

**A Design Methodology Investigation
and the Design
of a Material Handling System**

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Abstract

This dissertation is undertaken under the auspices of both the CSIR, Division of Mining Technology and the University of Natal, School of Mechanical Engineering. The CSIR have outlined two fundamental objectives of the dissertation. Firstly, the need for competent design engineers has become increasingly evident. To this end, an evaluation and research into the science of design methodology has been conducted and regarded as a significant component of the thesis. The rationale behind this aim is that the subject of design has been practiced for thousands of years, but an understanding of the process is comparably in its infancy. The importance of the steps involved in the mechanical design process can in no uncertain terms be overemphasized as the adherence thereto results in designs that are least likely prone to failure as well as the attainment of highly efficient product design time scales. This is vitally important more especially when the drive towards multifunctional multidisciplinary teams is rapidly developing in the global market place. Secondly, the CSIR, having done the appropriate market research, have defined the need for the design of a timber handling system to be implemented in a deep level mining environment. It is the authors expressed intent not to separate the theory from the design at hand but rather to allow this thesis to become, for the reader, forum where a holistic and integrated approach to design can be presented.

Declaration

I, Daryl Sebastian Govender, student number 953038711, hereby declare that the contents of this dissertation, is my own unaided work, unless otherwise stated. It is being submitted for the Degree of Master of Science in Engineering, to the University of Natal, Durban, Neither this dissertation, or part thereof, has been submitted before for any degree or examination at any other university.

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Chapter 1

Introduction

The objectives of this postgraduate undertaking are firstly to obtain an understanding of the design process in a broad context and secondly to allow for a manifestation of this process into the design of a timber handling system.

1.1. Engineering and Design

In the context of today's global market, design is a sophisticated process resulting in the manufacture of products, the development of processes, systems and subsystems. To understand the role of engineering design, a logical foundation would be the differentiation between science, society and engineering. This understanding of the differences should in itself highlight the interaction between them.

The Oxford Dictionary defines science as that branch of study which is concerned either with a connected body of demonstrated truths or with observed facts systematically classified and more or less colligated by being brought under general laws. From this definition one can deduce that the nature of science is to investigate the true nature of our surrounding and the mechanisms of change from one form to another. The scientist attempts to formulate theory which explains the observed facts and predicts the outcome of situations.

The scientific method depicted in Figure 1.1 consists of a repetition of observation, hypothesis and test eventually leading to the proposal of a general law. According to Chambers Encyclopedia Society is defined as the total system of recurrent actions, performed by an aggregate of human beings, who are differentiated by sex, age, role or status, variously linked to ties of kinship, sharing submissions to common authorities, distributed over a more or less

contiguous and boundary territory; and possessing continuity through time by virtue of biological reproduction and transmission of beliefs. The nature of human geography, history, climate, economy and psychology are the factors that are characteristic of a society.

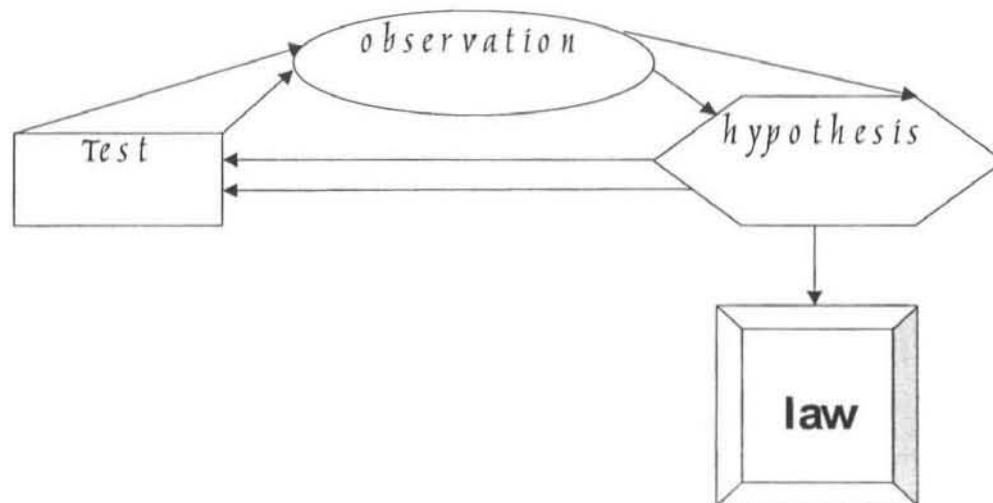


Figure 1.1. The Scientific Method (after Svensson, 1976)

It is the fluctuating balances between these factors in space and time that cause society to change from place to place and from time to time. Engineering is an iterative decision-making activity that generates plans to make optimum consumption of resources in order to satisfy human needs. Engineering as an activity is intended to benefit society. To this end, the needs of society should be understood. Crudely put, people desire food, shelter, transportation and recreation to live complete lives. The objective of engineering is to utilize scientific principles in order to develop practical solutions that are of value to society. The engineer must satisfy these said needs within the limitations imposed by social, economic and time constraints.

More than ever before, engineering is engaged in rapid development rarely matched by any other profession. To keep pace with engineering advances the engineer must devote much attention to technical journals and to make a study of

patents which concern himself. He must endeavor to apply for copies of catalogues and leaflets for information purposes. Engineers are also actively involved in research, writing theory and developing theoretical models. Therefore the work of the engineer does overlap into the science activity coining the term engineering science.

It is, however, the design discipline of engineering which exemplifies engineering in action. An important quality of an engineer is the ability to dichotomize problems. In this spirit, the definition of engineering given above has been broken down to obtain a clearer understanding.

Iteration : A satisfactory conclusion is achieved after several repeated attempts: information developed during the preliminary study is used in a second trial, and so on.

Decision Making : The engineer is frequently faced with one or more alternatives. This compels him to decide which of the many alternatives he must accept in order to solve the problem. The decision is based upon various working criteria and various ends or values which are exchanged from case to case. These exchanges are known as “trade offs.”

Conversion of Resources : Time, money, natural resources and ability are the important factors that need to be monitored to ensure effective and efficient modification of a resource into useful goods. These resources are always finite, and their nature and availability varies from situation to situation.

The word design comes from the Latin word *designare*, which means to designate or mark out. Pugh, 1990, defines design as the ‘total activity necessary to provide a product or process to meet a market need.’ Engineering design according to Asimow, 1962, is a purposeful activity directed toward the goal of fulfilling the needs of mankind. In the light of the worldwide drive for designs to be ecologically and environmentally friendly the author suggests a modification to Asimow’s definition. Design is a purposeful process directed toward the attainment of practical solutions that satisfy the needs of the living world. It is involved with the total situation embracing science and society.

It is the authors belief that engineering design is intellectually challenging. As a discipline it demands clear mindedness, creativity and logical thought. Somewhat paradoxical as it may seem -creativity and logical thought but it is here that the challenge lies. The design process requires the manifestation of creative thoughts into logical solutions. It rewards those who practice it with the opportunity to continually stimulate and enhance their creative problem solving ability and increase their knowledge on a daily basis.

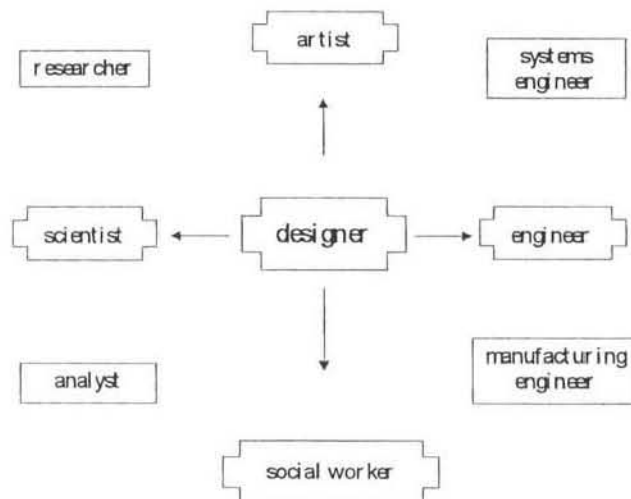


Figure 1.2. The engineering designer (after Pahl and Beitz, 1988)

Pahl and Beitz, 1988, have placed the role of the designer as the central sphere across the societal spectrum as shown in Figure 1.2. He must incorporate artistic and creative ability and be highly sensitive to the needs of the society in which he functions (the vertical spectrum, Figure 1.2). He must possess an inquiring mind ever ready to support and research his assertions, be practical in thought and numerate in his endeavors always seeking the most optimum solution, (horizontal spectrum, Figure 1.2).

An activity as important as design does, therefore, make great demands of those who undertake to practice it. It is not without justification that it is written, of the 'sweat and toil' of designing ; indeed the persistent unremitting struggle with the matter can call for the most strenuous efforts of which the human mind is capable. At the same time it demands the utmost concentration on detail and the understanding of wide ranging interrelations.

Design work forces the designer to give the most careful consideration to problems, to test, to compare and to make decisions - always with the realization that he or she may be making a mistake. Design calls, furthermore, for knowledge which is not confined to a narrow speciality but which ranges far into other fields.

As a result of the aforementioned arguments key words that come into play when engineering design is mentioned should be : Goal Orientated; Variform; Constrained; Evolutive; Probabilistic; Value Comparative; Compromising.

1.2. Why the Need to Study the Design Method

The February 2000 edition of the Engineering Management Journal reported on a survey indicating that between 35 % and 44 % of all products launched fail in the market place. These are catastrophic figures when one considers the waste

of time, money and resources associated with them. These figures only represent the products that are launched and not the products that fail to even make it to the production stage because of lack of foresight or poor designing practice. Companies are therefore in need of a method for the development of new and existing products and / or processes.

The design method presented here is not a far cry from what designers have been doing ever since the design of the potters wheel over five thousand years ago. What it does do, is that it replaces inconsistent, time consuming off the cuff decisions and practices with a structured approach. “Blind Designs” are no longer effective as products have become more sophisticated, consumer driven, government influenced and an increasing number of concepts are protected by patenting. The success of blind designs are therefore left to chance. Success in today's engineering world means highly efficient design to market time scales as well as the delivery of a working, reliable product to the market.

Sustained success requires systematic thoroughness and meticulous attention to detail from the beginning to the end of the design process. Design methodology is a framework within which the designer can practice with thoroughness. As mentioned already, design is the essence of engineering for it is an amalgam of technological skills, insight and the ever present need and desire to improve. To practice design successfully in today's fast paced, highly competitive market a formal method is fast becoming an industry demand.

To further entrench the dire need for a formal method associated with the design process one needs to consider that products have become so complex that most require a team of people with a multidisciplinary skills base to take a product from need through to design and production. In addition giving the design process a sense of direction is in itself an incentive to spur the designer to get on with the task at hand.

1.3. Creativity and the Design Engineer

One must feel that there must be some sort of creative activity in evolving a technical product. Something genuinely new occurs in design, when that which has never been seen, known or thought of before - i.e. a clear picture, backed up by drawings and perhaps by models as well, of a certain object with a certain shape, size and other characteristics, first comes into existence through intellectual effort.

Creativity can be defined as the successful step across the borderline of knowledge. The creative person has a driving curiosity, a willingness to explore the unconventional, to emphasize the unique as opposed to the traditional, to continually seek out the need for a device or product, and he believes that a truly unique solution exists to the problem at hand. Creative discoveries are more likely to occur when one lets one's imagination soar and then engineers it back to earth.

It is now the authors intention to formally list the ideal characteristics of an engineering designer. It should be borne in mind that if all designers possessed all the qualities listed below we would live in a perfectly engineered world. Rather every design is a manifestation of the designer's or design team's aspirations and personal skills.

1.3.1. Ability to identify problems

One of the basic characterial requirements of a successful designer is the ability to identify the problem at hand. Very often correct solutions are found to the wrong problem. Correct identification of the real problem is paramount in setting the project heading toward the right goal.

1.3.2. Imaginative

Strongly allied to this word is foresight. The designer must imagine or foresee the project from an understanding of what he has on paper to the stage when the product will be used and what could happen to the product in its operating environment. He must predict the changes in this environment and the consequences of the use of the product by both skilled and unskilled people.

1.3.3. Sense of Urgency

The designer must be able to think and act quickly but without losing his effectiveness by panic. He must also maintain a sense of balance between what is what is ideal and what is attainable within the restriction of time and finance. To this end, his skills must be awakened as reliable and quick judgment will allow his design to sustain and or acquire an often predetermined target market share.

1.3.4. Ability to Simplify Problems

Many situations lead more often than not to quite complicated analysis which needs to be carried out for predicting the performance of the product. Even with the drive that has made computers and engineering software an indispensable task simplification tool to the modern designer, he must guard against using the tool as a black box without understanding the logic and theory behind the electronic solution. The model that the designer submits for electronic analysis is very often a simplified one. Such simplification is coupled with the recognition of major facts and understanding, that is, the engineer must understand the consequences of his simplification. He must substantiate and account for the possible variation in

the solution to the simplified model and be ever ready to make adjustments if the answers seem unreasonable.

1.3.5. Numerate

Since there is a considerable amount of analysis of concepts in design, it is essential that the designer be adept at mathematical analysis. He must be capable of developing appropriate mathematical and computational procedures to solve equations which develop during the analysis. The use of high speed computers only means that the designers understanding of the numerical output, as mentioned above, should be heightened.

1.3.6. Decisive

The decision activity occurs at all stages of the design process. The designer must be psychologically capable of making a decision and have the courage to pursue the choices made. An absolute embargo should be placed on indecision and steps should be taken to avoid being overly conservative. The engineer at all times should be able to support his decision with calculations, tests, sound engineering principles and relevant case studies.

1.3.7. Open Minded

The design engineer must at all times be aware that he functions in a society where the only constant is change. Technology is advancing at an unprecedented rate and one cannot conceive new solutions if the mind is closed to the receptance of new things. He must be willing and able to have an open mind when his designs are criticized with properly supported arguments.

1.4. Overview of the Design Process

The purpose of this section is to set the stage for the chapters to follow. Points highlighted here have been treated and examined in greater depth later. Crudely put, design consists of examining a need and the development of a solution through sketches, models, brainstorming, calculations, working on the appropriate styling, ensuring that the product is suited to its working environment, that it as a sub-component or assembly is compatible with its parent and or co-systems. Further, care must be taken to ensure that the product can be manufactured and the costs incurred for the design are within the financial constraints.

The process of design can be represented schematically to levels of increasing formality and complexity in Figure 1.3 and Figure 1.4 below. Figure 1.3 represents the traditional approach associated with lone designers comprising of the generation of ‘bright ideas’ drawings and calculations - giving form to the idea, judgment of the design and reevaluation if necessary, resulting in the generation of the end product. Figure 1.4 shows a more formal description of the design process which might be associated with the designer operating as part of a design or management team. The following is a preview of the design process.

1.4.1. Recognition of need

Design is sometimes ignited with a simple idea that fulfills or satisfies a need and can be identified by any individual in an organization or a team of people dedicated to market research. As a direct implication of the aforementioned statement, design can begin when a potential market is identified for a product, device or process. Alternatively the ‘need’ can be established when a company decides to re-engineer one of its existing products.

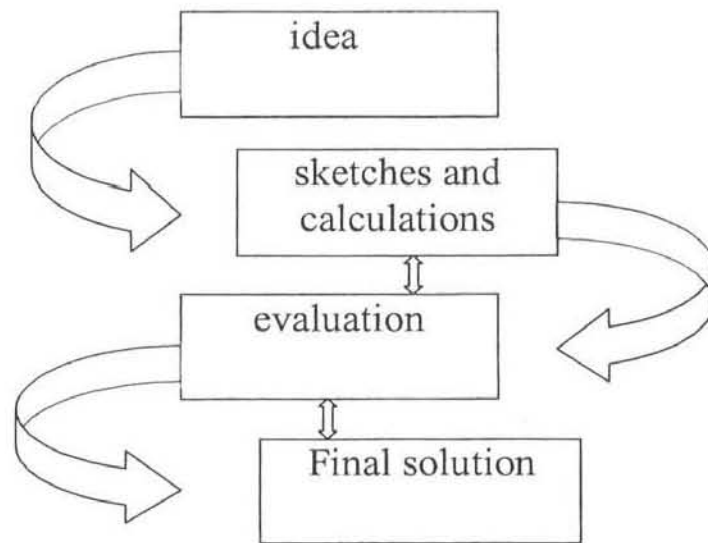


Figure 1.3. The traditional designers approach
(after Childs, 1998)

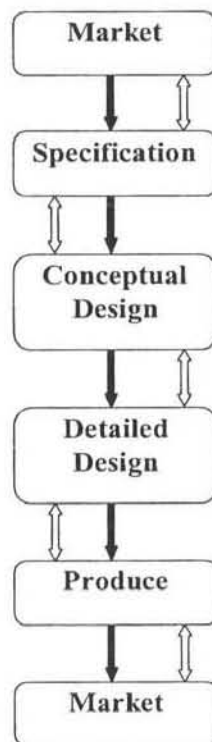


Figure 1.4. The total design core (after Pugh, 1990)

1.4.2. Problem definition

This stage encompasses and lists the characteristics desired of the product. It should include inputs and outputs, qualities, preliminary dimensions and limitations on quantities. In short it is that stage which results in the initial product specification.

1.4.3. Synthesis

This is that part of the design process in which ideas are generated and developed into a concept which offers a potential solution to the problem at hand. The potential solution to the problem should identify itself as closely as possible to the product specification.

1.4.4. Analysis

This is the truly technical stage of the design. It involves the application of engineering technology in earnest. The engineering tools, solid and fluid mechanics, dynamics, electrical technology and the like, are used to examine the design. Invariably analysis and synthesis go hand in hand.

1.4.5. Optimization

This is the process of repetitively refining a set of often conflicting criteria to achieve the best performance.

Chapter 2 - Part A

Methods of Problem Identification and Specification Development

2.1. Identifying the need

As highlighted in the previous chapter, the onset of design is more often than not ignited with the identification of a need or requirement which when given voice, will manifest itself in the design of a product or process unique to the established need.

It is important that as a foundation to the discussion and development of a rationale behind the 'needs identification of society', one must be aware of the total production-consumption cycle of mans socio-economic system. Figure 2.1. depicts the stages that material goods pass through in their cycling from raw materials to raw materials again, after it has served its intended function and been discarded.

Human beings as a result of urbanization and industrialization have managed to dissociate themselves in some instances from the cycle of the natural world. One such example is the case of inefficient recovery resulting in waste. In the natural world, this cycle is closed since waste material becomes raw material.

In Figure 2.1. goods flow counterclockwise while the information necessary for the production cycle generally flow clockwise. Design is a part of planning which is a part of production and the information about the needs of everyone involved in the cycle is of paramount importance to the design team or engineer. The four operating groups (production, distribution, consumption and recovery) are in fact

made up of sub- groups. The list in Table 2.1 is by no means exhaustive and a particular product, device or process will extract its own unique assembly of subgroups from the list.

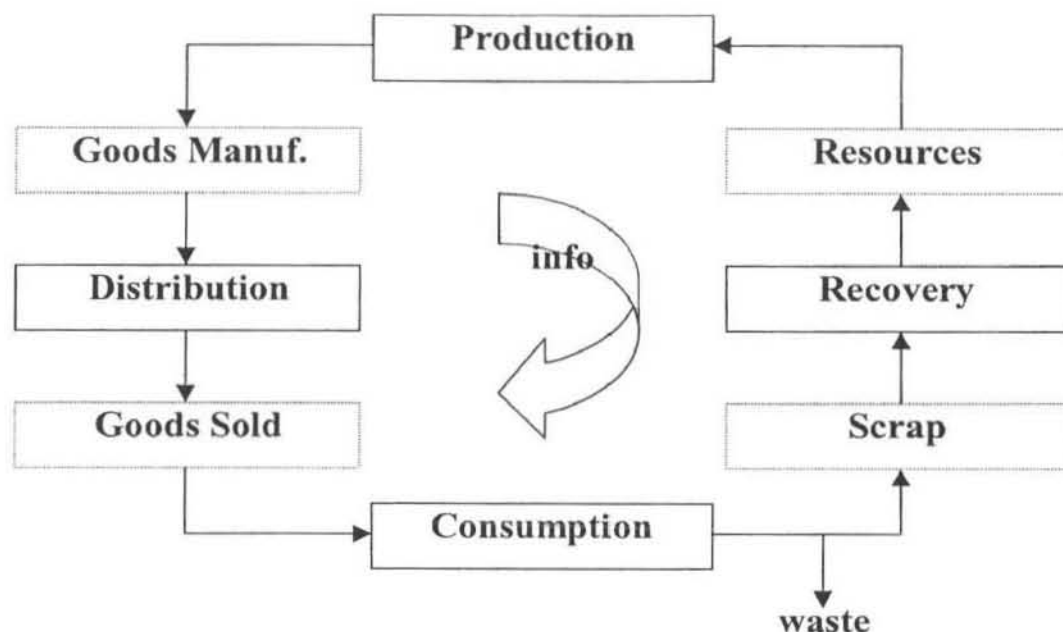


Figure 2.1. The production consumption cycle (after Woodson, 1966)

Table 2.1. Operating groups and their sub groups (after Woodson, 1966)

Production	Distribution	Consumption	Recovery
Planners and designers	Whole Sale Trade	Mass Consumers Indirect-Consumption	Salvage
Raw Material Suppliers	Retail Trade	Individual Consumers- Direct Consumption	Collection
Component Suppliers	Transportation		Conversion
Capital Equipment Suppliers	Finance		
Fabricators	Product Service		

Each individual group receives money for the effort made to ensure the perpetuation of the cycle but the fundamental drive for this said perpetuation comes from the consumer. He alone inputs money for goods supplied or services rendered. The other operating groups extract money and input effort. The entire motive power for the production cycle therefore comes from the consumer who may be an individual citizen, some governmental group, a company or some combination of these.

The design team management and or the marketing department may possibly provide the motivation for product design or re-engineering from endeavors made by competitors to increase their market share and hence threaten the survival of the existing product. In fact, the perceived need may be presented to a design team by anyone who has a vested interest in a solution to a particular problem. Market researchers and information scientists are sometimes referred to as front end designers as they have the responsibility of having a heightened awareness of consumer needs and they must present this information to the product designers.

2.2. Defining the Problem

The needs of the consumer more often than not are presented to those responsible for the technical development in simple (non technical) and sometimes ambiguous statement/s. These words become for the designer the “Primitive Statement Of Need”. They must be scrutinized removing all ambiguity and in doing so the engineer must address all viable meanings. The Primitive Statement is therefore a mere starting point for the engineer since, until they are examined and validated, can the designer or design team proceed with reasonable surety that the product to be developed will be one that the consumer wants badly enough to pay for.

It must be clearly understood that when the primitive statement is presented for development from a source other than a reliable market study, it is merely an expression of opinion and may in fact be based on extremely tenuous observations. There is no merit in providing a correct solution to a poorly defined problem. In effect, the designer is solving the wrong problem resulting in wasted time, money and effort. In this case, no monetary power will be provided by the consumer to sustain the production cycle resulting in the ultimate failure of the product in the market place.

To begin examining the problem the engineer must begin his thinking at the "Primitive Statement of Need". The engineer must be aware of the variability that may arise in the analysis that concerns the initial statement. These variations may be in description, interpretation and assessment. This said statement must be examined as follows :

- Where did it come from ? (its origin)
- Why is it deemed to be a need ?
- Whose need is it ?
- When does the need have to be satisfied ?
- For how long will it be desired ?
- Does it affect other needs of society and possibly the environment ?
- What were the observations that inspired the expression of the need ?

The answers to these questions will allow the need statement to be somewhat formalized having more substance than its original form. A very important part of the problem definition involves the careful definition of each word that appears in the statement. In this manner, trigger words promote inquiry and one question leads to another until the problem is defined sufficiently well to generate the technical specifications. The specification list must only be attempted once all uncertainty has been removed from the primitive statement.

2.3. The Product Design Specification

The justified and unambiguous need should be formally documented in the form of a brief. Having done this a next step would be to develop the technical specifications or the product blue print. This is referred to as the Product Design Specification (PDS). It is vitally important for the designer to utilize this document thoroughly throughout the design. Hence no amount of emphasis can be placed on the effort and time spent on the generation of this document.

The PDS acts as the controlling mechanism, mantle or envelope for the total design activity. Whatever the designer is concerned with, this document is his terms of reference. The PDS is a working document and is very often described as being a dynamic document because if at any time the occasion arises for change to the original PDS everyone concerned will be informed of the change, the proper justification thereof and the consequences of the change.

There are various formats for this document and it may be very much unique to a specific company. Pugh, 1990, suggests the format as shown in Figure 2.2. Figure 2.3. shows the aspects of information content of a PDS.

Date : _____ Product : _____

	Competition Best	Current Model (ours)	This design (intent)	World Class (target)
<i>Performance</i> (description)				
<i>Safety</i> (description)				

Figure 2.2. PDS suggested format (after Pugh, 1990)

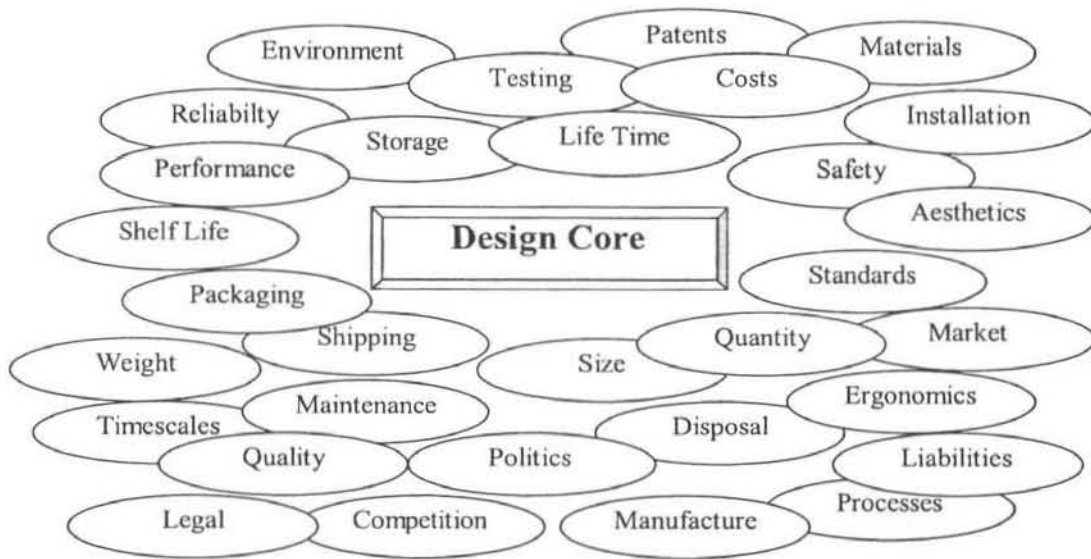


Figure 2.3. Possible aspects of the PDS (after Childs, 1998)

The format put forward by Pahl and Beitz (1988) is very similar to the one presented by Pugh (1990), however, they make it a point to distinguish between those items which are described as a *want* and those items that are a *demand*. A demand is that aspect of the design which under all circumstances should be satisfied, a want is an aspect that is desired of the product which are only met within the design constraints if possible. The differences between wants and demands are highlighted by the *Kano Model Of Customer Satisfaction*.

According to this model developed by Dr. Noriaki Kano in the early 1980's, there are three ways, or more precisely, three types of product qualities which satisfy the customer : Basic quality, Performance Quality and Excitement Quality.

