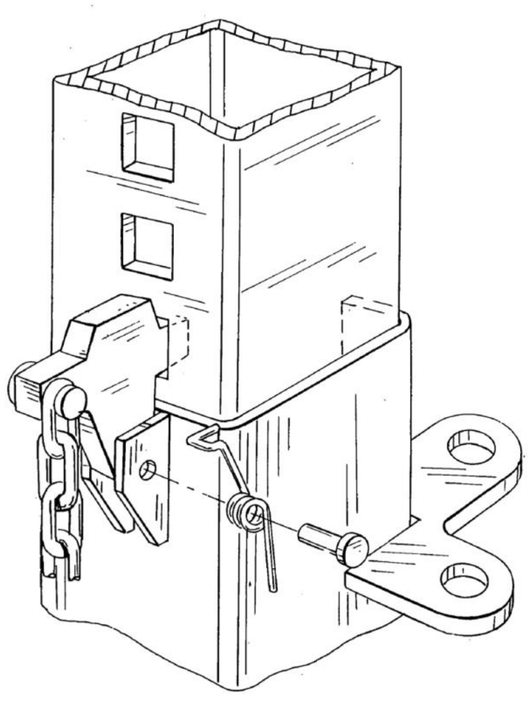


Figure 2.12 Solenoid released hoist safety latch.

2.5.8 Patent - hoist locking and release apparatus.

This patent (Jon S. Halstead 1998) describes a latch that uses a toothed plate to engage with the latch pawl. The difference here is the latch is fitted with a cam that releases the latch when the hoist is raised slightly. The latch remains disengaged until the hoist reaches the bottom, where the latch pawl 'resets', ready to engages with each tooth as the hoist is raised.

The key feature of this system is the latch is automatically released from its locked position when the hoist is raised slightly.

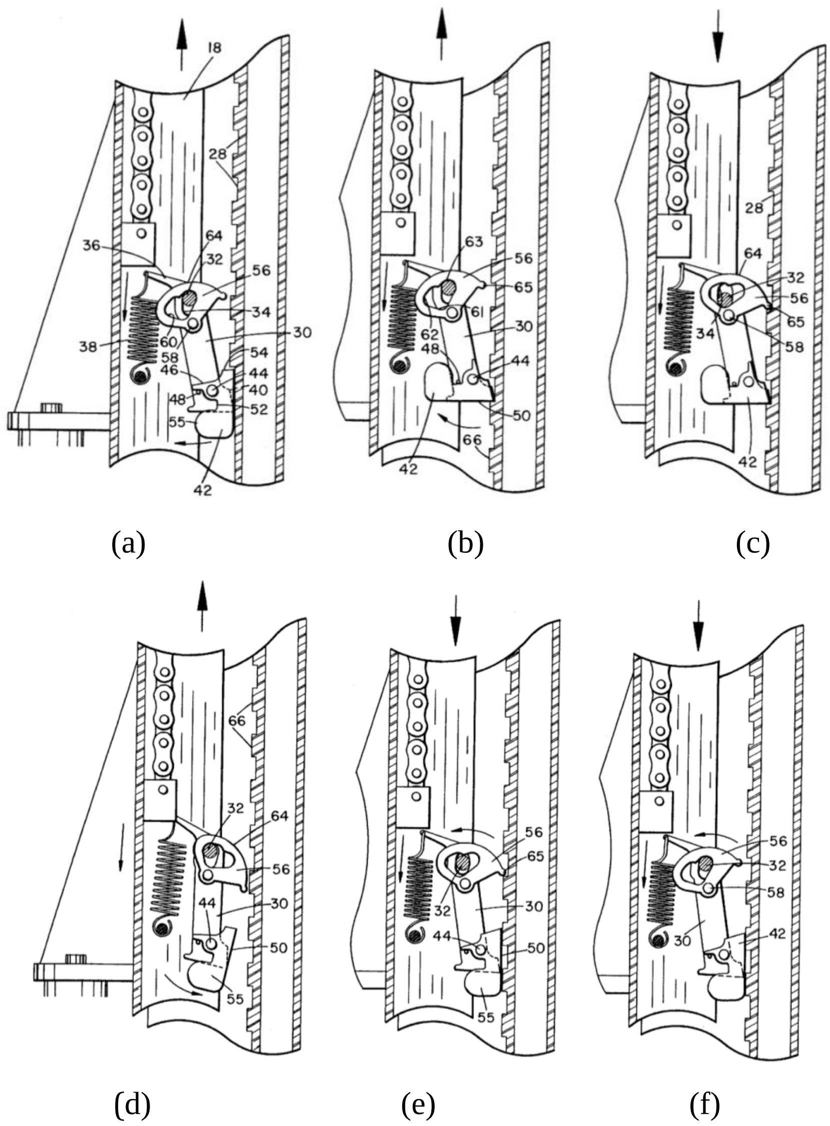


Figure 2.13 Six view of hoist safety latch operation (a) shows the hoist at the lowest position ready to be 'reset'. (b) shows the latch reset and engaging with each tooth as the hoist is raised. (c) the hoist is now lowered slightly to engage the latch. (d) to release the latch the hoist is raised slightly the cam disengages the latch pawl and allows the hoist to be lowered as shown in (d). (f) shows the cam and latch ready for resetting once the hoist is fully lowered.

2.5.9 Reimann & Georger Corp. Marine Products (RGC)

RGC Marine Products (Reimann & Georger Corp 2006) manufacture boat hoists such as the one below:

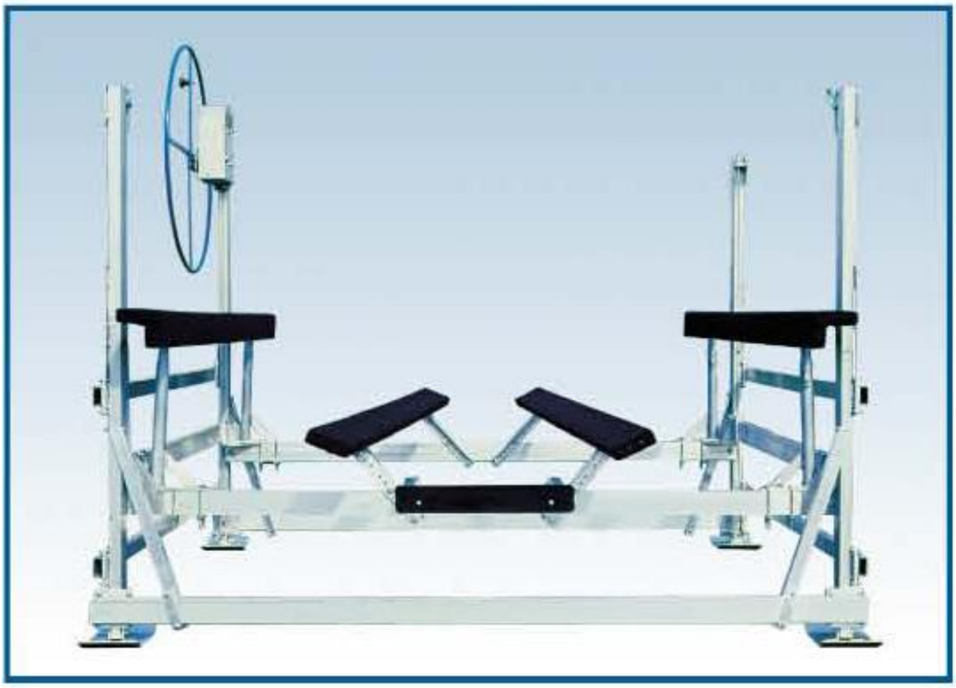


Figure 2.14 RGC vertical boat lift.

- Vertical Boat Lifts rated from 500kg to 4500kg.
- Frame Marine grade aluminium.
- Stainless steel cables and hardware.
- Polymer sheaves for maximum cable life.
- Hand powered or optional electric friction or electric power drive.
- Chain winch used for higher rated lifts.

Figure 2.15 RGCs AC or DC Friction drive attached to hand powered hoist winch.

Figure 2.16 RGC AC or DC remote control power drive.

2.5.10 Flat-plate boat hoist

This flat plate winch (Boat Hoist USA 2008) is specifically designed for boat hoists. Boat Hoist USA is also a retailer of boat hoists, parts and other marine equipment.

The flat plate boat hoist is an off the shelf electric powered winch drive. The electric motor drives a 5:1 Vee-belt reduction which in turn drives a 96 tooth worm wheel.

The winch plate is designed to be attached to a 2” diameter shed 40 pipe or grooved wire drum around which the hoist cable is wound. Any changes to this will void the lifetime warranty. The flat plate winch is not a load bearing unit, weight of the lift and lifting hardware needs to be supported by suitable bearings fitted to the winch pipe.

Figure 2.17 Boat Hoist USA BH70 7000 lb (3075 kg) flat plate winch.

2.5.11 Summary of hoist and hoist system designs

The hoisting system can be divided into sections for the purpose of evaluation. Below is a summary of potential solutions gathered by evaluation of existing products in chapter 3 and a general brainstorm for ideas.

Hoist power system. A study of the existing product available identifies the following methods of powering the hoist system:

- Hydraulic ram;

- screw thread, electrically driven, though could be hand driven;
- geared hand winch drum;
- geared electric drum winch; and
- direct mechanism i.e. rack, ratchet hoist, chain hoist.

Mechanical advantage achieved by:

- gearing in drive;
- hydraulic power;
- reeved wire rope; and
- screw thread.

Hoisting medium:

- wire rope;
- chain;
- soft strapping; and
- direct drive screw thread.

Power source:

- hand;
- electric 240Vac;
- electric low voltage DC; and
- electric hydraulic.

Levelling system:

- self levelling wires; and
- multiple lift points.

2.6 Boat manufacturers

Boat and trailer dimensions and weights are easily established from manufacturer's specification sheets. The manufacturers also recommend the maximum engine weight suitable for each of their boats.

	Length (M)	Depth (M)	Beam (M)	Height on trailer (M)	Weight boat only (kg)	Maximum Engine weight (kg)
Quintrex 395 Dart	4.00	0.91	1.69	1.5	95	90
420 Quintrex Dory Wide Body	4.27	1.07	1.87	1.6	164	110
435 Hornet trophy	4.43	0.94	2.02	1.7	280	120
Qunitrex trailers V-line Series	3.85-5.45	-	1.88-2.24	-	130-260	-
Alloy Craft 3.75 Open HD Cody	3.75	0.81	1.63		85	58
Ally Craft 3.95 Open HD - Avalon	4.00	0.90	1.70		106	87
Ally Craft 4.25m v-bow Shadow Deluxe	4.25	0.93	1.90		163	110
Ally Craft 4.25m v-bow Shadow Mirage	4.25	0.93	1.90		200	110
Ally Craft 4.45m Open HD - Sioux	4.45	1.16	2.25		223	120
Alloy Craft 4.95 Open HD - Abalone	4.95	1.22	1.95		270	178
Average	4.26	0.99	1.88		176	109
Maximum	5.45	1.22	2.24		280	

Table 2.1 Boat dimensions and weights.

This information is tabulated for review and establishing the design parameters for the hoist.

2.7 Component selection

The following section discusses some hardware that may be used for this hoist. The section is intended as background research to understand some of the characteristics of this equipment.

2.7.1 Power screws

2.7.1.1 Introduction to power screws

Power screws transfer rotary motion to linear motion in a smooth manner. They are extensively used for lathe lead screws, screw jacks, valve stems, screw type presses, clamps etc (A.D. Deutschmann 1975).

There are two distinct classes of power screws, one uses various thread forms such as square thread, ACME thread, buttress threads and modified versions thereof. The second type uses a ball or roller screw system where rolling elements are used. The big advantage of the ball screw is the much higher efficiency, up to 90%, as opposed to around 70% for an acme thread.

Each thread type has its own advantages and disadvantages. The following summary has been put together from (A.D. Deutschmann 1975) and www.roymech.co.uk (Roy Beardmore 2008).

2.7.1.2 Square thread form

- Highest efficiency all sliding thread forms.
- Most difficult to machine.
- Used for linear jacks and clamps.
- Not very compatible with split nuts (split nuts are used to control backlash and compensate for wear).

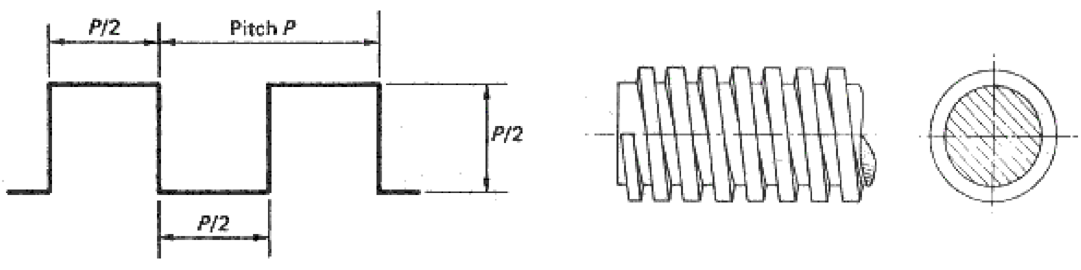


Figure 2.18 Square Thread.

2.7.1.3 ACME Thread

- Earliest type of power screw used for lead screws on lathes.
- Easier to manufacture than square thread.

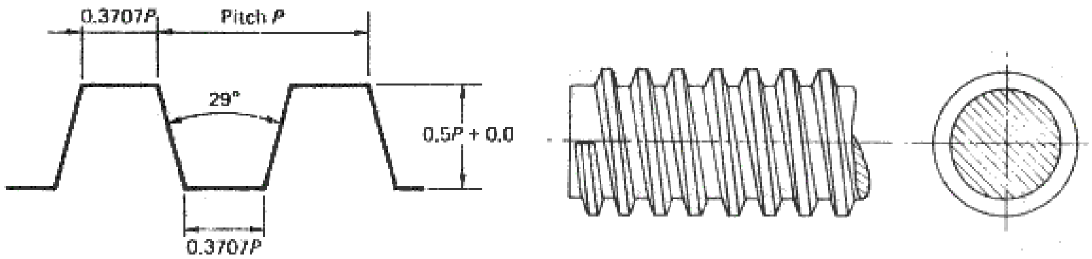


Figure 2.19 ACME thread.

2.7.1.4 Buttress thread

- Designed to resist loads in one direction only.
- Stronger than other forms for given size due to greater root thickness.
- Almost as efficient as the square thread.

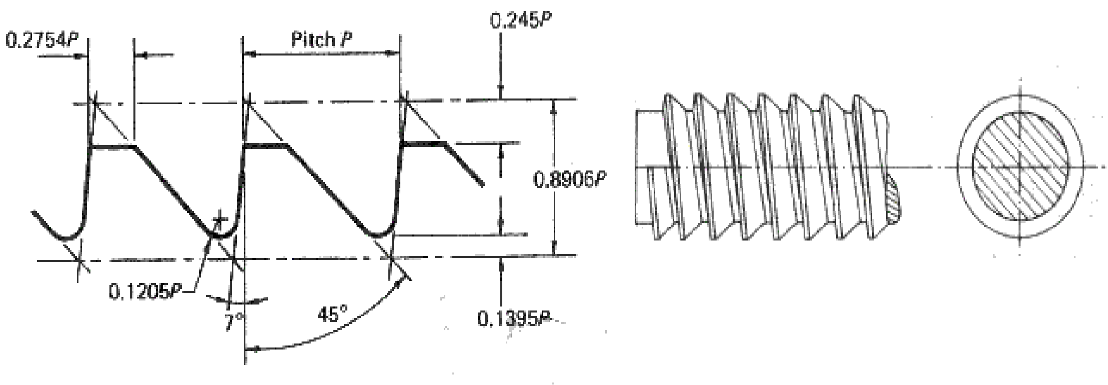


Figure 2.20 Buttress thread.

2.7.1.5 Ball and Roller Screws

Ball and roller screws use rolling elements to support the load. The screws are machined to suit either ball or roller bearings. And the nut allows the balls to circulate as the screw is rotated.

- High efficiency up to 90 per cent.

- Lower starting torque.
- Predictable life expectancy.
- Precise positioning and repeatability.
- Smaller ball nut due to higher load capacity of balls.
- Requires higher level of lubrication.
- Needs additional braking system due to the high efficiency they are prone to rolling under static loads.
- Susceptible to contamination.
- Because load capacity is higher, the screw is not as stiff as an equivalent thread screw.

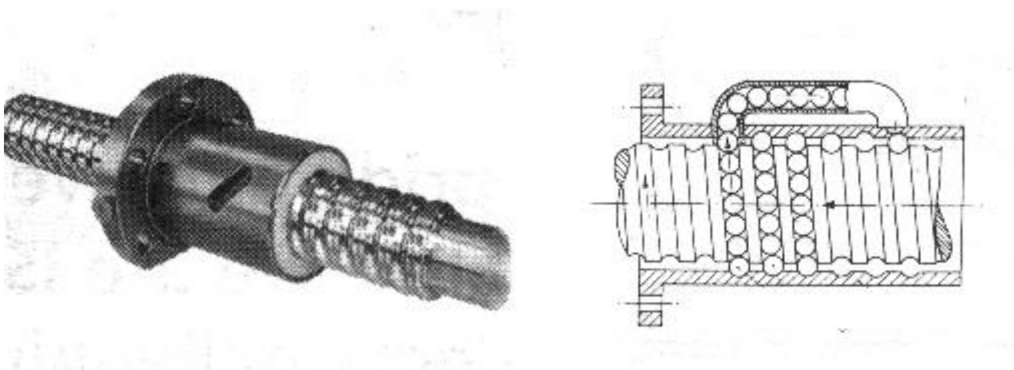


Figure 2.21 Recirculating ball screw.

Ball and roller screw manufacturers provide selection guides and example calculations for matching ball and roller screws to a particular task.

Ball screws nuts are not designed to support twisting forces on the nut. Therefore a nut guiding system will have to be considered.

2.7.2 Motor, gearbox and drive

Specialist advice was sought from SEW Eurodrive, Melbourne for motor, gearbox and drive selection. General requirements as follows:

- Single phase power supply.
- 3 phase motor (lower cost and more options than single phase motors).
- Brake motor, a disc brake assembly is mounted on the non-drive end of the electric motor. The brakes are generally engaged by spring pressure and engage

whenever power to the motor is removed.

- Inverter single phase supply to 3 phase output. Allows acceleration, and speed control.

2.7.3 Self leveling wires

Self levelling wires have been used in several of the hoist patents reviewed. The wires are designed to keep the hoist platform level when a hoisting force is being applied to a single point on the hoisting platform.

The wires interconnect two points on the hoisting platform. As one point of the hoist is moved by the lifting force, tension forces generated in the wire, create a resultant force acting upon the opposite corner, moving it in the same direction. The wire must be pre-tensioned and rated to full load as it is subject to the same force as the lifting wire.

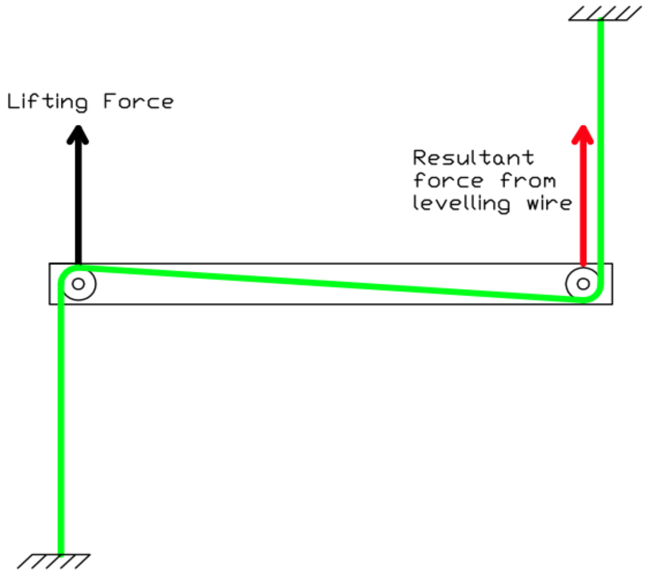


Figure 2.22 Self levelling wire schematic diagram.

2.8 Conclusions

This chapter has outlined the background information required to design the hoisting system concepts.

Existing hoists and patents have been reviewed, none of which have been designed or are capable of hoisting multiple boats at the same time.

A list of potential components for the hoisting system has been created and sorted into sections that make up a complete system. A combination of these items will be required to complete the design of the hoisting system. Each item will be evaluated for key features, advantages and disadvantages.

This information will be used in the next chapter to develop design concepts.

An important point regarding all of the hoists reviewed above is: they all use a single hoist unit i.e. one actuator. This overcomes the need for a system to synchronising multiple winches. A synchronising system would be required to maintaining a level hoist and ensure that variations in level do not accumulate over a number of winching cycles, resulting in lifting arms that are no longer level.

Chapter 3 - Hoist structure concepts

3.1 Introduction

This chapter outlines several design concepts, one of which will be chosen at the end and developed to the next stage, final design for prototyping. The design work has been divided into two sections, first the hoist arrangement, and second the hoisting system. This separation of the structure from the hoist system allows each distinct section to be assessed individually. Concepts will be detailed for each section and brought together in the decision phase.

At this time the structure drawings show the hoist concepts only and not the structural support, mounting or connections between the upright sections as this is considered to be outside the scope of this project.

To develop several design concepts the following process was undertaken:

1. Review existing products and other equipment for ideas.
2. Carry out a ‘brainstorming’ step to come up with distinct concepts with a high level of variation.
3. Reject concepts that are deemed not suitable.
4. Refine the chosen concepts during the drawing and detailing process.
5. Decide, which concept will be prototyped.