Basic Quality refers to customers requirements that are not verbalized as they specify assumed functions or product quality. The only time a customer will mention it is when it is absent from the product or process.

Performance Quality makes reference to those requirements that are verbalized in the form that the better the performance the better the quality. The customer will notice the change, if you will, and allow a mental differentiation between similar products to take place.

Excitement Quality are those qualities that are unspoken of by the customer in requesting such a product as he or she does not expect them to be met by the particular design. If they are absent, the customer is neutral but at the very least surprised at the additional functionality of the product.

The methods presented above to document the PDS are useful for small project teams as it makes the compilation of the technical specifications slightly simpler, as there are less people involved in the process. The method outlined below is greatly successful when applied to multifunctional multidisciplinary teams. The author has found (and shown in Part B) that some of the techniques allied to the process below, could also be used quite successfully in smaller teams or lone design efforts.

2.4. The Quality Function Deployment

It is in this light that the Quality Function Deployment (QFD) is used worldwide as a tool to generate engineering specifications. It is fast becoming the most popular tool used for design process teams. Surveys show that poor problem and specification definition is a factor in 80% of all time to market delays. In addition to this sentiment, getting a product to market later than the competitors can sometimes be more costly to a company than being over cost or having less than optimal performance.

The following is some factual information from a US survey of 150 companies. 69% of these companies used the QFD method. (71 % have begun during the nineties). 83% said that the use of the method improved customer satisfaction, 76% felt that it facilitated rational decision making. It is well documented, according to the Engineering Management Journal, that development time can be reduced by 50% and start up and engineering costs by 30%.

The QFD automatically documents this stage of the design process. The house of quality, Figure 2.4. serves as a design record and makes for an excellent comprehensive and annotated communication tool. A complex problem can be decomposed into simpler sub-problems for which a house can be constructed for each sub-problem. The QFD is a working document and since it is a people system the starting point is the voice of the customer.

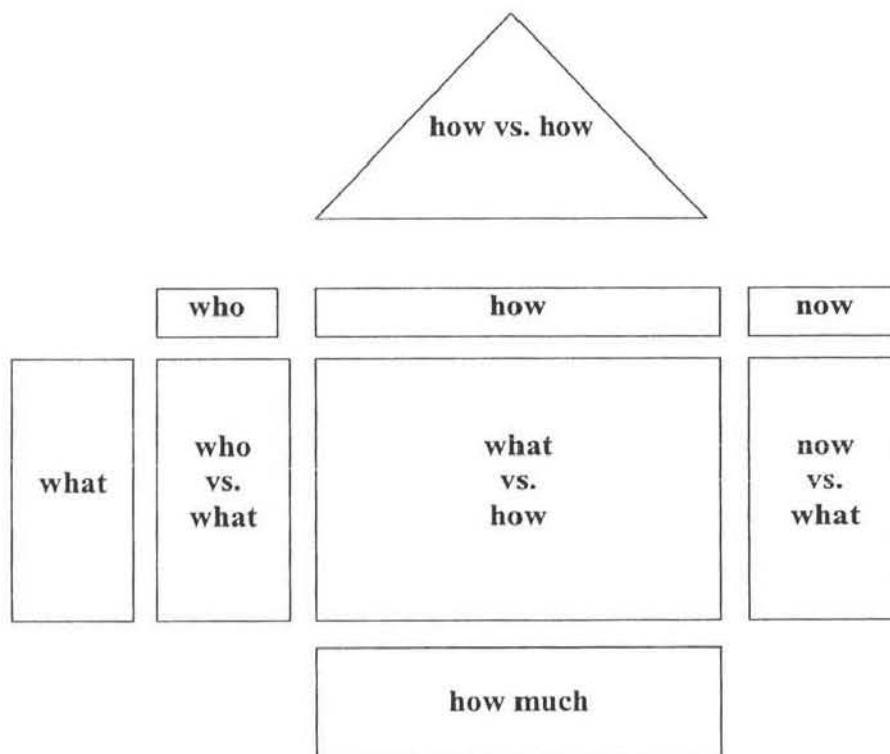


Figure 2.4 The QFD house of quality

Figure 2.4. can be explained by outlining several steps regarding the employment of this tool.

Step 1 : Identify the customers : Who are they ? (the 'who' box - Figure 2.4.)

The goal in understanding the design problem is to translate customer requirements into a technical description of what needs to be designed. The customer for the purposes of compiling the QFD may be regarded as anyone in contact with the device from conception through to operation as intended.

Step 2 : Determine Customer Requirements : What do they want ? (the 'what' box - Figure 2.4.)

Time magazine conducted a survey regarding consumer desires and published it in the November 1989 issue. The results of the survey appear in Table 2.2.

Table 2.2. Results of a consumer survey on product quality

What ?	Essential	Not Essential	Unsure
Works as it should	98	1	1
Lasts a long Time	95	3	2
Easy to maintain	93	6	1
Looks Attractive	58	39	3
Incorporates latest technology	57	39	4
Has many Features	48	47	5

Customer requirements can be broadly listed as :

Functional Performance

Human factors

Physical Requirements

Reliability

Time Requirements

Cost Requirements

Standards

Environmental Concerns

Manufacturing/ Assembly Requirements

Step 3 : Generating Quantitative Values of the Requirements : Who Vs. What (the 'who vs. what' box - Figure 2.4)

The generation of a weighting factor, will give a qualitative indication of how much time and effort to invest for a certain requirement.

Step 4 : Is the Competition Satisfying the customer (the 'now (competition) vs. what')

For each of the customer requirements the existing designs are assigned a number as follows

1. The design does not meet the requirement at all
2. The design meets the requirement slightly
3. The design meets the requirement somewhat
4. The design meets the requirement mostly
5. The design meets the requirement completely

Step 5 : Generate Engineering Specifications : How will the requirements be met?
(The 'how' box - Figure 2.4.)

These specifications are the restatement of the design problem in terms of parameters that can be measured and have target values.

Step 6 : Relate the Customer Requirements to Engineering Specifications :

How vs. What (the 'how vs. what' box - Figure 2.4.)

The following numerical values may be adopted to give the required relationship.

3	-	strong relationship
2	-	medium relationship
1	-	weak relationship
Blank	-	no relationship

Step 7 : Relate Engineering Specifications to each other (the 'how vs. how' triangle -Figure 2.4.)

This step also makes use of the 3-2-1 number system as outlined above

Step 8 : Set target Values (The 'how much' box - Figure 2.4.)

These values define an ideal product and must be based on what is tangible.

Chapter 2

Part B

The Timber Handling Problem Analysis

2.5. Identifying the Need

As a prelude to the needs identification a brief outline of mining is provided highlighting the use of support systems. The author is of the opinion that an understanding of basic concepts behind mining is imperative to the understanding of subsequent sections.

2.5.1. Preamble : Mining at a glance

Mankind has found uses for minerals found in the earth since prehistoric times. Materials like flint, for making fire, and pigment-bearing minerals like ochre and manganese ore were probably the first substances that people mined. The common names of archaeological periods like the Bronze Age and the Iron Age indicate significant use of those metals.

2.5.1.1. Locating Minerals in the Earth

Modern mining is a costly and complicated business. It begins with the locating of probable mineral veins that can produce enough of the desired substance to justify the high cost of extraction. Prospecting and exploration require a vast body of knowledge in the earth sciences to find likely mining locations. Geological evidence relates the age of sample rocks to the long-term mineralization processes of the Earth's crust. Geochemical evaluations are made as well as seismic tests,

magnetic analyses, and electrical studies to determine the geophysical character of a section of earth.

2.5.1.2. The Layout and Structure of a Mine

Once the approximate location and size of a vein or deposit are determined, mining engineers decide the best way to mine it. The familiar type of mine opening directly into the base of a hill or mountain on a horizontal plane is called an adit or crosscut tunnel. A straight vertical shaft may be drilled to one side of a large deposit, with horizontal tunnels running at periodic levels into the deposit. The cavernous openings where minerals are excavated are called stopes. Many times a vein of ore will run from the surface, where it appears as an outcrop, deep into the ground on a diagonal. An inclined shaft runs parallel to the vein with horizontal levels connected to the vein at intervals. Ore that is found at the surface is mined from an open pit or from long strip excavations in a process called strip mining.

2.5.1.3. The Process of Mine Excavation and Mineral Extraction

Excavating a mine and extracting mineral substances involves different combinations of drilling, blasting, hoisting, and hauling. Drilling a shaft itself may involve using a large circular bit that has a series of grinding wheels for cutting into rock. Drills are also commonly used for placing explosives. Drag-bit rotary drills use an abrasive, shaving action from diamond blades or steel shot, and the percussion drill pounds at the rock with a tungsten cutting edge that rotates at the same time. By carefully setting explosives in drilled holes, mining engineers can now use blasting agents of ammonium nitrate, which is effective at 40 percent the strength of dynamite. In some hard rocks, heat applied through a flame-jet or jet-piercing drill causes the rock surface to chip or spall.

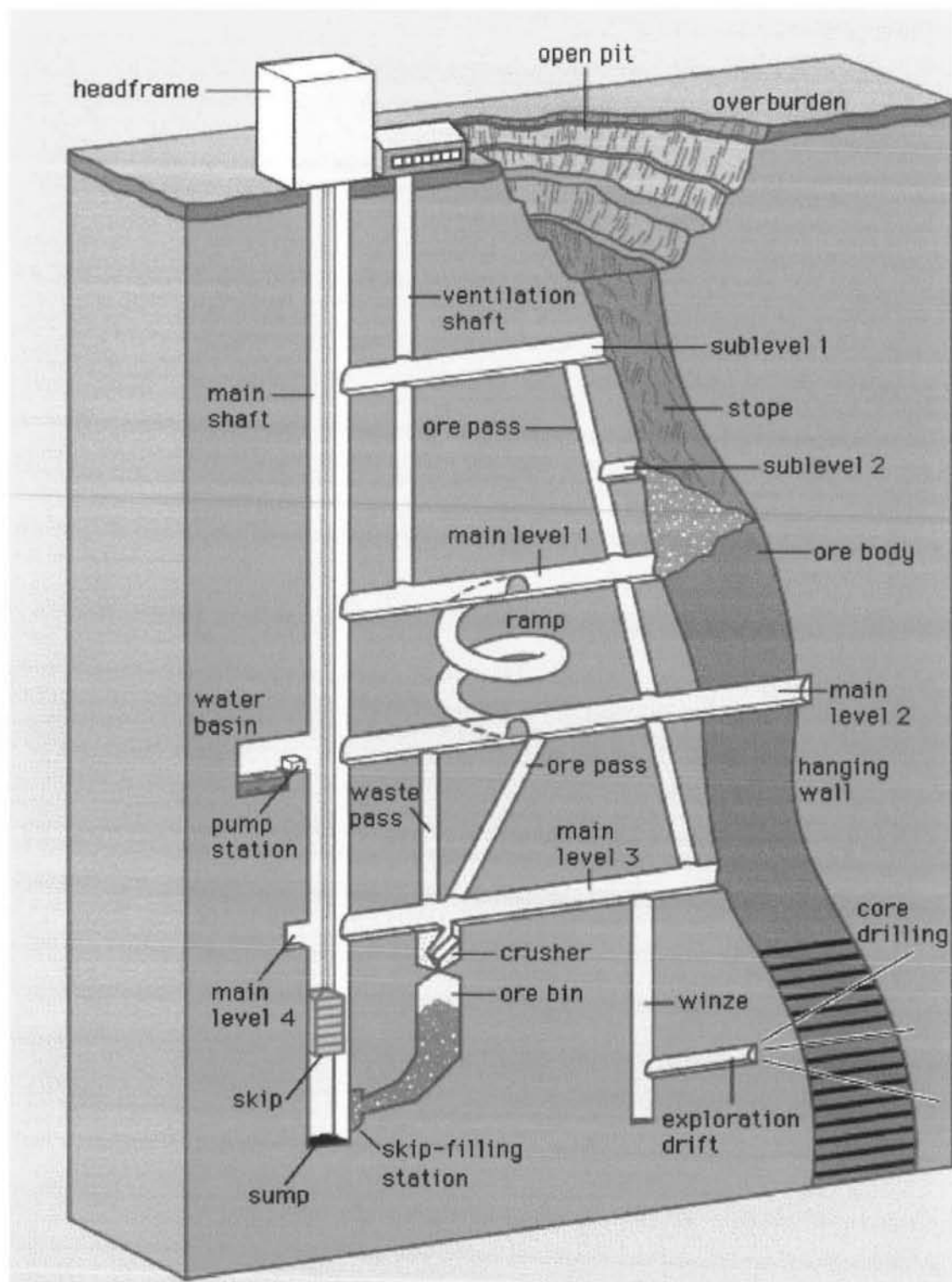


Figure 2.5. A hypothetical mine layout

2.5.1.4. Support Systems Employed in a Mining Environment

Careful consideration is given to the nature of the excavated sections to determine what support is needed in the mine. In some, the ceiling on an inclined shaft may not need support because of the lower ground weight from above in such a shaft. The ceilings of open stopes, the large cavities dug in the middle of deposits, may only need support from pillars of ore or waste rock. Rock ceilings with faults and fissures may be covered with shotcrete, a sprayed concrete that sets in and reinforces the cracks. Elsewhere, steel rods may be imbedded in the rock with large nuts threaded on the exposed end to hold up weakened segments of the ceiling.

Where the span of the ceiling and the ground weight above dictate, structural support is provided. This may range from a variety of timber supports to collapsible steel arches. Timber arrangements include simple beams, or stulls, with diagonal supports into the side walls and a wood ceiling; an inclined beam against the hanging wall and wedged against the opposing foot wall with another beam; a four-piece set that consists of a complete beam frame (top, bottom, and sides); the timber square set linking frames together; and a post that supports the ceiling. In loose ground, ceiling sections are short and built at angles, and at the digging face a top section of planking called the breast boards keeps the earth from sliding backward. Supported stopes like these move slowly and produce a lower volume of ore. In unsupported stopes a number of methods help speed up production. These include slicing broad sections from the stope ceiling and also allowing large sections to cave in.

2.5.1.5. Horizontal Transport

Removal of minerals from mines usually depends on trains of steel boxes drawn along tracks by an electric locomotive; vehicles running on treads or rubber

wheels are also used. In some mines, large hoe-like scrapers pull rubble out by cable. Hand carrying, or mucking, is still practiced where mechanical means cannot be used. The author also completed a comprehensive study involving horizontal and vertical transport systems as a sub- project in conjunction with the CSIR and Anglo American Technical Division.

2.5.1.6. Mine Ventilation

A vital consideration in mining is how to ventilate the underground tunnels and caverns, not only to provide fresh air to miners but also to disperse harmful gases. Some mines become unbearably hot and need cooling. Sometimes ventilation ducts arranged in certain combinations can create natural drafting of air; usually, powerful fans are necessary.

2.6. The Task

Suggest a Solution To the Problem of Inefficient Handling of Timber in an Underground Mining Environment.

Taking into account that the dissertation is part of the Deepmine initiative, which is characterized by its endeavor to develop guidelines and practices for the *Ideal Mine of the Future* designed for a depth of 5000m, the discussion that will follow has been provided for the purpose of answering the following questions :

- *Where did it come from ?*
- *Why is it deemed to be a need ?*
- *Whose need is it ?*
- *When does the need have to be satisfied ?*
- *For how long will it be desired ?*

- Does it affect other needs of society and possibly the environment ?
- What were the observations that inspired the expression of the need ?

The Council for Scientific and Industrial Research have identified impediments regarding the present system of handling timber used for the hanging wall (roof) support. This need is based on actual observations (outlined below) of the present handling system. It is also based on interviews with various mine managers and miners who are involved directly with the handling.

Timber is brought to the relevant horizontal tunnel via a vertical shaft and the horizontal transport of the timber along the tunnel is by flat car (see Figure 2.6 and Figure 2.7). A number of flat cars are joined to form a train driven by the appropriate locomotive.

The timber is stored in timber bays. Timber bays are 'rooms' used for the temporary storage of the timber until they are utilized for the aforementioned purpose. These bays are formed by blasting a cavity into the sidewall of the tunnel. The length of the *Future Mine* timber bay is assumed to be 20 m and the breadth 3 to 3.2 m, the depth of the bay is the same as the depth of the crosscut which is 3.5 m. The tunnel breadth is 6 m. Figure 2.8. is a representation of this description.

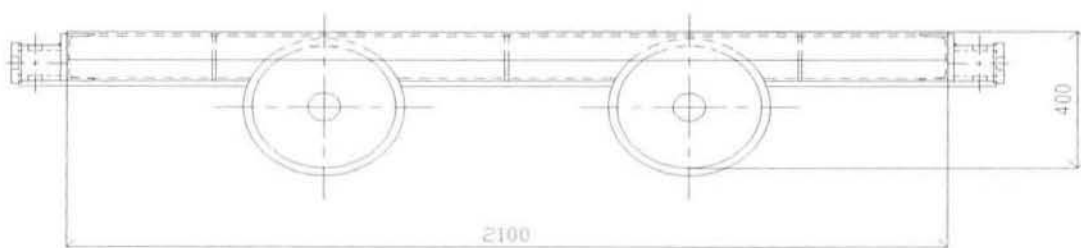


Figure 2.6. The side elevation of a flat car

Timber is supplied by Sappi, Mondi, and HS & L in quantities that are determined by mining engineer to suit the needs of a particular mine. The timber is manufactured as unit blocks of rectangular / square section and arranged as a pack consisting of a number of these blocks with alternate layers of timber blocks arranged at 90 degrees.

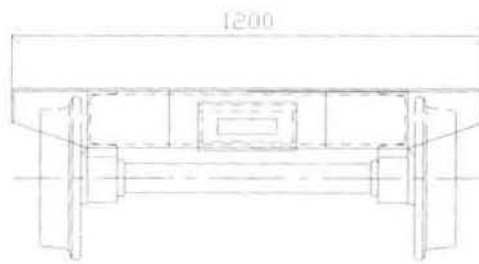


Figure 2.7. The end of the flat car

Whilst these suppliers do prepare packs in standard sizes they are not opposed to preparing the timber pack in sizes specified by a mining engineer as required and determined by the need of his particular mine.

Needless to say it is more time efficient (paper work, delivery time and the like) and cost efficient to order timber in substantial quantities and to store it . An added advantage of this is that the supplier (Sappi, Mondi, and HS & L) would be more susceptible to catering for the specific needs (timber pack sizes and quantity) when the order is substantial as this would make economic sense to them.

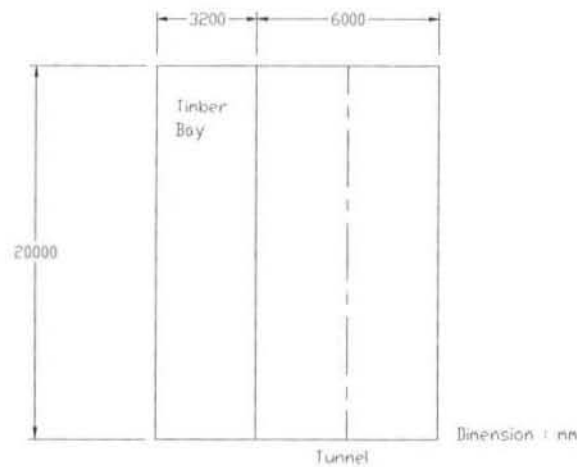


Figure 2.8. Layout in plan view of tunnel and timber bay

These advantages are, however, overshadowed by the disadvantages imposed by present method of offloading the timber from the flat car into the timber bay. Presently, manual labor is employed to offload the timber. For manual labor to offload 12 flat cars (the usual amount), this would entail the following :

If it was assumed for the purpose of this analysis that 1 laborer could lift one block of timber, taking its size and weight into account, he will lift it off the car carry it a minimum distance of 3m to the bay and 'place' it there, he then proceeds to back to the car and repeats the procedure. Presently, this exercise is conducted by 2 (possibly 3) laborers. Even with 3 laborers the process is grossly time consuming. It was further assumed for the purpose of this argument that the train was stationary during the off loading of all 12 cars. This would invariably mean that the track could not be used whilst offloading. Mining operations at the development end would bear the brunt as laborers, materials, rock and equipment / machinery that need to be transported to and from the development ends on a regular basis would have wait until the timber was offloaded.

One could possibly argue that, a quick fix to the solution would be to increase the labor input. This brought with it a host of problems. The transportation of more men down to the tunnel to assist the offloading was one concern, as the skip (mining 'elevator') can hold only a certain number of personnel and only travels at set times. Increase in the labor force increases the heat content of the mine exacerbated by the labor intensive nature of the task which would mean a possible revision of the air cooling system. In addition the Deepmine project is intent upon reducing the total labor force and at the same time increasing the amount of rock that would be mined. This serves only to lend itself to mechanized mining. Taking into account that the task was one of menial labor, no particular skill was involved, and that the added labor input would be needed only when the timber was required to be offloaded (timber is required at development ends in much smaller quantities at any one particular time), there would be no need for the personnel after the required period of offloading.

In addition to the aforementioned points, as a South African Engineer one needs to embrace the African Renaissance, and seek to uplift personnel through skills development and training. Increasing personnel numbers for the aforementioned task is in contradiction to this sentiment. Running parallel to this sentiment is the drive for the South African Mining community to become technically competitive with other mining giants world wide. The optimal use of our inherent primary resources, in South Africa, would be a step in the direction of improving exports and the exchange rate. Therefore some degree of urgency may be attached to the aforementioned drive. In light of these statements, it was desired to design a system that will aid in the handling of the timber and would allow the development of personnel so that it can be operated and or supervised.

The Deepmine Project intends to mine to depths of 5000 m. As a project it is in itself in it's conceptualization stage. The ethos behind the project is to develop optimum solutions regarding the mine layout, material handling, men handling

and ventilation . Development of techniques, methods and personnel is the present stage of the Deepmine project.

The next appropriate step was the scrutiny of the “Primitive Statement”,

Suggest a *Solution* To the *Problem* of *Inefficient Handling* of Timber in an Underground Mining Environment.

The definition of the italicized words were listed and honed down until ambiguity had as far as possible been removed, so as to obtain the definitions as presented in simple english.

Solution : the act or process of solving a problem

Problem : a matter of difficulty

Inefficient : not able to produce an effect without waste of time and/ or energy and/ or cost.

Handling : (i) denoting grasping by hand
(ii) dealing with a part, component, device or problem

Handling : the movement of material from where it is to where it is
(material) needed.

2.7. The Development of Specifications

The author sought to harness some aspects of the QFD method in the compilation of the design specifications.

2.7.1. Who are the Customers ?

This was a fundamental step in the development of the specifications. As mentioned the consumer is the all important driving force for the production consumption cycle. However, it was not uncommon in the past for the marketing department to carry out their required task and simply pass on the report to the technical department for the technical development who in turn would pass it on to the drawing office who would pass the result of their endeavor to the manufacturers without any real interaction between all of these functions. This type of practice has been termed 'over the fence' design and a total embargo has to be placed on its practice. For successful designs the compilation of sound specification lists must address the needs of all parties involved with the project so that ultimately the end consumer will be motivated enough to sustain the production consumption cycle.

With this in mind the customers will be regarded, for the purposes of compiling a specifications list, as all the functions involved in the design process. The functions that were considered to be the customers were :

- The Mine Manager : who would ultimately purchase the product
- The Manufacturer : would include the vendors, the fabricators of components, those involved in assembly (on and off site)

- Marketing : who identified the initial desire for the design and would be responsible for its sale to the mines
- The Operator : who would be involved in the maintenance and operation (the degree of interaction depending on the degree of autonomy)

These were the functions whose needs had to have been satisfied by the product specification so that they would have been able to carry out their individual tasks effectively.

2.7.2. What do the Customers Want ?

In order to ascertain what customers wanted the author dichotomized the needs as the functional performance, the human factors, the physical requirements, the reliability, the life cycle, the costs and the time scales. The following table represents the list of specifications used in the design, together with a qualitative value of its importance.

Table 2.3. The qualitative description of customer needs

Move sizable amount of timber	A	10
Least time to complete movement	B	10
Minimize the operation input (complexity)	C	8
Use minimal number of operators	D	7
Handle load safely	E	10
Make the device visible	F	7
Easy to manufacture	G	8

- Purchased components easily assembled - Assembly and fabrication off site		
Look operable and safe	H	6
Easily maintained	I	8
Ease in transporting to site	J	7
Include fault indication and protection	K	8
Design to be easily disposed of	L	6
Design to be rugged and strong	M	10
Minimize weight	N	9
Corrosion resistant	O	8
Wear resistant	P	8
Work within the specified envelope	Q	9
Design for fatigue	R	9
Use reliable off the shelf components	S	9
Reasonable purchase price	T	6
Minimize capital investment in design	U	6
Minimize design time - development time	V	7
Easily assembled (weld, bolt, etc.) and installed on site	W	8
Easily transported to timber bay from surface	X	7
Reasonable costs to - install - operate - maintain	Y	7
Minimize interruption of other mining activities	Z	8

The word reasonable was used to describe the costs above as the Deepmine Project endeavors to be more oppex (operation) sensitive than cappex (capital) sensitive. This sentiment was due to the serious capital repercussions that are presently experienced as a result of projects of yesteryear being cappex sensitive at the expense of its operation.

Each of the customer requirements were rated as to how important the requirement was to that particular function (customer). A 3-2-1-0 system was used to quantify this, with a value of 3 indicating a strong demand. The cumulative effect of each desire is expressed in the rightmost column. The maximum value that could have been ascribed to each specification was (3+3+3+3) 12.

2.7.3. The Engineering Specifications

A glance at the items in the previous table, Table 2.3., would convey that these items were qualitative descriptions of what the product was to do. This was of little use in the design of the system unless quantification of these descriptions were carried out. The following are a list of the measurable parameters that arose from the description.

Table 2.4. The measurable quantities for the specification

1	Load	N
2	Average operating speed	m/s
3	Total operation time	hrs
4	Operator tasks	#
5	Subsystems / Sub-components	#
6	Number of operators	#
7	Time to install on site	hrs
8	Number of steps to install on site	#
9	Labor needed to install on site	#
10	Time to assemble (weld, bolt, etc.) on site	hrs
11	Number of steps to assemble on site	#
12	Labor needed to assemble on site	#
13	Cost to install on site	R

14	Cost of purchased parts	R
15	Cost of fabrication	R
16	Time to manufacture, fabricate (if not purchased)	hrs
17	Number of steps to manufacture, fabricate	#
18	Number of manufacturing, fabricating processes	#
19	Maintenance cost	R
20	Maintenance Time	hrs
21	Number of steps to maintain	#
22	Ductility	% elongation
23	Stress	MPa
24	Corrosion resistance	% Cr /Zn depth
25	Mass	Kg
26	Compactness	m ³
27	Number of cycles	#
28	Height of tunnel	m
29	Width of bay	m
30	Life	yrs
31	Transport	type

As this was very much a working document throughout the design of the system the author has opted not to include the final specifications at this point but to allow the reader to see how the design process allows for the specification evolution. The author sought not to specify all of the items in the initial specification as complete lists were compiled during the conceptual design phases since each concept naturally offered different value ranges, advantages and disadvantages. This served to illustrate that the specification brief was indeed a working document and the design process highly iterative. The following table is a list of the initial specification.