The decision process ideally is as objective as possible to best satisfy the requirements of the design. This way each concept will be treated without favouritism, and will be assessed against the same criteria.

3.2 Design criteria

Review of existing hoist designs and the requirement of SSRS have been used to develop the design criteria. This criteria will be added to as the design evolves.

Rated lifting capacity, 500 kg per arm. Assuming 2 lifting arms, and the double sided configuration this equates to total rating of 2000 kg not including lifting hardware, safety factor, frictional losses etc. Because the weight distribution within the boat is likely to be uneven each lifting arm must be rated to support the entire load for each level i.e. 500 kg for each arm.

Stand alone structure. Hoist cannot depend on mountings for structural strength. This allows the hoist to be relocated by the user if required.

Separation between uprights, 2.75 m a compromise between the boat sizes in table

Separation between lifting arms, 1500 mm this allows sufficiently large tinnies with motors attached to be stored.

Lift vertical travel, 3000 mm this will allow the 1500 mm spacing between lifting arms.

Length of lifting arms, maximum of 2.4 m, this can be reduced in individual installations if required.

Minimal maintenance. Limited to greasing and inspection if possible.

3.3 Design concept one

3.3.1 Description

The key feature of this concept is the telescoped lifting arms where the upper lifting arm is guided inside the lower lifting arm. The lower lifting arm is guided within vertical structural sections.

The hoisting system is only attached to the upper lifting arm. When the upper arm is hoisted to approximately mid height any lost motion between the two lifting arms is taken up and the lower arm now lifts with the upper arm.

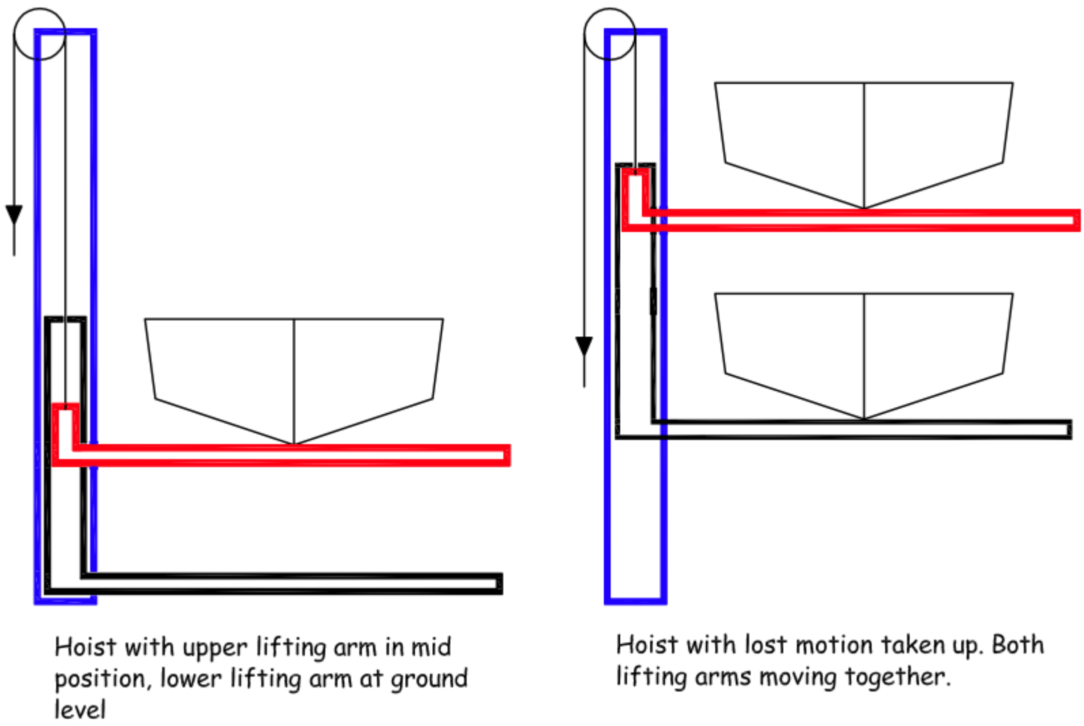


Figure 3.1 Two views of hoist concept 1.

3.3.2 Advantages

- Simple hoisting system where there is only a single connection between the hoist and the lifting arms.
- Lifting arm guides telescoped together.
- Minimal extra headroom requirement for vertical structural member.

3.3.3 Disadvantages

- Upper lifting arm is telescoped within the lower lifting arm. Two different sizes of vertical section required. Two sets of strength calculations.
- Potential for small amount of distortion to seize upper lifting arm in vertical guide assembly.

3.4 Design concept two

3.4.1 Description

Concept 2 is a variation of concept 1. As in concept one, the general arrangement is the same and the hoist system is connected to the upper arm only. The difference is the lost motion connection between the two lifting arms. In this concept the connection is a solid rod fixed to the lower arm and extending vertically. The rod passes thru a clearance hole in the upper arm, when the upper arm is hoisted to approximately mid height the stop block on the connection rod contacts the upper arm causing the two arms to lift in tandem.

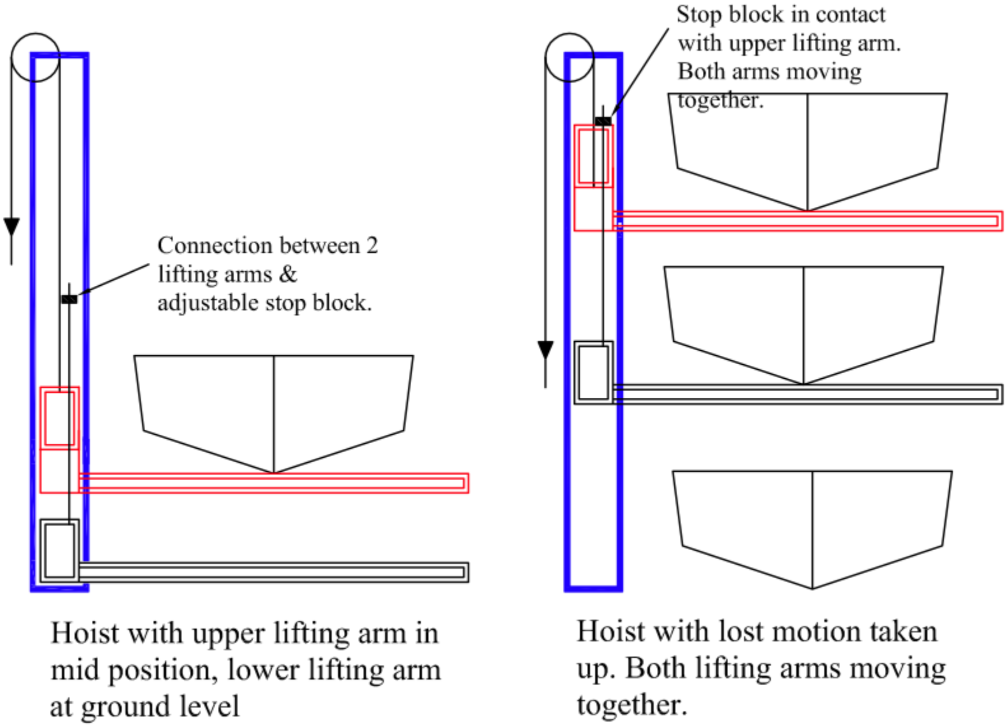


Figure 3.2 Two views of design concept 2.

3.4.2 Advantages

- Adjustable height between upper 2 levels either during fabrication or possibly after installation by hoist operators.
- Potentially more lifting arms could be added, each with lost motion solid rod connection.
- Possible savings in manufacture due to each upper and lower lifting arms having several common components.
- Minimal extra headroom requirement for vertical structural member.

3.4.3 Disadvantages

- Potential for wear and fatigue of the connection rod between the two lifting arms.

3.5 Design Concept three

3.5.1 Description

The main feature of this concept is the lifting arm bearing arrangement is external to the vertical guide post (dark blue). Each lifting arm is has two bearing assemblies linked by a plate section (aqua blue) with the (red) lifting arms attached. The two arm bearing assemblies overlap so that when the upper arm is lifted to approximately mid-height it makes contact with the lower lifting arm bearing assembly and both arms hoist together, in similar manor to concept 1.

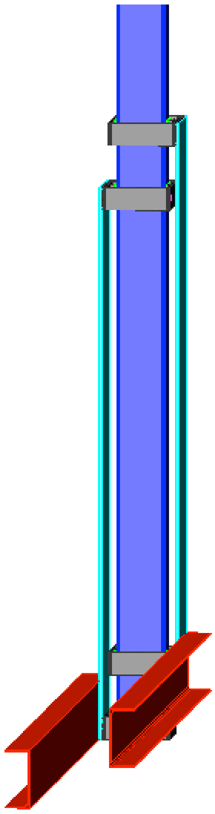


Figure 3.3 Hoist concept 3, view of one upright showing lifting arms and bearing assemblies.

3.5.2 Advantages

- Bearing assemblies are all the same reducing the overall number of components.
- Vertical structure is a single column.
- Large separation between bearing assemblies reduces forces on bearing assembly.
- External bearings and guides are easily inspected for maintenance.

3.5.3 Disadvantages

- Limited to 2 lifting arms only. More than 2 lifting arms would require either a much greater separation between each arms bearing assemblies or reduced head height between levels. Though it is possible this could be overcome by the use of additional sections to interlinked levels.
- Additional headroom requirement, vertical structural section must extend further in this design due to the upper lifting arm bearing carried design.

3.6 Conclusions

This chapter has explored and outlined three hoisting system concepts. No further work will be done as the hoist structure is outside the scope of this research project and sufficient information is now available to allow the hoist system to be designed.

Chapter 4 - Hoisting system options

4.1 General discussion

Hoist rated capacity is 2000 kg, this figure does not include lifting components, frictional losses, safety factor etc. Estimates of the lifting component weights, and the frictional losses, will be made as part of the final design. This will adequately hoist four boats with engines and typical equipment as detailed in Table 2.1, the largest boat, Alloy Craft 4.95 Open HD – Abalone, is considered to be the maximum size possible to hoist as the boat and engine have a combined weight of 448kg.

The use of wire rope is considered to be the most flexible and traditional method of connecting a hoisting system. Its advantages are flexibility in arrangement and easy adjustment. However, it does have a finite service life that varies greatly depending on environmental conditions.

The following hoisting system methods will be detailed for consideration:

- Off the shelf chain block.
- Electric flat plate winch.
- Power screw.
- Hydraulic cylinder.
- Electric drum winch.

4.2 Off the shelf chain block.

4.2.1 Description

A 3t SWL chain block such as the 'S' Series chain block described on the http://www.activelifting.com.au/liftingEquipment/chainBlock.htm requires 27kg of effort to lift the swl, this chain block is manufactured to AS1418.2

SSRS prefer not to use a chain hoist due to aesthetic reasons, however it does offer a simple off the shelf solution and therefore should be considered. The chain block could be housed improving aesthetic considerations.

The use of a chainblock will require a system for connection of the two lifting arms to ensure even and level hoisting operation.

4.2.2 Advantages

- Off the shelf complete solution.
- Designed to technical standard AS 1418.2.
- Internal gearing to achieve safe working load with minimal effort.
- Internal braking system.

4.2.3 Disadvantages

- Noisy.
- Aesthetically unappealing to SSRS, although it is anticipated this can be overcome by fitting the hoisting chain internally within the upright structure and fitting covers.
- Potentially difficult to adapt to powered operation, substitute electric chain hoist,
- Self levelling system required.

4.2.4 Estimated cost

Item	Cost
Chain block	500
Sheaves and wire rope	1000
extras	1000
Estimated Total	2,500

Table 4.1 Estimated costs of chain block hoist.

4.3 Electric flat plate winch

4.3.1 Description

This concept uses a 7000lb flat plate winch from BH USA as the basis for the hoisting system, see http://www.bh-usa.com/default.asp.

The hoisting plate consists of a dual voltage (110/240) electric motor driving a vee belt reduction which in turn drives a geared worm drive, see Figure 4.1. The plate is mounted with 2 bolts, and is designed to be connected to a length of 2” schedule 40 pipe, which is used as a cable drum. The winch plate is not designed to be load bearing, therefore the winch pipe must be adequately supported. A data table for a 7000lb winch plate is shown in Appendix D.

Figure 4.1 7000lb (3175kg) flat plate winch for boat hoists.

A typical installation for an overhead boat hoist is shown below in Figure 4.2. The winch pipe is fitted with the greased plain type support bearings either side of the lifting cables. This method is easily adapted to any of the 3 hoist concepts described above. A possible variation would be to mount the winch pipe at the base of the hoist, though it would be better to keep the winch and motor away from the floor.

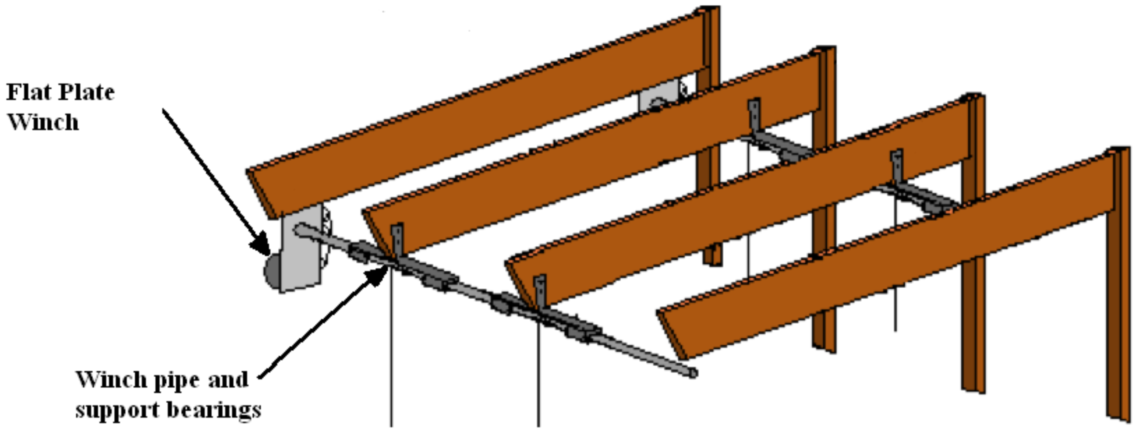


Figure 4.2 Example installation for flat plate winch and overhead hoist.

Summary of manufacturers installation recommendations from Boat Hoist USA electronic documents, (BoatHoistUSA 2008)

- Ratings based upon winch pipe 2” shed 40, max OD 2 3/8” (60.325mm).
- Unit can be mounted horizontally or vertically but must be at right angles to winch pipe.
- Winch and bearings must be regularly greased.
- Do not weld plate winch to structure, mount using any 2 of the 4 predrilled holes.
- Allow to cool after 2 lifting cycles.
- The hoist must be mounted at one end of the winch pipe.
- Support required each side of the lifting points and every 3m along the pipe.
- Cable winders are optional they increase cable life however decrease lifting capacity.
- Cables must be correctly attached to pipe, manufacturer recommends drilling an appropriate sized hole through the pipe, perpendicular to the cable passing the cable through the pipe to the end and fitting a cable clamp pulling the cable back till the clamp comes up against the pipe hole.

The BH70, 7000lb (3175kg) flat plate winch is available for $785 USD plus freight.

Alternatively the same gear plate used in the BH70 is available separately with no motor for $559 USD, this allows a locally sourced electric motor to be used saving freight costs.

4.3.2 Advantages of flat plat winch

- Off the shelf solution already widely used for hoisting boats for storage.
- Simple arrangement for mounting winch and winch drums.

4.3.3 Disadvantages of flat plate winch

- Slow, hoisting times is 6 minutes with a 50 hz power supply.

A significant problem has been identified with the use of this flat plate hoist, it relates to the diameter of the winch drum. The flat plate hoist is designed to operate with a 60.3 mm OD pipe as the wire drum, this diameter is much smaller than the minimum specified in AS 1418.1 (2002) Section 7.18.

Several variables were considered including reducing the service life of the wire to 5 years, and using a single reeved system to reduce wire tension. It is not possible to comply with the minimum drum diameters as specified by this standard using the 60.3 mm pipe.

The standard requires a drum to wire diameter ratio of 14. With a 12 mm wire this equates to 168 mm. Simply fitting the flat plate hoist with a 168 mm drum would reduce the lifting capacity by nearly 3 times, and void the manufacturers warranty.

4.3.4 Estimated cost

Although this is clearly the cheapest hoisting system no further work will be done because of the drum diameter problem discussed above.

4.4 Power screw

4.4.1 Description

US Patent 5,655,850 Floating Dock and Boat Lift uses a power screw for the hoisting system.

Power screw and ball screw thread forms and advantages and disadvantages are outlined in chapter 2. For this hoist a roller or ball screw is viewed as the best option due to the increased load carrying capacity of the roller screws.

Ball screw nuts are not designed to withstand twisting forces, therefore an additional guide system for the nut will be required.

4.4.2 Advantages of a power screw

- Excellent position accuracy.
- Speed control by specifying thread pitch and drive speed.
- High efficiency typically greater than 85per cent.
- Low maintenance.

4.4.3 Disadvantages of a power screw

- Cost.
- Braking system required because of the high efficiency, static load can rotate the screw.
- Loads need to be correctly guided as the screw nuts are not designed to support side loads.
- Lubrication is critical and contamination of bearing by debris will greatly affect service life.

4.4.4 Options for utilizing a power screw

Option 1. Fit a single power screw to one uprights and use this screw to hoist both lifting arms. Either by a self levelling wire or by 2 lifting wires, one attached to each lifting arm.

Option 2. Mount a single power screw horizontally within a structural member connecting the two uprights and use wires to connect to each of the lifting arms. This method would require increased spacing between the uprights and is therefore not suitable for this particular design.

Option 3. Each upright is fitted with a power screw. One power screw is fitted with a motor and gearbox, the other screw is driven by a roller chain from the output shaft of the gearbox.

This option will be more expensive than the first two options due to the extra power screw and chain drive. It does have some advantages such as reduced loading on the power screws, eliminating the need for wire rope and sheaves to connect the lifting arms of each upright, and reduced size of the horizontal structure between the uprights because they need only support the tension in the drive chains.

This option also allows more than 2 uprights and an increased upright spacing. And will be more attractive where these requirements exist.

Option 4. Each upright is fitted with a power screw, gearbox, motor and brake. The power screws must be synchronised to ensure level hoisting and ensure the lift remains level over a number of hoisting cycles. The likely method of synchronising is synchronous motors and a digital position control system.

This adds a layer of complexity to the hoist which is viewed as not suitable for this particular application. However, as with option 3 this becomes more attractive with a greater number of uprights and increased spacing.

SKF Australia have supplied a quote on their SL40x20R roller screw assembly. Linear Bearings Australia have quoted on a ‘Spiracon’ roller screw, a THK ball screw and a guide rail system that will be required regardless of selection.

Brand	Description	Price
SKF	SL40x20R roller screw assembly (no bearings or machining included)	A$4057.13 inc GST
THK	Ballscrew TS5016+3500LT	A$ 3,130 + GST.
	Ballscrew Nut BTK5016ZZ	A$ 1,170 + GST.
	Fixed bearing BK40	A$ 1,022 + GST.
	Supported bearing BF40	A$ 282 + GST.
	Machining for Support Bearings	A$ 550 + GST.
	Total	A$ 6,769 inc GST
Spiracon	40x24 Roller Screw # SPT-040-24-3-RH-3500-3300-1-0 (based on 0.43 A$ to GBP	$41,000
	Guide rails and bearings for power screw nut	A$2000-3000

Table 4.2 Component cost for power screw options.

The THK and SKF ball screws have a lower efficiency and load carrying capacity compared with the ‘Spiracon’ roller screw however for this application it doesn’t justify the price difference.

Linear Bearings Australia have provided a complete quote and screw selection data based upon the following data:

- Vertical mounting.
- Load mass of 2500 kg.
- Hoist velocity 0.025 ms-1.
- Stroke length of 3000 mm.
- Acceleration time 0.5 seconds.
- Fixed bearings at bottom end, support bearings at top.

Their recommended ball screw assembly has the following characteristics:

- Rotational speed required 94 rpm, permissible speed 560 rpm.

- Calculated loading 24.7 kN.
- Buckling load limit 58.6 kN.
- Compressive load limit 213 kN. Which gives a static safety factor of 12.7.
- Service life at loaded condition 7200 hours which is 7 times the estimated running hours.
- Driving torque is estimated as 160 Nm and power 1.1 kW.

The THK selection data sheet is included in Appendix D.

Linear bearings Australia have also specified a suitable guide rail system for the ball screw nut. An assembly of this type is required for power screws to prevent any twisting forces being transferred to the power screw nut. A schematic drawing has been included in Appendix D, as well as the calculation sheets provided by email on 24th November 2008.

4.4.5 Screw drive motor, gearbox, inverter.

SEW Eurodrive were contacted for their recommended drive system consisting of a gearbox, brake motor, and inverter for the hoist with the following requirements:

- Torque 160 Nm.
- Power 1.5 kW.
- Output shaft speed 94 rpm.
- Single phase to 3 phase inverting drive.
- 3 phase brake motor.
- 90 degree gearbox to keep arrangement compact.
- Duty cycle 20 mins per day.

SEW have offered four gear motor options and two drive inverter options.
Ranging from 1.5 to 2.2 kW, to gain the torque required the best option is as follows:

Helical-worm gear unit with AC Squirrel-cage motor.