Table 2.5. Initial specifications

Load	Approx. 2000 kg
Average operating speed	The average speed 15 - 25 m/ min
Total operation time	Off loading of Twelve cars in < 2hrs
Operator tasks	Dependent on concept : to be minimum
Subsystems / Sub-components	Dependent on concept : to be minimum
Number of operators	Dependent on concept : to be minimum
Time to install on site	< 5 hrs
Number of steps to install on site	Dependent on concept : to be minimum
Labor needed to install on site	Qualified and trained personnel
Time to assemble (weld, bolt, etc.) on site	< 5 hrs
Number of steps to assemble on site	Dependent on concept : to be minimum
Labor needed to assemble on site	Dependent on concept : to be minimum
Cost to install on site	Operation minimally capex sensitive
Cost of purchased parts	Operation minimally capex sensitive
Cost of fabrication	Operation minimally capex sensitive
Number of steps to manufacture,	Dependant on concept : to be minimum
Number of manufacturing, fabricating processes	Dependent on concept : to be minimum
Maintenance cost	Minimum
Maintenance time	< 1 hr
Number of steps to maintain	Dependent on concept : to be minimum
Ductility	% elongation > 5 %
Stress	Dependent on concept : to be appropriately chosen with suitable safety factors

Corrosion resistance	Galvanized steel work, Non corrosive materials
Mass	½ - 1 ton
Compactness	To facilitate transportation, handling
Number of cycles	~10 ⁶ - 10 ⁷
Height of tunnel	3.5 m
Width of bay	3 m
Life	15 yrs
Transport	Standard transport vehicles to transport to site, able to be transported to bay along tunnel

Chapter 3 - Part A

The Theory and Methods behind Concept Genesis

3.1. Introduction

Conceptual design has different meanings to different people. To some it may represent the description of all subsystems and component parts which go up to make the system as a whole. To others it may be an idea that has been sufficiently developed to evaluate the physical principles that govern its behavior, confirming that the product will satisfy the customer need.

Another way to define conceptual design would be to describe it as an iterative process comprising of a series of generative and evaluative stages which converge to the preferred solution. Whilst there is much merit in each of the above definitions the author is of the opinion that the following simple statement best describes conceptual design :

Conceptual design is the generation of solutions to meet the specified requirements.

3.2. Concept Genesis

It is of paramount importance that as many ideas and concepts as is economically expedient, are generated. An absolute restraint should be placed on accepting the first seemingly promising solution and thereafter proceeding to its design and development. Whilst the possibility that this design could very well be the one chosen for further development, that is exactly what it is - a possibility. In order to

increase the odds against the design failing in the market, designers and or design teams simply cannot afford the luxury of such chances.

McGrath, 1984 states that concepts are most efficiently generated by working individually and then regrouping with other members of a design team in order to evaluate the cumulative efforts. Each concept warrants scrutiny and the advantages weighed against the disadvantages so that the best concept may be selected. Upon having done this should the designer or team decide that the concepts as presented do not satisfy the need sufficiently well, the process may be repeated and new concepts generated. Alternatively, the advantages or strengths of several concepts could be combined giving rise to an amalgamated concept. The latter has much merit as taking into account that different designers could have placed slightly more emphasis on different aspects of the design. In this way ideas and concepts that fulfill as many requirements of the product as possible can be combined into one workable concept.

The author has found that it is good practice to list the diverse techniques available in a table according to the manner in which they assist memory scan. The techniques listed under problem recognition are used most effectively when they follow directly from the problem definition phase as they are focused on the formation of the first layer on top of the foundation (problem definition).

Table 3.1. Conceptualization techniques

Problem Recognition	Idea Simulation	Groups
Attribute Listing	Fertile Words	Brain Storming
Input-Output	Analogy	Synetics
Morphological Analysis	Empathy	Authors Suggestion
	Conceptual Inventory	
	Morphological Analysis	

3.2.1. Problem Recognition

3.2.1.1. Attribute Listing

The attributes of a design are those qualities and or quantities that distinguishes the product from others. By identifying these attributes the designer will empower himself with the ability to focus on a particular aspect of the product. Having stated that, the attribute listing method can be most appropriately diagnosed for projects concerned with product enhancement where the basic concept of the product has already been established and the enhancement is required to eliminate defects or the optimization with respect to product economics, functional operation, material expenditure or clarification and expansion of market potential.

It is quite possible and sometimes an absolute necessity for repetition of the attribute listing procedure until the fundamental features which characterize every feature of the product are known. By doing this the product may be dichotomized into more comprehensible aspects. The attribute listing method forces the designer to expand his horizons and thereby stimulate the creation of alternative ideas.

3.2.1.2. Input - Output Analysis

This technique attempts to erect a bridge to link the system input to the system output with the freedom to chose the means by which the input may be transformed into the output. In other words there is no prepossessed solution to the problem.

As a technique it is was originally formalized for the design and development of products which perform as physical task or function. As mentioned above, the method embraces three abstract aspects of a product viz. *input* (in going power or

force), *output* (corresponding action or force), *specification* (additional information that may govern the nature of the input and output).

The input-output analysis method has been developed considerably since its conception. The philosophy behind the refined technique is that structure follows function. At this point, it is important to understand what the term function means in a design context and to differentiate it from the form of the design. *Function* communicates *what* the product should do and not *how* it is to be done. The *form* of a product will be a disclosure of *how* it will be designed to fulfill *what* it must do.

For a complete understanding of the technique it must be understood that function can also be described as the logical flow of energy, material and or information. Those functions associated with energy flow can be described by both energy type and by the effect that it has on the system. Energy flowing through a system may be stored, transformed, conducted, supplied and dissipated. With this in mind the words used to describe the flow of energy are verbs.

The flow of material may be best described by defining its subdivisions. *Through flow* is a material conserving process, *diverging flow* can be described as dividing the material into two or more components, *converging flow* is an assembly or joining of materials. The functions associated with information flow can embody mechanical, electrical and or fluid signals.

As a technique the input output analysis can be used for the genesis of new concepts and the enhancement of existing ones. To give direction to the utilization of this technique the four steps listed below need to be followed.

Step 1 : *Find the overall function that needs to be accomplished*
- Be aware of the system boundaries

- Remember that energy and materials are conserved
- All objects that interface with the system must be listed
- The information flows may be included in the diagram by asking the question : How will one know if the system is performing?

Step 2 : Create sub function descriptions

- consider what and not how !
- break down the function as finely as possible
- consider the operational sequence
- list alternative functions
- include the function box or function table; the action word (verb); the object on which the verb acts; the known material, energy and information.

Step 3 : Order the sub function

- the flow should follow a logical or temporal order
- redundant functions should be identified and eliminated or combined.

Step 4 : Refine Sub functions

- decipher whether the functions listed above are truly basic functions i.e. in their simplest form
- make sure that all assumptions regarding the product are documented and properly voiced.
- make sure that the function is directed at what and not how unless the how was assumed at the onset
- ensure that the domain of knowledge required of the product is sufficient.

3.2.1.3. Morphological Analysis

This technique entails firstly the listing of the all the parameters involving what the product should do, in other words the listing of the device functions, and secondly the listing of ways and means of fulfilling these required functions. It is important that these said ways and means are of the same level of abstraction as different levels would place this technique under different sub-categories according to which concepts are generated. For example, if the task at hand was to lift an object from point A to point B, then if one offered potential solutions as being fluid power, electrical power, or possibly using a mechanical advantage these items would be on the same level of abstraction falling under the sub-category of *problem recognition*. A hydraulic motor, a hoist or mechanical linkages would be a list of the same abstraction level and would fall under the category of *idea simulation*. The author has found that in practice a listing similar to the latter of the two is a simpler means of visualizing the final device, provided that a valid general and functional abstraction has been carried out.

The morphological chart is presented as a tabular matrix vertically listing the required functions on the extreme left and each one in turn followed by a horizontal listing of the possibilities available in conceptual fulfillment of the product. It follows sequentially from the effort invested in the input output technique, however takes it to the next level by considering the methods available to achieve the required function. A morphological analysis will prompt the mind into considering the total problem at one time.

The value of this technique lies in the fact that numerous solutions to the problems become immediately available with the number of solutions limited only by the number of alternatives available in fulfillment of the function. The draw back of the technique is oddly enough the same as its advantage. The generation of too

many solutions might in fact be bewildering. It takes good engineering insight to decipher the practicality of a possible solution. The designer should ask himself whether or not the combination as put forward can gel together as a working unit. He should also be aware of the implications of gelling such sub systems.

3.2.2. Idea Stimulation

3.2.2.1. Fertile Words

This method entails the listing of one word relevant to the problem and following it up by words that have similar meaning. The method relies on the ability of the human brain to attach an idea to a word or series of words. The ideas however abstract should be documented. Often the most successful designs have been rooted in abstract ideas, after sufficient refinement, of course. The range and number of ideas are limited only by the human imagination and is therefore boundless. This is one of the simplest and most effective techniques available.

3.2.2.2. Analogy

This method is similar to the method outlined in section 3.2.2.1. with the difference that instead of extracting ideas from words with similar connotations, ideas are stimulated by extracting information from previous experience and observation that have bearing on the problem at hand. A common fuel for the analogy technique is often found in other designs whether similar in classification or not to the task at hand. One of the richest sources of the analogy method is nature. There is an immeasurable number of principles, practices, structures and mechanisms in nature.

In order to develop the facility for the application of analogies to engineering design a heightened awareness is desired. That is to say that it is required that the

power of observation and the intensity of interest which enables the designer to recognize and understand the principles governing objects, whether natural or man made, must be brought to the fore. The designers' inquisitive and inquiring ability is required here. His ability to ask why? how? where? what? about the situation he is faced with will form the basis for the inexhaustible reservoir of knowledge that he will possess when he is faced with the opportunity to design and conceptualize.

3.2.2.3. Empathy

This is a method of idea development by considering a personal involvement in the project. By definition empathy is the "power of projecting ones personality into the task". The essential characteristic of the approach is to imagine that one is a part of the device or some phenomenon influencing its development. This method is sometimes useful when one examines the effort that would be invested by a human (thought, motion and control) in carrying out a specific task. The next question that would require answering would be, how can this effort be carried out by a machine or what product would make this task more pleasant and simpler.

3.2.2.4. Conceptual Inventory

The conceptual inventory method as is also known as the **check list** method is used as a means to stimulate creativity. The ethos behind the method is based on the adage that *in order to get the right answers one must ask the right questions*. With this in mind one can define the method as being a means of examining ideas and rearranging thoughts in an effort to expand or optimize the concept. The process is closely allied to the method of **boundary shifting** in which the constraints and parameters as identified in the Product Design Specification are challenged in order to proceed with reasonable surety to the next stage.

The suggested list of questions follows :

- | | |
|--------------------------------|---|
| <i>1. Put into other uses?</i> | In what way can an existing product or material be used differently- either as it is or after modification? |
| <i>2. Adapt?</i> | What other product either man made or of nature satisfies a similar problem? |
| <i>3. Magnify?</i> | What are the consequences if the product was designed to be stronger, higher, faster, with greater frequency, extra value, etc. |
| <i>4. Modify?</i> | Can the shape, color, motion, odor, sound be changed. What are the consequences. |
| <i>5. Minify?</i> | The effort made to limit quantities or parameters possibly to the extent of eliminating them. |
| <i>6. Substitute?</i> | Replacement of parts, processes energy method delivery methods which are able to perform similar tasks. |
| <i>7. Rearrange?</i> | Can one interchange components, temporal orders, transpose cause and effect? |
| <i>8. Reverse?</i> | Can the process, machine, device run backwards, upside down, stop the moving, move the fixed? Implications thereof? |

9. Combine?

In what way can products, devices, portions of concepts be combined?

These are examples of questions that one could ask. The intention is to force the mind into a situation where the memory has a direction in which to act. Once the mind is opened a whole new vista may appear and vast changes in the design concept become evident.

3.2.3. Organized Group Activities

These methods can be applied when there is a group of people gathered to confer, contribute, develop and stimulate ideas. Anyone involved in group activity of any kind will be aware that organization and direction are not desires but absolute musts. Seeing as how the move toward group activity or group orientated product development is rife in industry these methods allow the efficient employing of such activity. The justification behind this drive is that a much greater memory store is available with more than one person and when managed and practiced adequately will prove to be beneficial to the total design process. The following two techniques were developed with the intention to organize the group activity.

3.2.3.1. Brainstorming

A gathering of persons with knowledge about the problem at hand is desired. In the authors opinion it is not necessary that all of persons involved have the facility, training or education to develop the product, as long as they have the insight into the situation, ability to think quickly, open mindedly and without inhibition. At the same time persons with technical and marketing insight are of indispensable value to the team when it comes to the engineering of the ideas into

feasible solutions. There are strict guidelines which should be adhered to ensure success of the method.

- lay down an already sufficiently defined problem
- a total restriction is to be placed on the criticism or passing of judgment on ideas during the ideation phase, as the goal of the group becomes sidetracked if the attacked ideas are ridiculed or have to be defended. Often people are reluctant to express their ideas at the expense of them being scoffed at for them.
- it must be well known that all ideas are welcome and documented. Often wild ideas ignite another thought or line of thought which leads to a successful solution, even though original idea may have been rooted on a tangent to the problem.
- the development of as many ideas as possible is prudent as the number of ideas are proportional to the probability of obtaining ideas warranting further development.
- combination of ideas should be investigated and is encouraged since it is here that wild ideas may prove useful.

3.2.3.2. Syntetics

This technique involves a slightly more rigid agenda as compared to the preceding one. For this technique the members of the group must have particular knowledge and skills in a particular area. The members are invited by the chairperson of the

meeting because they possess these abilities. The method hinges on the fact that at the outset only the chairperson is aware of the problem in its entirety.

The steps governing synetics are :

- the chairman leads a broad general discussion on a topic central to the problem.
- he then provides some guidance and stimulation for further discussion by questioning the statements/ideas and or declaring significant but not problem revealing information.
- when the chairman is satisfied that the discussion has led to or at least close enough to a lucrative solution, he reveals the problem to the team. In this way the many 'loose' ideas generated prior to this will be given a substrate and further discussion will allow a firm rooting of these ideas so that the finalization of the concept can be realized.

3.2.4. Suggestion of the Author : Assessment of Group Activity

The advantages, as mentioned above, of a group effort over that of the lone designer are indeed important, especially in present engineering design environments. However this requires the proper coordination and management to ensure success. It could very well be the case that a group discussion may go round in circles with people trying to voice their opinions and beliefs rather than attempting to direct their thought process toward the attainment of the goal. This could be because in group activity the pressure to speak and be heard could be greater than the pressure to think. This could also be because some people think best when they are alone and purposeful ideas flow more easily when in isolation.

It might very well be that there are some who in fact have a lot to share, but prefer to give it more thought before volunteering the information. Does this mean that these persons should not function at the conceptual design level or at any level requiring group activity? The reality is that there is without a doubt more memory store and ability available in a group of people. The following method, suggested by the author, embraces the advantages of group activity and attempts to bias the intention of group activity towards constructive thought followed by presentation of them to a group.

- having identified and defined the problem the design team manager informs his team of the problem possibly by memorandum or by a formal meeting for which the sole purpose is to present the team with a formal representation of the problem. This should take the form of the PDS.
- at this stage there should be a no questions asked policy adopted as the manager should see to it that all the necessary information is presented in the PDS. It is the onus of the manager and possibly his superior/s to decide on how open to leave the problem.
- he should inform his team that, on a predetermined date, each one should have a document prepared containing many off the cuff ideas or one or two ideas that the member has chosen to expand slightly. This is left to team members' discretion. The author is identifying the possibility that some people may prefer to choose 1 or 2 ideas and develop them to the next level of abstraction and some may prefer to generate many ideas on some lesser level.
- further each member must be prepared to present his document to the team. The team members have the freedom to choose from the list of techniques above as to how they will generate ideas.

- on the day of presentation, each member should have in front of them a copy of the other members document and should make notes while someone is presenting.
- there is to be no verbal criticism of ideas at any point.
- at this point the members may disband until the next meeting or the next step may in fact occur at the same meeting.
- the manager or coordinator must ask each member to select with motivation one concept other than their own, or seek to combine one or more ideas into a new concept. It is also important that if one opts to select an off the cuff idea that they would be obliged to make suggestions for the next abstraction level.
- this again should be completed in documented format and presented to the team. This document should also list the criteria used by the team member in selecting the ideas of the others.
- The onus is then upon the coordinator to draft a selection table containing the ideas selected by the members together with a set of criteria against which they can be further scrutinized.
- this preliminary selection table is presented to the team, where further input is requested for the selection criteria and thereafter each member is asked to complete the selection table.
- the next step would entail the finalizing of the concept by discussion from the results obtained from the selection matrix.

Chapter 3 - Part B

Conceptualization Techniques Applied to the Design of a Timber Handling System

3.3. Introduction

It is important that the designer be well aware of the techniques presented in the theory section of this chapter and with this awareness he may possibly seek to harness certain aspects of some techniques and order the techniques at his discretion for a particular situation. With this in mind the following method has been presented at the discretion of the writer as being appropriate for the design at hand, to carry out conceptual design.

It is important to list at the onset of conceptual design all assumptions that will have a bearing on the conceptualization process. It is reasonable to make assumptions provided that there is an awareness that the assumption is being made.

3.4. Product Function

3.4.1. What should the device do ?

Simply put the device had to move timber from a flat car into the timber bay. The initial assumptions were that the timber arrives at the timber bay on a rail driven flat car , the dimensions of the tunnel and bay were as stated in chapter two. A further assumption was that the timber will be moved as a unit comprising of many blocks. This is often known as the unit load principle of material handling

and is considered good practice when items have to be moved in large quantities in short time spaces. The use of a pallet for the handling would seem appropriate given the block nature of the constituent material being moved.

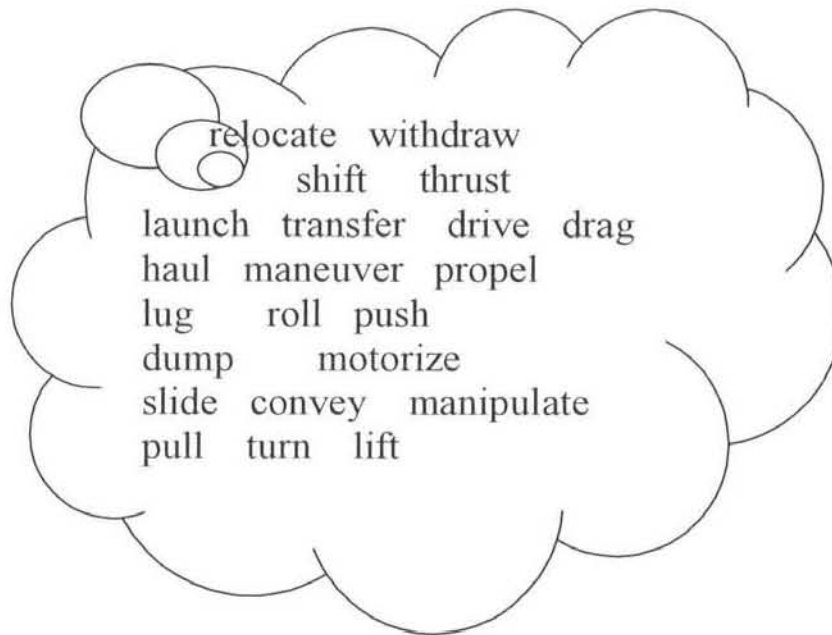


Figure 3.1. Application of the fertile word technique

Upon examination of this simple statement the verb, move, in the statement deserved scrutiny. A list of allied words was compiled and appears in Figure 3.1. Reference is made to the fertile word technique. The next step was the compilation of various scenarios on a somewhat superficial level that were identified as being potential solutions to the problem. These scenarios attempted to embrace the meanings of some of the words in Figure 3.1.

3.5. Subsequent Attempts at Conceptualization

This section is presented only for the purpose of illustrating the conceptual design process. The true nature of the iterative process of conceptualization for the sake of brevity could not adequately presented, as the ideas, however feasible or unfeasible, were indeed numerous. The scenarios presented here are in fact second and third stage iterations and may list more than one idea that were grouped and in some instances modified to be incorporated into a single scenario.

3.5.1. Scenario One

Have a rolling mechanism fitted onto the flat car or possibly use a detachable rolling system that could be placed on the flat car. The system may have been manually, electrically, mechanically or fluid actuated. It could have had a braking system incorporated. The system could have been automated. There would have been a roller system in the timber bay so that the palletized timber could be moved along in the bay. Some modification could have been made to the path of the flat car rail so that it passed closely enough to the bay to allow for the transfer of the pallet from the rollers on the cart to the rollers in the bay.

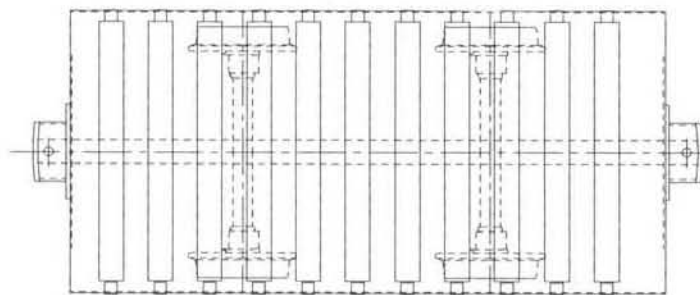


Figure 3.2. Flat car shown in plan view with rollers

3.5.2. Scenario Two

Set up a gantry system to work within the specified envelope. Have some sort of manipulator to raise and support the load off the cart and place it in the bay. Various options of control and degree of autonomy would be available for the purpose. Cartesian path options would be x-only, x-y, or x-y-z. This would include gantry cranes, single and double girder options for cranes on runways. Options for gantry systems range from runway cranes, overhead gantry systems and gantry cranes. An overhead runway x-y-z gantry is shown below.

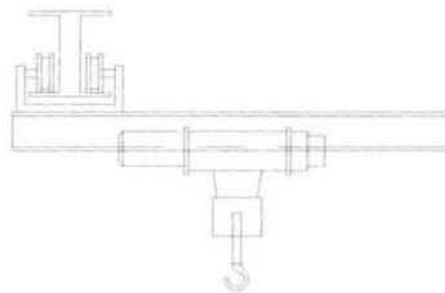


Figure 3.3. An overhead gantry - one of the gantry options

3.5.3. Scenario Three

Have a custom designed mining low lift ‘truck’ to raise the loaded pallet sufficiently off the cart, transport it horizontally to the timber bay and place it. This could have manifested itself as a manually operated and propelled pallet ‘truck’ that would possibly incorporate some means of lifting the large load. The alternative being a motorized version equipped to raise the load with fluid or electrical power. The power supply, for lifting and traversing, could have been on

board the truck. This truck would have had to be steered appropriately and could have been designed to be either walk along or rider operated.

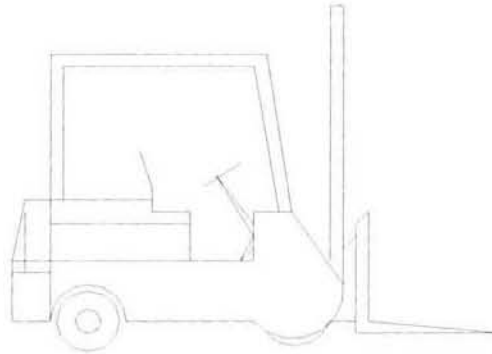


Figure 3.4. A driven pallet car

3.5.4. Scenario Four

Used the idea in scenario one to alter the path of the track so that it passed the bay very closely. The car could be modified to dump the load into the bay by either elevating the entire car or a platform on the car to some angle so that the load can slide off or role off the car into the bay.

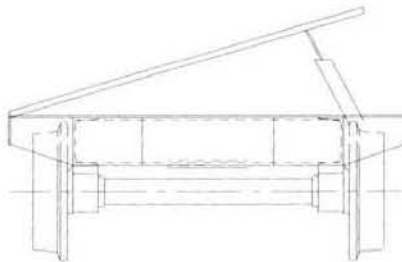


Figure 3.5. The elevating platform car

3.5.5. Scenario Five

Used the idea in scenario one which made use of rollers. This scenario made use of a conveyor concept. However, instead of changing the path of the track, it could have been designed such that the conveyor reaches the track. The timber bay would have had a conveyor system so that, from the time the pallets leave the flat car on arrival, through storage in the bay, and finally dispatch back onto a car, they would be on a system of rollers. The system could have been automated or driven by an operator. In addition, for flexibility the conveyor could incorporate a track switch so that some pallets could have been diverted to a secondary line.

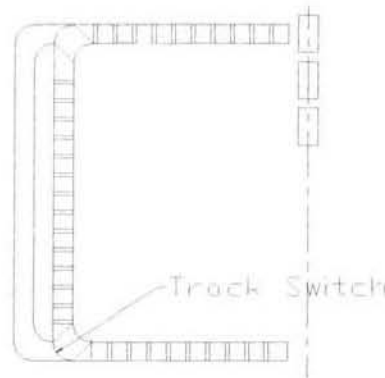


Figure 3.6. The foot wall conveyor

3.5.6. Scenario Six

Allied to scenario five and scenario two an overhead conveyor system could have been implemented with some device equipped for lifting and supporting the load traveling along the path. The supply could have employed electric, fluid or mechanical power. This could consist of a series of trolleys or wheels supported from or within an overhead track and connected by an endless propelling means-possibly chain, cable or some other linkages.

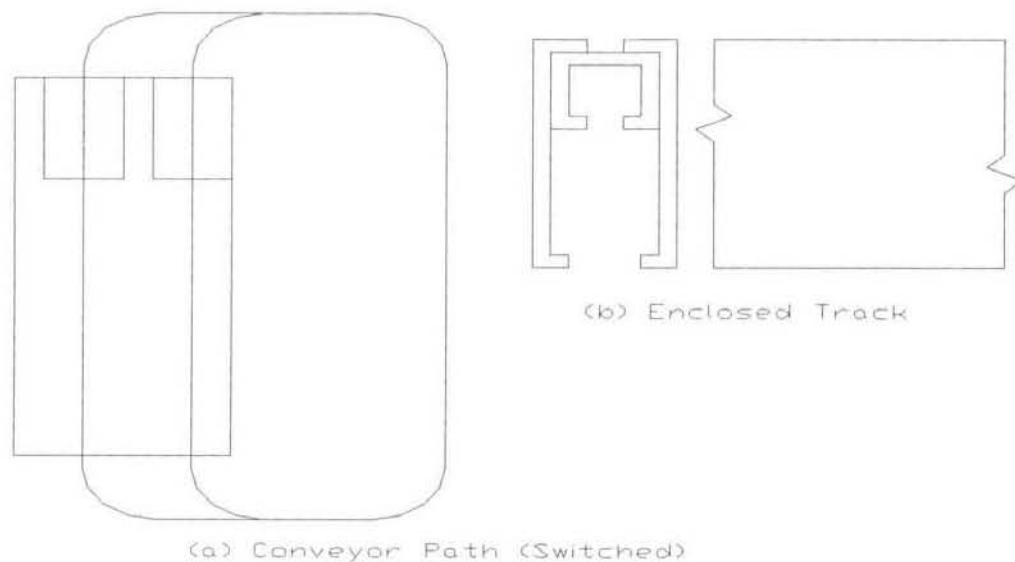


Figure 3.7. Overhead conveyor system

3.5.7. Scenario Seven

Similar to the pallet truck idea above, this idea embraces the capability of the truck to be designed for path guidance on the floor. The degree of automation could have been pronounced in the design and would fall under the sub-category of guided pallet trucks of the automatic guided vehicle family. As an implication the vehicle would be driverless. Guidance for the vehicle could have been effected electrically or magnetically or possibly optically.

3.5.8. Scenario Eight

Would have consisted of a pivoted post and a carrying boom, on which some device for lifting and supporting the load could travel. The post could have been attached to a column so that it could swing through approximately 270° . The

dimensions of the bay would have possibly dictated the need for two such assemblies.

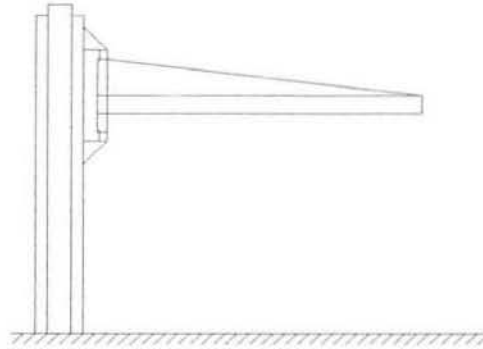


Figure 3.8. The cantilever arm concept (jib crane)

3.6. Scenario Evaluation

The scenarios above were evaluated using a tabular matrix. This table is essentially a series of criteria against which the concepts are scored. The tabular matrix is a structured technique of evaluation as the significance of the criteria is numerically weighted and the most suitable concept is chosen as the one that has scored the highest. This only served as a pivot point for the conceptual design process as the concept that was chosen underwent further conceptual design as is evident in sections to follow. The scenarios were highlighted under the broad headings of functional performance (operation), human factors, physical requirements, life cycle, the costs, and handling (assembly, installation, implementation). The values were obtained by multiplying the rated score (a score out of ten) by the percentage value.

Table 3.2. The concept selection matrix

		S1	S2	S3	S4	S5	S6	S7	S8
Criteria	Wt %	score	score	score	score	score	score	score	score
functional	22	6	8	7	4	6	7	6	5
human	17	5	6	5	5	5	6	6	6
physical	18	5	7	7	5	5	7	5	4
life cycle	12	5	6	3	6	5	6	4	6
cost	12	3	5	4	4	5	3	4	5
handling	19	2	7	6	3	6	5	5	6
score		4.41	6.69	5.63	4.40	5.41	5.85	5.15	5.30
rank			1	3			2		

The criteria listed in Table 3.2. were weighted according to the initial specification. Each of the options were evaluated and the merit of each, with respect to the criteria listed, was given a numerical value. A final score for a particular option was obtained by summing the individual products of the *criteria weighted percentage* and the corresponding *numerical rating* of the option.

3.7. Streamlining the Best Concept

3.7.1. What should the Gantry System do?

The Gantry should move the largest amount of timber possible off a flat car and into the timber bay in the shortest space of time.

1. **Locate** the correct *timber pallet* on the *cart*
2. **Move** to the *pallet*
3. **Stop** the at an appropriate position.
4. **Actuate** to hold *pallet*
5. **Secure** *pallet*
6. **Clear** *pallet* off *cart*
7. **Stop** at adequate clearance
8. **Locate** position in *timber bay*
9. **Move** *pallet* to this position
10. **Stop** at an appropriate position
11. **Actuate** to release *pallet*
12. **Clear** *pallet*
1. **Locate** the next correct *timber pallet* on the *cart*

3.7.2. Functional Input Output Analysis

The following table, Table 3.3. is provided in lieu of providing a structured effort for functional decomposition. Even though there might be tendency to scoff at it's simplicity, there is indeed merit in stating and documenting what may seem obvious so that the conceptualization to follow will have not only a solid foundation but a strong sense of direction. The author has opted to present this

initial stage of the conceptualization process in tabulated form as opposed to presenting the functions in block diagram form.

Table 3.3. Functional decomposition of tasks

Verb	Energy Input	Energy Output	Material Input	Material Output	Infor- mation Input	Infor- mation Output
locate	optical, electrical	optical, electrical	timber	timber	pallet located ?	pallet located !
move	electrical, mechanical, fluid	mechanical	device	device	position obtained ?	position obtained !
stop	electrical, mechanical	mechanical	device	device	stopped ?	stopped !
act- uate	electrical, mechanical, fluid	mechanical	device	device	actuated ?	actuated !
secure	electrical, mechanical, fluid	mechanical	timber, pallet, device	timber, pallet, device	secured ?	secured !
clear / lift	electrical, mechanical, fluid	mechanical	timber, pallet, device	timber, pallet, device	cleared ?	cleared !

3.7.3. Scenarios for Gantry Systems

The scenarios presented in this section are of level two possibly level three abstraction and have followed from an initial inspection and evaluation of level one abstractions. To reiterate, a level one abstraction are the physical laws and or phenomenon governing a task. This is very general. Level two abstraction would be the developed or tangible components available for the realization of a level one abstraction. Level three would describe a specific component or device type. It is however possible for the levels to overlap into each other in some cases.

Table 3.4. Level two morphological chart

Support	Track	Wheels	Air cushion	Slides	Pedi-pulator	Beam
Propulsion	Driven wheels	Air thrust	Moving cable	Linear induction	Hand Chain	
Power	Electric	Diesel or Petrol	Hydraulic	Bottled Gas	Steam	Manual force
Transmission	Belts	Chains	Gears and shafts	Hydraulic	Flexible cable	
Stopping	Brakes	Reverse Thrust	Ratchet			
Lifting	Hydraulic Ram	Pneumatic Ram	Rack and Pinion	Chain Or Rope Hoist	Linkage	
Locate	Photo Sensor	Proximity Switch	Human Vision			

Using Table 3.4. above together design handbooks the following gantry concepts have been presented.

3.7.3.1. Single Girder Overhead Runway Cranes

This consisted of an I-beam supported by four wheels attached to a carriage traveling on a runway. The trolley traveling on the lower flanges carries the electric chain hoist, forming the lifting unit. The crane could have been moved by hand chain turning a sprocket wheel, which is keyed to a shaft. The pinions on the shaft could be designed to mesh with gears, keyed to the axles of two wheels. An under slung construction could have also been used with the pairs of wheels at each corner which riding on the lower flange of the I-beam rails as shown above. One wheel axle on each carriage could be coupled directly to a shaft which transmits power from a gear reducer. Or each of the two sets of wheels on opposite ends of the girders could be driven by a separate motor accompanied by the appropriate gearboxes and braking systems. Figure 3.9 below shows a crane girder and traveling unit that travels on and is supported by an erected runway. The traveling unit concept is depicted in Figure 3.10.

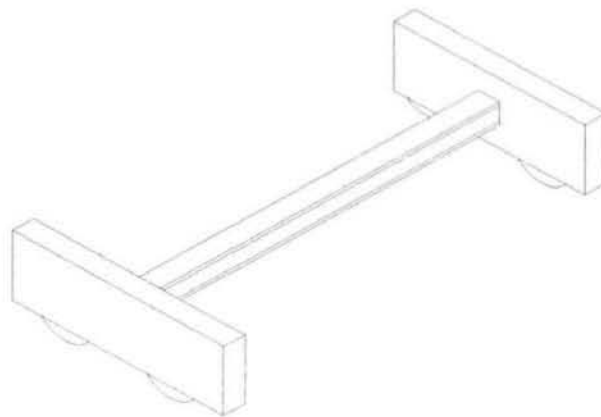


Figure 3.9. Single girder runway crane

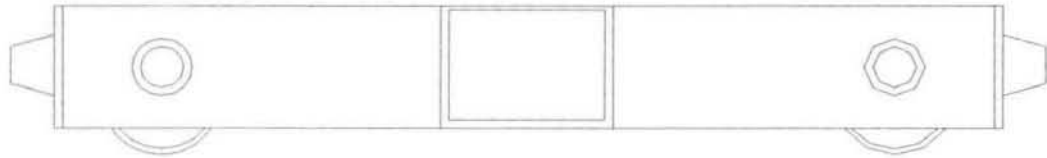


Figure 3.10. The traveling unit concept

3.7.3.2. Double Girder Cranes

This scenario consisted of two bridge girders, as opposed to one in Figure 3.9, traveling on parallel runways, on top of which were rails on which the self contained hoisting unit, called the trolley, traveled, see Figure 3.11. The girders were supported at the ends by trucks (carriages) with two wheels. The crane would be designed to move along the track by a motor, through shafting and gearing to the truck wheels. The bridge girders could be of the I-beam or box section, with the latter being employed to give torsional and lateral stiffness for long spans.

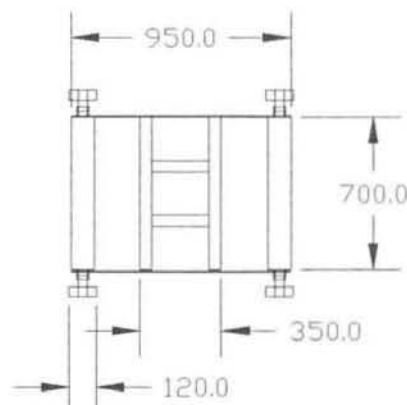


Figure 3.11. The trolley for a double girder gantry
(standard dimensions-mm)

The girders were designed for rigid attachment to the carriage, which carried/housed double flanged wheels for supporting the bridge. As a measure of safety the girders would project over the rail so that in the case of a broken wheel or axle, the girder would rest on the rail. The wheels could be driven as outlined in the preceding section.

The trolley would consist of a frame which carried the hoisting machinery and is supported on wheels for travel along the bridge the bridge rails. The wheels are coupled to the trolley traverse motor through suitable gear reduction. The hoisting machinery would consist of a motor, motor brake, load brake, gear reduction, and rope drum.

Wire rope winding in helical grooves on the drum could be designed to be reeved over sheaves in the upper block and lower hook block for additional mechanical advantage. Limit switches could be provided to stop the motor when the limits of travel are reached.

3.7.3.3. The electrics involved for the Double and Single girders

Current could be brought to the crane by sliding or rolling collectors in contact with the conductors attached to or running parallel to the runway. Current to the trolley could be effected in a similar manner. Festooned multiconductor cables were also available as an option for the transfer of current.

The motors, either alternating or direct current, used for the application would be those intended and designed for crane application. Direct current motors would probably be of the series wound type and alternating current could be of the wound rotor or squirrel cage type.

3.7.3.4. Gantry Cranes

This scenario was a modification of the traveling cranes of the single and double girder type. It could find application where there would be difficulty in erecting an overhead runway. The bridge/s would be carried at the ends by legs, supported by carriages on wheels at either end so that the crane could traverse. As with options above the hoisting unit would be attached to the lower flange in the case of a single girder and attached to a trolley in the case of a double girder. The crane could be driven by a motor through gear reduction to shaft, which drives vertical shafts through bevel gears. Bevel and spur gear reductions connect the axles of the wheels with the vertical shafts. As an alternative, the crane could have been built without the cross shaft, using separate motors, brakes and gear reducers at each end of the crane. The carriage travel unit concept for the gantry crane was essentially the same as that depicted in Figure 3.10.

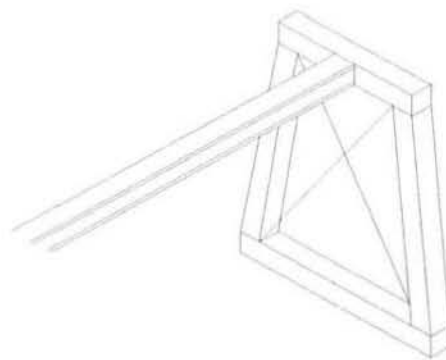


Figure 3.12. The gantry crane

FEM analysis, preliminary calculations and consulting with material suppliers and catalogues revealed firstly that, there was no need for a double girder set up, as the load was under 3 t. Secondly that the deflection was directly proportional to the length of the girder. The runway crane girder Figure 3.9 extended from the timber

bay right across the tunnel to the opposite side. There was little room for experimentation except perhaps to consider the use of stiffeners to decrease the deflection. As mentioned the gantry crane in Figure 3.12 was to be implemented when it became inconvenient to erect an overhead runway.

Whilst it would not be entirely impossible to do design and erect a runway on the side wall, one would have a host of constraints to contend with. Amongst which were the vibration and stresses associated with rock masses. The gantry crane had the flexibility to have it's length modified as it could have been designed to extend into the tunnel by a distance fitting for it's purpose. A track could be layed alongside the main track to facilitate the operation of this structure. Therefore the gantry crane was chosen for development.

3.8. Handling the pallet

The options selected and discussed under the single girder gantry system were chosen for further development and design. The single girder handling system employed a lift unit and a special purpose handling unit. The question that was addressed was how to attach / handle the pallet and timber on it.

The words *grab, grip, fork, clasp, clutch, clench, seize* were words that were used to generate the concept for a device that would have been able to operate with a minimum amount of operator input. The design would have had to provide for the attachment to the lift unit. It would have to have made provision for the clasping movement desired and perform the tasks outlined already within specified constraints.

The aforementioned functional input output analysis and the morphological charts allowed for the generation of the following general concepts of a gantry grab unit

and a fork lift unit both of which had manifested within them a number of possibilities.

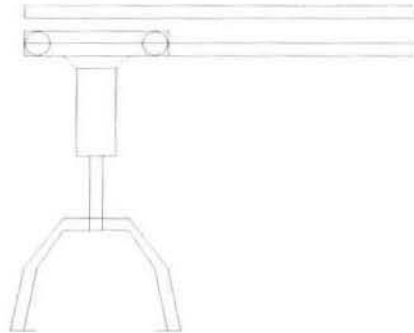


Figure 3.13. A grab unit concept



Figure 3.14 A fork lift concept

The general grab unit concept maximized the use of the two degree of freedom gantry system with its addition of the a third degree in the vertical direction. The fork lift system required a rotational degree. That is to say, as soon as the pallet had been forked, the loaded fork had to rotate by 180° so that the pallet could have been delivered to the bay. From a technical point of view, preliminary calculations revealed that the moment associated with the fork at critical points were undesirable. Whilst this design could have been made practicable its design would

have been complicated by trolley acceleration, misalignment of wheels and skewing. It was considered that the grab if properly designed would transmit a uniform dynamic (moving) load to the trolley moving along the girder and therefore to the girder.

The author, after much consideration, then listed the following blue print, reference is made to Figure 3.15. The **arms** would be those members that would make direct contact with the pallet. They had to be designed adequately for strength and form. The optimization of the strength to weight ratio was also an important factor. Functionality, fabrication, assembly and cost were key criteria.

The member onto which the arms were fixed was the **shaft**. The shaft would have had to function in the support of the arm and the actuation of the arms to effect the grabbing action. The shaft had to be designed for weight, strength, functionality, displacement, assembly and fabrication.

The **chassis**, was that structural member which would house the two shafts, with two arms on either side. The chassis also had to provide for the attachment to lifting unit and attachment of the actuating mechanism. Its design for functionality, fabrication and assembly were of great importance.

The actuation of the arms was a key design aspect and is treated in Chapter 5. The lifting unit utilized was, after much consultation with suppliers of industrial cranes, to be an electric rope hoist for actuation in the vertical direction. The selection of the vertical actuating unit is also treated in Chapter 5.

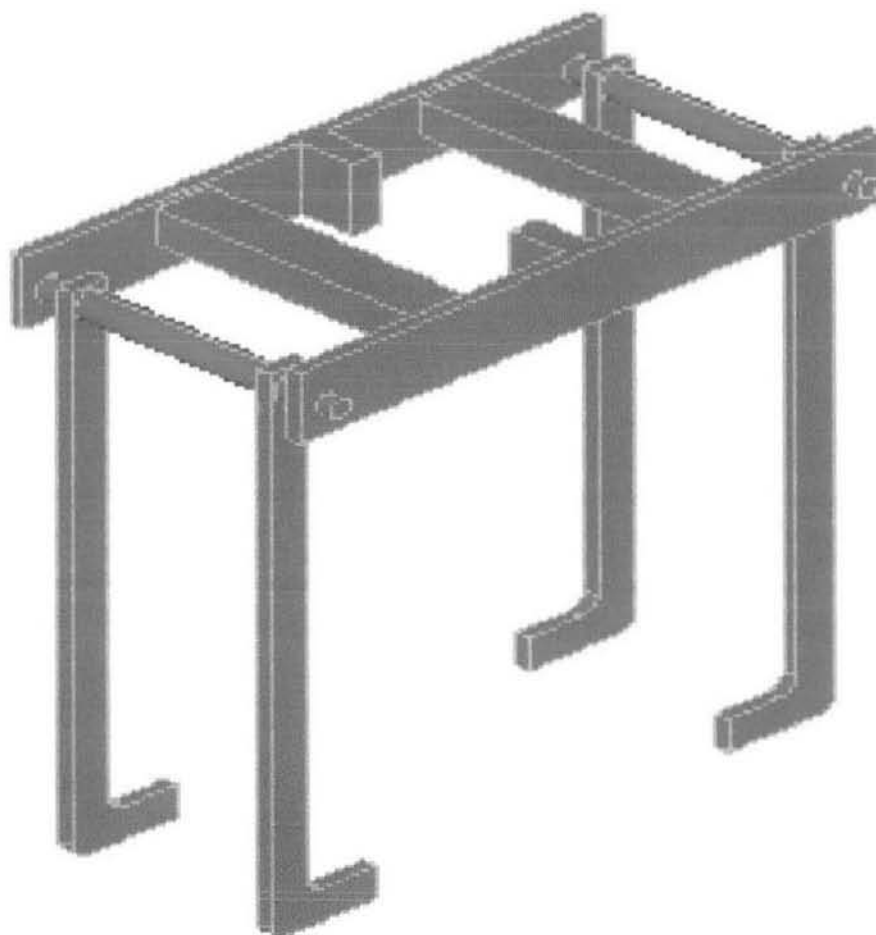


Figure 3.15. The 4 Arm Gripper

Chapter 4

Aspects of Design Theory and Elements of Mechanical Design

4.1. Overview of Structural Computer Modeling

4.1.1. The Finite Element Technique

The computer modeling technique most often used for the analysis of structures is that of finite element analysis. In this technique the geometry of a structure is defined by a series of elements, the shape of which is determined by the form of the structure. For each condition of loading - for example static, dynamic or thermal - the behavior of each of the individual elements can be defined by a set of simultaneous differential equations. These equations are solved by the computer using a series of matrix operations through which the behavior of all the individual elements are combined to give deformations and stresses within the structure. The results obtained can be displayed as colored images identifying areas of high and low stress.

As in any analysis, the results are only as good as the model and must be checked by either using a verification model or by physical testing to confirm their validity.

4.1.2. The NASTRAN Finite Element Software

The MacNeal-Schwendler Corporation (MSC) has been supplying sophisticated engineering tools since 1963. MSC is the developer, distributor and supporter of

the most complete and widely used structural analysis program in the world, MSC/NASTRAN.

4.1.3. Introduction to Finite Element Theory

Two different finite element approaches to analyzing structures are the force method and the displacement method. In both methods, equilibrium, compatibility and stress-strain relations are used to generate a system of equations that represent the behavior of the structure.

4.1.3.1. The Force Method

The member forces are the basic unknowns in the system of equations.

4.1.3.2. The Displacement Method

The nodal displacements are the basic unknowns in the system of equations. Both methods can be used to solve structural problems. The displacement method is easier to adapt to electronic computations. MSC/NASTRAN uses the displacement method approach to finite element analysis.

4.1.4. Fundamental Structural Analysis Requirements

All structural engineering analysis must satisfy the following three general conditions :

- (i) Equilibrium of forces and moments :

$$\Sigma F = 0, \Sigma M = 0 \quad (4.1)$$

- (ii) Strain-Displacement Relations : (also known as compatibility of deformations) Ensure that the displacement field in a deformed continuous structure is free of voids or discontinuities
- (iii) Stress-Strain relations: (also called constitutive relations) for a linear material, the generalized Hooke's Law states :

$$\{\sigma\} = [E]\{\varepsilon\} \quad (4.2)$$

where :

$$\{\sigma\} = \{\sigma_x \sigma_y \sigma_z \tau_{xy} \tau_{yz} \tau_{zx}\}$$

$$\{\varepsilon\} = \{\varepsilon_x \varepsilon_y \varepsilon_z \gamma_{xy} \gamma_{yz} \gamma_{zx}\}$$

$$[E] = 6 \times 6 \text{ Matrix of elastic constants}$$

- A homogenous, isotropic material $[E]$ reduces to two independent material constants E and ν
- For such a material under uniaxial load,

$$\sigma = E \varepsilon \quad (4.3)$$

These three conditions can be used to generate a system of equations in which the displacements are unknown (the displacement method).

4.1.5. Basic Equation Of the displacement Method

- The basic equation of the displacement method are derived from :
 - the equilibrium of nodal forces;
 - the compatibility of displacements (at grid points and within elements);
 - the force-displacement relationship.

- The compatibility condition correlates the external grid point displacement to end deformations of the elements.
- The force displacement relationship is established between the member end forces and displacements, and between the grid point forces and displacements.
- The stiffness matrix $[K]$ is used to relate the forces acting on the structure and the displacements resulting from these forces in the following manner :

$$\{F\} = [K]\{u\} \quad (4.3)$$

where

- $\{F\}$ = forces acting on the structure
- $[K]$ = stiffness matrix $[k_{ij}]$ where each k_{ij} term is the force of a constraint at coordinate i due to a unit displacement at j with all other displacements set equal to zero
- $\{u\}$ = displacements resulting from $\{F\}$

- Boundary conditions are applied to prevent rigid body motions, and the system of linear equations is solved for the unknown $\{u\}$.

4.1.6. Interpretation of Elemental Stiffness Matrix $[K]$ and Stiffness Coefficients (k_{ij})

- Physically, $[K]$ describes how force is transmitted through the element.
- for elastic problems, Maxwell's Law requires that the stiffness matrix is symmetric.

- a single term of the stiffness matrix k_{ij} is called a stiffness coefficient. The units of k_{ij} are load / displacement.

4.1.7. Discretization of Continuous Structures

- A continuous structure may be divided into discrete grid points connected by elements.
- each grid point has six independent degrees of freedom (DOFs). A degree of freedom is defined as an independent component of translation or rotation at a grid point
- $\{u\}$ = vector of displacements = $\{u_x \ u_y \ u_z \ \theta_x \ \theta_y \ \theta_z\}$, where the term displacement is a general term describing a component of translation or rotation.

4.1.8. Constraining the Structure - Rigid Body Motion

- The solution of the structural equation

$$\{F\} = [K]\{u\} \quad (4.4 \text{ a})$$

requires inversion of the $[K]$ matrix

$$\{u\} = [K]^{-1}\{F\} \quad (4.4 \text{ b})$$

- Inversion of the $[K]$ matrix requires that $[K]$ be square and that $[K]$ have a nonsingular determinant.
- If the rigid body motion or mechanisms are not constrained, the structure is unstable and the stiffness matrix will be singular.

4.2. Defining Systems of Stress

Stress can be defined as the force per unit area, and is usually expressed for engineering purposes in megapascals (MPa). When a component is loaded every element, however small, could possibly experience different stresses at the any one time.

Serving the purpose of this discussion these ‘small’ elements could be described as being vanishingly small cubes. Normal stresses act at 90° to the face of the cube in either tension or compression. Shear stresses act in the planes of the faces as couples on opposite faces.

These normal and shear stresses that act perpendicular and parallel to the faces make up nine values of the 3-D stress tensor of the second order.

$$\begin{bmatrix} \sigma_{xx} & \tau_{xy} & \tau_{xz} \\ \tau_{yx} & \sigma_{yy} & \tau_{yz} \\ \tau_{zx} & \tau_{zy} & \sigma_{zz} \end{bmatrix} \quad (4.5)$$

where

σ = the normal stress

τ = the shear stress

The first of the subscripts denotes the direction of a normal to indicate the surface and the second the direction in which it is acting.

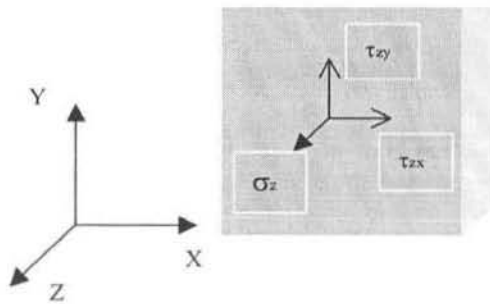


Figure 4.1. Stress on an elemental cube (after Gere and Timoshenko, 1991)

Figure 4.1. is a representation of the infinitesimally small element whose faces are chosen be parallel to the planes generated by an arbitrarily chosen xyz coordinate frame. For illustration purposes the stresses on the z face are depicted. The components depicted are shown in the positive direction.

For combined loading conditions, there is bound to be a continuous distribution of stress around a point chosen for analysis. The normal and shear stresses at this said point varies with direction. The planes on which the shear stresses are zero are defined as being the principle planes of a system, and the stresses acting normal to these planes are designated as principle stresses. Principle axes are defined as the directions of the surface normals to the principle planes.

The axes that are set orthogonally to the principle axes are those axes along which the shear stresses are maximum. The planes on which the principle shear stresses act are set at 45° to the planes of the principle normal stresses. The subject of much of the work presented in this dissertation is concerned with designing against failure. Failure will occur when the a component is stressed past some value which is a property of the component material. It is therefore of the utmost importance to find out what the maximum stresses in the material, and to take the appropriate design measures to minimize the possibility of failure of this type.

The matrix equation relating the applied stresses to the principle stresses given by :

$$\begin{bmatrix} \sigma_x - \sigma & \tau_{xy} & \tau_{xz} \\ \tau_{yx} & \sigma_y - \sigma & \tau_{xy} \\ \tau_{zx} & \tau_{zy} & \sigma_z - \sigma \end{bmatrix} \begin{bmatrix} n_x \\ n_y \\ n_z \end{bmatrix} = 0 \quad (4.6)$$

σ = the principle stresses magnitude and n_x, n_y, n_z are the direction cosines of the unit vector \mathbf{n} , which is normal to the principle plane.

In order to calculate the principle stresses the following steps are undertaken.

- Find the determinant of the coefficient matrix
- Set it equal to zero and solve

Having expanded this said determinant the following polynomial of the third degree is obtained.

$$\sigma^3 - C_2\sigma^2 - C_1\sigma - C_0 = 0 \quad (4.7.1)$$

where

$$C_2 = \sigma_x + \sigma_y + \sigma_z \quad (4.7.2.)$$

$$C_1 = \tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2 - \sigma_x\sigma_y - \sigma_y\sigma_z - \sigma_z\sigma_x \quad (4.7.3.)$$

$$C_0 = \sigma_x\sigma_y\sigma_z + 2\tau_{xy}\tau_{yz}\tau_{zx} - \sigma_x\tau_{yz}^2 - \sigma_y\tau_{zx}^2 - \sigma_z\tau_{xy}^2 \quad (4.7.4.)$$

The roots of the polynomial are the principle normal stresses $\sigma_1, \sigma_2, \sigma_3$ and are ordered in algebraic magnitude as listed from largest to smallest.

The principle shear are calculated from the values obtained for $\sigma_1, \sigma_2, \sigma_3$.

$$\tau_{13} = \frac{|\sigma_1 - \sigma_3|}{2} \quad (4.8)$$

$$\tau_{21} = \frac{|\sigma_2 - \sigma_1|}{2} \quad (4.9)$$

$$\tau_{32} = \frac{|\sigma_3 - \sigma_2|}{2} \quad (4.10)$$

In cases where the structure can be analyzed in a two dimensional coordinate frame the stress tensor in 2D is given by

$$\begin{bmatrix} \sigma_x & \tau_{xy} \\ \tau_{yx} & \sigma_y \end{bmatrix} \quad (4.11)$$

with the equations in (4.7) reducing to

$$\sigma_{1,3} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \quad (4.12)$$

where

σ_1 is the largest algebraic principle stress,

σ_3 is the smallest algebraic principle stress, maintaining the notation had the values been calculated from equations (4.7).