- Model: S67DRE100M4BE2
- Rated power [kW]: 2.2 Output torque [Nm]: 210

- Motor speed [rpm]: 1425 Reduction ratio : 15.6
- Output speed [rpm]: 91
- Price per unit excluding GST $1491 AUD

And the drive inverter:

- MOVITRAC B Frequency Inverter, type MC07B 0022-2B1-4-00
- 200V-240 V, 2.2 kW, 8.6 A
- Dimensions (W×L×H) 80mm*274mm*165mm
- Weight 2.2kg
- Unit Price (Ex GST) $556

4.4.6 Estimated cost

Item	Cost
Power screw, bearings, nut	7000
Motor and gearbox	1491
Inverter	556
Guide system	2-3000
Wires and sheaves	1000
Extras	2000
Estimated Total	14,000

Table 4.3 Estimated costs of power screw hoisting system

4.5 Hydraulic cylinder

4.5.1 Description

The design patent reviewed in section 3.2 utilised a hydraulic cylinder mounted in the upper horizontal member. The cylinder in this design is connected to a reeved wire rope system. The reeved system increases the hoist travel for a given cylinder travel, however the arrangement also results in increased load on the hydraulic cylinder.

The same approach would also be required in this hoist. Due to the same cylinder travel restrictions.

The hydraulic cylinder can be mounted on one of the uprights or in the horizontal structural section.

For 3 m lift, a 4 turn reeved system is considered the best option to limit the length of the cylinder required. Required cylinder travel is approximately 0.75 metres, minimum cylinder power is 4 times maximum tension in the wire.

4.5.2 Advantages of hydraulic cylinder

- Excellent control of motion in particular acceleration, deceleration, speed and maximum lifting capacity by hydraulic control valves.
- Option to vary hoisting speed by specifying pump capacity.
- Easily adapted to hand pumping if required. Although this would be quite slow.
- Robust able to absorb some shock loading.
- Flexible location of hydraulic pump unit.
- Possible to use low voltage motor.
- No wire rope drum required.
- Fault finding of hydraulic can be performed by hydraulic technician.

4.5.3 Disadvantages of a hydraulic cylinder

- Increased complexity associated with hydraulics.
- Hydraulic system maintenance, i.e. filters and fluid changes, in addition to greasing of wires and bearings.
- Limited by stroke length of cylinder, one solution is to use the reeved system reviewed in section 3.2. This increases the load on the cylinder, however the loading is well within the capability of hydraulics.
- Hydraulic fluid leakage potential.
- Some form of hoist latching is required to ensure hoist does not creep. This latching requirement can be incorporated into the fail safe system.
- Piston rod needs to be protected from the environment.

4.5.4 Estimated cost

Item	Cost
Hydraulic ram and power pack	5000
Wire ropes and sheaves	2000
Fabrication	2000
Control system	2000
Extras	2000
Estimated Total	13,000

Table 4.4 Estimated costs of hydraulic cylinder lifting system.

4.6 Electric drum winch

4.6.1 Description

A wire rope drum winch, driven directly by a reduction gearbox and electric motor is a simple system. Connection to the hoisting arms via wire rope and sheaves as required. Several different arrangements are possible and shown on the next page:

4.6.2 Advantages

- Low cost.
- Minimal individual components required.
- Numerous arrangements possible.

4.6.3 Disadvantages

- Weight, 200kg estimated for gearbox and motor.

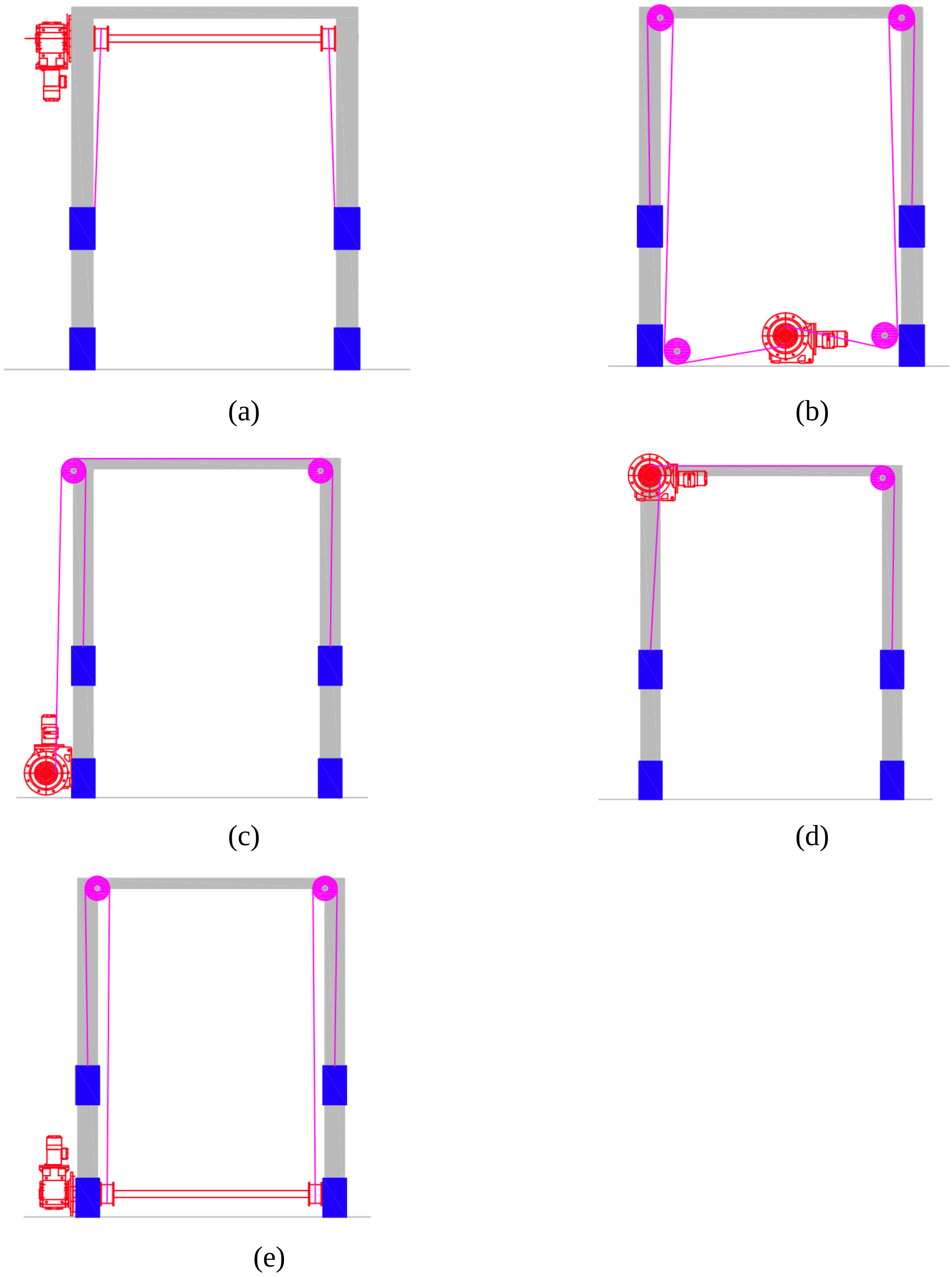


Figure 4.3 Potential arrangements for wire rope winch. (a) Winch assembly mounted at the top of one upright, two wire drums driven by a long shaft. (b) Winch assembly mounted between the two uprights with sheaves to guide the wires. (c) Winch assembly mounted at the base, connected to the lifting arms with wire rope via sheaves. (d) Winch assembly mounted at top of one upright, direct connection to one lifting arm, second arm connected via sheave. (e) Winch assembly at base. Driving two wire drums connected with a long drive shaft.

The arrangements shown in figure 4.3 show several possible winch arrangements. With the following design comments:

- Arrangement (a) requires no sheaves and is the simplest arrangement.
- Arrangement (b) the upper horizontal section is not load bearing, an additional section is required at the base between the uprights to support the structure and mount the winch assembly.
- Arrangement (c) is a good simple system, however mounting of the winch at the base of one upright will require additional consideration to avoid interference with the lifting arm guides.
- Arrangement (d) overcomes the winch mounting problems discussed in the previous point.
- Arrangement (e) uses the two rope drums used in arrangement (a) access to the winch assembly at ground level is excellent. Additional consideration is required to ensure the winch mounting and the wire drums and drive shaft don't interfere with the lifting arm guides.

SEW Eurodrive Melbourne has been contacted by phone requesting a quote for a gear motor and inverter drive suitable for this purpose.

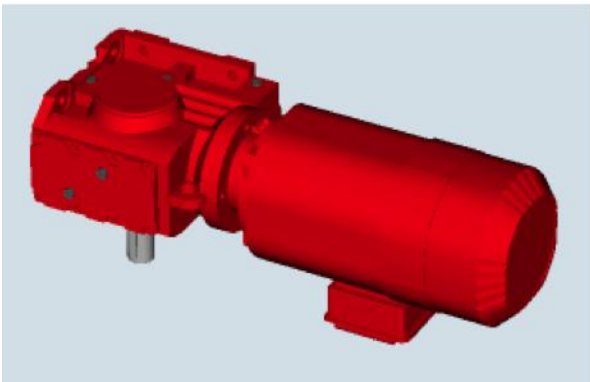


Figure 4.4 Example of SEW gearmotor.

4.6.4 Estimated cost

Item	Cost
Gearmotor	5500
Inverter	600
Shaft	500
Bearings	1200
Wire rope drums	2000
Wire rope	200
extras	2000
Estimated Total	12,000

Table 4.5 Estimated costs of wire rope winch.

4.7 Decision process

The hoist system concepts were discussed with SSRS, evaluating the advantages and disadvantages of each, as outlined below.

4.7.1 Hoist structure

Although outside the design work of this project the preferred hoist structure uses external bearing assemblies and a single section upright, probably an ‘I’ beam for the prototype.

The best system for connection of the two lifting arms is the solid rod system outlined in hoist concept 2.

The hoisting system will be mounted within a single structural section which will bolt on top of the 2 upright posts. This system will allow the hoist system to be prefabricated and assembled ready for installation. It will also eliminate any alignment problems with the shaft bearings and the gearbox during installation.

4.7.2 Hoisting system

Initially the preferred hoisting system was the flat plate winch described in section 4.3 because this was the simplest, cheapest and an off the shelf solution. However, after further research it became apparent this hoist would not comply with Australian Standards due to the small wire drum diameter specified by the winch manufacturer.

The next preferred method was to utilise a power screw mounted vertically within one of the upright sections. This power screw would be connected to the lifting arms using wire rope via sheaves.

With this design it was initially anticipated twisting forces on the ball screw could be controlled by the lifting arm guides and careful positioning of the cable attachment points. However, it became apparent these methods of controlling forces imposed onto the ball screw nut would be insufficient and therefore an additional ball screw nut guiding system would be required.

The additional hardware required to guide the ball screw nut increased the total cost of the ball screw lifting system to $11,000 not including wires, sheaves etc. This option was now considered to be too expensive for it to be effectively priced for marketing. And therefore alternative systems would have to be considered.

The concept of a direct drive wire rope winch was revisited. Gearbox and motor assemblies were investigated and it became apparent units such as SEW Eurodrives K series Gearmotor could deliver the required torque and lifting speed for approximately $5300 + GST. Due to cost and the simplicity of the arrangement this is the preferred hoisting system for prototype development.

Several hoist system arrangements are shown in Figure 4.3, after discussions with SSRS arrangement (a) is considered to be the best for prototyping for the following reasons:

- Winch assembly and wire rope drums don't interfere with upright foundations supports and lifting arm guides and bearing assemblies.
- Minimal components in lifting system, because no sheaves required to guide wire ropes.

- Horizontal upper section can be prefabricated containing the gearmotor mounting flange, winch shaft bearings etc. for fastening on top of the uprights during installation. This controls the bearing alignment issues associated with long drive shafts.

4.7.3 Latching system

A self engaging latching system is considered to be an essential part of the design. Two systems designs have been reviewed in chapter 2, both ratchet during hoisting and then engage fully when slightly lowered. To release the hoist is raised slightly then the latch disengaged. Both these latches offer no protection during lowering operations.

Other options under consideration are:

- A wedge shaped friction brake to engage between the lifting arm guide rollers and the frame. The brake would engage by spring force when the wire rope tension is removed. Because of the wedge design, downward force on the roller would act to increase the clamping force.
- Wire rope clamp. This spring loaded device clamps onto a wire rope when lifting rope tension is removed, i.e. when the locking pins are engaged and the lifting wire is slackened or if the lifting wire breaks. Additional wire ropes would be required as this clamping will damage the rope over time. One such system is shown in US patent no. 4,195,332.

Chapter 5 - Final Design

5.1 Introduction

This chapter will detail the calculations and design criteria for selecting the hoisting system components and the prototype design.

5.2 Hoist Loads

Hoist loads are discussed in section 2.4.2 and are tabled and calculated below.

Load Group	Loads	Value
Principle loads	R1 loads due to mechanism (assumed initial value)	500kg
	R2 Loads due to deadload of parts acting on mechanism	Included in R1
	R3 Loads due to mass acting on crane hook	2000 kg
	R4 Loads due to dynamic affect of load on hook	100 N
	R5 Loads due to dynamic affect of crane mechanisms including inertia.	25 N
	R6 Loads from frictional forces	100 N
	V1 Loads due to in service wind acting horizontally in any direction	-
	V2 Loads in out of service condition due to wind	-
Additional loads	Wind, snow, ice, temperature extremes, oblique travel	-
Special loads	B1 loads due to collision with buffers	-
	B2 loads due to emergency conditions	-

Table 5.1 Categorisation of loads on mechanisms.

R4 and R5 are calculated from the specification of the hoist and the hoist drive.
Hoisting velocity (V_f) is determined by distance over time. Hoisting time of 3 metres in 2 minutes has been selected.

Acceleration time (t_{acc}) has been selected as 0.5 seconds (acceleration time is controlled by the electric motor drive inverter). For the purpose of calculation it is assumed the acceleration is constant. Calculations as follows:

Using $F = ma$, (5-1)

Where $a = V_f / t_{acc}$ ms-2 (5-2)

$V_f = s / t$ (5-3)

$= 3/120 = 0.025$ ms-1

$m = 2000$ kg

Substituting equation (5-2) and (5-3) into (5-1)

Gives F (R4) = 2000 x (0.025 / 0.5) = 100 N

The same method and formula are used to calculate R5.

F (R5) = 500x(0.025/0.5) = 25 N

5.3 Classification of mechanisms

As discussed in section 2.4.1.1 Mechanical classification.

Class of utilization is the total hours of use for the entire life of the hoist component. From AS 1418.1 (2002) table 7.3.4.2.

It is anticipated this hoist will complete an average of 2 lifting cycles per day, over a 10 year life. Assuming 2 cycles per day at 4 minute per cycle (2 minutes up, 2 minutes down) this translates to approximately 973hours over 20 years. This equates to a Class of Utilization of T3.

A loading factor (K_m) was selected as 0.50 because the hoist will be hoisting only one level (half rated load) for 50% of a complete lifting cycle. From AS 1418.1 (2002) table 7.3.4.3 this gives a state of loading of L3 – Medium, which is described in the table as ‘frequently subject to maximum loads and normally to loads of moderate magnitude’.

Using the values of T3 and KM in AS 1418.1 (2002) table 7.3.4.4 the mechanisms are classed as M4.

The tables referred to in this section have been included in appendix F.

5.4 Component Selection

This section will perform calculations required to specify the hoisting system components and detail the selections made.

5.4.1 Wire Rope selection

Wire rope maximum tension is a sum of the force required to accelerate the load plus the gravity force on the load plus the friction force. Calculated as follows:

Tension in hoist cable (kN) = (m*a + m*g + friction)/1000 (5-4)

Where a = hoist acceleration = 0.05 ms-2

g = acceleration due to gravity = 9.81 ms-2

m = Hoist load + mechanism weight = R1 + R3 = 2500 kg

Thus T_{cable} = (2500*0.05 + 2500*9.81 + 100)

= 24.57 kN

Note friction component is an estimate as the rotational speeds are low, and there is little in way of sliding friction within the hoist design.

The above calculation is supported by the ball screw data sheet provide by Linear Bearings Pty Ltd.

As discussed in section 2.4.5 Wire Ropes

- Classification of the wire rope mechanism from above = M4.
- Minimum coefficient of utilization Z_P from AS 1418.1 (2002) table 7.16.2.1 = 4.0
- Maximum wire tension from above = 24.75 kN

Using the following formula for ‘minimum wire rope breaking load’ (F_O):

$$F_O = S_R \times Z_P \tag{5-5}$$

Where S_R = maximum wire rope tension in newtons.

Gives: $F_O = 24.75 \times 4.0 = 99$ kN

A. Nobles & Son, Sydney Office have recommended a 12 mm, 19x7 ‘Rotation Resistant’ Rope, Grade 1960, this wire rope has a minimum breaking force of 105.9 kN and a weight of 0.608 kg/m. Appendix C shows the data table for this rope. Their quote was emailed (A. Nobles & Son 2008) as follows:

- 12mm 19x7 non rotating wire rope B1960 minimum breaking load 105.9 KN $ 6.50 Per Metre + gst.
- option of swaged thimble eye one end and jaw / stud end rigging screw on other end. Approx cost $ 195.00 + gst.

5.4.2 Winch drum

Winch drum design was discussed in section 2.4.7. The winch drum will be fabricated from readily available steel plate, grade AS 3678-250. Drum dimensions are calculated as follows:

Minimum wire drum pitch diameter is given by the formula (2-5)

$(D_d) \geq h_d \times d_{min}$

Where h_d for M4 mechanism = 16

d_{min} from section 5.4.1 = 12 mm

Gives $D_d = 16*12 = 192$ mm

Drum thickness calculations as discussed in section 2.4.7.2 using equation (2-6), (2-7) and (2-8):

$$T_D = (T_{DB}^2 + T_{DB*} T_{DC} + T_{DC}^2)^{1/2}$$

T_{DB} = Minimum thickness considering bending stress in mm

$$T_{DB} = \frac{1250 \times M}{{D_{DM}}^2 \times F_B}$$

Where M = maximum bending moment in Nm,

D_{DM} = mean diameter of drum shell in mm,

F_B = permissible bending stress in MPa

= 0.67 times yield stress.

Gives $T_{DB} = \frac{1250 \times 4187}{184^2 \times 0.67 \times 250}$ = 0.923

And T_{DC} = minimum thickness allowing for compressive stress in mm.

$$T_{DC} = \frac{1000 \times K_{RL} \times P_{RS}}{p \times F_C} + 0.15 \times d$$

Where K_{RL} = rope lay factor = 1.0 for single layer,

P_{RS} = maximum unfactored static rope load in kN,

p = pitch of rope coil in mm,

F_C = permissible compressive stress from AS 1418.1 table 7.19.5

= 125 MPa,

d = nominal rope diameter in mm.

Gives $T_{DC} = \frac{1000 \times 1.0 \times 24.57}{192 \times 125} + 0.15 \times 12$ = 2.82 mm

Therefore minimum drum thickness as follows:

$T_D = (T_{DB}^2 + T_{DB*}\, T_{DC} + T_{DC}^2)^{1/2}$

$= (0.923^2 + 0.923*2.82 + 2.82^2)^{1/2}$ = 3.38 mm

This thickness will be rounded up to 6mm plate for this winch.

5.4.3 Winch gearbox and motor

Calculations required to select a suitable winch gearbox and motor are performed in this section. The selected components are detailed.

Hoisting speed is calculated from the following:

$$\text{Winch rpm} = (S/D_{circumference})/t \qquad (5\text{-}6)$$

Where S = hoisting distance = 3000mm

$D_{circumference}$ = drum pitch circumference

$$= \pi D_d = \pi * 192 = 603 \text{ mm} \qquad (5\text{-}7)$$

t = hoisting time = 2 minutes

Gives winch rpm = (3000/603)/2 = 2.48 rpm

Drum width is discussed in section 2.4.7.2 and given by equation (2-9)

Width = (hoisting distance/drum pitch circumference + 3)* wire diameter

= 95.7mm rounded up gives 100mm

Required Torque calculation:

$$T = T_{cable} * \text{drum radius} \qquad (5\text{-}8)$$

Where T_{cable} = 24.75 kN from section 5.4.1

Gives Torque required = 24.75 * 0.192/2 = 2370 Nm

With the above information SEW Eurodrive Melbourne were contacted for quoting on a gearmotor with the following requirements:

- Suitable for hoisting winch application.
- Output shaft speed 2.5 rpm.
- Output torqure required 2370 Nm.
- 90 deg gearbox.
- Single phase to three phase inverter.
- Brake motor.
- Hollow output shaft.

SEW Eurodrive have quoted the following Helical-bevel gear unit with AC squirrel-cage motor:

- Model: KAF97R57DRE80M4BE2
- Rated power 0.75 kW
- Motor speed 1435 rpm
- Output speed 2.5 rpm
- Output torque 2630 Nm
- Brake torque 20 Nm
- Output flange 450 mm
- Price $4704.92 excluding GST

Figure 5.1 SEW Eurodrive K series gear motor.

SEW Eurodrive Melbourne were contacted by phone (SEW Eurodrive Melbourne 2008, pers. comm. 25th Nov 2008) to discuss mounting options for their flanged output gearbox. In particular if the flange can be used to mount the wire drum bearings via a flanged spool piece.

Their advice as follows:

- Primary function of the flange is for gearbox mounting.
- Overhanging loads on the gearbox shaft must be controlled.