4.3. Static Failure Theories

Material properties are usually determined from tests in which specimens are subjected to simple stresses under static or fluctuating loads. This section has been reserved only as an outline for the presentation of the theories used when a material fails under static conditions.

Attempts to apply the strength properties of a material to three dimensional and two dimensional states of stress fields has spurred engineers to postulate various theories which could use the results of the simple tests to predict with reasonable accuracy whether or not failure will occur.

The following symbols appear in the sub sections to follow

U_d = the energy due to distortion

σ_{ut} = the ultimate tensile stress

σ_{uc} = the ultimate compressive stress

σ_y = the yield stress

μ = poissons ratio

E = Young's Elastic Modulus

4.3.1. Maximum Stress Theory (Rankine)

This theory assumes that failure occurs when the largest principle stress reaches the yield stress of tension or compression specimen.

For tension

$$\sigma_1 = \sigma_y \quad (4.13 \text{ a})$$

for compression

$$\sigma_1 = -\sigma_y \quad (4.13 \text{ b})$$

4.3.2. Maximum Shear Theory (Coulomb)

Assumes that yielding occurs when the maximum shearing stress equals that in a simple tension or compression specimen at the point of yielding.

$$\frac{|\sigma_1 - \sigma_3|}{2} = \sigma_y \quad (4.14)$$

4.3.3. Maximum Strain Energy Theory (Beltrami)

This theory postulates that failure occurs when the energy absorbed per unit volume equals the strain energy per unit volume in a tension (or compression) specimen at yield. This can be mathematically expressed by the following equation.

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\mu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1) = \sigma_y^2 \quad (4.15)$$

4.3.4. Maximum Distortion Energy Theory (von Mises)

The theory predicts that yielding occurs when the distortion energy equals that in simple tension at yield. The distortion energy, which can be defined as that portion of the total energy which causes distortion rather than volume change is given as

$$U_d = \frac{1+\mu}{3E}(\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - (\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)) \quad (4.16)$$

And failure can be defined by

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - (\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1) = \sigma_y^2 \quad (4.17.1)$$

This is often conveniently expressed in terms of the applied stresses.

$$\frac{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)}{2} = \sigma_y^2 \quad (4.17.2.)$$

4.3.5. Internal Friction Theory (Mohr)

Experimental data proves the distortion-energy theory best for ductile materials where the strength in tension and compression is highly comparable. However, for brittle materials, when the compressive strength exceeds the tensile strength by a considerable amount, the internal friction theory best predicts the behavior of the material at static failure. It is worth noting that when the tensile and compressive strengths are comparable for a brittle material the internal friction theory reduces to that of the maximum shear theory. The theory contends that :

For principle stresses of opposite sign failure is defined by :

$$\sigma_1 - \left(\frac{\sigma_{uc}}{\sigma_{ut}} \right) \sigma_2 = -\sigma_{uc} \quad (4.18)$$

If the signs are the same

$$\sigma_1 = \sigma_{ut} \text{ OR } -\sigma_{uc} \quad (4.19)$$

If the principle stresses are both either tension or compression , then the larger one say the first principle stress must equal the ultimate tensile stress when this stress is in tension or it must equal the ultimate compressive stress when the stress is in compression.

4.4. Aspects Of Fatigue Failure Theory

According to Fuch and Stephens, 1980, between 50 % and 80 % of all mechanical failures are fatigue failures. This challenges the engineer to consider fatigue at the design stage. Fatigue can be defined as the sudden failure of a part, component or structure due it being cyclically loaded. Fatigue can be described as consisting of three stages : crack initiation, crack propagation and component failure.

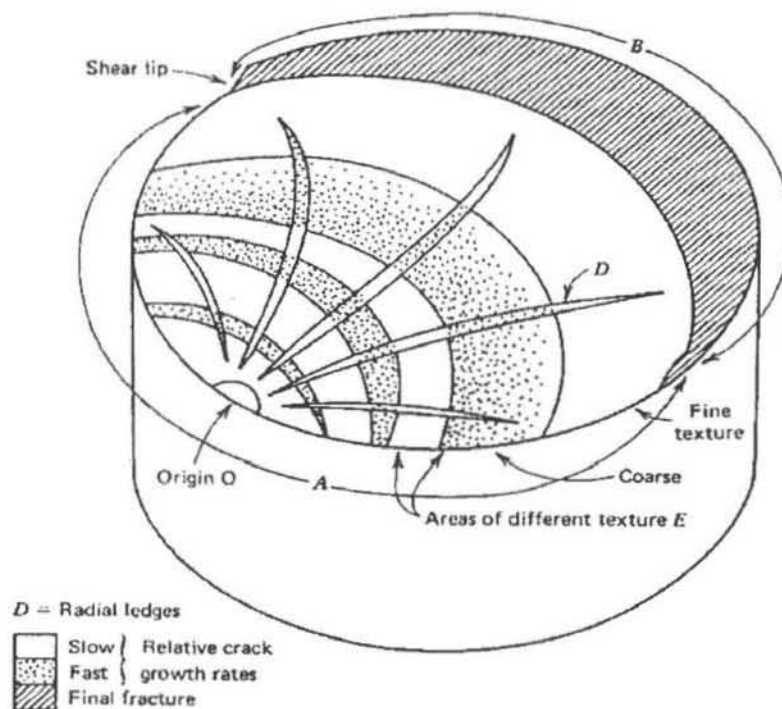


Figure 4.2. Fatigue crack propagation (after Collins, 1981)

A highly localized region where slip cycling is prominent has the tendency to develop sub microscopic cracks. The growth of these tiny cracks give rise to bigger ones until fracture occurs. The three distinct regions shown in Figure 4.2, characterize what has become known as the progressive failure phenomenon of fatigue failure. Before any further discussion of the topic the following terms need to be defined.

Endurance Limit is that strength or stress level below which the component can be cycled to infinity without failure by the mechanism of fatigue. It is denoted by S_e .

Fatigue Strength is that stress level corresponding to a particular number of load cycles. It is denoted by S_f .

Many low strength steels and alloy steels some stainless steels exhibit the fatigue behavior as depicted in Figure 4.3. and Figure 4.4. The figures are a plot with the fatigue strength plotted on the y-axis and the number of cycles plotted as the independent variable on the on the x-axis. The Figure 4.3. shows that the fatigue strength of the materials mentioned, drops linearly, when a log scale or semi log scale is used, until a leveling off of the curve occurs. For most steels, this leveling off occurs between one and 10 million cycles. The stress level corresponding to this leveling off is known as the endurance limit.

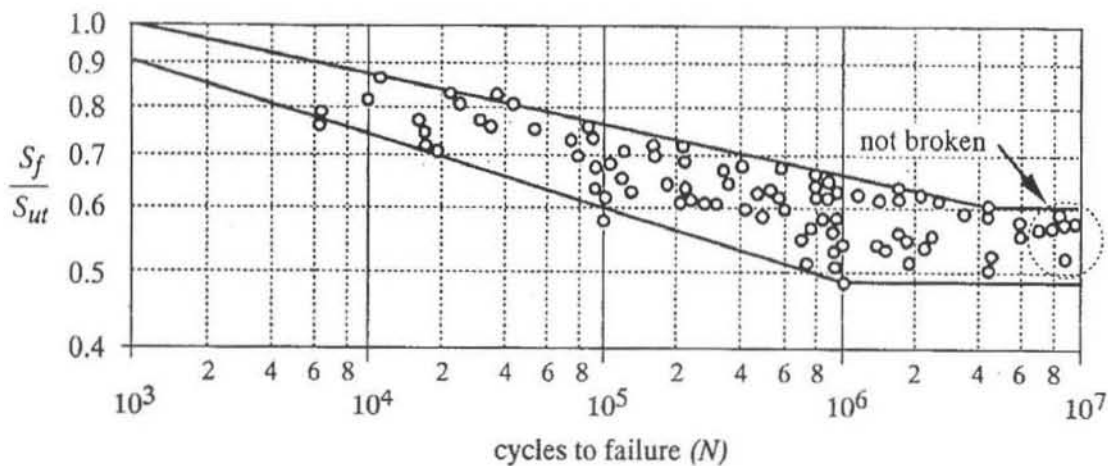


Figure 4.3. Characteristic stress cycle diagram for ferrous metals loaded in reverse bending (after Lipson and Juvinall, 1963)

The best information on the fatigue characteristics of a material is based on the actual testing of the component in the desired environment (or at least simulated as closely as possible). However, as it is not always possible to achieve this, laboratory tests carried out on test specimens of a particular material may be used to estimate the fatigue properties/ability of a component when it is placed in the desired environment.

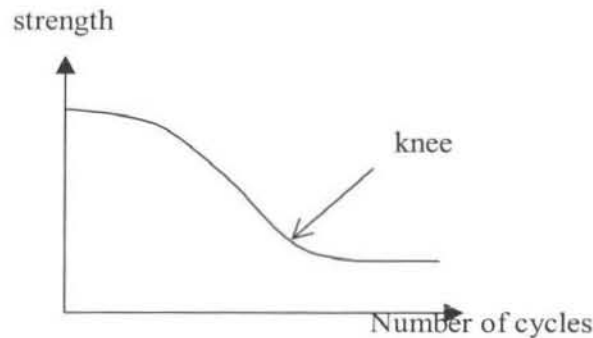


Figure 4.4. A typical strength cycle diagram for various steels

The endurance limit for steels can be estimated by the following equation :

$$S_e = \frac{1}{2} \sigma_{ut} \quad (4.20)$$

This is valid when the ultimate tensile strength is less than 1400 MPa. When the Ultimate tensile strength is greater than or equal to 1400 MPa, the endurance limit is taken as 700 MPa.

As the test is usually conducted on a small polished 8 mm diameter specimen subjected to a bending load, the endurance limit given by equation 4.20 needs to be modified to account for the difference in loading conditions, different size of component, the difference in surface roughness, different operating temperature and the difference in the reliability warranted by the application.

The modified endurance limit is given by

$$S_{em} = C_{load} C_{size} C_{surf} C_{temp} C_{rel} S_e \quad (4.21)$$

where

C_{load} = the loading correction factor

C_{size} = the size correction factor

C_{surf} = the surface correction factor

C_{temp} = the temperature correction factor

C_{rel} = the reliability correction factor

The relevant charts required for the attainment of these values are listed in appendix B.

Fatigue loading may be broadly subdivided into fully reversed and fluctuating cases of loading. The fluctuating case may be further divided into the case when the mean stress is equal to the alternating stress and when it is either greater or less than the alternating stress. The cases are depicted in Figure 4.5.

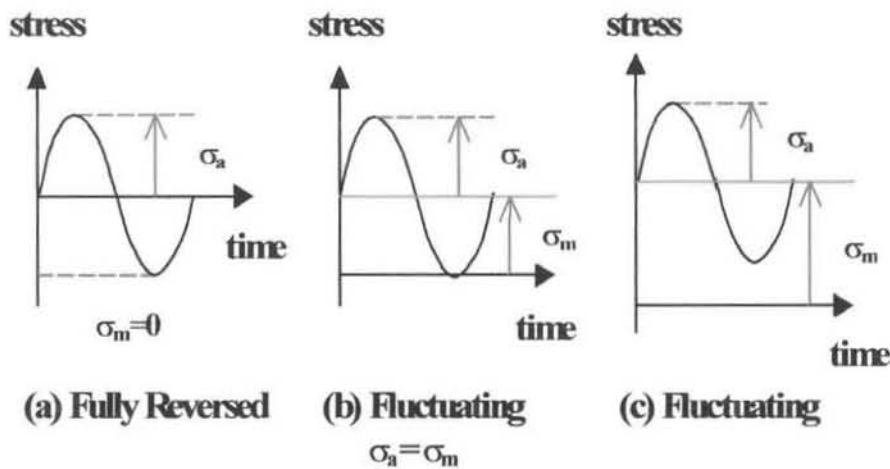


Figure 4.5. The types of fatigue loads (after Norton, 1998)

Equation 4.21. provides information about the materials fatigue strength in the high cycle region of the stress cycle diagram. To construct a diagram of this nature one would require information about the materials behavior in the low cycle region. Experimental data reveal that the strength at approximately 10^3 cycles, denoted by S_m , correlates extremely well to the following equations.

$$S_m = 0.9S_{ut} \quad \text{for bending} \quad (4.22a)$$

$$S_m = 0.75S_{ut} \quad \text{for axial loading} \quad (4.22b)$$

To reiterate, most ferrous metals exhibit a knee point at approximately 10^6 cycles. With this in mind the estimated S-N diagram may be plotted on log-log axis, and the line from S_m to S_{em} is a straight line joining the two points and extending to infinity (sometimes taken to be 10^9 cycles) as a line from S_{em} parallel to the x-axis. This situation is depicted in Figure 4.6.

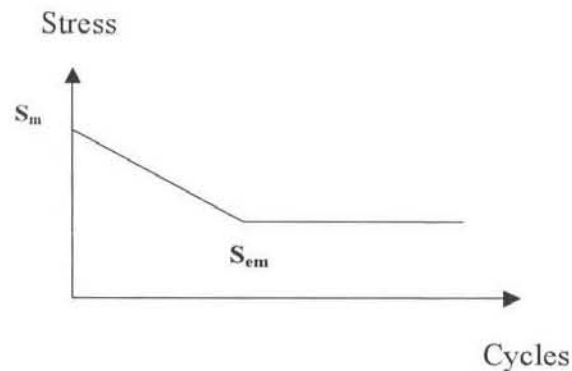


Figure 4.6. The estimated S-N curve (log-log axes) for materials with knee characteristic

The equation for the line joining S_m to S_{em} expressed in logarithmic form is

$$\log S_n = \log a + b \log N \quad (4.23)$$

With a and b , regarded as constants defined by the boundary conditions, defined as follows :

$$b = \frac{1}{z} \log \left(\frac{S_m}{S_{em}} \right) \quad (4.24)$$

with,

$$z = \log N_1 - \log N_2 \quad (4.25)$$

and,

$$\log a = \log(S_m) - 3b \quad (4.26)$$

4.5. Overview of Form Design

4.5.1. The general principle of form design

Design to obtain uniform stress.

The mechanical loading of a component will have direct repercussions on its stress and hence its life, that is until the point of failure. This loading is the designers first reference in deciphering the form of the component. Every part of a machine or piece of equipment transmits forces which largely dependent on the layout. The ethos of form design is that geometrical form influences strength.

The following steps may assist the designer in establishing the geometrical form of his design.

- Try to secure a simple force transmission arrangement which avoids region of high stress.
- Place the material so that it follows the same direction as the lines of force through the component.
- Where possible use shapes that can be expected to result in low conditions of stress.
- Determine the external and internal forces present. As far as possible, give cognizance to additional inertia forces, elastic deformations, impact or shock loads that act within an already established system boundary.
- Design a component from the inside outwards.

- Calculate and dimension using classical theories of material mechanics. Very often the theory is used to verify the coherence and consistency of advanced computational tools available for this purpose.
- Identify Maximum Stresses on the model.
- Inquire as to what material could be used to sustain these stress levels. This question should be asked iteratively with those inquiring as to how these stress levels could be altered (reduced) by altering the geometry.
- Having placed the required emphasis on static and dynamic model studies the final model should emerge as the best trade off as identified by the designer or design team.

4.5.2. Corollaries to the general principle of form design

4.5.2.1. The Tetrahedron-Triangle Principle

The use of tetrahedron and triangle shapes results in uniform stresses in tension and compression.

4.5.2.2. Uniform Shear or Hollow Shaft Principle

Uniform shear is obtained with a hollow shaft or tube, which allows for the transmission of greater torsion loads with less weight.

4.5.2.3. The Force Loading Principle

Certain portions of the component may exist for the purpose of forcing other portions to be loaded essentially on tension, compression or uniform shear. For

example webs in I-beams and C-channels force the flanges to be loaded essentially in tension and compression.

4.5.2.4. The Mating Surface Principle

Matching the surface of contacting surfaces of mating parts produces a more efficient transmission of force between parts.

4.5.3. Optimizing Strength to Weight Ratio

Components with uniform stress throughout may often be improved by selecting the optimum strength to weight ratio. Consider, for example, the case of bending where the moment can be expressed as

$$M = \sigma_{\max} \left(\frac{I}{c} \right) \quad (4.27)$$

where

M = the bending moment

I = the moment of inertia

c = the distance from the neutral axis to the point of interest

σ_{\max} = the maximum bending stress

For a given stress, σ_{\max} , the bending moment strength is proportional to the section modulus, $\left(\frac{I}{c} \right)$. Furthermore the weight of a element is directly proportional to it's cross sectional area. Thus the ratio of section modulus to area is an accurate indication of the elements strength to weight ratio. With all this in the designers minds eye, iteration in calculation may yield a highly lucrative solution with respect to the uniformity of stress and good strength to weight ratios.

4.6. Material Selection in Mechanical Design

4.6.1. Overview Of Material Selection

Material selection should in no uncertain terms be a task that is retro fitted to the designed component. Rather the design engineer is challenged to keep abreast of the factors involved in material selection during the entire design process of designing his component. He may make certain assumptions provided that these assumptions are warranted and or justified.

At all times the engineer should have candidate materials for his design in his minds eye. As a logical conclusion one would therefore be able to say that the designer must be aware of mitigating factors and properties of his materials. To do this the design engineer must have somewhat of a personal repertoire of engineering materials so that he may effectively select the best material for the task at hand.

The author has presented the following section in lieu of providing such a repertoire and to highlight the factors that should be borne in mind when selecting engineering materials. The reader should appreciate that the information presented is simply an example of such a repertoire, chosen by the author, and every such listing whilst similar in many respects will depend very much on the individual designer.

4.6.2. Factors Affecting Material Selection

The author sought to list some of the questions that should run through the mind of the designer. The author felt that it was worth stating that whilst the answers to these questions may change over the design process the initial answers will be of

great benefit to the design engineer as they will manifest themselves as the selection terms of reference.

The questions are :

- What is the components general shape and size ?
- Is the mass critical ? Is a high density or low density material better ?
- Does the component need to have good thermal or electrical conductivity?
- Does the component need to have anti corrosive properties ?
- What dimensional accuracy is required of the component ?
- What are the quantities in which the product will be produced ?
- If the operating temperature of the product will vary - Will expansion or contraction of the material present problems ?
- Does the component need to be hardened to prevent wear or indentation ?
- Is the component subject to static and or fatigue loads ?
- What mechanical strength is required of the component ? Is high tensile / compressive strength required ?
- What are the intended methods of fabrication ?

- What are the time scales of the project ? Is the candidate material available within this constraint ?
- What are the project economic issues ? How much of a constraint is this ?

These questions have arisen from four basic sub headings, namely, properties, availability, economics and miscellaneous issues. The author has opted to briefly discuss each of these in turn.

4.6.2.1. Availability

It is important that candidate materials are available within the time frame of the project. Whether the material is available on hand or it must be obtained within a stipulated period is certainly worth considering. The number of suppliers from whom the product is available is also important. Taking into account that a product/s may be manufactured over a period to follow, obtaining a material that is exclusive to one supplier, will place the designer at the mercy of the supplier for cost and possibly delivery.

It is therefore prudent to use materials that are readily available, if the design permits, as this would help to ensure efficient time to market scales. It is a little known fact in the commercial world that the producer who has introduced his product to the market before his competitors will have the initial sales burst necessary to balance the usually high cost of development.

4.6.2.2. Economics

In order to discuss this topic, one must understand that the relative price of raw materials varies from country to country. This is because different countries have

different materials in abundance and different amounts in which the 'shortages' occur. Taking the aforementioned statement into account and the variability of the economy both on a world wide scale and a intra governmental scale, it is quite a difficult task to compile a price list that would be of any value over even a short period of time. What the author recommends is that the designer make frequent contact with suppliers via either personal calls or through design publications pertinent to a specific country. The electronic media available on the subject is of great help in this regard.

There may be cases where the product requires to function in a particular way that may only be fulfilled by a material with a particular , special or rare quality. In this case the designer must be able to justify that the usual added cost associated with this special material will be ultimately of benefit to the consumer and that this added functionality will add value to the product that the customer is willing to pay for.

The other points worth discussing is the quantities in which the material will be required and the time for which the product will be in service. If only a few products are required, the design of the project may therefore be considered to be oppex sensitive as opposed to being cappex sensitive. If a substantial quantity is required the selection of a material should be influenced by a comprehensive cost study using current material costs.

Cost is allied also to the cost of treatment processes associated with some materials. It may be the case that a cheaper material that has to be treated will ultimately end in up being more costly than selecting a more expensive material which has the property that the cheaper one was treated for. A cost analysis in this respect is not just useful but necessary.

4.6.2.3. Material Properties

Fundamentally, the properties that a material possesses determine its viability for a specific application. A materials properties may be broken down into mechanical properties, physical properties, chemical properties and dimensional properties. Each of these headings may be even further subdivided. Some of the latter sub headings are explored but again the list is by no means exhaustive.

4.6.2.3.1. Mechanical Properties

- | | | |
|--------------|---|---|
| Strength | - | Is the ability of a material to resist fracture under load. |
| Hardness | - | Is the ability of a material to resist scratching and indentation |
| Elasticity | - | Is the ability to return to its original shape and dimension after being subjected to a load that caused or tended to cause, its deformation. |
| Stiffness | - | Is the measure of how well a material resists deformation. |
| Plasticity | - | The ability of a solid material to undergo permanent deformation without rupture. |
| Malleability | - | Is the ability of a material to be deformed by predominantly compressive stresses. |
| Ductility | - | The ability of a material to be plastically deformed by predominately tensile stresses. |

4.6.2.3.2. Physical Properties

- Density - Is defined, serving all useful needs, as the ratio of a materials mass to its volume.
- Conductivity - Is the affinity of a material to provide for the process of (electrical) charge transfer through it.
- Conductivity - Is the affinity of a material to provide for the internal (thermal) transfer of energy from particle to particle.
- Melting Point - the temperature at which the material state changes.

4.6.2.3.3. Chemical Properties

Environmental Resistance, Composition, Bonding and Structure :

The elemental, atomic and sub atomic makeup and behavior of the material that provides for its suitability to a particular environment.

4.6.2.3.4. Dimensional Properties

Flatness, Surface Finish, Stability, Tolerances

These terms refer either to a materials 'raw property' or its susceptibility to undergo fabrication or treatment to attain such a state.

4.6.2.4. Miscellaneous Properties

This heading should be reserved for those issues which cannot be categorized above but nonetheless must be borne in mind during the material selection process.

- Recyclability
- Health hazards (Carcinogens, Toxic, Flammable)
- Emissions (Heavy metals, Volatile solvents, Ozone depletion)
- Waste Disposal (Burn, Landfill)
- Product Liability
- Code Compliance

4.7. Application of Safety Factors

There are several ways in which a factor of safety is applied to a design. However it is important to provide a definition for the safety factor before the methods are discussed. Ideally designers must design their components or products to resist failure during its designated life. This means that the loads a structure or component will support during its working or service life should be less than the capacity for which the component is designed. With this in mind we relate the actual strength, the required strength and the factor of safety according to the following formula.

$$factor\ of\ safety = \frac{actual\ strength}{required\ strength} \quad (4.28)$$

It is common to :

- define a factor of safety for a particular application;

- determine whether or not the material used in the application is ductile or brittle;
- define a working stress or an allowable stress based on the yield strength in the case of ductile materials and the ultimate tensile strength in the case of a brittle material;
- utilize this working stress to compute appropriate geometry for the applied load.

It is also possible to :

- identify areas of maximum stress;
- calculate this maximum stress value based on the actual load on the structure;
- choose an appropriate factor of safety based on the circumstances of the application ;
- multiply the maximum stress obtained by the factor of safety and the resultant stress value should be just less than or at least equal to the strength of the material selected;
- again the ductility or absence thereof must be taken into account when specifying a material.

There is much merit in both approaches and a designer should be familiar with both of them. The author has used both methods (independently) in the design of the 4 Arm Gripper and Gantry System.

Chapter 5

The Design of a Four Arm Gripper and Gantry System

5.1. The Method of Computational Modeling Employed

Before the analysis on the actual models could be conducted, a preliminary test was carried out using MSC/NASTRAN.

5.1.1. Summary

A simple computer prototype test specimen was used to investigate whether modeling different strengths of a material, keeping all other model parameters the same namely: mesh size, geometry, loading and constraints, had any effect on the value and pattern of the stresses in the material as computed by the software. This test will serve to verify the latter of the two approaches in section 4.8 and illustrate the Nastran modeling process step by step, using this model for simplicity.

This was beneficial to the structural design process, as the geometry could be experimented with and thereafter a suitable material strength selected for that specific geometry provided that the material chosen had essentially the same elastic modulus, density and Poisons ratio. The breakdown of the modeling process serves as a foundation for the various models analysis to follow.

5.1.2. Model Description

A rectangular bar with length* breadth* depth = 400* 40*40 is depicted in Figure 5.1.1.

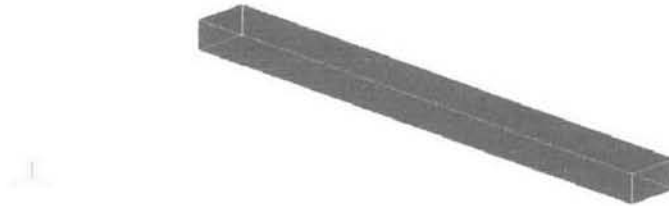


Figure 5.1.1. Test specimen

5.1.3. The Method of Finite Element Modeling

The geometry (in mm) was defined as a rectangle with corners (0,0,0) and (400,40,0). The rectangle defined as a boundary surface. The boundary surface was then extruded to a depth of 40 mm so as to define the solid. The materials as listed in Table 5.1.1. were saved to the Nastran material library.

The solid meshing process could be broadly subdivided into 3 steps. The selection of the hex meshing option, the designation of the element size as 10 which determines the degree of mesh coarseness and consequently the result accuracy. The actual meshing of the model which was preceded by the assignment of a material. The former 2 steps constitute the mesh control and were responsible for the setting of the mesh seeds. This resulted in the creation of the elements and 1025 nodes.

The bar was loaded as depicted in Figure 5.1.2. The hex mesh resulted in the creation of 25 nodes on each side face. A 50 kN load was applied to the model as shown using the load - nodal - on surface command. The bar was constrained as shown in Figure 5.1.2. The hex mesh produced 205 nodes on each of the four lateral surfaces. Two of these surfaces were assigned fixed, i.e. no rotation and no translation, constraints.

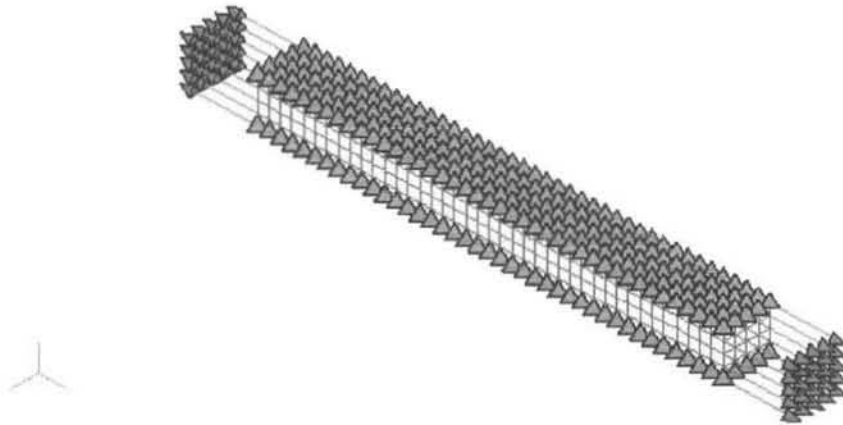


Figure 5.1.2. Loaded and constrained model

With reference to Figure 5.1.2. the blue colour represents the fixed constraints and the green the applied load. The hex meshed, loaded and constrained solid model was submitted for a static analysis. The resulting output set was opened. The Maximum solid von Mises stress option was selected and resultant value recorded. The change in material and associated properties had no effect on the stress contour and deflection. The stress contour is available in Figure 5.1.3. The software does, however, have the ability to display elements that fail, i.e. elements where the yield stress (for a ductile material) specified for the analysis was exceeded by the applied stress, this however is not included in the stress contour. The explanation is that all alloys of the same base metal have essentially the same modulus of elasticity, the same density and Poisson's ratio. The bar was loaded in

tension. The tensile stress was constant due to the constant load and area used on all models. One can conclude that higher strength alloys typically only provide higher yield or break strengths as stress is dependent on loading and geometry. A deflection analysis has the modulus of elasticity as the only material parameter included in the analysis.

Table 5.1.1. Materials selected for preliminary test and the resulting maximum stress

Test Number	Material	Elastic Modulus (GPa)	Ultimate Stress (MPa)	Condition	Max. Stress (MPa)	Density (kg/m³)
	<i>Carbon steels</i>					
1a	En 8	206	540	normalized	1342	7849
1b	En 9	206	700	normalized	1342	7849
1c	En 9(T)	206	850	H and T	1342	7849
	<i>Alloy steels</i>					
1d	En 12	203	540	normalized	1342	7849
1e	En 24(T)	203	850	H and T	1342	7849
1f	En 100(V)	203	1000	H and T	1342	7849
	<i>Case hardened steels</i>					
1g	En 33	203	*	*	1342	7849
1h	En 39A	203	*	*	1342	7849
1i	En 57	208	850	H and T	1342	7849

MSC Nastran was also able to produce deformation plots. An exaggerated example of this type of plot is depicted in Figure 5.1.4. Even though the model seems grossly distorted the actual displacement was of the order of 10^{-7} mm.

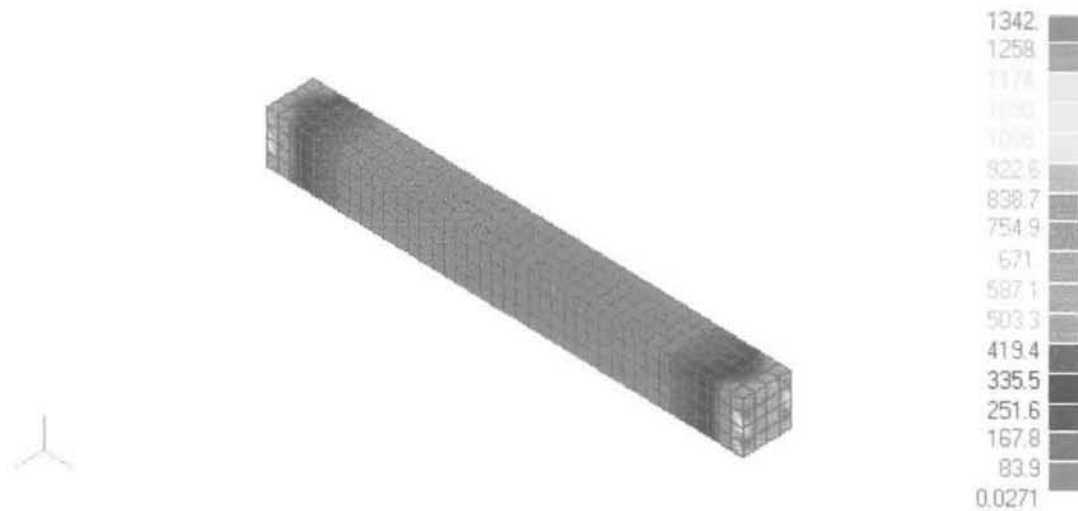


Figure 5.1.3. The maximum von Mises stress contours (Megapascals)

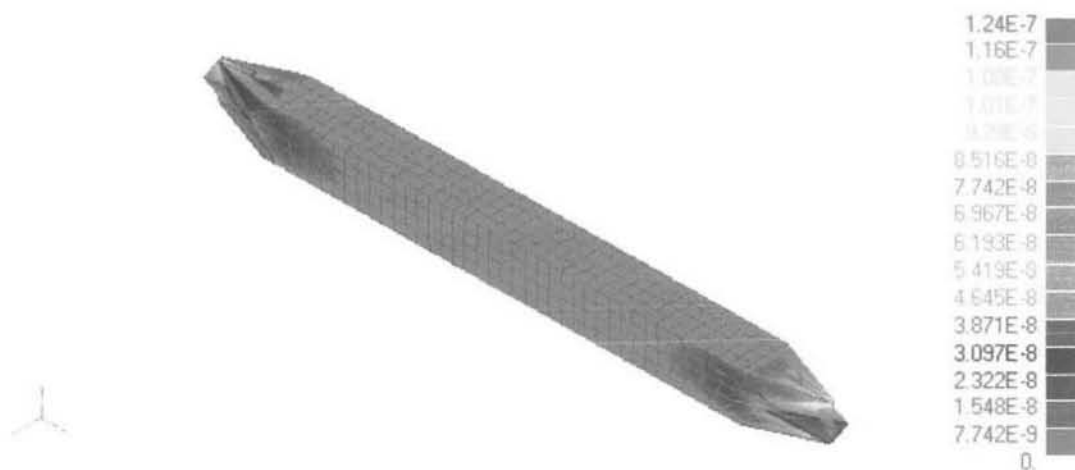


Figure 5.1.4. The deformation plot of the model (millimetres)

5.2. Arm Design

**Stress, Displacement and Mass Investigation of the Arm
: a basis for optimum geometrical selection and form design**

5.2.1. Summary

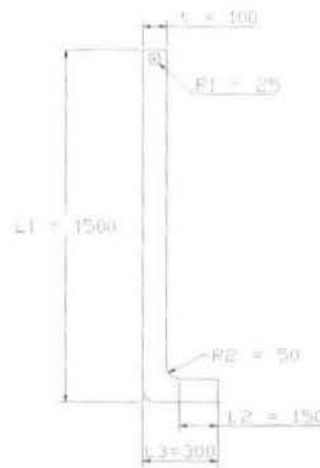


Figure 5.2.1. A general arm geometry

The central theme of most design work is the minimization of weight and at the same time the prevention of failure by designing a suitably strong component or machine. With this in mind, the selection of optimum geometrical and load bearing configurations cannot be overemphasized. The general arm scheme appears in Figure 5.2.1. together with what was deemed to be an initially suitable set of variables, taking into account the initial size specification of the timber pack.

5.2.2. Arm Design Considerations

5.2.2.1. The Arm Model

Using MSC Nastran as the tool of analysis, displacements and areas of stress concentration were obtained. The models were used as a qualitative and quantitative reference so that redesign could have been effected if necessary. Simplicity of the arm geometry accompanied by the fact that the ratio of the

parameter $L1$ to the parameter t being equal to 15, lent itself to the modeling of the arm using plate elements. Whilst this would provide an elegant FEM solution to the problem as finer mesh could be obtained leading to a more accurate result the author has also decided to provide a model using solid elements and to use the opportunity to draw a parallel between the plate and solid modeling options. A typical model submitted for analysis is depicted in Figure 5.2.2. This Figure showing the depth of the model can be obtained using both plate and solid elements. Fixed constraints were placed at the desired mating contact area between the shaft and arm. This was a reasonable assumption as no relative movement was to be permitted between the shaft and the arm. The load was intended to act over the load bearing length $L2$ (see Figure 5.2.1.) as this was intended to be the mating contact area between the palette and the arm.

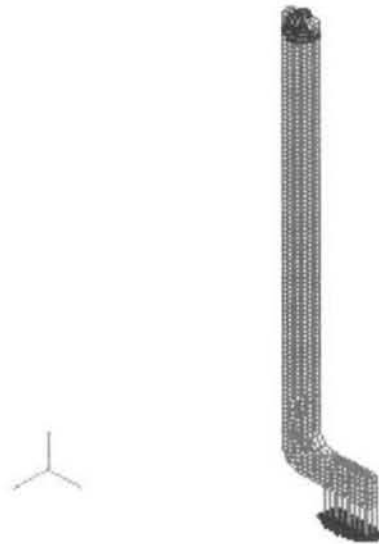


Figure 5.2.2. Model submitted for analysis

5.2.2.2. Loading

As mentioned the timber pack will be handled using a pallet with a length of 2 m and a width of 1.1 m. For an arm height of 1.5m, timber can be stacked to a height

of 1.3m. The stack of timber on the pallet was specified initially to have the following dimension : 1.3 by 1 by 1.8 meters. The hard woods used for mine support have an approximate density of 800 kg/m^3 . The load on the pallet is therefore 1872 kg. In an attempt to apply reasonable safety factors in to the load, the author sought first to investigate the conditions arising when the load was 2.5 ton and when a few models were eliminated loads of 3 ton and 3.5 ton were applied. The reason for investigating the application of the loads listed above will become apparent when the parameter L1 is experimented with later on. A reduction in this parameter would lend itself to a decrease in the actual load as dictated by the subsequent decrease in the stack height. With this in mind, design loads within the range of 2.5 t and 3.5 t would be used comfortably with a safety factor of 2. Therefore in lieu of consistency and accurate comparison the aforementioned loads are used in tests 5.2.1. to follow. The use of this safety factor is justified by the use of a computer stress modeling process as well as a control calculation.

5.2.3. The tests

5.2.3.1. Test 5.2.1 (a)

For the initial test, a load of 2.5 tons was borne by a length of $L_2 = 150 \text{ mm}$. The thickness of the geometry in Figure 5.2.1. was varied and the subsequent results recorded in Table 5.2.1. This initial test is hereinafter referred to as test 5.2.1.

It should be noted that the plate mesh quality is far superior to the solid mesh quality more especially in the regions of the lower fillet radii and the upper hole. This is not to say that the solid mesh would be an inadequate representation of the model. The solid mesh would have to be made finer (i.e. more nodes especially in

regions of geometrical discontinuity) which would invariably mean a greater analysis / solution time.

Table 5.2.1. Stresses, masses and displacements obtained for test
5.2.1. using solid elements and plate elements

Thickness (mm)	Stress (Mpa)	Mass (kg)	Maximum Displacement (mm)
5.0	174.779	6.555	14.000
	181.234		14.600
10.000	87.642	13.111	7.320
	93.339		7.530
15.000	58.300	19.665	4.680
	60.413		4.880
20.000	43.725	26.221	3.510
	45.310		3.660
25.000	35.237	32.777	2.810
	36.248		2.930
30.000	29.149	39.330	2.340
	30.207		2.440
40.000	21.786	52.440	1.750
	22.655		1.830
50.000	17.396	65.550	1.400
	18.124		1.460

Note : where two values appear in a block, the upper value represents the solid element model result and the bottom the plate element model result.

It was realized that some of the models tested in test 5.2.1.(a) could be omitted as viable contenders for the arm geometry. The mating surface area between the arm and shaft was given voice as the models with 5 mm and possibly the 10 mm whilst displaying stress patterns that could be accommodated, the cost factor aside, by using a material with the relevant yield stress, provided for an insufficient contact area which in turn could have very well lead to fulcrum type rocking at the joint even if the joint was only slightly imperfect.

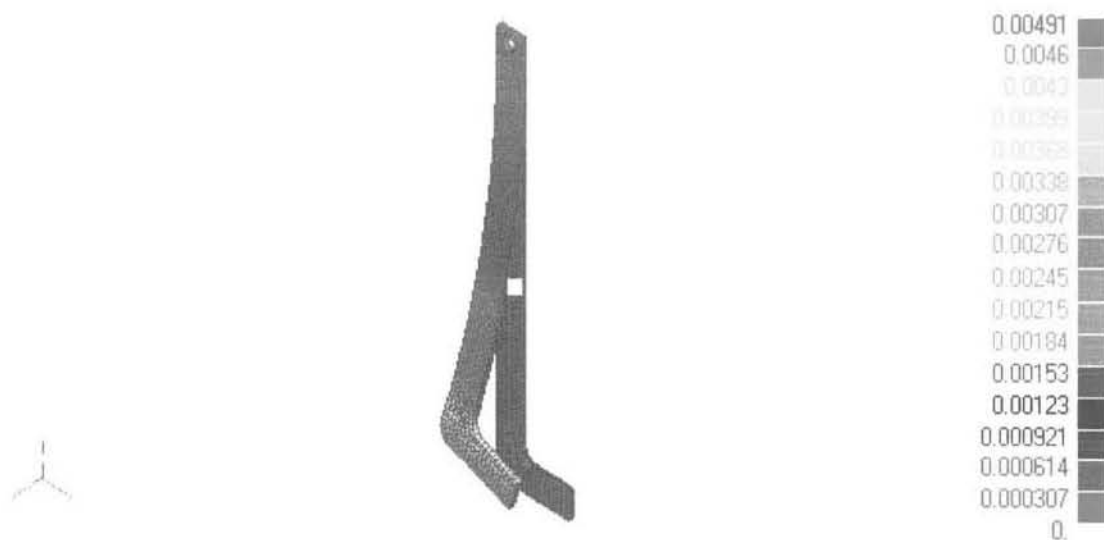


Figure 5.2.3. An exaggerated deflected model (unit : metres)

For clarification, a magnified view of the maximum arm deflection in Figure 5.2.3. has been shown in Figure 5.2.4. If the mass for the 30 mm thick model is noted, one would immediately become aware that 4 of these arms would constitute a mass of approximately 160 kg. This is unsuitable in terms of the initial specification regarding the design mass as masses for the shafts, the chassis and actuators had at that point yet to be included. Needless to say, the 40 mm and

50 mm thick models would be negated for the same reason even though the stress values seemed highly lucrative.

5.2.3.2. Test 5.2.1. (b) and Test 5.2.1.(c)

It was desired to increase the load from 2.5t to 3t and 3.5t and investigate the effect of this change on the stress and maximum displacement. The results of the 3t and 3.5t load test appear in Table 5.2.2. and Table 5.2.3. respectively.

Table 5.2.2. Output obtained by increasing load to 3t : an extension of test 5.2.1

Thickness (mm)	Stress (MPa)	Mass (kg)	Maximum Displacement (mm)
10.000	108.744	13.111	8.780
15.000	72.496	19.665	5.850
20.000	54.372	26.221	4.390
25.000	43.497	32.777	3.510

The concern over mating contact area allows not only the arm-shaft contact area problem to surface but also the mating contact area between the arm and the pallet, which also for stability, safety and the prevention of fulcrum rocking needed to be somewhat substantial. As from economic point of view, i.e. cost of the material and the manifestation of this rise per size in actuation costs, providing a chunky depth for the geometry in Figure 5.2.1, after having given credence to the masses in Table 5.2.1, would have not been feasible unless some geometrical compromise could have been attained.

Table 5.2.3. Output obtained by increasing load to 3.5t : an extension of test 5.2.1

Thickness (mm)	Stress (MPa)	Mass (kg)	Maximum Displacement (mm)
10.000	126.868	13.111	10.200
15.000	84.578	19.665	6.830
20.000	63.434	26.221	5.120
25.000	50.470	32.777	4.100

5.2.3.3. Test 5.2.2. - Parameter t

The geometry was firstly modified by changing the parameter 't' in Figure 5.2.1. from 100 mm to 70 mm. This necessitated a change in the parameter 'L3' from 300 mm to 270 mm so that the mating contact area between the arm and the pallet along with all other parameters would have been kept constant. The arms were loaded by a 3t force. The values for test 5.2.2 (t = 70 mm) are tabulated in Table 5.2.4. As the reduction in the parameter t lent itself to a reduction in mass, the 30 mm thick model came back into contention.

The results of test 5.2.2. showed a distinct reduction in mass as compared to the models in test 5.2.1. for the 3t load. However, endeavors to possibly reduce the models with 25 and 30 mm depths to masses of under 20 kg had to be made.

Table 5.2.4. Reduction in parameter 't' : results of test 5.2.2

Thickness (mm)	Stress (MPa)	Mass (kg)	Maximum Displacement (mm)
15.000	117.137	13.779	15.200
20.000	87.853	18.372	11.400
25.000	70.915	22.965	9.090
30.000	61.579	27.559	7.880

5.2.3.4. Test 5.2.3. - Parameter L1

It was desired to investigate the effect of reducing the parameter L1 to 1.2 m on the outputs discussed thus far. The reason for choosing 1.5 m as an initial length for L1 was that a greater timber stacking height of the timber on the pallet could have been achieved. The justification for the reduction is twofold. Firstly, the costs (material and actuation would be have been reduced), and secondly the vertical spatial constraint of a mining tunnel, 3.5m as specified, impinged on the allowable design height of the gantry. The results are available in Table 5.2.5 and designated as test 5.2.3. The reduction in this parameter allows the timber to be stacked to a height of 1 m. So a load of 1440 kg was expected and with a safety factor of 2, the design load of 3 t was justified. The employment of a safety factor of 2 can be explained by the use of a finite element package for the modelling as well as a verification calculation.

Table 5.2.5. Arm length reduction to 1.2 m : the results of test 5.2.3

Thickness (mm)	Stress (MPa)	Mass (kg)	Maximum Displacement (mm)
15.000	117.410	11.306	9.780
20.000	88.057	15.076	7.340
25.000	70.802	18.845	5.870
30.000	59.045	22.614	4.890
40.000	44.031	30.152	3.670

The mass of the arms that were investigated up to that point were cause for concern. A mass per arm of 20 kg or less, as mentioned earlier, would be suitable. Attention to Table 5.2.5, rendered models with depths 15, 20 and 25 mm more or less suitable based on the aforementioned criteria. However, as is also evident from Table 5.2.5, thickness is proportional to the mass and inversely proportional to stress and displacement. Having stated this, the author has chosen to use the 25 mm thick model for the advantage of the shaft arm mating contact area and the pallet arm gripping surface.

5.2.3.5. Test 5.2.4. - Parameter L3

It should be noted that the 25 mm thick model has a displacement of 5.87 mm for which attempts at minimization had to be looked into. A closer inspection at the deformation model at the points of maximum deformation appear in Figure 5.2.4.

It is apparent from Figure 5.2.4. That the maximum displacement value, indicated by the red displacement contour, occurs towards the end of length L3. An attempt

was made to lessen the local maximum displacement. An investigation whereby the length L3 was modified, was undertaken keeping the parameters t and R2 equal to 70 mm and 50 mm respectively.

Regarding to the statement made above, the length L3 was reduced to 220 mm, allowing the mating contact area between the arm and pallet to have a length of 100 mm. The result of the stress-mass-displacement investigation was that the stress was reduced to 59.798 MPa, the mass to 18.157 kg and the displacement to 4.81 mm.

5.2.3.6. Test 5.2.5. - Parameter R2

The nastran tests carried out on the arm in order to obtain the optimum geometric configuration could have not been brought to a fruitful conclusion without having experimented with the parameter R2. The parameter was reduced from 50 mm to 30 mm and thereafter increased to 70 mm. In order to obtain a result that could be compared to the tests conducted up to that point, the effective arm pallet mating contact area had to be kept constant. To this end, the parameter L3 had to be altered. The reader should be aware that the relationship between the parameters L3, L2, R2 and t is

$$L3 = L2 + R2 + t$$

The parameters together with the resulting outputs appear in Table 5.2.6.

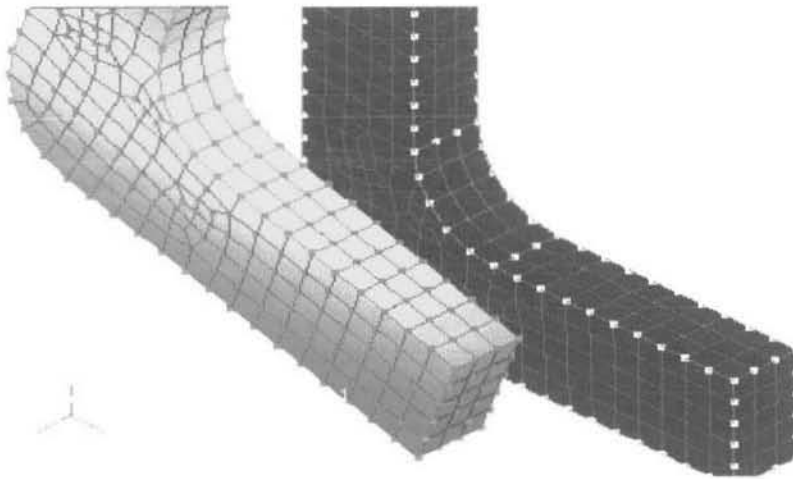


Figure 5.2.4. The maximum arm deflection (exaggerated)

Figure 5.2.4. which shows clearly the maximum deflection, is a magnification of the relevant area. The scale is available in Figure 5.2.3.

Table 5.2.6. Effect of Parameter R2 on Stress, Displacement and Mass

R2 (mm)	L3 (mm)	L2 (mm)	t (mm)	Stress (MPa)	Displacement (mm)	Mass (mm)
30.000	200.000	100.000	70.000	47.983	4.070	17.880
50.000	220.000	100.000	70.000	59.798	4.810	18.157
70.000	240.000	100.000	70.000	68.200	5.570	18.433

5.2.3. The Arm Geometry

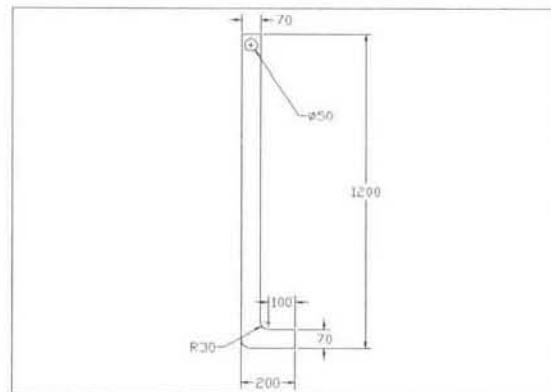


Figure 5.2.5. A geometrical manifestation of the parameter
Experimentation (dimensions : mm)

Even though extensive experimentation has been carried out on the arm geometry, the author has opted to resist the temptation to include further, albeit slight changes that had to be made to the arm in light of the shaft design as this would not serve to illustrate the reality of the iterative process of a design. Figure 5.2.5 is a manifestation of the form design investigation outlined above.

5.2.4. The Control Calculation For Arm Stresses

An attempt was made to verify the result provided by the NASTRAN computer simulation. To this end the following stress calculation is presented.

Title :

Calculation of Arm Stress

Given :

The geometry in Figure 5.2.5. The model simplified as in Figure 5.2.6. and a stress element point D identified as the point of examination. The arm is loaded and

and modeled for the purpose of the stress calculation as in Figure 5.2.7. The arm is assumed to be rigidly attached to the shaft.

Calculation :

A 3 ton load design load, as justified in the preceding discussion regarding the arm design, will be carried by the 4 arms. Each arm will therefore carry a force of 7357.5N. It is assumed that the force acts at the center of the load bearing surface of the arm.

Reference to Figure 5.2.7. (b)

at A :

$$\Sigma F_y = 0 \rightarrow R_{YA} = 7357.5 \text{ N} \quad (5.2.2)$$

$$\Sigma F_x = 0 \rightarrow R_{XA} = 0 \quad (5.2.3)$$

$$\Sigma M_z = 0 \rightarrow M_A = 7357.5 \times 0.115 = 846.1125 \text{ Nm} \quad (5.2.4)$$

Reference to Figure 5.2.7. (c)

at D :

$$\Sigma F_y = 0 \rightarrow R_{YA} = R_{YD} = 7357.5 \text{ N} \quad (5.2.5)$$

$$\Sigma F_x = 0 \rightarrow R_{XD} = 0 \quad (5.2.6)$$

$$\Sigma M_z = 0 \rightarrow M_D = 846.1125 \text{ Nm} \quad (5.2.7)$$

where

ΣF_i = the sum of forces in the i direction

ΣM_i = the sum of moments in the i direction

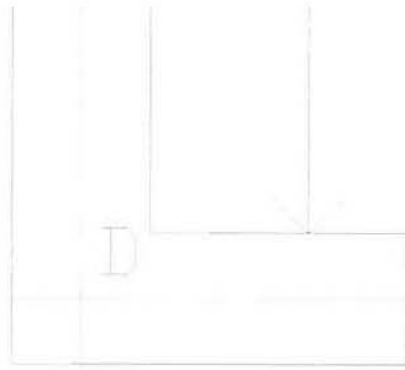


Figure 5.2.6. Simplified Model For Calculation

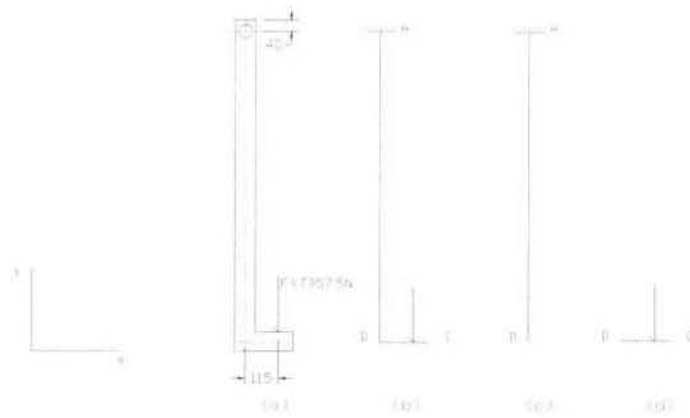


Figure 5.2.7. Arm Model for Calculation

on AD

$$\sigma_y = \sigma_{AD} = \sigma_{axial} + \sigma_{bending}$$

$$\sigma_y = \frac{P}{A} + \frac{Mc}{I} = 45.647 \text{ MPa}$$

where

σ_y = the stress on element at D in the direction of the y axis

P = the axial force = 7357.5 N

A = the cross sectional area of the modeled beam = 0.07 * 0.025 m²

M = the moment at point D = 846.1125 Nm

c = the distance from the neutral axis to point D = 0.035 m

I = the moment of inertia of the cross section = $bh^3/12 = 7.14583e-7$ m⁴

Reference to Figure 5.2.7.(d)

at D :

$$\Sigma F_y = 0 \rightarrow F = R_{yD} = 7357.5 \text{ N} \quad (5.2.10)$$

$$\Sigma F_x = 0 \rightarrow R_{xD} = 0 \quad (5.2.11)$$

$$\begin{aligned} \Sigma M_z = 0 \rightarrow M_D &= \text{load} \times \text{length from load point to point D} \\ &= 7357.5 \times 0.08 = 588.6 \text{ Nm} \end{aligned}$$

Where (5.2.12)

R_{ij} = the reaction force in the i direction at point j

on CD :

$$\sigma_x = \sigma_{CD} = \sigma_{bending} \quad (5.2.13)$$

$$\sigma_x = \frac{Mc}{I} = 28.829 \text{ MPa} \quad (5.2.14)$$

$$\tau_{xy} = \tau_{bending} \quad (5.2.15)$$

$$\tau_{xy} = \frac{3V}{2A} = 6.306 \text{ MPa} \quad (5.2.16)$$

where

σ_x = the stress on element at D in the direction of the x axis

M = maximum bending moment along CD = 7357.5 * 0.08 = 588.6 Nm

c = the distance from the neutral axis to point D = 0.035 m

I = the moment of inertia of the cross section = 7.14583e-7 m⁴

τ_{xy} = the bending shear stress acting in the yz plane in the direction of the y axis

V = the shear force at D

A = the cross sectional area of the modeled beam

Combination of Stresses : According To The Maximum Distortion Energy Theory (MDET) the equivalent stress

$$\sigma_e = \left(\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau_{xy}^2 \right)^{\frac{1}{2}} = 42 \text{ MPa} \quad (5.2.17)$$

The result calculated was excepted, since the value produced by the finite element software was computed to be 48 Mpa. The difference could however be attributed to the fact the the software employed a numerical method for the solution. In addition to this the geomtry of the model employed for the verification calculation was simplified at point D.