5.4.4 Winch shaft calculations

Winch shaft shear and bending moments are calculated below, based on the following arrangement.

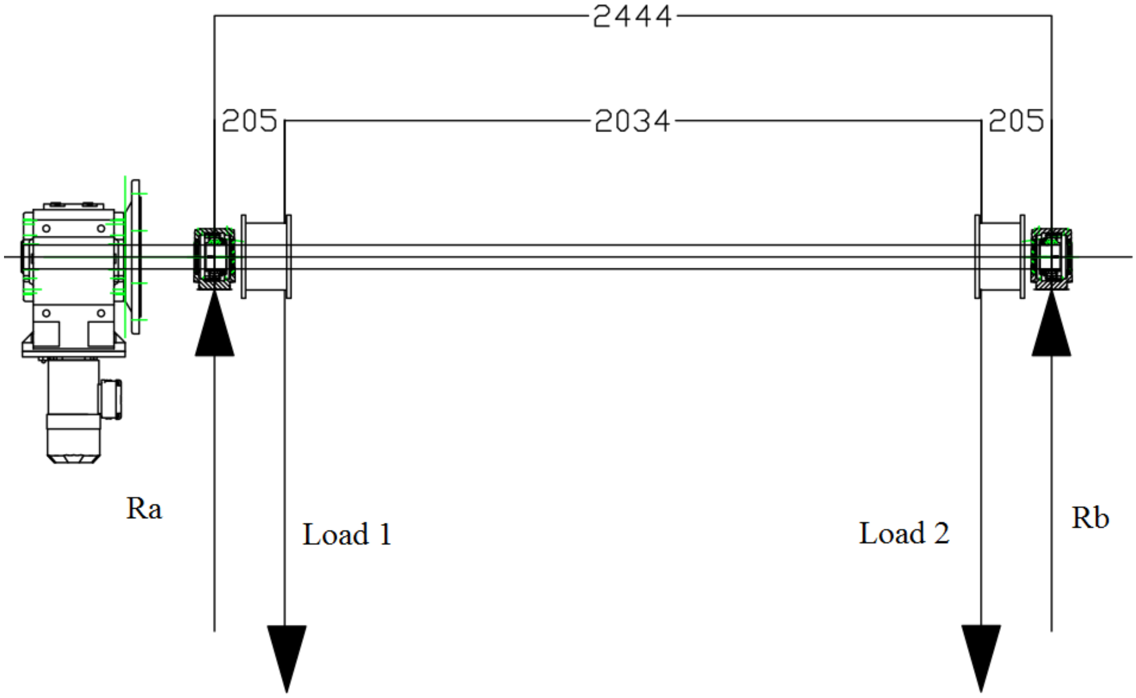


Figure 5.2 Winch shaft loading and bearing reaction forces.

The greatest shear force and bending moment in the vertical axis occurs then all the load is carried by one of the wire rope sheaves ie Load 1 = 2250 kg, load 2 = 250 kg.

The gearbox only provides torque it will be mounted with a flange and bear no weight onto the shaft. Horizontal components due to bearing friction will not be considered as they are small in relation to the vertical loadings.

Calculating the moments about bearing Ra

$$\sum M_z : \text{Load1}*g*0.205 - \text{Load2}*g*2.239 + Rb*2.444 = 0 \qquad (5\text{-}9)$$

And sum of the forces in the vertical direction:

$$\sum F_y : \quad Ra + Rb - \text{Load 1}*g - \text{Load 2}*g = 0 \qquad (5\text{-}10)$$

Where g = acceleration due to gravity = 9.81 ms-2

Load 1 = rated load + lifting arm mass = 2000 + 250 kg

Load 2 = lifting arm mass = 250 kg

From equation (5-9)

-2250*9.81*0.205 – 250*9.81*2.239 + Rb*2.444 = 0

Gives Rb = 4098 Nm

Substituting into equation (5-10)

Ra + 4098 – 2250*9.81 – 250*9.81 = 0

Gives Ra = 20,427 Nm

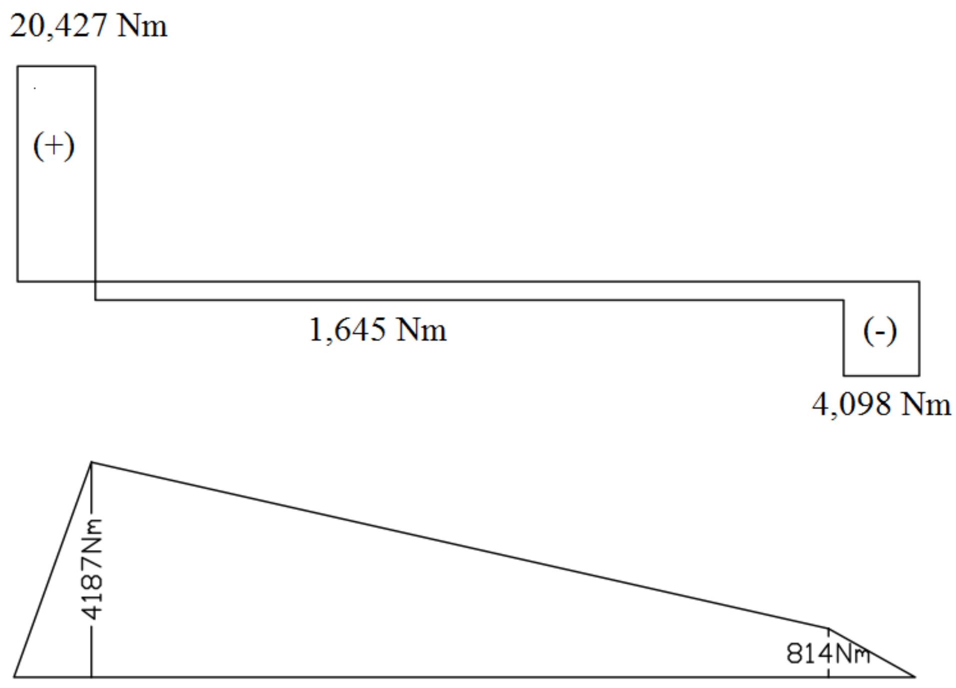


Figure 5.3 Shear force and bending moment diagrams for the vertical axis.

Minimum diameter of shafts used in cranes hoists and winches as discussed in section 2.4.8. using equation (2-10) and variables as stated below are obtained from previous calculations, (Standards Australia 2004) and material data from (Atlas specialty metals 2005).

$$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{4}{T_q}^2}$$

Where D = minimum shaft diameter (mm)

F_S = safety factor = 1.2 (as per AS 1403, Table 2, note 5)

F_R = endurance limit of shaft material, reversed bending during rotation or 0.45*tensile strength.

K_S = Size factor as per AS 1403, clause 8.1 = 1.6

K = Stress raising factor as per AS 1403, clause 8.2

M_q = bending moment at shaft cross section in (Nm)

P_q = maximum axial tensile force at shaft cross section (N)

T_q = Maximum torque at shaft cross section (Nm)

To limit the diameter and shaft machining required, stress raising factors will be limited as far as possible. Analysis of the stress raising factors in AS 1403 (2004) indicate keyways are higher stressed than other types of locking devices. Therefore it is proposed the SEW gearbox is connection to shaft via their TorqLOC system requiring no keyways, shaft bearings mounted with adapter sleeves, and winch drums mounted preferable with taper lock style couplings, without additional manufactures safety factor information at this time is assumed K = 3.5

Selected steel grades:

- 4140 has UTS 930 MPa
- 6580 UTS 1200 MPa

T_q calculated in section 5.5.1 = 2370 Nm,

P_q = 0 as shaft is not subject to axial tensile force,

M_q = 4187 Nm as calculated above,

Using steel grade 4140 gives minimum shaft diameter = 66.2 mm.
Using steel grade 6580 gives minimum shaft diameter = 60.8 mm.

The proposed shaft diameter is 70mm this calculation will need to be checked when the exact type and stress raising factor of the compression couplings has been determined.

Shaft torsion was checked using the following formula from (A.D. Deutschmann 1975).

$$\text{Shaft torsion } \theta = (584*T*L) / (\phi^4*G) \tag{5-11}$$

Where T = shaft torque = 2,370,000 Nmm2
L = shaft length in mm = 2444 mm
ϕ = shaft diameter = 70 mm
G = torsional modulus of elasticity = 80 N/mm2

Gives maximum angular deflection $\theta = 1.76°$

This maximum angular deflection is considered satisfactory since based on a 196 mm wire drum pitch diameter translates to a difference in lifting arm positions of 2.9 mm. The maximum deflection occurs when full load is on a single wire furtherest from the gearbox.

5.4.5 Shaft bearings

Capital Bearings Fyshwick (local SKF distributer) were contacted for advice on suitable bearings for supporting the winch shaft. In my phone call (pers. comm. 15/11/2008) initial estimates of a 70 mm shaft OD were provided (output hollow shaft diameter of gearbox). Loads as discussed above.

Their advice as follows:

- 22216 EK spherical roller bearing mounted on adapter sleeve.
- 516-616 split bearing housing with standard lip seals.
- FBR 12.5/140 Locating ring for fixed end.
- No locating rings required for free end.

- Estimated cost for 2 bearings with adapter sleeves, 2 housings, seals and locating rings $1184 inc. GST.
- Bearing load rating.

Spherical roller bearings are self aligning and capable of taking heavy loads, the above bearing arrangement is used in heavy industry. Split plummer block housings are best mounted with the load acting vertically towards the base plate of the bearing. Other positions are possible and outlined in SKF technical information on SNL bearing housings (SKF 2004). The mounting width required for this type of bearing mounting led to a search for a more compact arrangement.

Further research of bearing housings has located a flanged bearing housing capable of supporting a minimum of 140 kN. The housing part number is FNL 516 B manufacturers recommended mounting is by 4 of 8.8 grade M16 bolts tightened to 200 Nm.

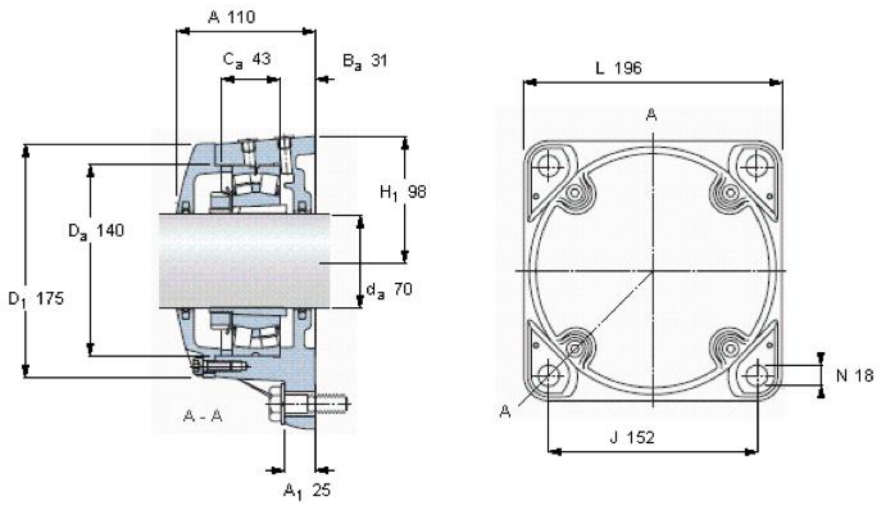


Figure 5.4 SKF FNL 516B flanged bearing housing.

The bearing and housing data tables have been included in Appendix I.

5.5 Design and Drawings

The hoisting system structural member is based around the Onesteel 250 PFC (parallel flange channel), (Onesteel 2007) this channel will support and protect the entire hoisting system and allow the hoisting system to be preassembled for bolting on top of the upright sections during installation.

The fabricated box sections housing the bearings and the gearbox flange are made from 12mm plain steel plate. This thickness corresponds with the flange thickness of the parallel channel. The base of these box sections is a flat square plate with drilled holes for fastening to corresponding plates on top of the upright sections. The overall dimensions of this these sections have been minimised as they affect the total height of the hoist.

The gearbox mounting flange is spigoted to locate the gearbox concentric with the bearings.

The bearing closest to the gearbox will be fitted with locating rings to position the shaft axially. The other bearing will have no locating rings and has some axial clearance to allow for any expansion. Each bearing housing is mounted with 4 of grade 8.8 M16 bolts, clearance holes will be drilled in the structure to allow some radial movement of the bearing housings during the shaft alignment process before dowelling the housings.

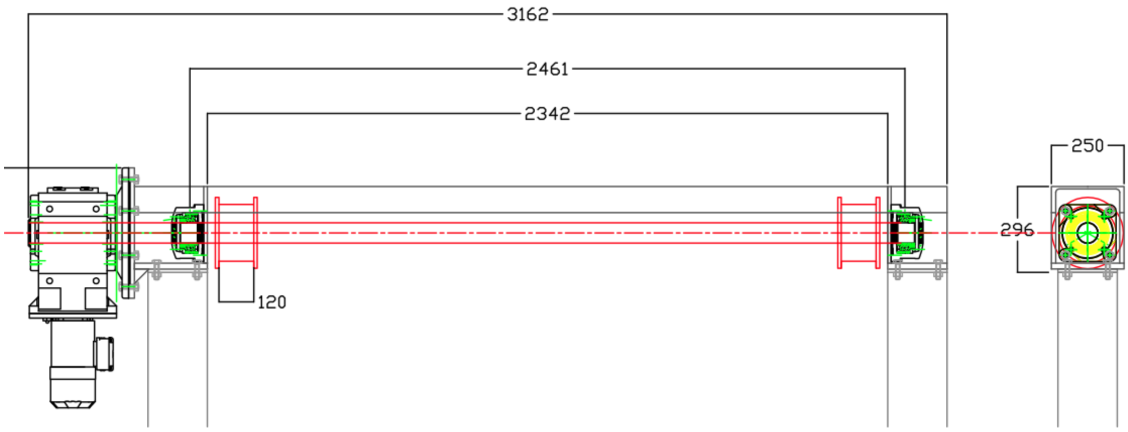


Figure 5.5 Plan and elevation view of hoist structure with components mounted.

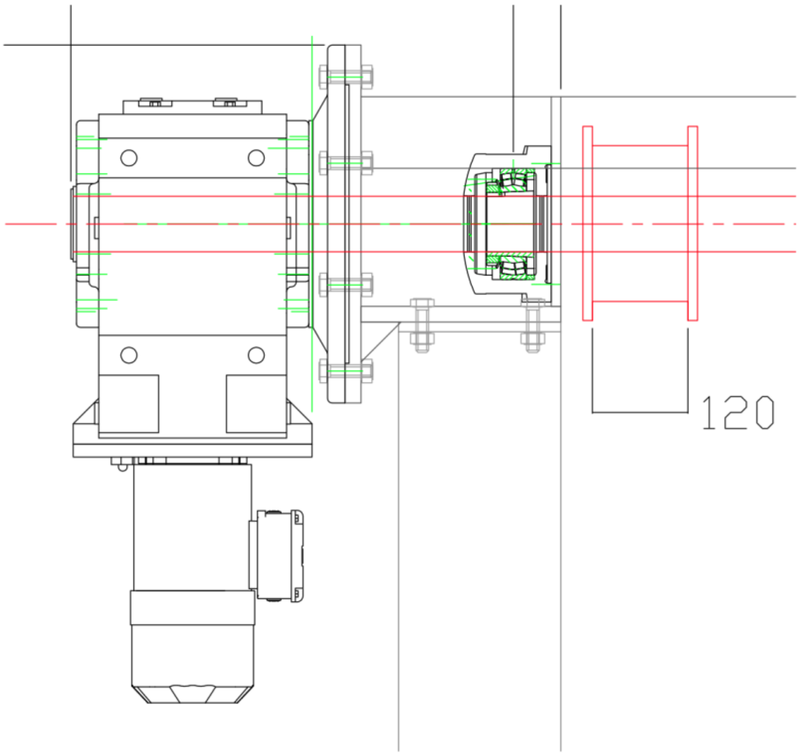


Figure 5.6 Plan view of drive end.

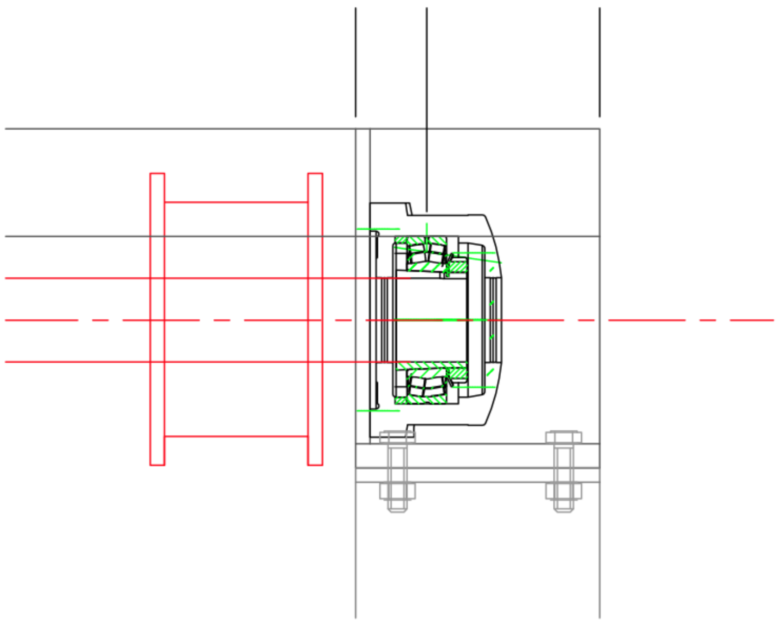


Figure 5.7 Plan view of non-drive end.

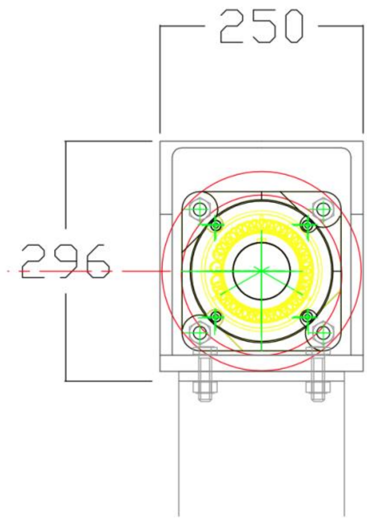


Figure 5.8 Elevation view of non-drive end.

5.6 Prototype costing

Known costing have been tabled below, the unknowns are the steel and fabrication costs. Estimates have been made, though they are based somewhat on my personal experience working in remote areas i.e. Dampier & Karratha where cost of trades is significantly higher than other regions of the country.

Item	Cost
Gearmotor	4704
Bearings	1184
Wire and fittings	195
Inverter drive	556
Shaft (est.)	500
Wire drum fabrication (est.)	2000
Support structure fabrication (est.)	2000
General fitting and machining (est.)	2000
Total	$ 13,139

Table 5.2 Cost estimates for winch components.

5.7 Conclusions

This chapter has designed and specified the components required for the hoisting system. Chapter 6 will summarise the achievements of this project, further work required and future development of this hoisting concept.

Chapter 6 - Conclusions and Further Work

6.1 Introduction

This is the final chapter in this report, and will summarise the achievements of this project against the ‘Project Specifications’ document included in Appendix A. Further work required will be discussed as will future developments associated with this project.

6.2 Summary of objectives achieved

A large part of the project work was research, gathering information on legal, and technical aspects of hoist design.

- Manufacturers and suppliers information from websites, product brochures and online patented design documents has been reviewed and summarised.
- NSW and Queensland Government Occupational Health and Safety legislation and regulations were researched for the legal requirements for hoist design.
- Australian standards were identified and searched for information required for this design.
- General mechanical design information has been gathered from other sources such as books, websites, and personal communications with manufacturers and suppliers.
- The Project sponsor (SSRS) has been involved in the critical review and decision making process along the way to guide the outcome of this work.
- Calculations have been researched and performed to define the design parameters of key components of the hoisting system.
- A design has been settled upon and the lifting system and housing designed and drafted in CAD.

Overall the objectives of the project have been achieved, however it is felt additional drawings will be required for the fabrication of the prototype. Time was not available for this as the chosen lifting system was initially rejected for various reasons and was only revisited very late in the design process.

6.3 Further Work

The minimum winch shaft diameter needs to be checked as this work is recommending the use of ‘taperlock’ style couplings for mounting the wire drums and gearbox. These mountings affect the stress raising factors used in shaft diameter calculations. AS 1403 (2004) recommends manufacturers should be consulted to confirm the stress raising factors of their couplings and the calculations performed with the new information.

The fall arresting and latching system requires prototype design, two patented designs of safety latches used in automotive hoists have been reviewed in chapter 2, and two other ideas have been briefly discussed in section 4.7.3.

Prototype design of the lifting arm bearing and guide assembly has not been included in this work. This assembly is complex due to the requirements for multiple lifting arms that lower flat to ground and the need to work around the upright, support and foundation structure. Current thinking is the arms will be offset, and therefore detailed design and stress analysis will be required. The design of this assembly has been discussed with SSRS and is included briefly in the hoist structure design concepts.

Design and load analysis of the hoist frame and foundation structure, it is planned the uprights will be fabricated from either a ‘universal column’ or ‘Universal beam’ these are both industry terms for ‘I’ beams, these sections are off the shelf and allow the lifting arm bearings to be inside the section thus reducing overall dimensions and keeping the assembly ‘tidy’. Some variation of the lifting system housing the maybe required depending on the dimensions of the chosen upright section.