5.3. Theoretical Approach to Shaft Design

5.3.1. Shaft Design in the Context of the 4 Arm Gripper Design

The detail design phase of the design process was carried out as layed out in this thesis. With the design of the arm preceding the design of the shaft which was succeeded by the design of the chassis. As should be evident at this point, the parameters of the arm were available for experimentation and this experimentation manifested itself in the geometrical configuration selected.

The shafts purpose or functionality was twofold. Firstly to permit the rigid attachment and support of the arms and secondly to rotate or more correctly to undergo some predetermined angular displacement so as to provide for the gripping action.

The author approached the detail design of the shaft in somewhat of a different light to that of the arm. The reason being, that firstly, few geometric parameters were available to be experimented with and secondly the design of a simple shaft, need not be analyzed using an FEM package such as NASTRAN as the theory governing the design of this machine element is covered both extensively and sufficiently well in text books and in any undergraduate mechanical engineering degree.

Options for hollow or solid shafts were indeed given voice. The dimension of the hole in the arm, which has a direct relationship to the shaft diameter, was in no uncertain terms an arbitrary choice. It bore testimony and was therefore reserved for discussion under the design of the shaft to illustrate the iterative or “back and forth” process that is a benchmark characteristic of the design methodology already outlined.

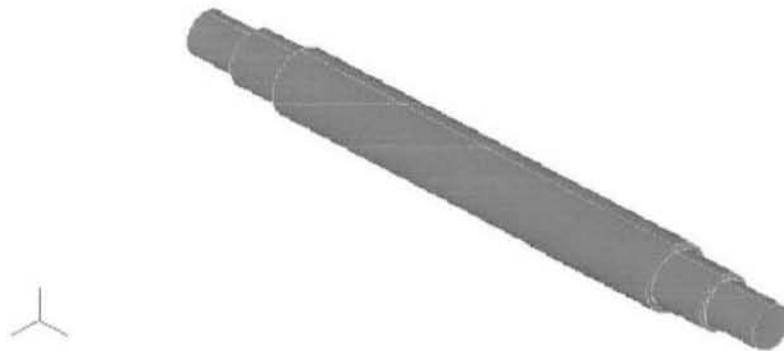


Figure 5.3.1. A 3 dimensional layout of the shaft

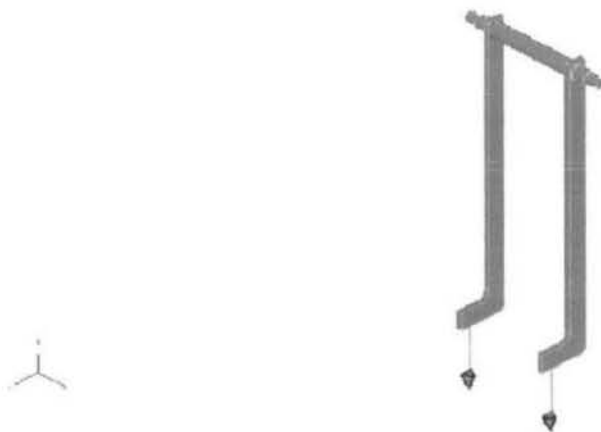
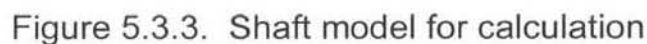


Figure 5.3.2. Shaft arm arrangement

As is evident from the Figure 5.3.1. the shaft has two stepped diameters. From a practical standpoint the shaft had to be stepped as this would provide a surface for the abutment of the arms as well as the bearings. This is important more especially since attempts made to minimize mass of the arm resulted in a compromised arm thickness and shaft arm mating contact area.

Figure 5.3.2. depicts a preliminary shaft and loaded arm configuration. The shaft supported by bearings housed in the chassis is not depicted in the figure. The bearings provide the reaction force to the load transmitted along the arm length and the due to the designated arm shape and the position of the load on the arm, the shaft is subject to a torsional moment along its longitudinal axis. In order to conduct a deflection analysis on the shaft the shaft was modeled as in Figure 5.3.3.



In general any bending load will cause the beam to deflect since it is made of an elastic material. It is assumed that the transverse-shear component is small compared to that due to bending seeing as how the beams length to depth ratio was designed to be less than 10.

Let : the distance between R1 and R2 = L
 the distance between R1,R2 and P = a

Summing forces in the vertical direction (reference to Figures 5.3.2. and 5.3.3.) it is found that $P = 7357.5 \text{ N}$.

The torsional moment on the shaft is as calculated by multiplying the perpendicular distance between the load line and the center of the hole on the arm by the value of the load. Therefore the torsional moment,

$$M_t = 7357.5 * 0.115 = 846.1125 * 2 \text{ for two arms on a shaft} = 1692.225 \text{ Nm.}$$

5.3.2.2. Hollow shaft vs. solid shaft

Up to this point no mention has been made as to whether the section should be hollow or solid. A qualitative look at this aspect was of great significance to further analysis.

The maximum shear stress of a shaft

$$\tau_{\max} = \frac{M_t r_o}{J} \quad (5.3.1)$$

For a solid section the polar moment of inertia is

$$J_{\text{solid}} = \frac{\pi r_o^4}{2} \quad (5.3.2)$$

For a hollow section the polar moment of inertia is

$$J_{\text{hollow}} = \frac{\pi(r_o^4 - r_i^4)}{2} \quad (5.3.3)$$

where

τ_{\max} = the maximum shear stress

M_t = the torsional moment

r_o = the outer radius

r_i = the inner radius
 J = the polar moment of inertia

For the situation above permit the torsional moment as well as the outer radius to remain constant. So that the relationship between the polar moment of inertia and the maximum shear stress is one of inverse proportion.

The shaft as depicted in preceding figures has two stepped diameters. For the scenario of the solid shaft the smallest of these diameters had been considered and was taken as twice r_0 . For the hollow shaft the inside diameter was d_i and had been taken as twice r_i . The outside diameter of the hollow shaft was twice r_0 . Further both hollow and solid shafts are made from the same material and were of the same length.

For $r_i = 0.5 r_0$

$$J_{hollow} = \frac{\pi(r_o^4 - (0.5r_o)^4)}{2} = \frac{\pi r_o^4}{2}(0.9375) \quad (5.3.4)$$

Therefore the ratio for the maximum shear stress in the hollow shaft to the solid shaft was found to be

$$\frac{\tau_{hollow}}{\tau_{solid}} = \frac{1}{0.9375} = 1.06667 \quad (5.3.5)$$

Similarly when $r_i = 0.8 r_0$, $r_i = 0.75 r_0$, $r_i = 0.7 r_0$ and when $r_i = 0.6 r_0$

$$\frac{\tau_{hollow}}{\tau_{solid}} = 1.6938, 1.4629, 1.3160, 1.1489 \quad (5.3.6)$$

respectively. This means that for $r_i = 0.5 r_0$; $r_i = 0.6 r_0$; $r_i = 0.7 r_0$; $r_i = 0.75 r_0$ and

$r_i = 0.8 r_o$ the stress increase for solid shaft to hollow shaft was 7%, 15 %,31%, 46% and 69 % respectively. The mass of the shafts in both scenarios had been given attention. Mass is equal to the product of density and volume. Seeing as how the material and by implication the density had together with the shafts lengths been equal, the relationship between mass and the area of the cross section was one of direct proportion.

With the area for a hollow shaft given by

$$A = \pi(r_o^2 - r_i^2) \quad (5.3.7)$$

And that of a solid shaft

$$A = \pi r_o^2 \quad (5.3.8)$$

Therefore for :

$r_i = 0.5 r_o$; $r_i = 0.6 r_o$; $r_i = 0.7 r_o$; $r_i = 0.75 r_o$ and $r_i = 0.8 r_o$ the weight of the hollow shaft was 75%, 64 %, 51%, 43.75 % and 36% of the weight of the solid shaft respectively.

5.3.2.3. Preliminary Layout and Material Selection

In selecting a preliminary layout for the shaft, stress calculations performed prior to that which is presented in the dissertation dictated that the smallest stepped diameter had to be over 30 mm, as this together with smaller diameters produced very high stresses. A leading factor in the selection of the preliminary layout was an appreciation of what is commercially available. For the task at hand, that is of a shaft with two steps, the range of shaft dimensions and materials available were extremely limited. After preliminary calculations, a 63 mm outside diameter and

32 mm inside diameter bar was commercially available. The steel, readily available locally, was sourced from overseas markets and had a yield strength in the region of 510 MPa to 610 MPa depending on the chemical composition of which it was composed. The chemical composition of the steel was such that it contained a maximum of 0.2 % Carbon, 1.5 % Manganese and 0.5 Silicon.

the stepped diameters	= d1, d2, d3
where	d1 < d2 < d3
corresponding to moments of inertia's	I1, I2, I3
with internal diameter d _i	= 0.8d1

For the calculations to follow let [d_i d1 d2 d3] = [32 40 50 63]. The selection of the step from 40 mm to 50 mm was based on an initial estimate of 5 mm for the abutment height required by a bearing. The author was well aware at the time that the layout and subsequent calculations were susceptible to reiteration and has attempted to illustrate this in sections to follow.

5.3.2.4. Shaft Stresses

For the calculation of shaft stresses the investigation included the shaft being loaded in both bending and torsion. The load distributions over the shafts length were determined and manifested themselves in the drawing of shear, bending moment, and torsional moment diagrams for the shaft.

Reference is made to Figure 5.3.3, with a = 0.05m, L = 0.7m :

The maximum shear force	$V_{\max} = 7357.500 \text{ N}$
The maximum bending moment	$M_{\max} = 367.875 \text{ Nm}$
The torque the shaft	$M_t \text{ or } T = 1692.225 \text{ Nm}$

As was evident from Figure 5.3.4. the most heavily loaded cross sections are at F and G. To this end, a section has been taken at F corresponding to maximum combinations of torsional moment and shear force and at G corresponding to maximum combinations of bending moment, torsional moment and shear force and the stresses that existed by virtue of the interference fit between the shaft and the arm.

For the calculations that follow, the following notation was used, reference is made to Figure 5.3.4, E_{ij} which is read as follows : a stress element 'E' at the point 'j' on a cross section taken at 'i'.

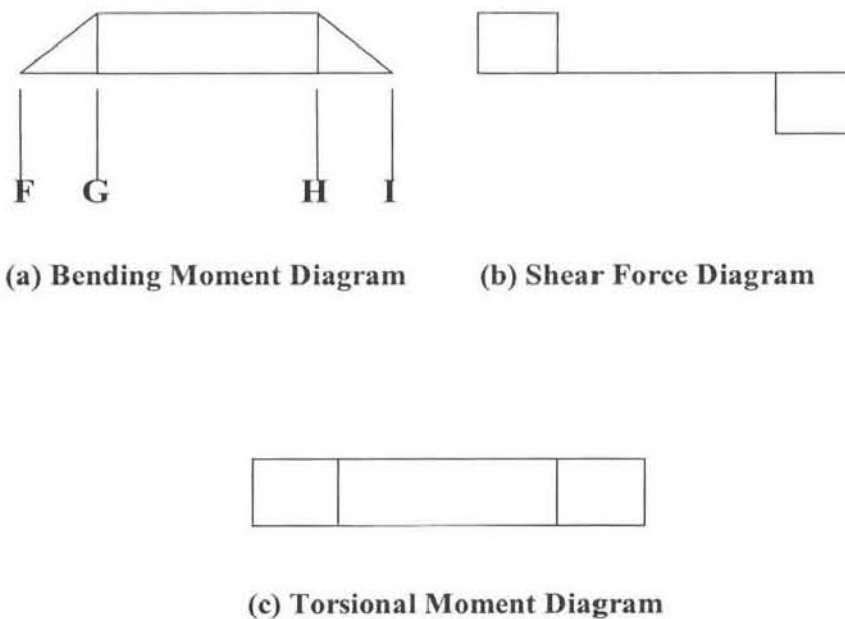


Figure 5.3.4. Shaft loading diagrams

The shear stress due to torsion is proportional to the radius and is zero at the center and a maximum at all points on the outer surface. The bending stress magnitude is proportional to the distance y from the neutral plane and is maximum at only top and bottom of the section. The shear stress due to transverse loading is a maximum at all points in the neutral plane and is zero at the outer fibres.

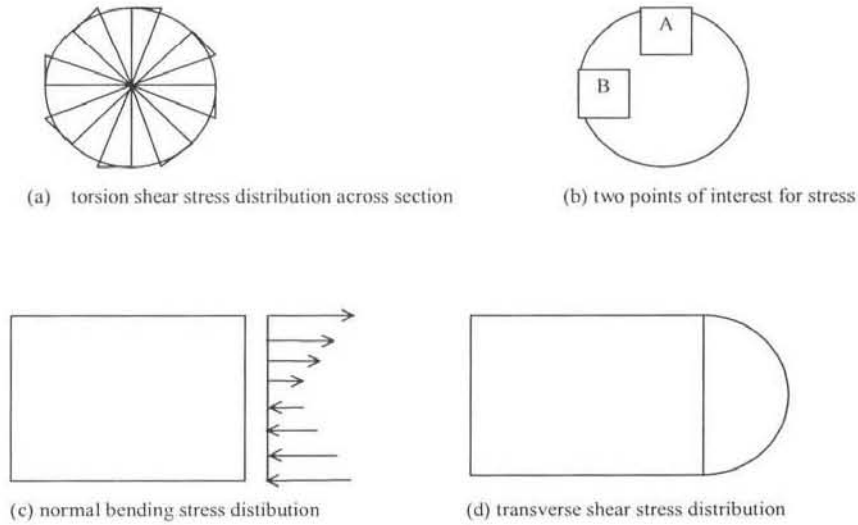


Figure 5.3.5. Stress distribution on cross sections

Four points of interest were selected for computation of stresses. Two points for the section at F and two points for the section at G (see Figure 5.3.5. (b)).

For a section at F the first point under consideration was an element at A i.e. E_{FA} . At this element both the bending stress and the transverse shear stress are zero so the element is subjected to a torsional stress. This was calculated as

$$\begin{aligned}
 \tau_{nom} = \tau_{torsion} &= \frac{M_t r_0}{J} \\
 &= \frac{1692.225 \times 0.02}{1.4838 \times 10^{-7}} \\
 &= 228.0877 \text{ MPa}
 \end{aligned} \tag{5.3.9}$$

where

τ_{nom} = is the nominal shear stress for the element
 $\tau_{torsion}$ = is the torsional shear stress on the element
 M_t = is the torsional moment as calculated in section 5.2.3.2
 r_o = is the distance of the element in question from the neutral axis
 J = the polar moment of inertia of the first step = $\frac{\pi}{32}(0.04^4 - 0.032^4)$

Also for a section at F, an element at B (see Figure 5.3.5. (b)) was considered. This element was subjected to a transverse shear stress and the torsional stress as calculated above. For E_{FB} :

$$\begin{aligned}
 \tau_{nom} &= \tau_{torsion} + \tau_{transverse\ shear} = \tau_{torsion} + \frac{4V}{3A} \\
 &= 228.0877 + \frac{4 \times 7357.5}{3 \times 4.5239 \times 10^{-4}} \quad (5.3.10) \\
 &= 228.0877 + 21.6848 \\
 &= 249.7725 \text{ MPa}
 \end{aligned}$$

where

τ_{nom} = is the nominal shear stress for the element
 $\tau_{torsion}$ = is the torsional shear stress on the element
 $\tau_{transverse\ shear}$ = is the transverse shear stress for the element
 A is the cross sectional area = $\frac{\pi}{4}(0.04^2 - 0.032^2) \text{ m}^2$
 V is the shear force = 7357.5 N

For the section at G, an element at A is subjected to a combination of bending, torsional as well as stresses due the interference fit see section 5.3.2.

The bending stress had been calculated with the following formula.

For E_{GA}

$$\begin{aligned}\sigma_{bending} &= \frac{M_b y}{I} = \frac{P a y}{I} \\ &= \frac{(7357.5 \times 0.05) 0.025}{2.5532 \times 10^{-7}} \\ &= 36.0203 \text{ MPa}\end{aligned}\quad (5.3.11)$$

where

M_b = the bending moment

P = the external transverse load on the shaft

a = the distance between the load and the support

y = the distance from the neutral to the point A

I = the section moment of inertia = $\frac{\pi}{64}(0.05^4 - 0.032^4) \text{ m}^4$

The torsional stress at all cross sections on the 50 mm OD was calculated as

$$\begin{aligned}\tau_{nom} = \tau_{torsion} &= \frac{M_t r_0}{J} \\ &= \frac{1692.225 \times 0.025}{5.106487 \times 10^{-7}} \\ &= 82.8468 \text{ MPa}\end{aligned}\quad (5.3.12)$$

where : unless otherwise stated below the symbols have their usual meaning as have been previously defined

$$J = \frac{\pi}{32}(0.05^4 - 0.032^4) \text{ m}^4$$

E_{GB} experiences torsional shear, transverse shear and the stresses that exist by virtue of the interference fit (discussed in section 5.2.3.8.)

The shear stresses on the element have been computed as follows :

$$\begin{aligned}
 \tau_{nom} &= \tau_{torsion} + \tau_{bending} = \tau_{torsion} + \frac{4V}{3A} \\
 &= 82.8468 + \frac{4 \times 7357.5}{3 \times 1.1592 \times 10^{-3}} \\
 &= 82.8468 + 8.4623 \\
 &= 91.3091 \text{ MPa}
 \end{aligned} \tag{5.3.13}$$

where : unless mentioned below all values have the same meaning and assume the same properties as previously defined.

$$A = \frac{\pi}{4}(0.05^2 - 0.032^2)$$

Looking at the stresses calculated in perspective, or more specifically the shear stresses, computed are very high. Realizing that a stress concentration factor had yet to be applied and that the material had a minimum yield strength of 510 MPa for which the shear yield strength was 0.58 of the yield strength, an attempt had to be made to reduce the stress. The author sought to increase the moment of inertia by increasing the outside diameter of the shaft at the bearing-shaft step surface to 45 mm as this is the next standard bearing inner diameter available commercially.

The shear stress on E_{FA} was reassessed as follows :

$$\begin{aligned}
 \tau_{nom} &= \tau_{torsion} = \frac{M_t r_0}{J} \\
 &= \frac{1692.225 \times 0.0225}{2.99963 \times 10^{-7}} \\
 &= 127.0718 \text{ MPa}
 \end{aligned} \tag{5.3.14}$$

where

τ_{nom} = is the nominal shear stress for the element

$\tau_{torsional}$ = is the torsional shear stress on the element

M_t = is the torsional moment as calculated in section 5.2.3.2

r_o = is the distance of the element in question from the neutral axis

J = the polar moment of inertia of the first step = $\frac{\pi}{32}(0.045^4 - 0.032^4) \text{ m}^4$

The shear state of E_{FB} was recalculated as follows :

$$\begin{aligned}\tau_{nom} &= \tau_{torsion} + \tau_{bending} = \tau_{torsion} + \frac{4V}{3A} \\ &= 127.0718 + \frac{4 \times 7357.5}{3 \times 7.8618 \times 10^{-4}} \quad (5.3.15) \\ &= 127.0718 + 12.4780 \\ &= 139.5498 \text{ MPa}\end{aligned}$$

where : unless stated or recalculated below the symbols have their usual meaning

$$A = \frac{\pi}{4}(0.045^2 - 0.032^2)$$

In order to have calculated the stresses on the next step the diameter of the next step had to be determined. The step height and by implication the diameter was directly influenced by the bearing that was selected which would have dictated a minimum abutment height. As a first step toward the bearing selection the shaft deflection and angular displacement had to be computed as this was deemed to be an indispensable step in selecting the type of bearing that would be used for a given misalignment. The loading capacity, the other vital aspect of concern when selecting a specific bearing type of bearing has been discussed in section 5.3.2.6.

5.3.2.5. Application of Castigliano's Theorem for Shaft Deflections

The Strain Energy, U , in a straight beam subject to bending moment, M , is

$$U = \int \frac{M^2 dx}{2EI} \quad (5.3.16)$$

Castigliano's strain energy equation can be applied to problems involving shafts of non-constant section.

Reference to Figure 5.3.3.

The total strain energy due to bending is

$$U = \int \frac{M^2 dx}{2EI} = U_1 + U_2 + U_3 \quad (5.3.17)$$

where

U_1 is the energy from $x = 0$ to a

U_2 is the energy from $x = a$ to $L-a$

U_3 is the energy from $x = L-a$ to L

The strain energy is considered for each section of the shaft

Resolving vertical loads

$$R_1 + R_2 = 2P \quad (5.3.18)$$

Taking moments about the left hand bearing

$$R_1 = R_2 = P \quad (5.3.19)$$

Splitting the beam between $x = 0$ and $x = a$, the moment can be expressed as Px .

$$U_1 = \int_0^a \frac{(Px)^2}{2EI_2} = \frac{P^2 a^2}{6EI_2} \quad (5.3.20)$$

Splitting the beam between $x = a$ and $x = L-a$, the moment can be expressed as Pa .

$$U_2 = \int_a^{L-a} \frac{(Pa)^2}{2EI_3} = \frac{P^2 a^2}{6EI_3} (L - 2a) \quad (5.3.21)$$

Splitting the beam between $x = L-a$ and L , the moment can be expressed as $R_2(L-x)$

$$U_3 = \int_{L-a}^L \frac{(R_2(L-x))^2}{2EI_2} = \frac{P^2}{2EI_2} \left[\frac{L^3}{3} - L^2(L-a) + L(L-a)^2 - \frac{(L-a)^3}{3} \right] \quad (5.3.22)$$

And the total strain energy is

$$U = \frac{P^2 a^2}{6EI_2} + \frac{P^2 a^2}{6EI_3} (L - 2a) + \frac{P^2}{2EI_2} \left[\frac{L^3}{3} - L^2(L-a) + L(L-a)^2 - \frac{(L-a)^3}{3} \right] \quad (5.3.23)$$

To obtain the deflection at the load P , the partial derivative of U with respect to P is calculated :

$$\frac{\partial U}{\partial P} = 2P \left\{ \frac{a^2}{6EI_2} + \frac{a^2}{6EI_3} (L - 2a) + \frac{1}{2EI_2} \left[\frac{L^3}{3} - L^2(L-a) + L(L-a)^2 - \frac{(L-a)^3}{3} \right] \right\} \quad (5.3.24)$$

I_2 is the section moment of inertia between the load and the support and I_3 is the section moment of inertia between the loads. It was assumed that the diameter at the shaft_arm interface was 50 mm. The following analysis was therefore regarded as a worst case scenario and would be revisited once the bearing had been selected and the abutment height found. This, once again serves to illustrate the iterative nature of the design process.

$$I_2 = \frac{\pi}{64}(d_o^4 - d_i^4) = \frac{\pi}{64}(0.05^4 - 0.032^4) = 2.553243e - 7 \text{ m}^4$$

$$I_3 = \frac{\pi}{64}(d_o^4 - d_i^4) = \frac{\pi}{64}(0.063^4 - 0.032^4) = 7.217999e - 7 \text{ m}^4$$

With $a = 0.05\text{m}$; $L = 0.7\text{m}$; $E = 203 \text{ GPa}$

The deflection (d) under load is obtained by substituting a , L , E , I_2 , and I_3 into equation (5.3.24). The result was a deflection of $1.9953e-4 \text{ m}$ and the angular deflection can be calculated as $\arctan(d / a)$ to be 0.229° . Clearly the use of self aligning bearings seemed apt. However as the relative cost these bearings is more or less twice the cost of rolling element bearings, the author sought to make some attempt to minimize the deflection, in order to verify that the use of such bearings was absolutely necessary. This was done by experimenting with the lengths a and L .

Table 5.3.1. Experimentation with parameter 'a'

a(m)	d(m)	$\Theta = \arctan(a / d)$ (deg)
0.030	7.279E-05	0.14
0.040	1.285E-04	0.18
0.060	2.854E-04	0.27
0.070	3.859E-04	0.32

Attempts to modify the parameter 'a' in regard to the aforementioned aim were unsuccessful. As the depth (thickness) of the arm chosen was 25 mm, 30 mm was the smallest practically conceivable length for this parameter. A length of 0.07 was deemed to be unsuitable as this would shorten the distance between the arms and invariably place the load further from the support. To this end, the author had decided take the parameter 'a' to be 40 mm. Remembering that a change in this parameter would affect the bending stress as calculated in the previous section.

The length L was the next parameter under investigation. The results of this investigation appear in Table 5.3.2. All parameters remained as originally outlined so that an appropriate comparison could be made i.e., the value of a remained as 50 mm. Table 5.3.3. contains the deflections when L is varied and 'a' is kept at a constant value of 40 mm.

Table 5.3.2. Experimentation with parameter 'L' for a = 0.05

L(m)	d(m)	$\Theta = \arctan(a / d)$ (deg)
0.500	1.744E-04	0.20
0.600	1.870E-04	0.21
0.800	2.121E-04	0.24

Table 5.3.3. Experimentation with parameter 'L' for a = 0.04

L(m)	d(m)	$\Theta = \arctan(a / d)$ (deg)
0.500	1.125E-04	0.16
0.600	1.205E-04	0.17
0.800	1.366E-04	0.20

The trend from the Table 5.3.2. and Table 5.3.3. is that both the deflection and the angular deflection increase with an increase in the length L . However it is not practical to limit the length L to less than 0.5 m and even this value is questionable. The rationale behind this is that the length of pallet is 2 m and a 'reasonable' distance between the arms on the shaft was required in order to facilitate the safe handling of the pallet. It was also apparent that the attempt to reduce the angular deflection to 0.04° had not materialized.

In light of the above discussion and analysis, the author has chosen ' L ' and ' a ' to be 800 mm and 40 mm respectively. The next task was that of selecting the self aligning bearing.

5.3.2.6. Bearing Selection

Seeing as how the bearing was stationary for the most part of its operation and slowly slewing when moving, serving all useful needs and purposes it was regarded as being statically stressed.

The assessment of the index of static stressing, f_s , is a safety factor against prohibitive permanent deformations on the contact areas of rolling elements. A bearing for the gripper application had to satisfy a normal demand and could have assumed a value of 1.2. However credence must be given to the environment in which a definitive maintenance policy could very well be absent and to this end the author has chosen a value of 1.35.

C_0 , the static load rating is a listed value in bearing tables. This load causes a total permanent deformation of 0.0001 of the rolling element diameter at the most heavily loaded element / raceway contact. P_0 , the equivalent static load rating and represents the external load on the bearing.