Electric motor starter and the electrical control system has not been designed in this project. There are a numerous possible control elements that could be incorporated such as dead man switches, limit switches, direction reversal delay timers, audible and visual warnings of hoist starting and operating, emergency stops etc. Design of the control system is best performed by a specialist in this area.

6.4 Future Development in Design

SSRS has a viable customer for this hoist, therefore a prototype will be fabricated in the near future for testing.

The use of the long winch shaft and twin wire drums may need to be reviewed as a single wire drum on a short shaft will likely be a more economical system. Some design work of the arrangement shown in figure 4.3(d) has already been undertaken.

SSRS also is working on a variation of this hoist to be used for racking rowing shells. The new hoist will require more levels of lifting arms (6 or more), 2 or 3 uprights and increased upright spacing of 6 metres. The long winch drive shaft will be unsuitable for this hoist, therefore a variation will be required.

Figure 6.1 Future storage requirements. These rowing 8s are 18 m long.

6.5 Conclusion

The prototype design in this project will satisfy the design criteria put together during the research phase. When the further work as outlined above is complete, the design will be ready for prototype fabrication and testing.

Bibliography

A.D. Deutschmann, W. J. M., C.E. Wilson (1975). Machine Design Theory and Practice, Macmillan Publishing Co., Inc,.

A.Nobles&Son (2006). Nobles Blocks and Swivels.pdf, A. Noble & Son Ltd.

Atlas specialty metals (2005). Technical Handbook of Bar Products.

Boat Hoist USA. (2008). "BH70 7000 lb Boat Hoist." from http://www.bh-usa.com/product.asp?pID=18894&cID=96.

BoatHoistUSA (2008). BHUSA Overhead Lift Guide.pdf.

BoatHoistUSA (2008). BHUSA Revised Equipment Guide.pdf.

Byron L. Godbersen (1983). *Boat hoist*. United States Patent no 4401335.

Erik Oberg, F. D. J., Holbrook L. Hornton, Henry H. Ryffel (2008). Machinery's Handbook 28th Edition. New York, Industrial Press.

Holmgren, R. F. (1997). *Floating Dock and Boat Lift*. United States Patent no 5655850.

J.L. Meriam, L. G. K. (1987). Engineering Mechanics, Volume 1, Statics, John Wiley & Sons.

James A. Endres, F. J. D., Gideon Hecht, (1997). *Programmable Boat Lift Control*. United States Patent no 5593247.

Jon S. Halstead, A. J. H. (1998). Hoist locking and release apparatus. United States Patent no. 5803206.

Layton J. Reprogle, T. B. R. (1987). *Boat Dock and Lift*. United States Patent no 4641596.

Onesteel (2007). Hot rolled and structural steel products.

Reimann & Georger Corp (2006). Marine Products Catalog 2007,

Roy Beardmore. (2008). "Power Screws." Retrieved 25/10/2008, 2008, from http://www.roymech.co.uk/Useful_Tables/Cams_Springs/Power_Screws.html.

SKF (2004). SNL plummer block housings solve the problem,

Standards Australia (2004). AS 1403 - 2004 Design of rotating steel shafts,.

Wayne G. Floe, D. C. M. (2006). *Powered Boatlift with Electronic Controls*. United States Patent no 7059803 B2.

Wolfgang Raffler, W. S. (2006). Electromagnetic release. U. S. P. a. U. A1.

Additional reading

A. Nobles & Son (2008). Wire Rope & Strand, A. Nobles & Son,.

Burton DeWolfe Bishop (1976). Safety latch for automotive hoist or the like. United States Patent no 3934680.

Charles I. Hubert (1991). Electric Machines
Theory, Operation, Applications, Adjusment, and Control Macmillan publishing company

Dr Frank Young (2008). Eng 4111/4112 Research Project Project Reference Book, University of Southern Queensland.

E.P. Popov (1978). Mechanics of Materials SI Version, Prentice-Hall Inc.

Engineers Australia (2000). Code of Ethics 2000, Institute of Engineers Australia.

G.H. Ryder, M. D. B. (1990). Mechanics of Machines Second edition, Macmillan Education Ltd.

J. Ang, K. E. (1997). Mechanical Design 4 Subject Book. Churchill VIC, Distance Education Centre Monash University.

J.L. Meriam, L. G. K. (1987). Engineering Mechanics, Volume 2, Dynamics, John Wiley & Son.

James A. Endres, F. J. D., Gideon Hecht, (1997). *Programmable Boat Lift Control*. United States Patent no 5593247.

Power Jacks Ltd (2008) Spiracon Roller Screw Product Catalogue. Volume, DOI:

Richard L. Thompson (1980). Luminaire hoist safety device with automatic brake means adjacent to top cable guide. U. S. P. n. 4195332.

Sea Jay Boats.

Standards Australia (1993). AS 2550.1 - 1993 Cranes - Safe use - General. AS 2550.1. A. Standards.

Standards Australia (2004). AS 2550.1-2002/Amdt 1-2004. A. Standards.

Standards Australia (2004). AS 2759 – 2004 Steel wire rope—Use, operation and maintenance. . S. Australia.

State of Queensland (1997). Workplace Heath and Safety Regulation 1997, State of Queensland.

Stephen P. Bulmann (2004). *Boat Hoist Hydraulic Lifting Device*. United States Patent no 6695533 B1.

Steven Michael Sullivan (2007). Research, design and develop a pipe latching/unlatching device that can be used to help automate the pipe tripping process. Toowoomba, Australia**.**.

WorkCover NSW Cranes, Hoists and Winches.
Guidance for the provision of cranes, hoists and winches under OH&S legislation in NSW.

Appendix A – Project Specification

ENG 4111/4112 Research Project

PROJECT SPECIFICATION

FOR: Foster Robert LEAR

TOPIC: Hoist system for small boat stacker

SUPERVISOR: Professor David Ross

SPONSERSHIP: Doug Lumsden, Space Saver Oar Brackets.

CONFIDENTIALITY: The concept is subject to a standard confidentiality agreement and thus no material can be published in the public domain unless specific permission is obtained in writing from the owner of the concept.

PROJECT AIM: To develop a hoisting system design concept suitable for small boat racking and storage.

PROGRAM: (Issue B, 14th November 2008)

1. Research information relating to small boats and trailers such as dimensions and weights
2. Research of relevant Australian Standards, and OH&S Legislation.
3. Research of existing products and designs.
4. Concept development of structure, lifting arms & guides, hoisting mechanisms and fail safe devices.
5. Assessment of concepts against design criteria.
6. Develop force and moment diagrams and perform design calculations.
7. Design of hoisting system to prototype stage only.
8. Drafting in CAD format to a standard suitable for prototype manufacture.

As time permits.

1. Design of critical components, lifting arm and guide assembly, connection between uprights and foundation frame.
2. Costing of prototype manufacture.
3. Manufacture of prototype.

AGREED: ______________________ (Student) ____________________(Supervisor)

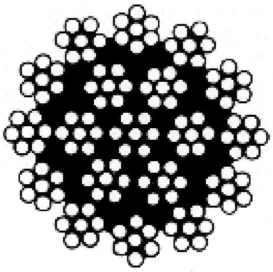
Date ____________________ ______________________

Examiner/Co-examiner: ____

Appendix B - Boat hoist USA 7000 lb winch plate data table

Manufactured	Longview, Texas
Capacity	7000lb Straight Line Pull
Warranty	Lifetime Warranty on Gear Plate Assembly, 1 Year Limited Warranty on the Motor & Wire Harness (See BHUSA Equipment Guide for Details)
Back Plate Options	Powder Coated (Standard), Hot-Dipped Galvanized
Back Plate Dimensions	12" x 24" and 1/4" Thick
Back Plate Mounting Hole Dimensions	9/16" Hole for1/2" Bolt GR5
	10" Center to Center
Electric Motor	1.5 hp Reversible Dual Voltage Motor
Electric Motor Options	Leeson (Standard) or East Bay Stainless Steel
Motor Mounting	56 Frame
Amps Required	19.4/115, 9.7/230 Leeson (Standard) 15.2/115, 7.6/230 East Bay Stainless Steel
Belts Used	AX-34 Cogged Drive Belt on Dual Cast Aluminum Pulleys
Switch Options	Maintain Switch (Standard) or Spring Loaded Switch
GFCI Options	110volt In-Plug GFCI (Standard) 110volt In-Line GFCI 220volt In-Line GFCI

Appendix C - A. Nobles & Son 19x7 Rotation resistant wire rope table



2160 Grade tensile is non-preferred. 1960 Grade Wire Ropes are recommended as they still provide very high breaking loads but the slightly lower tensile ensures an excellent service life.

19 x 7 Construction Wire Ropes are not recommended in diameters above 18mm. 35 x 7 ropes should be used above 12mm diameter.

Rotation Resistant 19 x 7

Nominal Diameter	Approximate Mass	Minimum Breaking Force	
		Grade 1960	Grade 2160
mm	kg/100m	kN	kN
7	20.6	36.1	39.7
8	26.9	47.1	51.8
9	34.1	59.6	65.6
10	42.1	73.5	80.9
12	60.8	105.9	116.5
14	81.9	143.8	158.2
16	107	188.5	207.4
18	137	238.6	262.5
20	170	294.2	323.6
22	195	355.2	390.7
24	237	422.8	465.1
26	276	495.8	545.4

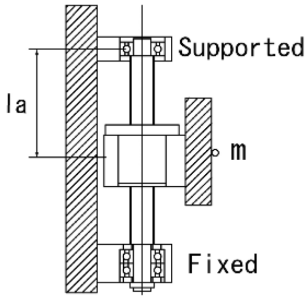
Appendix D - THK ball screw data and sheets

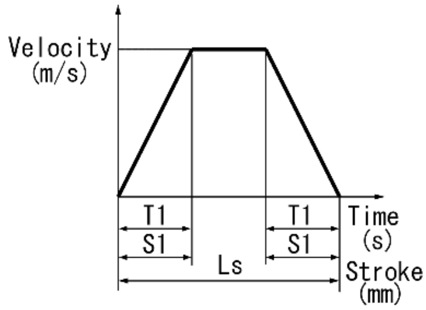
D-1 THK ball screw calculation sheet

THK Calculate the service life of Ball Screw

Title: Lear - BTK5016 Date: 2008-11-18

Operating Conditions





Installation direction		vertical	Mass	m=	2500kg
Mounting method		Fixed/supported	Velocity	v=	0.025m/s
Ball screw classification		Rolled shaft/Non-preloaded	Drag force	f=	100N
Model number		BTK5016-5.3	Distance between mounting positions	la=	3400mm
Basic dynamic load rating	Ca=	93800N	Stroke length	Ls=	3000mm
Basic static load rating	Coa=	315200N	Acceleration time	T1=	0.5s
Thread minor diameter	d1=	42.9mm	Number of reciprocations per minute	n=	0.2min^{-1}
Ball center-to-center diameter	dp=	52.7mm	Friction coefficient	μ=	0.005
			Load factor	fw=	1.2

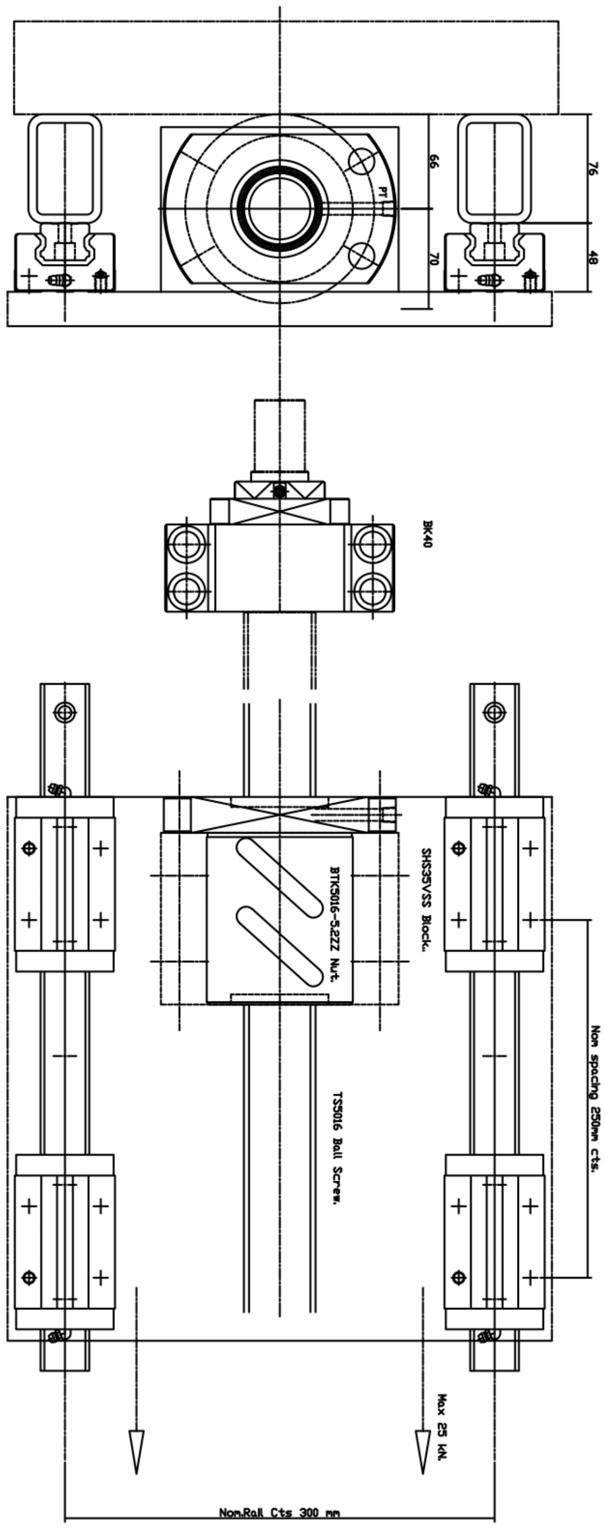
Calculation result

Ball screw rotational speed	Ns=	94min^{-1}	**Static safety factor**	**fs=**	**12.7**
DN Value	DN=	4941	**Mean load**	**fm=**	**24517.9N**
Permissible rotational speed	NC=	560min^{-1}	**Nominal life**	**L=**	**3.24E+07rev.**
Maximum load	F=	24742.5N	**Running life**	**L=**	**5.18E+02km**
Bucking load	P1=	58601N	**Service life time**	**Lh=**	**7.20E+03h**
Permissible tensile-compressive load	P2=	213488N			

Note:

* Ex: 2.35E+06 is displayed in exponential notation as 2.35×10^6.
* Calculation results for this software are based on the theoretical calculation in accordance with your input conditions. Under actual operation, they may vary depending on the operating conditions like operating environment, lubricating condition, mounting accuracy, rigidity & etc.
* Nominal life and static safety factor may be calculated using differing raceway equivalent loads.
* When the static safety factor (fs) is less than 1.0, since it is over the basic static load rating (C0a), judge it to be unusable and enter "---" for the calculation result of the service life. Please change the model and/or operating conditions so that you can ensure a static safety factor of over 1.0, and then calculate again.
* When the number of revolutions to use is over the permissible numberof revolutions, or when the DN value is over the permissible value, judgeit as unusable, and enter "-----" as the result of the service lifecalculation. Change the conditions and then calculate again.

D-2 Ball screw guide rail and bearing assembly

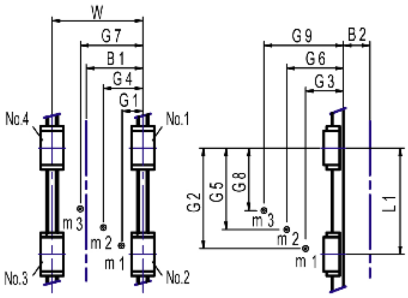


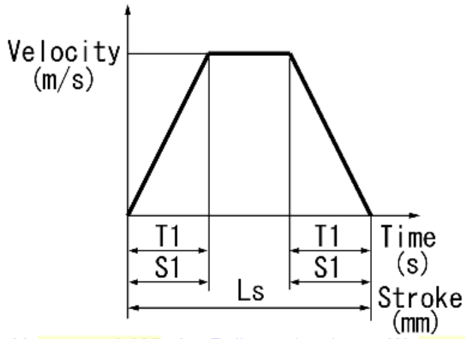
D-3 Ball screw guide assembly life calculations

THK Calculate the service life of LM Guide

Title: SHS30 table bearings | Date: 2008-11-20

Operating Condition





Mounting Orientation:	Vertical installation	Velocity:	V=	0.025m/s	Rail span(mm):	W=	300
Model Number:	SHS30	Acceleration time:	T1=	0.25s	Block span(mm):	L1=	250
Basic dynamic load rating:	C= 44800N	Load factor:	fw=	1.2		L2=	---
Basic static load rating:	Co= 66600N	Stroke length:	Ls=	3000mm		L3=	---
Revised Coefficient:	Ka_MR= ---	Number of reciprocations per minute:	N1=	0.2min^{-1}	Thrust point(mm):	B1=	150
	Ka_ML= ---	Acceleration:	α=	0.100m/s^2		B2=	10
	Kb_M= ---	Acceleration,Deceleration:	S1=	3.13mm	Block width(mm):	BW=	60
		Velocity:	S2=	2993.75mm	Block length(mm):	BL=	106

Mass(kg)	m1= 2500	m2= 0	m3= 0		
Gravity points(mm)	G1= 250	G2= 300	G3= 60	G4= 0	G5= 0
	G6= 0	G7= 0	G8= 0	G9= 0	

Calculation Result

Applied load(N)

		No.1	No.2	No.3	No.4
(1) During acceleration	Radial direction	-3467.4	3467.5	3467.5	-3467.4
	Horizontal direction	4953.5	-4953.5	-4953.5	4953.5
	Equivalent load	8421	8421	8421	8421
(2) In uniform motion	Radial direction	-3432.4	3432.5	3432.5	-3432.4
	Horizontal direction	4903.5	-4903.5	-4903.5	4903.5
	Equivalent load	8336	8336	8336	8336
(3) During deceleration	Radial direction	-3397.4	3397.5	3397.5	-3397.4
	Horizontal direction	4853.5	-4853.5	-4853.5	4853.5
	Equivalent load	8251	8251	8251	8251
(4) Mean load	Pm=	8336N			

Static safety factor
fs= 7.9

Nominal life
L= 4491km

Service life time
Lh= 62382h

Notes:

* Ex: 2.35E+06 is displayed in exponential notation as $2.35x10^6$.
* Calculation results for this software are based on the theoretical calculation in accordance with your input conditions. Under actual operation, they may vary depending on the operation conditions like operating environment, lubrication condition, mounting accuracy, rigidity & etc.
* As for the equivalent load, we show the value for the specific raceway for which the nominal life becomes the shortest.
* Nominal life and static safety factor may be calculated using differing raceway equivalent loads.
* When the static safety factor (fs) is less than 1.0, since it is over the basic static load rating (C0a), judge it to be unusable and enter "---" for the calculation result of the service life. Please change the model and/or operating conditions so that you can ensure a static safety factor of over 1.0, and then calculate again.
* For operation at high speed (over V>2 m/s), as the value may vary because of other factors, we recommend to verify durability & etc. with the actual operating machine.
* This software is made generally assuming carrier equipments with standard clearance and C1 clearance. Therefore, we have no calculation emphasizing preload. For operation with C0 clearance emphasizing rigidity, or when any influence from outside factors are expected, please confer with THK.
* For operation with C0 clearance, it is possible that the nominal life and the service life may become shorter than the calculation results obtained from this software.

Appendix E - Website URLs accessed for research information

Boat hoists:
http://www.bh-usa.com/
http://www.boatliftwarehouse.com/
http://www.boathoistdirect.com/
http://www.boatliftdistributors.com/

Legislation:
http://www.legislation.nsw.gov.au/scanview/inforce/s/1/?TITLE=%22Occupational%20Health%20and%20Safety%20Act%202000%20No%2040%22&nohits=y
http://www.legislation.nsw.gov.au/scanview/inforce/s/1/?SRTITLE=%22Occupational%20Health%20and%20Safety%20Regulation%202001%22&nohits=y
http://www.deir.qld.gov.au/workplace/law/legislation/act/index.htm
http://www.deir.qld.gov.au/workplace/law/legislation/regulation/index.htm

Ball and roller screw suppliers and information:
http://www.roymech.co.uk/Useful_Tables/Cams_Springs/Power_Screws.html
http://www.linearbearings.com.au/Home/tabid/36/Default.aspx
http://www.powerjacks.com/Power-Jacks-Screw-Jacks.htm
http://www.linearmotion.skf.com/

Hydraulics:
http://www.atos.com/index_nosuono.html

Lifting hardware – wires and sheaves etc:
http://www.nobles.com.au/
http://www.superlift.com.au/index.htm
http://www.activelifting.com.au/

Electric motor and gearbox information:
http://www.sew-eurodrive.com.au/produkt/A13.htm
http://www.weg.net/au/Products-Services/Motors/Low-Voltage-Motors-IEC-Brake-Motor
http://www.reliance.com/prodserv/motgen/motgenhome.htm

Boat manufacturers:
http://quintrex.com.au/
http://www.seajayboats.com.au/
http://www.allycraft.com.au/Home.aspx

Other
http://www.onesteel.com/
http://www.winch.com.au/winch/home/home.html

Appendix F – AS 1418.1 2002 tables

F-1 AS 1418.1 2002 table 7.3.4.2

TABLE 7.3.4.2

CLASS OF UTILIZATION OF MECHANISMS

Class of utilization	Total duration of use H	Description of use
T_0	H ≤ 200	Infrequent use
T_1	200 < H ≤ 400	
T_2	400 < H ≤ 800	
T_3	800 < H ≤ 1600	
T_4	1600 < H ≤ 3200	Fairly frequent use
T_5	3200 < H ≤ 6300	Frequent use
T_6	6300 < H ≤ 12 500	Very frequent use
T_7	12 500 < H ≤ 25 000	Continuous or near continuous use
T_8	25 000 < H ≤ 50 000	
T_9	50 000 < H ≤ 100 000	
T_{10}	100 000 < H	

F-2 AS1418.1 2002 table 7.3.4.3

TABLE 7.3.4.3

NOMINAL LOAD SPECTRUM FACTOR AND STATE OF LOADING FOR CRANE MECHANISMS

Nominal load spectrum factor (K_m)	State of loading	Description of use
0.125	$L1$—Very light	Mechanisms subjected very rarely to the maximum load and, normally, to very light loads
0.25	$L2$—Light	Mechanisms subjected fairly frequently to the maximum load but, normally, to rather light loads
0.50	$L3$—Medium	Mechanisms subjected frequently to the maximum load and, normally, to loads of moderate magnitude
1.00	$L4$—Heavy	Mechanisms subjected with high frequency to the maximum load

F-3 AS1418.1 2002 table 7.3.4.4

TABLE 7.3.4.4

GROUP CLASSIFICATION OF CRANE MECHANISMS

1	2	3	4	5	6	7	8	9	10	11	12
State of loading	**Nominal load spectrum factor (K_m)**	**Group classification of crane mechanism**									
		Class of utilization									
		T_0	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9
*L*1—Light	0.125	M1	M1	M1	M2	M3	M4	M5	M6	M7	M8
*L*2—Moderate	0.25	M1	M1	M2	M3	M4	M5	M6	M7	M8	*
*L*3—Heavy	0.50	M1	M2	M3	M4	M5	M6	M7	M8	*	*
*L*4—Very heavy	1.00	M2	M3	M4	M5	M6	M7	M8	*	*	*

NOTE: Where class utilization calculations give a crane mechanisms group classification of greater than M8, as indicated by an asterisk (*), the mechanism shall be designed for the required rated life.

F-4 AS1418.1 2002 table 7.9

TABLE 7.9

LOAD COMBINATIONS FOR CRANE MECHANISMS

Load group	Line number	Loading condition		Frequently occurring load combinations				
		Load type		Vertical motion	Horizontal motion			Horizontal and vertical motion (see Note)
		Description	Symbol	Raise or lower	Traverse	Travel	Slewing	
				1	2	3	4	5
Principal loads	1	Dead load of mechanism	R_1	R_1	R_1	R_s	R_1	R_1
	2	Dead load of parts of crane acting on mechanism or component	R_2	—	R_s	R_2	R_s	R_2
	3	Hook load mass (payload)	R_3	R_3	R_3	R_3	R_3	R_3
	4	Dynamic effects of payload	R_4	R_4	—	—	—	—
	5	Dynamic effects due to inertia of mechanism	R_5	R_5	R_5	R_5	R_5	R_s
	6	Frictional forces	R_6	R_6	R_6	R_6	R_6	R_6
	7	Service wind (acting horizontal)	V_1	—	—	—	—	—
	8	Out of service-wind (acting horizontal)	V_2	—	—	—	—	—
Additional load	9	Wind, snow, ice, temperature extremes, oblique travel	—	—	—	—	—	—
Special loads	10	Collision forces with buffers	B_1	—	—	—	—	—
	11	Emergency conditions	B_2	—	—	—	—	—
γ_c				1.0				

NOTE: Combined horizontal/vertical motion occurs during the following in service conditions:

(a) Luffing or telescoping with a non-level luffing crane.

(b) Travel or traverse on an inclined plane.

(c) Slew on an inclined plane.

F-4 (continued)

TABLE 7.9 (*continued*)

Infrequently occurring load combinations					Rarely occurring load combinations																	
Vertical motion	Horizontal motion			Horizontal and vertical motion (see Note)	Vertical motion	Horizontal motion									Horizontal and vertical motion (see Note)		
Raise or lower	Traverse	Travel	Slewing		Raise or lower	Traverse			Travel			Slewing					
6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Not applicable	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1	R_1
	R_2	R_2	R_2	R_2	—	R_2	R_2	R_2	R_2	R_2	R_2	R_2	R_2	R_2	R_S	R_2	R_2
	R_3	R_3	R_S	R_S	R_3	—	R_3	R_3	—	R_3	R_3	—	R_3	R_3	—	R_3	R_3
	—	—	—	—	R_4	—	—	—	—	—	—	—	—	—	—	—	—
	R_5	R_5	R_5	R_5	R_5	—	—	—	—	—	—	—	—	—	—	—	—
	R_6	R_6	R_6	R_6	R_6	—	—	—	—	—	—	—	—	—	—	—	—
	V_1	V_N	V_1	V_1	—	—	—	—	—	—	—	—	—	—	—	—	—
	—	—	—	—	—	V_2	—	—	V_2	—	—	V_2	—	—	V_2	—	—
	—	—	—	—	—	—	B_1	—	—	B_1	—	—	B_1	—	—	B_1	—
	—	—	—	—	B_S	—	—	B_2	—	—	B_2	—	—	B_2	—	—	B_2
0.9					0.75												

F-5 AS1418.1 2002 table 7.16.2.1

TABLE 7.16.2.1

MINIMUM COEFFICIENT OF UTILIZATION (Z_p) FOR REEVED SYSTEMS

Classification of mechanism	Minimum coefficient of utilization (Z_P)
M1	3.15
M2	3.35
M3	3.55
M4	4.0
M5	4.5
M6	5.6
M7	7.1
M8	9.0

F-6 AS1418.1 2002 table 7.18

TABLE 7.18

RATIOS OF DRUM AND SHEAVE PITCH DIAMETERS TO ROPE DIAMETER

Classification of mechanism	Minimum ratio of drum and sheave pitch diameter to steel wire rope diameter (D/d)		
	Drums (h_d)	Sheaves (h_s)	Rope equalizer sheaves (h_e)
M1	11.2	12.5	11.2
M2	12.5	14.0	12.5
M3	14.0	16.0	12.5
M4	16.0	18.0	14.0
M5	18.0	20.0	14.0
M6	20.0	22.4	16.0
M7	22.4	25.0	16.0
M8	25.0	28.0	18.0

F-7 AS1418.1 2002 table 7.15

TABLE 7.15

MINIMUM COEFFICIENT OF UTILIZATION (Z_p) FOR OTHER THAN REEVED SYSTEMS

Classification of mechanism	Minimum coefficient of utilization (Z_P)
M1	2.5
M2	2.5
M3	3.0
M4	3.5
M5	4.0
M6	4.5
M7	5.0
M8	5.0

Appendix G – Wire rope anchorages AS 2759

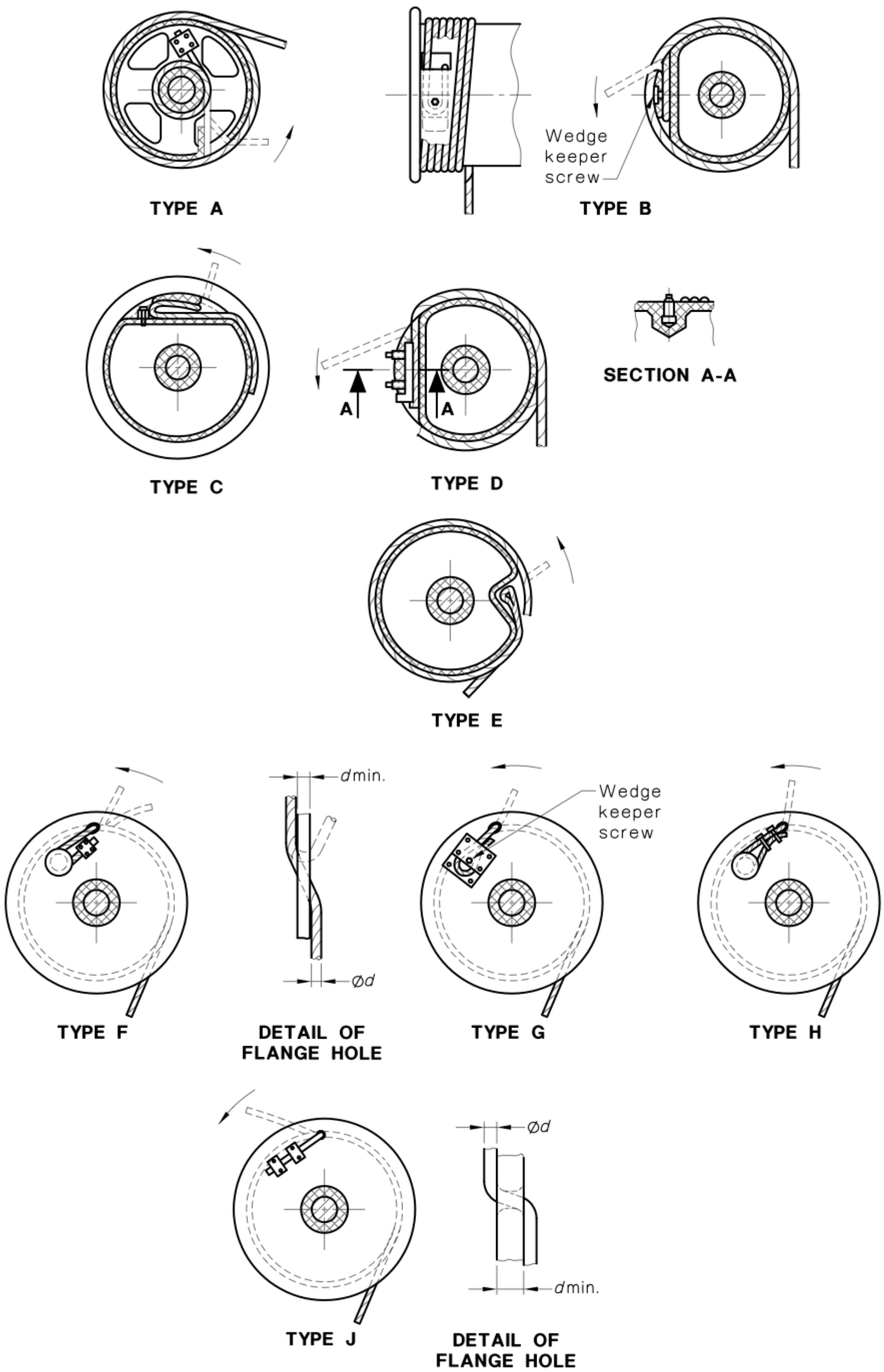


FIGURE 9.2(A) TYPES OF ANCHORAGE ADAPTABLE FOR REVERSE WINDING

Appendix G (continued)

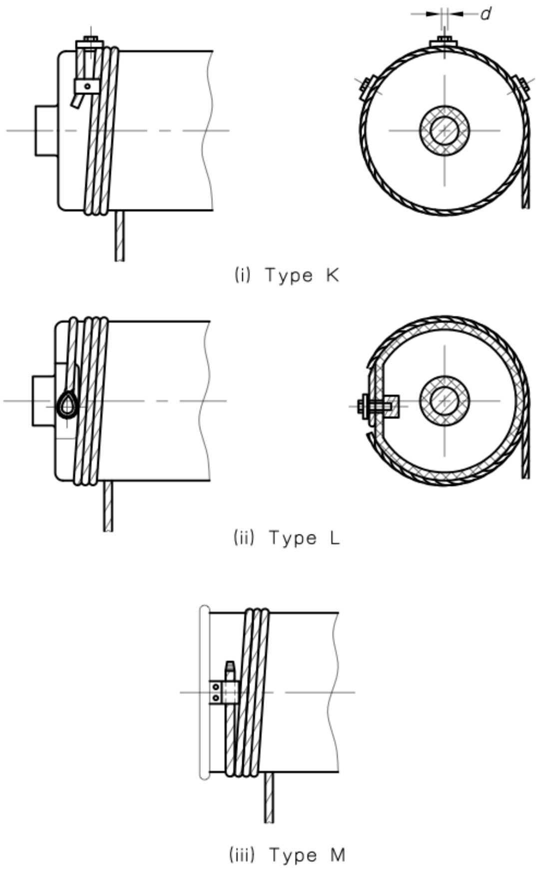


FIGURE 9.2(B) TYPES OF ANCHORAGE NOT ADAPTABLE FOR REVERSE WINDING

Appendix H – Quotes

H-1 Quote SEW Eurodrive gearmotor and inverter

SEW EURODRIVE

SEW-EURODRIVE Pty. Ltd. ACN 006 076 053
27 Beverage Drive, Tullamarine, Victoria, 3043
Tel: (03) 9933 1000 Fax (sales): (03) 9933 1003

Date: 1/12/2008
No of pages: 1

Quotation number: ML241108/01

All correspondence regarding this quotation must make reference to the quotation number.

SEW Contact person: Martin Luvara E-Mail: martin.luvara@sew-eurodrive.com.au

Company: Phone: Fax:

Attention: Foster Lear Email: flear@bigpond.com

Subject: Your ref:

Helical-bevel gear unit with AC squirrel-cagemotor

Quantity: **1** Catalogue Description: **KAF97R57DRE80M4BE2**

Rated power	[kW]:	**0.75**	Output torque	[Nm]:	2630
Motor speed	[rpm]:	1435	Reduction ratio	:	573
Output speed	[rpm]:	**2.5**	Permissible overhung load	[N]:	40000
Service factor	:	1.65	Frequency	[Hz]:	50
Rated voltage	[V]:	230/400	Cos (phi)	:	0.78
Rated current	[A]:	1.68	Duty type	:	S1-100%
Wiring diagram	:	R13	Thermal classification	:	F
Enclosure	:	**IP55**	Mounting position	**:**	**Please advise**
Brake torque	[Nm]:	**20**	Terminal box position	[°]**:**	**Please advise**
Brake control	:	BG	Output flange	[mm]:	**450**
Brake voltage	[V]:	**400**	Output shaft	[mm]:	**70**
Weight	[kg]:	198.7			

Included options
BE2- SEW - disk brake

Standard delivery would be 4-5 working days from order. Price per unit excluding GST $ 4704.92 AUD

MOVITRAC B Frequency Inverter			
Device Type	MC07B 0008-2B1-4-00	Unit Price (Ex GST)	$322
Quantity	1		
Part Number	8284954		
Technical Specifications			
Input Voltage	200V-240V		
Rated Output Power	0.75kW		
Rated Output Current	4.2A		
Enclosure	IP 20		
Dimensions (W×L×H)	80mm*185mm*163mm		
Weight	1.5kg		

Option Cards				
Keypad Slot	FBG11B Keypad		Price	$50
Interface Slot	FIO11B Analog Module + Communication Interface		Price	$133
Option Slot				
Accessories		**Part Number**		
Brake Resistor - BW072-005 - 72 Ω , 0.5 kW, For 0005…0014		826 060 5	Price	$125
Delivery; 1-2 days from order date		**Total Unit Price (Ex GST)**		**$630**

FREIGHT:

A $20 + GST delivery charge will apply for each order shipped within the metropolitan area of Melbourne, Sydney, Adelaide, Brisbane and Perth. For delivery outside of these areas, please nominate your freight carrier and account details when placing the order.

NOTE: To ensure correct supply of goods all characteristics in bold type and the quotation number must appear on your purchase order.

DXF/DWG drawings of the above unit can be generated and downloaded at SEW Drivegate (registration required). 3D model's in SAT, STEP and IGES format are also available at SEW Drivegate online.

Regards

Martin Luvara

This quotation is valid for 30 days from the above. SEW Eurodrive terms and conditions apply, a copy of which is available upon request. No returns will be accepted after 60 days from invoice date or without prior approval. Returns may be subject to a restocking fee.

H-2 Quote SKF power screw

Quotation

"Quality Conscious and Safe"

To :	Foster Lear	**Fax :**	
Company :			
From :	Goran Danilovic	**Fax :**	+ 61 3 9269 0712
	SKF Bearing Supplies Pty Ltd	**Tel :**	+ 61 3 9269 0830
	AMC - Melbourne Australia	**E-Mail :**	Goran.Danilovic@skf.com
Date :	Monday, 1 December 2008	**Page :**	100 of 1
Subject :	Price and availability	**Quote Ref :**	FL1011GD

Price and availability as requested - quote reference must be used when ordering:

QTY	ITEM	PRICE EACH (excludes GST)	AVAILABILITY
1	SL40X20 R NUT	$1756.45 ea + gss	Ex stock Melb
1	VF40X20 R 3000/3000G9	$643.95 per metre	1-2 Working weeks
			ex production France
			Plus shipping
1	Air Freight ex France	$810.00	7-10 W/days

Delivery - Air freight ($810.00) 7-10 WORKING DAYS

All prices quoted are your costs unless marked otherwise and do not include GST. GST will be added to the price and charged at the rate of 10%.

Deliveries quoted are subject to prior sales.

Quotation validity: 30 days unless marked otherwise.

Appendix I - SKF data tables

I-1 Data sheet SKF Spherical roller bearing on adapter sleeves

SKF

Product tables | Search | IMP | PDF | Print | Close

Spherical roller bearings, on an adapter sleeve

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation
			dynamic	static		Reference speed	Limiting speed		Bearing + adapter sleeve
d_1	D	B	C	C_0	P_u				* - SKF Explorer bearing
mm			kN		kN	r/min		kg	-
60	150	51	400	430	45	3400	4500	5,35	**22314 EK + H 2314 ***
65	130	31	212	240	26,5	4800	6300	2,45	**22215 EK + H 315 ***
65	160	37	285	325	34,5	4000	5600	4,50	**21315 EK + H 315 ***
65	160	55	440	475	48	3200	4300	6,50	**22315 EK + H 2315 ***
70	140	33	236	270	29	4300	6000	3,00	**22216 EK + H 316 ***
70	170	39	325	375	39	3800	5300	5,30	**21316 EK + H 316 ***
70	170	58	490	540	54	3000	4000	7,65	**22316 EK + H 2316 ***
75	150	36	285	325	34,5	4000	5600	3,70	**22217 EK + H 317 ***
75	180	41	325	375	39	3800	5300	6,20	**21317 EK + H 317 ***
75	180	60	550	620	61	2800	3800	8,85	**22317 EK + H 2317 ***
80	160	40	325	375	39	3800	5300	4,55	**22218 EK + H 318 ***
80	160	52,4	355	440	48	2800	3800	6,00	**23218 CCK/W33 + H 2318 ***
80	190	43	380	450	46,5	3600	4800	7,25	**21318 EK + H 318 ***
80	190	64	610	695	67	2600	3600	10,5	**22318 EK + H 2318 ***
85	170	43	380	450	46,5	3600	4800	5,45	**22219 EK + H 319 ***
85	200	45	425	490	49	3400	4500	8,25	**21319 EK + H 319 ***
85	200	67	670	765	73,5	2600	3400	12,0	**22319 EK + H 2319 ***
90	165	52	365	490	53	3000	4000	6,15	**23120 CCK/W33 + H 3120 ***
90	180	46	425	490	49	3400	4500	6,40	**22220 EK + H 320 ***
90	180	60,3	475	600	63	2400	3400	8,75	**23220 CCK/W33 + H 2320 ***
90	215	47	425	490	49	3400	4500	10,5	**21320 EK + H 320 ***
90	215	73	815	950	88	2400	3000	15,2	**22320 EK + H 2320 ***
100	170	45	310	440	46,5	3400	4300	5,75	**23022 CCK/W33 + H 322 ***
100	180	56	430	585	61	2800	3600	7,70	**23122 CCK/W33 + H 3122 ***
100	200	53	560	640	63	3000	4000	8,90	**22222 EK + H 322 ***

I-2 Data sheet SKF split plummer block housings

Product tables | Search | IMP | PDF | Print | Close

Select design (all)

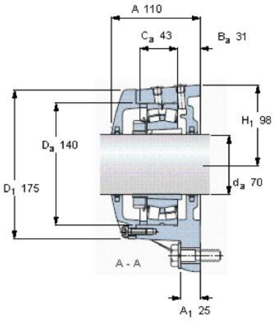
Split plummer block housings, SNL series for bearings on an adapter sleeve, with standard seals

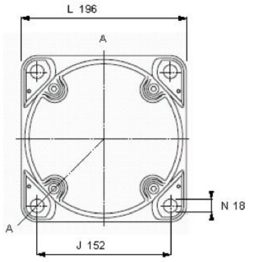
Shaft	Appropriate bearings (basic designation)					Housing Principal dimensions				Mass	Designations				
	Self-aligning ball bearings		Spherical roller bearings		CARB						Housing only	Housing complete with			
d_a					bearing	A	L	H	H_1			lip seals	V-ring seals	labyrinth seals	Taconite seals
mm	-					mm				kg	-				
61,913	1315 K	2315 K	21315 K	22315 K	C 2315 K	140	345	194	100	12,5	**SNL 518-615**	**SNL 615 TGA**	**SNL 615 TAE**	**SNL 615 TSA-**	
63,5	1215 K	2215 K	-	22215 K	C 2215 K	115	280	156	80	7,00	**SNL 515-612**	**SNL 515 TGE**	**SNL 515 TA**	**SNL 515 TSE-**	
63,5	1215 K	2215 K	-	22215 K	C 2215 K	115	280	156	80	7,00	**FSNL 515-612**	**FSNL 515 TGE**	**FSNL 515 TA**	**FSNL 515 TSE**	-
63,5	1315 K	2315 K	21315 K	22315 K	C 2315 K	140	345	194	100	12,5	**FSNL 518-615**	**FSNL 615 TGE**	**FSNL 615 TA**	**FSNL 615 TSE**	-
63,5	1315 K	2315 K	21315 K	22315 K	C 2315 K	140	345	194	100	12,5	**SNL 518-615**	**SNL 615 TGE**	**SNL 615 TA**	**SNL 615 TSE-**	
65	1215 K	2215 K	-	22215 K	C 2215 K	115	280	156	80	7,00	**SNL 515-612**	**SNL 515 TL**	**SNL 515 TA**	**SNL 515 TS**	**SNL 515 TND**
65	1215 K	2215 K	-	22215 K	C 2215 K	115	280	156	80	7,00	**FSNL 515-612**	**FSNL 515 TL**	**FSNL 515 TA**	**FSNL 515 TS**	**FSNL 515 TND**
65	1315 K	2315 K	21315 K	22315 K	C 2315 K	140	345	194	100	12,5	**SNL 518-615**	**SNL 615 TG**	**SNL 615 TA**	**SNL 615 TS**	**SNL 615 TND**
65	1315 K	2315 K	21315 K	22315 K	C 2315 K	140	345	194	100	12,5	**FSNL 518-615**	**FSNL 615 TG**	**FSNL 615 TA**	**FSNL 615 TS**	**FSNL 615 TND**
68,263	1216 K	2216 K	-	22216 K	C 2216 K	120	315	177	95	9,50	**SNL 516-613**	**SNL 516 TGA**	**SNL 516 TA**	**SNL 516 TSA-**	
68,263	1216 K	2216 K	-	22216 K	C 2216 K	120	315	177	95	9,50	**FSNL 516-613**	**FSNL 516 TGA**	**FSNL 516 TA**	**FSNL 516 TSA**	-
68,263	1316 K	2316 K	21316 K	22316 K	C 2316 K	145	345	212	112	13,7	**FSNL 519-616**	**FSNL 616 TGA**	**FSNL 616 TA**	**FSNL 616 TSA**	-
68,263	1316 K	2316 K	21316 K	22316 K	C 2316 K	145	345	212	112	13,7	**SNL 519-616**	**SNL 616 TGA**	**SNL 616 TA**	**SNL 616 TSA-**	
69,85	1216 K	2216 K	-	22216 K	C 2216 K	120	315	177	95	9,50	**SNL 516-613**	**SNL 516 TL**	**SNL 516 TA**	**SNL 516 TSE-**	
69,85	1216 K	2216 K	-	22216 K	C 2216 K	120	315	177	95	9,50	**FSNL 516-613**	**FSNL 516 TL**	**FSNL 516 TA**	**FSNL 516 TSE**	-
69,85	1316 K	2316 K	21316 K	22316 K	C 2316 K	145	345	212	112	13,7	**FSNL 519-616**	**FSNL 616 TG**	**FSNL 616 TA**	**FSNL 616 TSE**	-
69,85	1316 K	2316 K	21316 K	22316 K	C 2316 K	145	345	212	112	13,7	**SNL 519-616**	**SNL 616 TG**	**SNL 616 TA**	**SNL 616 TSE-**	
70	1216 K	2216 K	-	22216 K	C 2216 K	120	315	177	95	9,50	**SNL 516-613**	**SNL 516 TL**	**SNL 516 TA**	**SNL 516 TS**	**SNL 516 TND**
70	1216 K	2216 K	-	22216 K	C 2216 K	120	315	177	95	9,50	**FSNL 516-613**	**FSNL 516 TL**	**FSNL 516 TA**	**FSNL 516 TS**	**FSNL 516 TND**
70	1316 K	2316 K	21316 K	22316 K	C 2316 K	145	345	212	112	13,7	**SNL 519-616**	**SNL 616 TG**	**SNL 616 TA**	**SNL 616 TS**	**SNL 616 TND**
70	1316 K	2316 K	21316 K	22316 K	C 2316 K	145	345	212	112	13,7	**FSNL 519-616**	**FSNL 616 TG**	**FSNL 616 TA**	**FSNL 616 TS**	**FSNL 616 TND**
74,612	1217 K	2217 K	-	22217 K	C 2217 K	125	320	183	95	10,0	**SNL 517**	**SNL 517 TL**	**SNL 517 TA**	**SNL 517 TSA-**	
74,612	1217 K	2217 K	-	22217 K	C 2217 K	125	320	183	95	10,0	**FSNL 517**	**FSNL 517 TL**	**FSNL 517 TA**	**FSNL 517 TSA**	-
74,612	1317 K	2317 K	21317 K	22317 K	C 2317 K	160	380	218	112	17,6	**FSNL 520-617**	**FSNL 617 TG**	**FSNL 617 TA**	**FSNL 617 TSA**	-
74,612	1317 K	2317 K	21317 K	22317 K	C 2317 K	160	380	218	112	17,6	**SNL 520-617**	**SNL 617 TG**	**SNL 617 TA**	**SNL 617 TSA-**	

I-3 Data sheet flanges bearing housing data sheet

Flanged housings, FNL series for bearings on an adapter sleeve

Shaft	Appropriate bearings (basic designation)				Housing Principal dimensions				Mass	Designation
	Self-aligning ball bearings		Spherical roller bearing	CARB bearing						
d_a					A	L	H	H_1		
mm					mm				kg	-
70	1216 K	2216 K	22216 K	C 2216 K	110	196	196	98	10,5	FNL 516 B



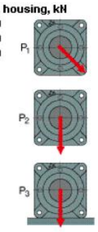


Appropriate SKF bearings and accessories

Bearing	Adapter sleeve	Locating rings
1216 K	H 216	FRB 8.5/140 (2)
2216 EKTN9	H 316	FRB 10/140 (1)
22216 EK	H 316	FRB 10/140 (1)
C 2216 K	H 316 E	FRB 10/140 (1)

Breaking loads, housing, kN

P_1	150
P_2	140
P_3	340



Included housing components

Contact seals	TFL 516 (2)
Cover bolts	M 8x25 (4)
Rec. tightening torque, Nm	24

Appropriate attachment bolts

Size, mm	M 16
Rec. tightening torque, Nm	200

Grease quantities, kg

Initial fill	0,3

Appendix J - Tables and figures for design of rotating steel shafts

J-1 AS1403 2004 table 2

TABLE 2

FORMULAS FOR CALCULATING MINIMUM DIAMETER OF SHAFT D

Number of mechanism starts per year	Number of revolutions of shaft per year	Torque application conditions	Formula	Formulas	
≤600	≤900	Manually or power applied	$D^3 = \frac{10^4 F_S}{F_Y}\sqrt{\left(M_q + \frac{P_q D}{8000}\right)^2 + \frac{3}{4}T_q^2}$	1	
≤600	>900	Power applied	$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{4}T_q^2}$	2	See Notes 2 and 3
>600	>900	Power applied torque reversals	$D^3 = \frac{10^4 F_S}{F_R}K_S K\sqrt{\left(M_q + \frac{P_q D}{8000}\right)^2 + \frac{3}{4}T_q^2}$	3	See Notes 2 and 3
>600	>900	Power applied, no torque reversals (see Note 1)	$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{16}\left[(1 + K_S K)\,T_q\right]^2}$	4	See Notes 2 and 3

NOTES:

1 Where the magnitude of the torque in one direction is not greater than 0.1 times the torque in the other direction, the torque application conditions may be considered as being non-reversing.

2 These formulas are extracted from a paper titled 'Shortcuts for Designing Shafts' by H.A. Borchardt, published in *Machine Design*, Vol. 45, No.3, 8 February 1973, pp 139-141.

3 The value of F_R is based on 10^6 stress cycles and is applicable for numbers of revolutions of shafts per year greater than 50 000.

4 For numbers of revolutions of shafts per year from 50 000 down to 900, formulas 2, 3 and 4 result in progressively more conservative values for the theoretical diameter of shaft.

5 The value of F_s is 2.0 for Formula 1 and 1.2 for Formulas 2, 3 and 4. Where severe injury, death or extensive equipment damage is likely to occur because of the failure of the shaft, higher factors of safety may be used.

6 The values of K_s, K and the term $P_q\ D/8000$ may require the calculation of a 'trial' diameter (see Appendix A).

J-2 AS1403 2004 shaft diameter calculations, Size factor

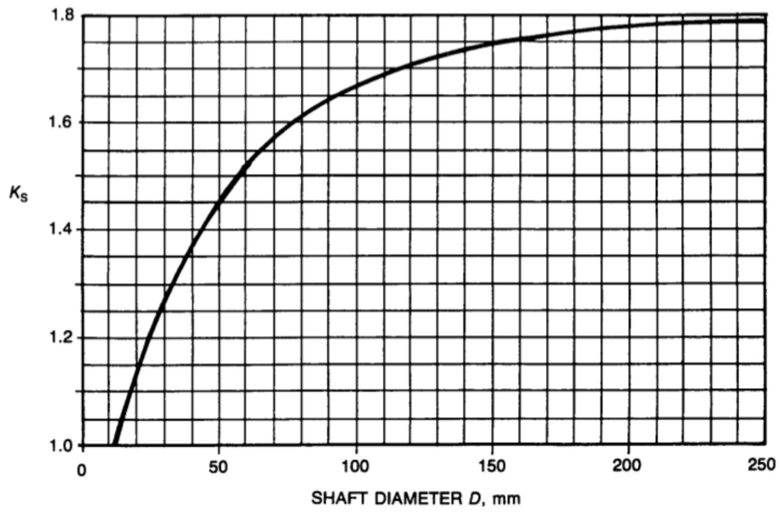


FIGURE 1 SIZE FACTOR K_s

J-3 AS1403 2004 shaft diameter calculations, stress raising figure

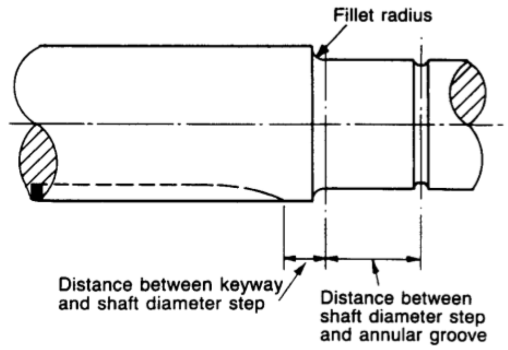


FIGURE 2 TYPICAL EXAMPLES OF STRESS-RAISING CHARACTERISTICS

J-4 AS1403 2004 shaft diameter calculations, stress raising factor figures 3-10

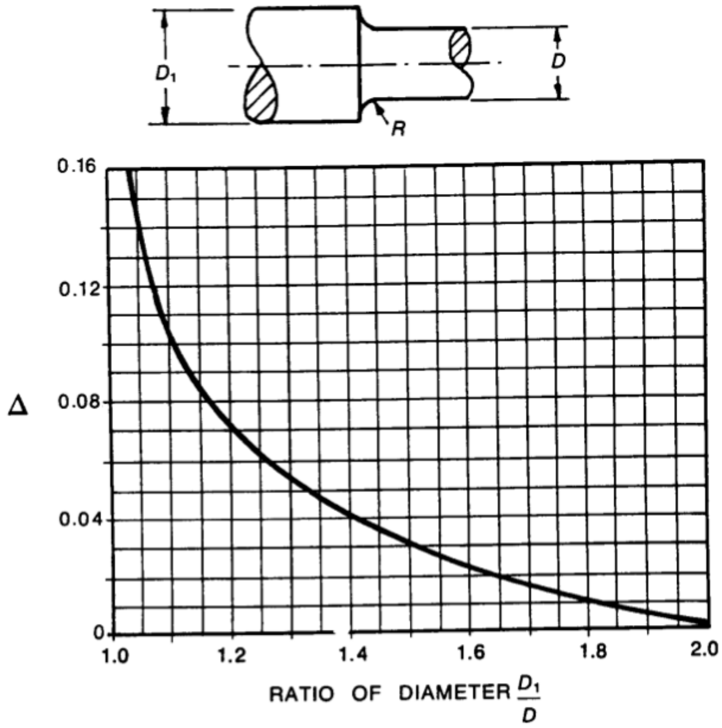


FIGURE 3 CORRECTION FACTOR Δ

J-4 (continued)

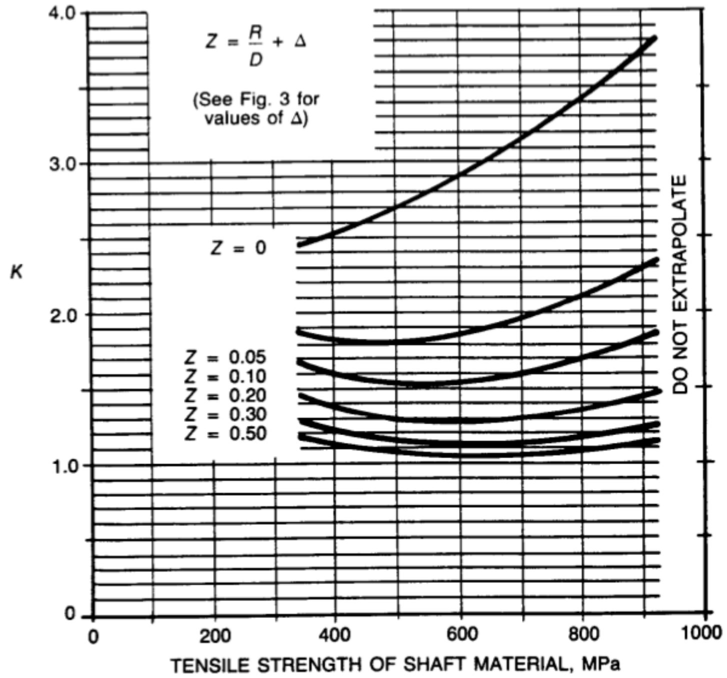
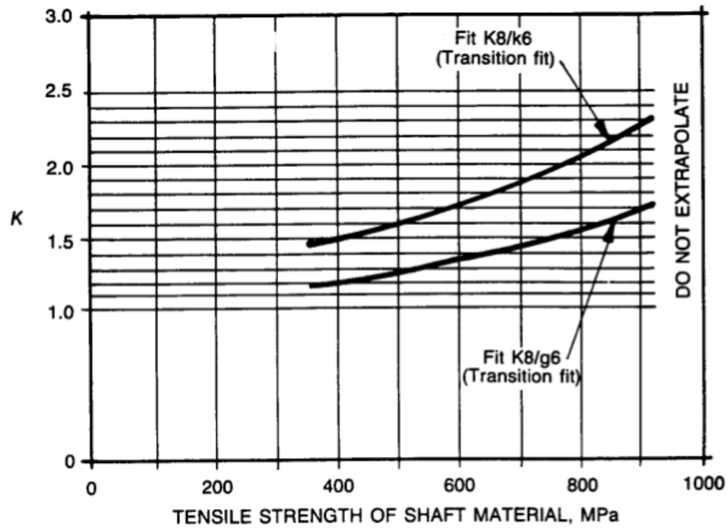


FIGURE 4 STRESS-RAISING FACTOR K FOR STEPPED SHAFT



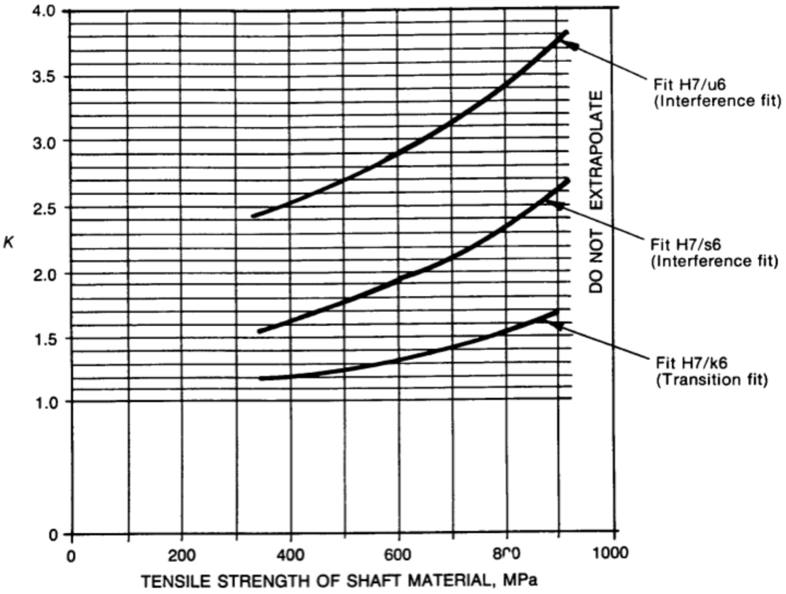
NOTES:

1 Values may be interpolated for fits between K8/k6 and K8/g6 which are recommended by the bearing manufacturers.

2 For tolerances, see AS 1654.

FIGURE 5 STRESS-RAISING FACTOR K FOR SHAFT FITTED WITH ROLLING ELEMENT BEARING

J-4 (continued)



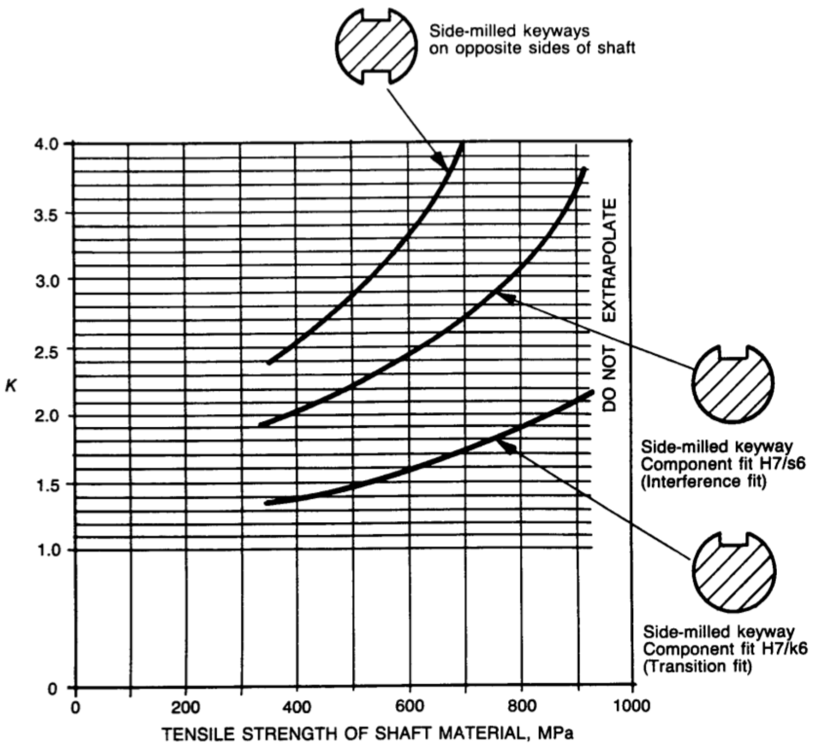
NOTES:

1 Values may be interpolated for other fits between H7/u6 and H7/k6.

2 For tolerances, see AS 1654.

3 For stress-raising factors of locking devices refer to the manufacturer's recommendations.

FIGURE 6 STRESS-RAISING FACTOR *K* FOR FITTED COMPONENT WITHOUT KEY OR SPLINE



NOTES:

1 For end-milled keyway with blind-end, multiply factor *K* by 1.1.

2 Values may be interpolated for fits between H7/s6 and H7/k6.

3 For tolerances, see AS 1654.

FIGURE 7 STRESS-RAISING FACTOR *K* FOR KEYED COMPONENT

J-4 (continued)

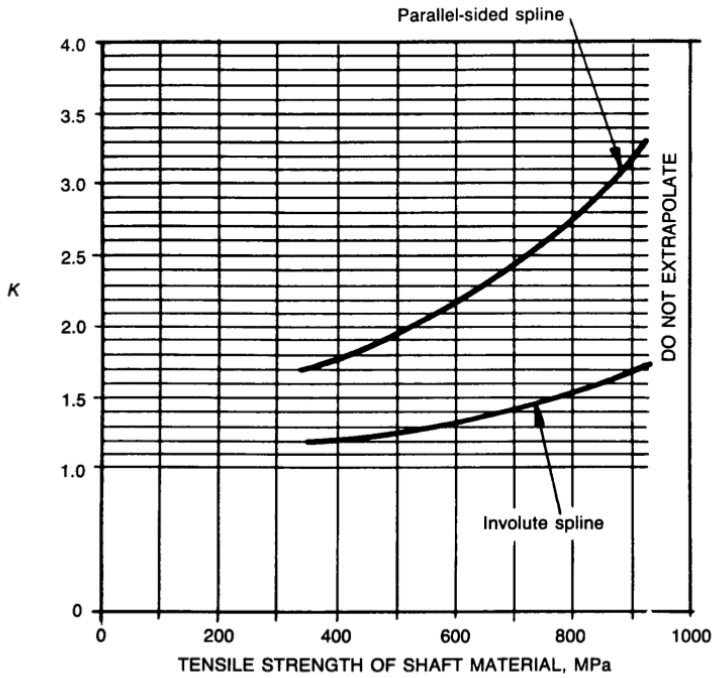


FIGURE 8 STRESS-RAISING FACTOR *K* FOR SPLINED SHAFT

J-4 (continued)

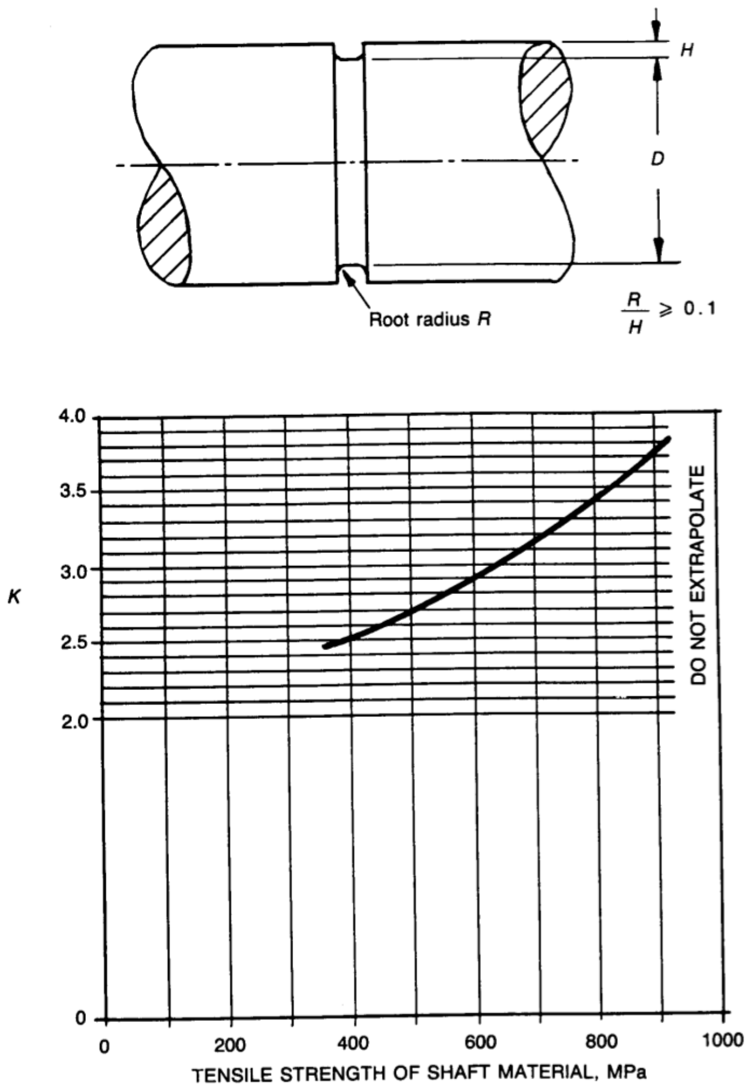


FIGURE 9 STRESS-RAISING FACTOR K FOR SHAFT WITH ANNULAR GROOVE

J-4 (continued)

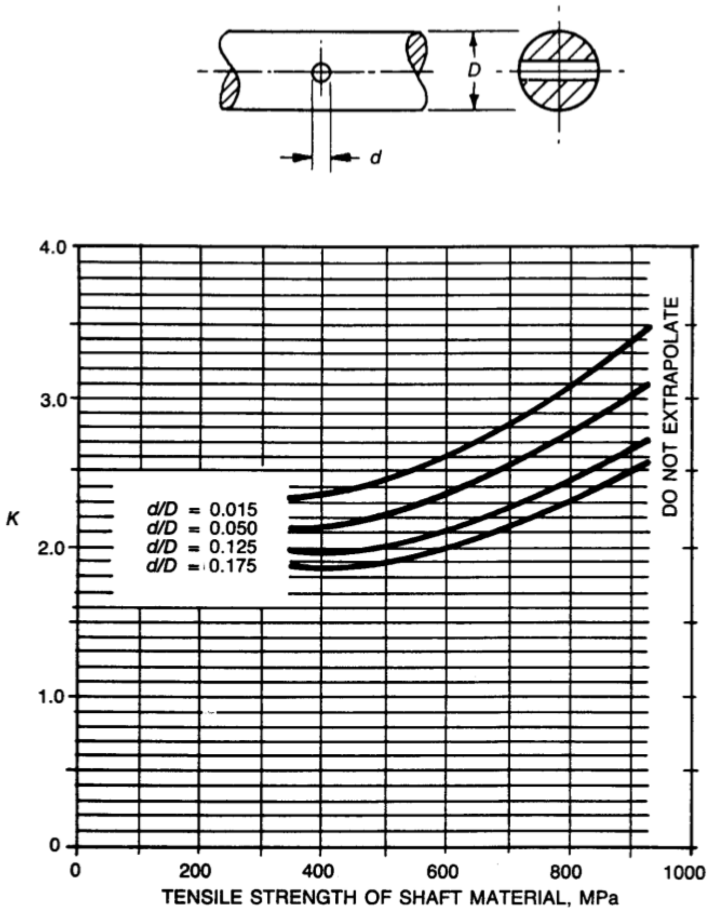
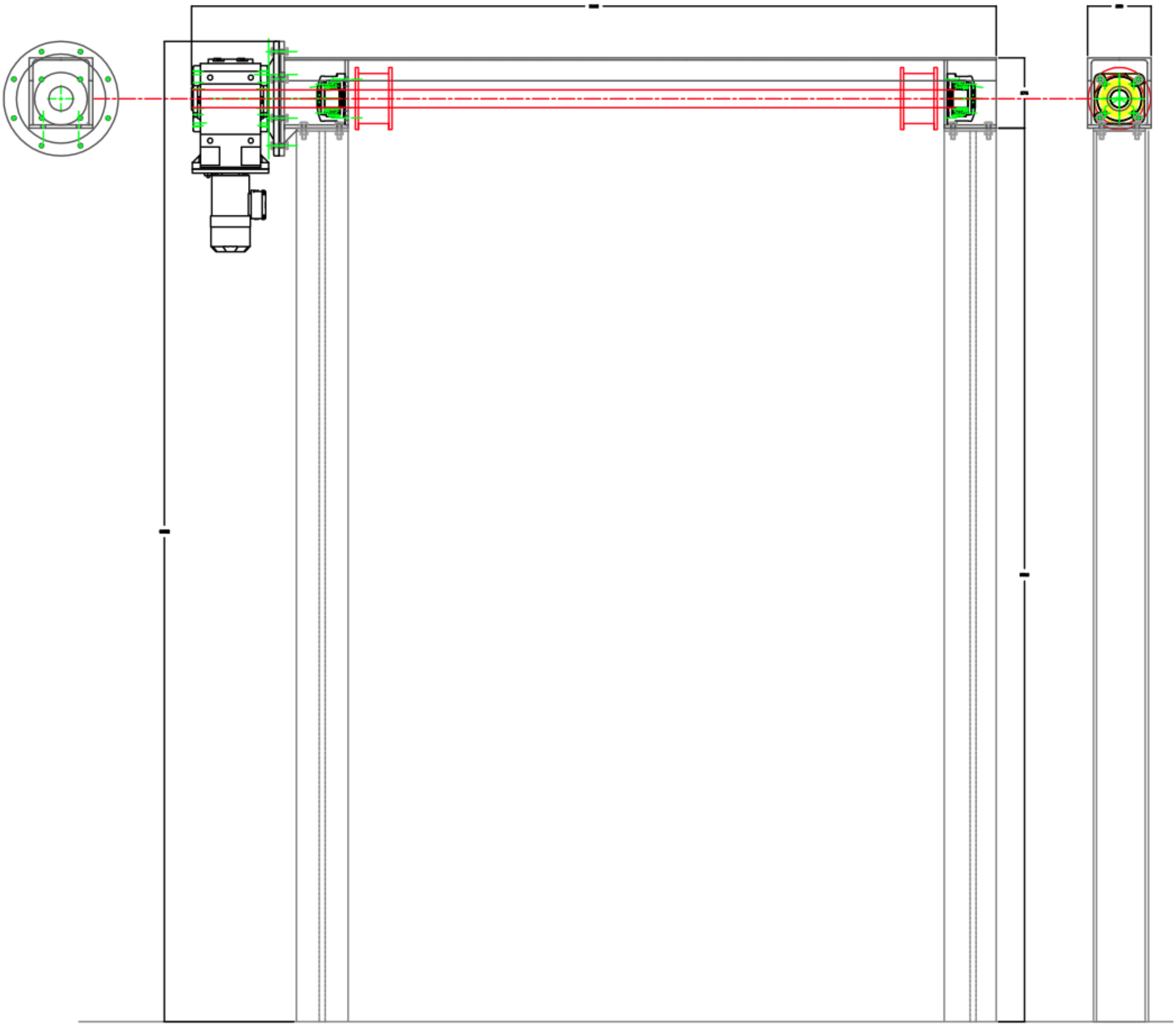


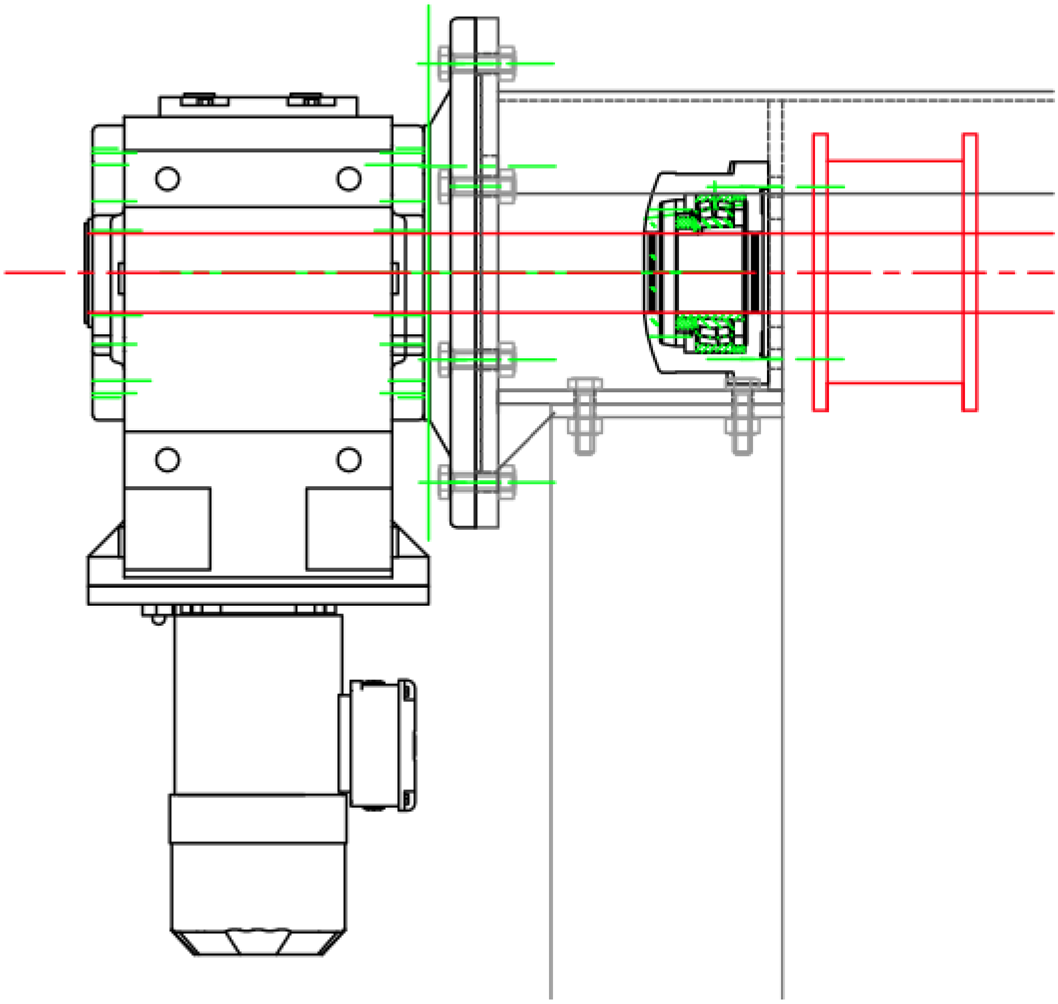
FIGURE 10 STRESS-RAISING FACTOR *K* FOR SHAFT WITH TRANSVERSE HOLE

Appendix K – Drawings

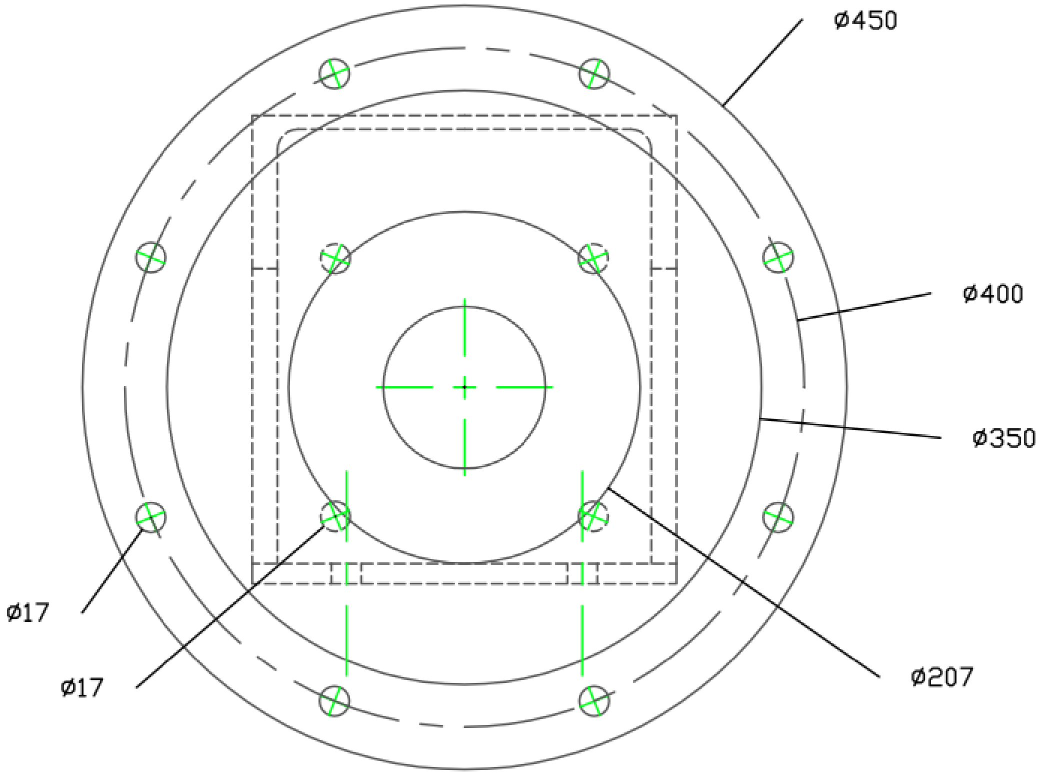
K-1 Hoist assembly

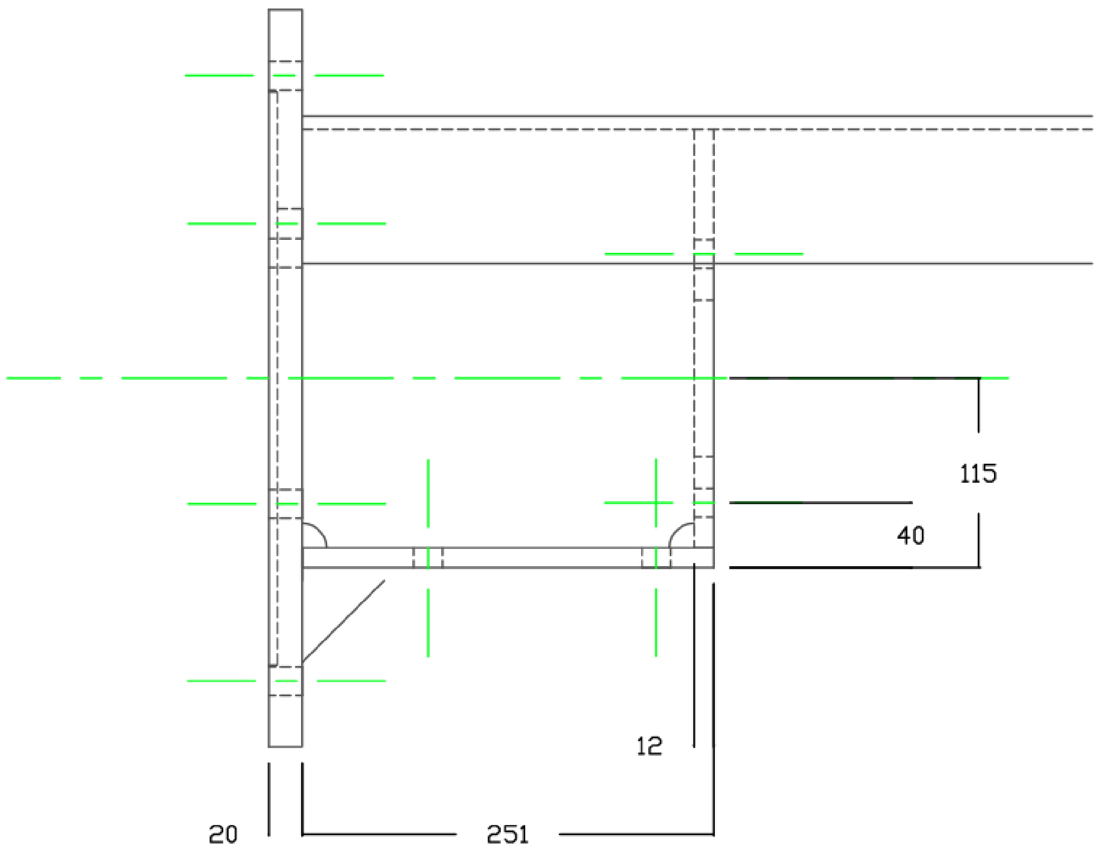


K-2 Drive end drawings showing assembly and beam construction

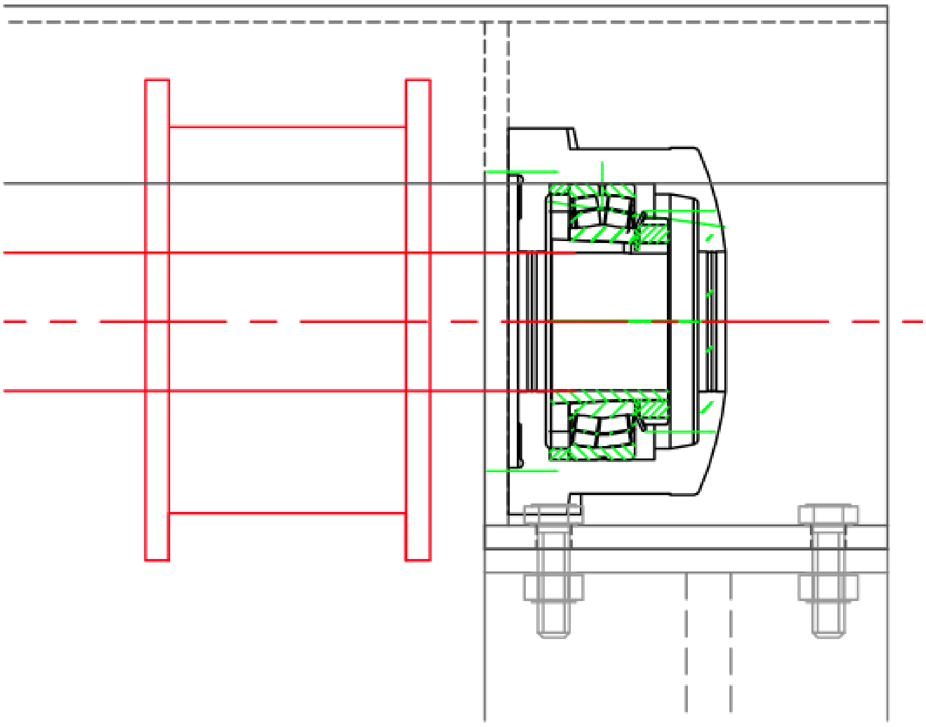


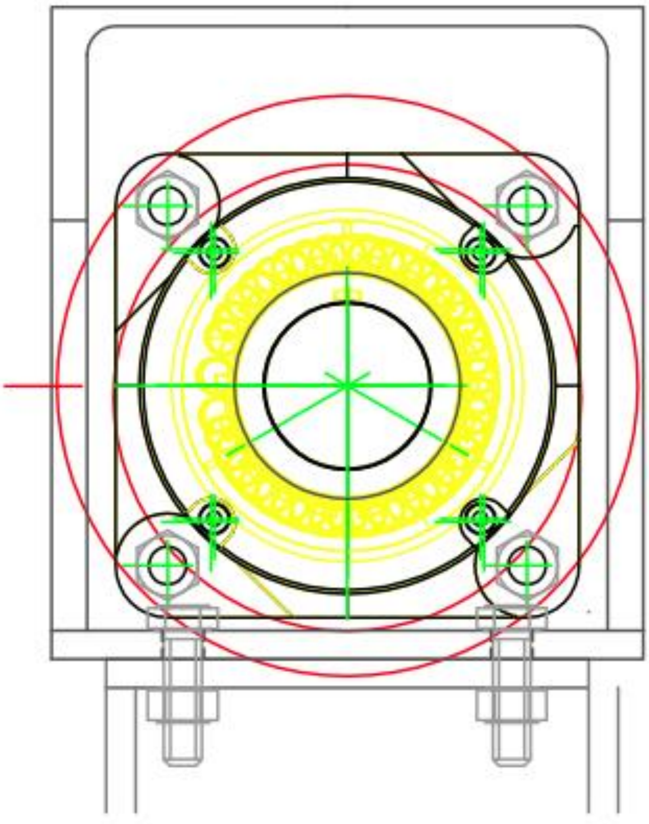
K-2 (continued)





K-3 Non-drive end assembly and beam construction





K-3 (continued)