The index of static stressing f_s , the equivalent static load P_o and the static load rating, C_o are related by the following equation.

$$f_s = \frac{C_o}{P_o} \quad (5.3.25)$$

From this equation :

$$C_o = f_s P_o = 1.35 \times 7357.5 = 9.93 \text{ KN} \quad (5.3.26)$$

For an internal diameter of 40 mm and a static load rating of 9.93 KN. Bearing catalogues were consulted and the appropriate bearing selected.

As already mentioned a self aligning bearing would be used to cater for the angular deflection of the shaft. Self aligning bearings are able to cater for a misalignment of 4° , however the environment in which the bearing would have to operate in, dictated the use of a sealed self aligning bearings which cater for a misalignment of up to 1.5° . Sealed self aligning bearings incorporate rubbing seals on both sides. They are lubricated for life at manufacture, and this property is well suited to the minimal maintenance specification.

The bearing dimensions of the bearing selected is depicted in Figure 5.3.6. The fillet radius on the shaft is dictated by the choice of bearing. The bearing rings were required to closely fit the shaft or housing shoulder, and it was not allowed to foul the shoulder fillet. As a result the fillet radius r_g , of the shaft had to be smaller than the minimum chamfer r_{smin} , of the bearing. For the selected bearing r_g was chosen to be 1 mm against a tabulated r_{smin} of 1.1 mm. The significance of the abutment height in the specification of the shaft_arm interface diameter has already been mentioned. The bearing selected dictated that the minimum abutment

height, h_{\min} had to be 3.5 mm. This would have meant that the diameter in question would have to be 52 mm. Having found this value, the following reiterative calculations were carried out :

- the stresses that existed due to the interference fit;
- the calculation of the stresses on a section at G (see section 5.3.2.5)
- the combination of these stresses
- a revisited analysis of the arm : changing the hole diameter and loading the hole arm with the pressure that existed by virtue of the interference fit and the torsional load

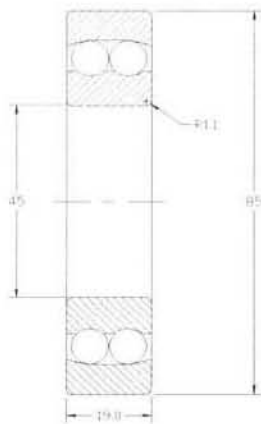


Figure 5.3.6. Bearing dimensions

5.3.2.7. Calculation of the Interference Fit and Associated Stresses

The torque which can be transmitted between components by an interference fit can be estimated by :

$$T = 2fP\pi b^2L \quad (5.3.27)$$

where :

- T = torque, Nm
 f = coefficient of friction
 P = interference pressure, N/m²
 b = interference radius, m
 L = length of interference fit, m

Equation (5.3.27) can be rearranged to evaluate the pressure required to transmit the torque :

$$\begin{aligned}
 P &= \frac{T}{2f\pi b^2 L} \\
 &= \frac{1692,225}{2 \times 0.78 \times \pi \times 0.026^2 \times 0.025} \\
 &= 20.4313 \text{ MPa}
 \end{aligned} \tag{5.3.28}$$

Even though the materials used for the arm and the shaft maybe different grades of steel, catering for different stress level, the modulus of elasticity is essentially the same. Therefore the following equation, used to find the diameter interference, for the case when both shaft and hub are of the same material can be used with sufficient accuracy.

$$\begin{aligned}
 \delta &= \frac{4b^3(c^2 - a^2)P}{E(c^2 - b^2)(b^2 - a^2)} \\
 &= \frac{4(0.026^3)(0.035^2 - 0.016^2)(20.4313 \times 10^6)}{203 \times 10^9(0.035^2 - 0.026^2)(0.026^2 - 0.016^2)} \\
 &= 2.9736 \times 10^{-5} \text{ m} \\
 &= 0.03 \text{ mm}
 \end{aligned} \tag{5.3.29}$$

where (as illustrated in Figure 5.3.6.) :

- P = pressure at the mating surface, N/m²
 δ = total diameter interference, m

- a = internal radius of shaft, m
- b = outer radius of inner shaft, m
- c = outer radius of outer component
- E = Young's modulus, N/m²

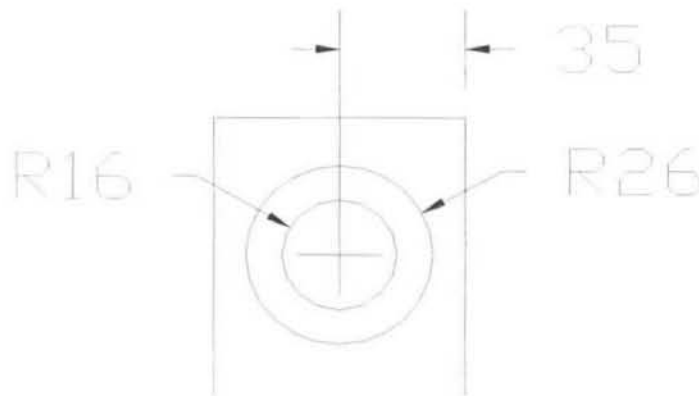


Figure 5.3.7. Radii used in the interference fit

Therefore the required interference fit was found to be 0.03 mm. This was specified using the H7/s6 tolerance band, (see Appendix A, Figure A1 : selected ISO fits, holes basis).

The shaft dimension was specified as : $52^{+0.072}_{+0.053} \text{ mm}$

The hole dimension was specified as : $52^{+0.03}_0 \text{ mm}$

5.3.2.8. Shaft Deflection Revisited

As mentioned once the diameter of the shaft_arm interface had been determined, this diameter would be used to determine a more accurate deflection under load as compared to the previously assumed value of 50 mm. Having found the

diameter, as discussed in section 5.2.3.7. the section moments of inertia were re-calculated as follows :

$$I_2 = \frac{\pi}{64}(d_o^4 - d_i^4) = \frac{\pi}{64}(0.052^4 - 0.032^4) = 3.074363e-7 \text{ m}^4$$

$$I_3 = \frac{\pi}{64}(d_o^4 - d_i^4) = \frac{\pi}{64}(0.063^4 - 0.032^4) = 7.217999e-7 \text{ m}^4$$

For which substitution of these values along with a, L, and E equal to 40 mm, 800 mm and 203 GPa into equation (5.3.24) yields a deflection under load of $1.2324e-4 \text{ m}^4$ and a corresponding angular deflection of 0.17° .

5.3.2.9. Shaft Stresses Revisited

The author mentioned in section 5.2.3.5. that the recalculation of the stresses at points A and B on a cross section taken at G (see Figure 5.3.4. and Figure 5.3.5.) had been postponed until the appropriate diameter had been selected. Section 5.3.2.6 dictated the use of a minimum diameter at cross section of 52 mm.

For E_{GA} , the bending stress was given according to the equation as :

$$\begin{aligned} \sigma_{nom} = \sigma_{bending} &= \frac{M_b y}{I} = \frac{P a y}{I} \\ &= \frac{(7357.5 \times 0.04) 0.026}{3.074363 \times 10^{-7}} \\ &= 24.8891 \text{ MPa} \end{aligned} \quad (5.3.30)$$

where

σ_{nom} = the nominal stress, MPa

$\sigma_{bending}$ = the bending stress, MPa

- M_b = the bending moment, Nm
 P = the external transverse load on the shaft, N
 a = the distance between the load and the support, m
 y = the distance from the neutral to the point A, m
 I = the section moment of inertia = $\frac{\pi}{64}(0.052^4 - 0.032^4) \text{ m}^4$

And the torsional shear recalculated according the equation :

$$\begin{aligned}
 \tau_{nom} = \tau_{torsion} &= \frac{M_t r_0}{J} \\
 &= \frac{1692.225 \times 0.026}{6.148726 \times 10^{-7}} \\
 &= 71.5560 \text{ MPa}
 \end{aligned} \tag{5.3.31}$$

where :

- M_t = the torsional load or torque, Nm
 r_0 = the distance from the center to the point of interest, m
 J = the polar moment of inertia of the section = $\frac{\pi}{32}(0.052^4 - 0.032^4) \text{ m}^4$

Regarding E_{GB} , the stress state was reassessed as follows :

$$\begin{aligned}
 \tau_{nom} = \tau_{torsion} + \tau_{bending} &= \tau_{torsion} + \frac{4V}{3A} \\
 &= 71.5560 + \frac{4 \times 7357.5}{3 \times 1.319469 \times 10^{-3}} \\
 &= 82.8468 + 7.4348 \\
 &= 90.028160 \text{ MPa}
 \end{aligned} \tag{5.3.32}$$

where

τ_{nom} = is the nominal shear stress for the element

$\tau_{torsional}$ = is the torsional shear stress on the element

$\tau_{bending}$ = is the transverse shear stress for the element

A is the cross sectional area = $\frac{\pi}{4}(0.052^2 - 0.032^2)$ m²

V is the shear force = 7357.5 N

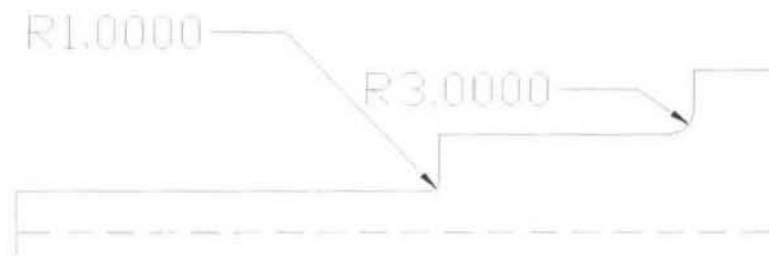


Figure 5.3.8. Shaft with fillet radii

The penultimate task in the shaft stress analysis was the application of stress concentration factors to the stresses calculated above.

Reference is made to Appendix A, Figure A2 and Figure 5.3.8.

For the step from 45 mm to 52 mm :

$$\frac{D}{d} = \frac{52}{45} = 1.155 \quad (5.3.33)$$

$$\frac{r}{d} = \frac{1}{45} = 0.0222 \quad (5.3.34)$$

For which $K_{ts} = 1.7$

where

D = the larger of the step diameters
 d = the smaller of the step diameters
 K_{ts} = the torsional stress concentration factor

Therefore the maximum shear stress on E_{FA} :

$$\begin{aligned}
 \tau_{\max} &= \tau_{nom(FA)} \times K_{ts} \\
 &= 127.0718 \times 1.7 \\
 &= 216.0221 \text{ MPa}
 \end{aligned}
 \tag{5.3.35}$$

And the maximum shear stress on E_{FB} was calculated as :

$$\begin{aligned}
 \tau_{\max} &= \tau_{nom(FB)} \times K_{ts} \\
 &= 139.5498 \times 1.7 \\
 &= 237.2347 \text{ MPa}
 \end{aligned}
 \tag{5.3.36}$$

For the step from 52 mm to 63 mm

$$\frac{D}{d} = \frac{63}{52} = 1.2115
 \tag{5.2.37}$$

$$\frac{r}{d} = \frac{3}{52} = 0.058
 \tag{5.2.38}$$

For which $K_{ts} = 1.55$ and $K_t = 1.95$

where

K_t = the bending stress concentration factor

Therefore for E_{GA} the maximum torsional shear stress was given by :

$$\begin{aligned}\tau_{\max} &= \tau_{nom(GA)} \times K_{ts} \\ &= 71.5560 \times 1.55 \\ &= 110.9118 \text{ MPa}\end{aligned}\tag{5.3.39}$$

And the maximum bending stress had been calculated as :

$$\begin{aligned}\sigma_{\max} &= \sigma_{nom(GA)} \times K_t \\ &= 24.8891 \times 1.95 \\ &= 41.3159 \text{ MPa}\end{aligned}\tag{5.3.40}$$

For the shear state of E_{GB} :

$$\begin{aligned}\tau_{\max} &= \tau_{nom(GB)} \times K_{ts} \\ &= (7.4348 + 71.5560) \times 1.55 \\ &= 122.435 \text{ MPa}\end{aligned}\tag{5.3.41}$$

For the section taken at G, point A the equivalent stress was found using the von Mises formula for finding an equivalent pseudo tensile stress was employed. The pressure, $\sigma_y = -20.43 \text{ MPa}$, due to interference fit acted in the along the negative y axis at element A. The maximum bending stress, $\sigma_x = -41.31 \text{ MPa}$, put point A in compression and therefore acted along the negative x axis. The shear stress, $\tau_{xz} = 110.91 \text{ MPa}$, acted on the x face of the element and in the direction of the z axis. The equivalent stress at element A at the shaft arm interface was

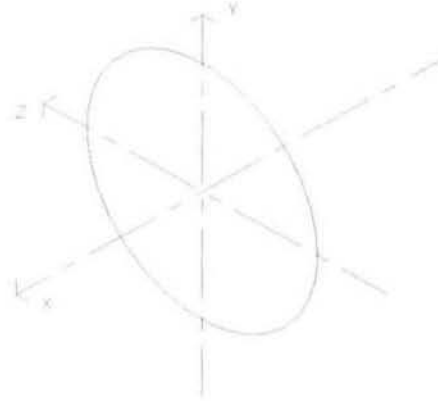


Figure 5.3.9. Coordinate frame for the shaft

$$\sigma_e = \left[\frac{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)}{2} \right]^{\frac{1}{2}} \quad (5.3.43)$$

$$= 199.6 \text{ MPa}$$

For element B at the shaft arm interface, the combined shear, $\tau_{xy} = -122.435 \text{ MPa}$ of bending and torsion, was together with the interference fit pressure, $\sigma_z = -20.43 \text{ MPa}$ used to find the equivalent stress.

$$\sigma_e = \sqrt{\sigma_z^2 + 3\tau_{xy}^2} \quad (5.3.44)$$

$$= 213 \text{ MPa}$$

Elements on the section F at A and B were both, according to the assumption of the calculation, in states of pure shear, with the greater of the two being 237 MPa. With a material yield stress of 510 MPa the shear yield being taken as 0.58 of this value, which was 295 MPa, the design passes the strength criteria.

5.4. Chassis Design Considerations

5.4.1. The Chassis Design Outline in Context

The chassis was regarded as the ‘glue’ of the design. The author felt that this was appropriate as it was that structural element which housed the bearings which supported the shaft and the shaft had the arms rigidly attached to it. The chassis was also that structural element to which the 4 Arm Gripper was attached to the hoisting unit. It was also that element which would provide for attachment of the gripping actuating system.

One can therefore understand that the chassis had to be suitably designed for strength and rigidity and for the attachment of the subsystems mentioned. The latter of these two will be discussed in the chapter on Design for Manufacture and Assembly. The actuation system as well as the hoist selection have also been treated in subsequent sections. The focus of the present section is therefore the design for strength and rigidity. The reader should bear in mind that the design for strength and the design for subsystem attachment, assembly and manufacture was not treated in isolation but treated in the spirit of total design.

The design for strength and rigidity was conducted with the finite element software and stress values produced by software at critical sections were verified.

5.4.2. The Chassis Structural Layout

Figure 5.4.1. shows a skeleton structural layout. The geometry of the chassis had to satisfy the requirements for strength and mass.

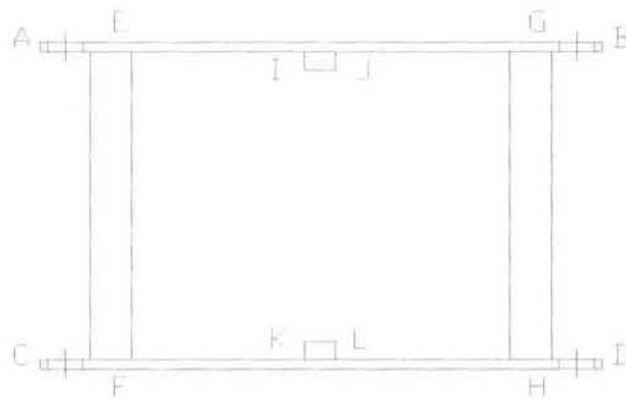


Figure 5.4.1. The chassis geometry

Table 5.4.1. Shows the possibilities for the structural sections that were investigated using the software. The following were noted in the geometrical layout :

- provision had to be made for 4 bearing holes, two per member - AB,CD,
- the thickness of members AB and CD must be greater than 19 mm as dictated by the bearing width,
- the members IJ and KL were to provide for the attachment of the 4 arm gripper to the hoist. Their length was determined by the pulley housing arrangement associated with a 4 rope fall. The shortest distance separating IJ and KL was dictated by the pulley housing to be 630 mm,
- consideration had already been given to the distance AC and BD in the section on shaft design. This distance was found to be 800 mm,
- the distance AB and CD , was influenced by the width of the flat car and timber pallet,

- provision had to be made for the attachment of the actuating system, shown as protrusions from the otherwise rectangular front view in Figure 5.4.2.

Table 5.4.1. Options for structural cross sections for the chassis members

Member	Section 1	Section 2	Section 3
AB and CD	Rectangular (Plate)		
EF and GH	Square Tube	Round Tube	I Section
IJ and KL	Square Tube	Round Tube	I Section

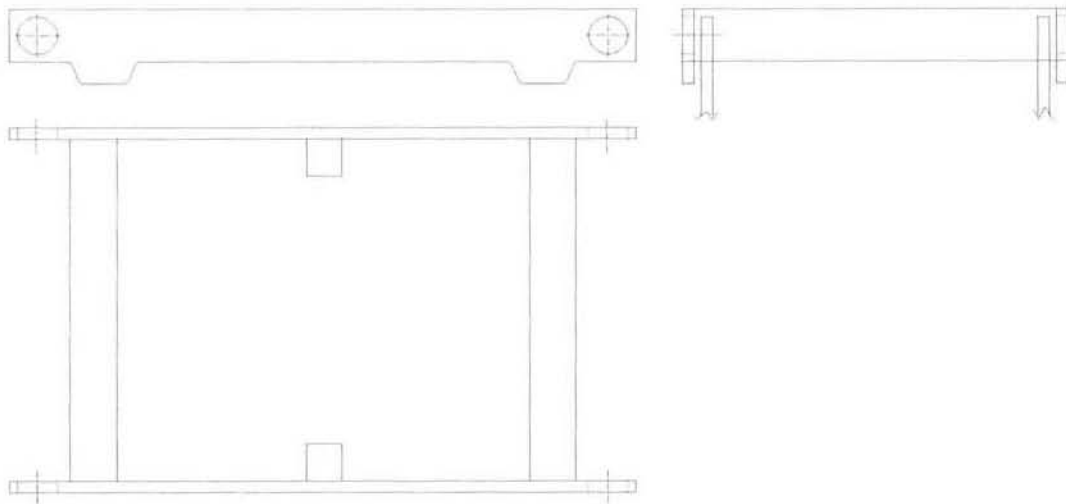


Figure 5.4.2. The basic chassis configuration in left, top and front view

5.4.3. Computational Finite Element Approach To Design

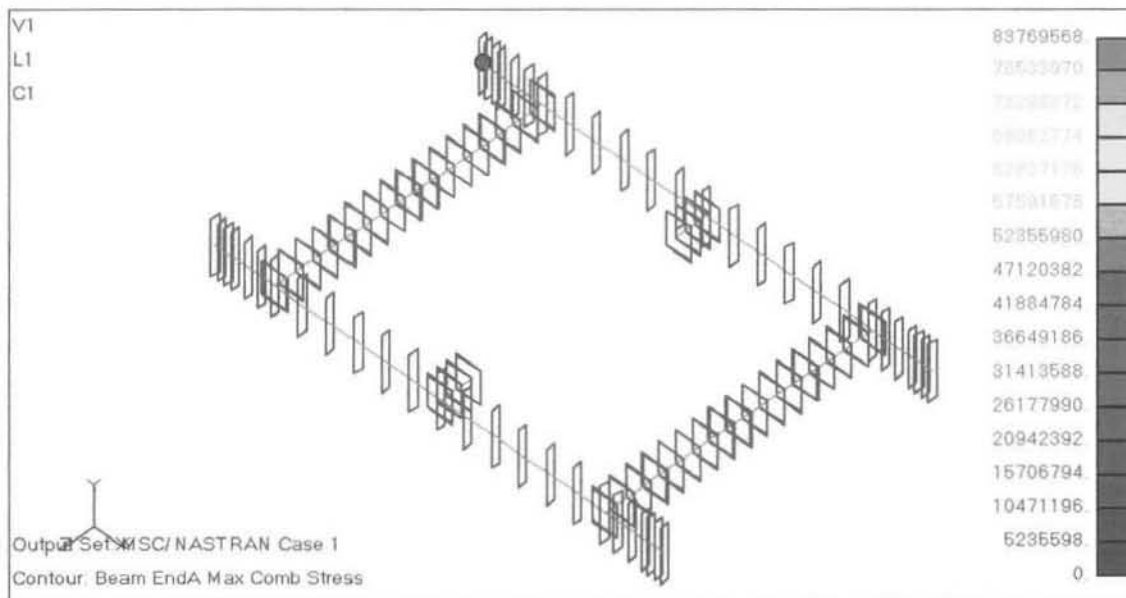
The approach adopted by the author was twofold. Firstly, to utilize beam elements and to treat the chassis as a frame structure. The second modeling approach utilizing plate elements, was used as the beam model verification as well as to model the favorable characteristics of various structural components as identified when modeling was carried out using beam elements.

Modeling using beam elements.

The author modeled the chassis using beam elements with the following in mind :

- The load exerted by the actuator (weight) was small in comparison to the timber, therefore, the model presented omitted the protrusion for the attachment of the actuator. The author did in fact verify that the omission was justified.
- The chassis would be rigidly attached to the pulley housing of the hoist, and therefore the model assumed that there was no degree of freedom at points I through L. The points A through D (Figure 5.4.1.), were be attached independantly to the pulley housing through a structural tie. The author desired to investigate the effect of utilizing only the former of the two constraint systems as the latter was designed primarily a precautionary measure.
- Many combinations of structural constituents were investigated. With reference to Figure 5.4.1. member AB, EF, IJ and their symmetrical counterparts will hereinafter be referred to as the member M1, member M2 and member M3, respectively.

5.4.3.1. Option 1



Note: The scale (Pa) applies to the stress contour plot which is depicted in Figure 5.4.4

Figure 5.4.3. Structural cross sections for option one

Figure 5.4.3. serves to depict the cross sectional make up of option 1. The contour plots for the stress and deflection , together with their respective scales appear in Figure 5.4.4. and Figure 5.4.5. Option 1 had the following cross sectional make up : (dimensions in mm)

Member 1	:	rectangular	:	height 125
				width 25
Member 2 and 3	:	square tube	:	side 76
				thickness 3

The following is a maximum stress contour plot obtained from the computational analysis. The author was aware that utilization symmetry was possible in modeling. However, as a full model using beam elements requires relatively

minute number of nodes as compared to other modeling elements, the author sought to use a full model for the graphic advantage of the reader. The use of symmetry was however used in the plate models. The exaggerated deflection contour appears in Figure 5.4.5.

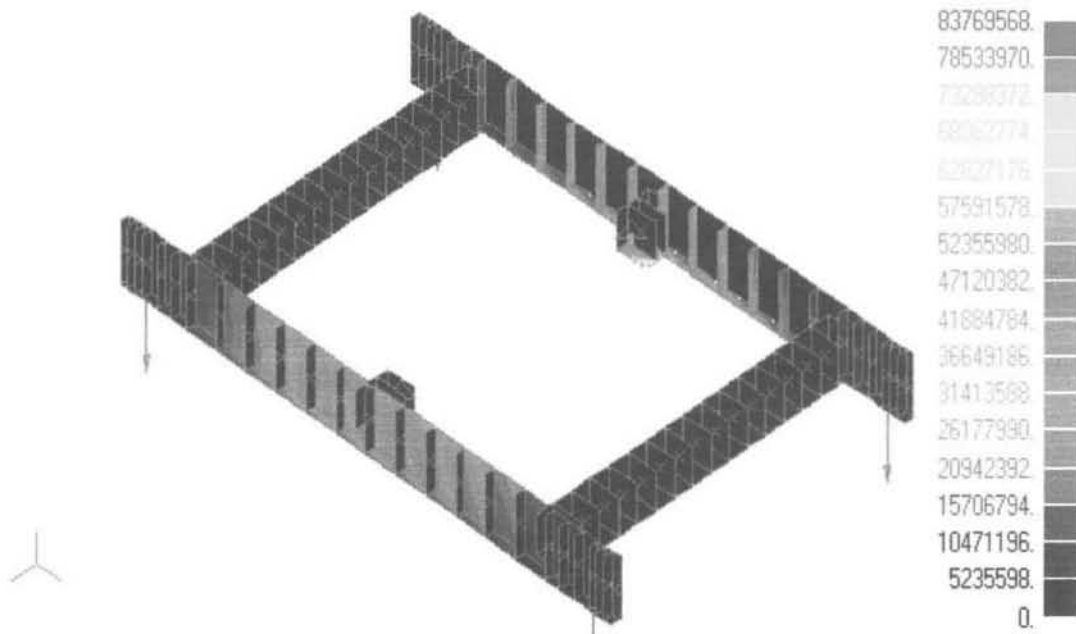


Figure 5.4.4. Maximum stress contour plot (unit : Pascals)

As is evident from the stress contour plot in Figure 5.4.4 the maximum stress is concentrated at the center of M1 and at the end of M3. The value of the maximum stress which occurred at the end of M3 was computed to be 83 MPa. The deflection of less than 1 mm was well within that which was expected.

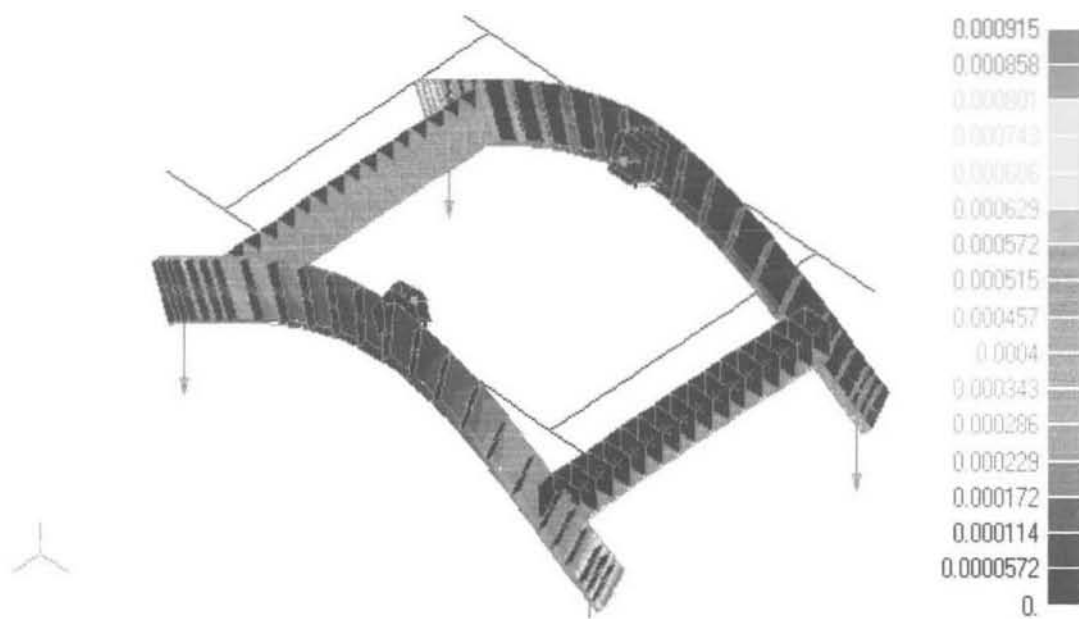


Figure 5.4.5. Deflection of the chassis frame (unit : metres)

5.4.3.2. Option 2

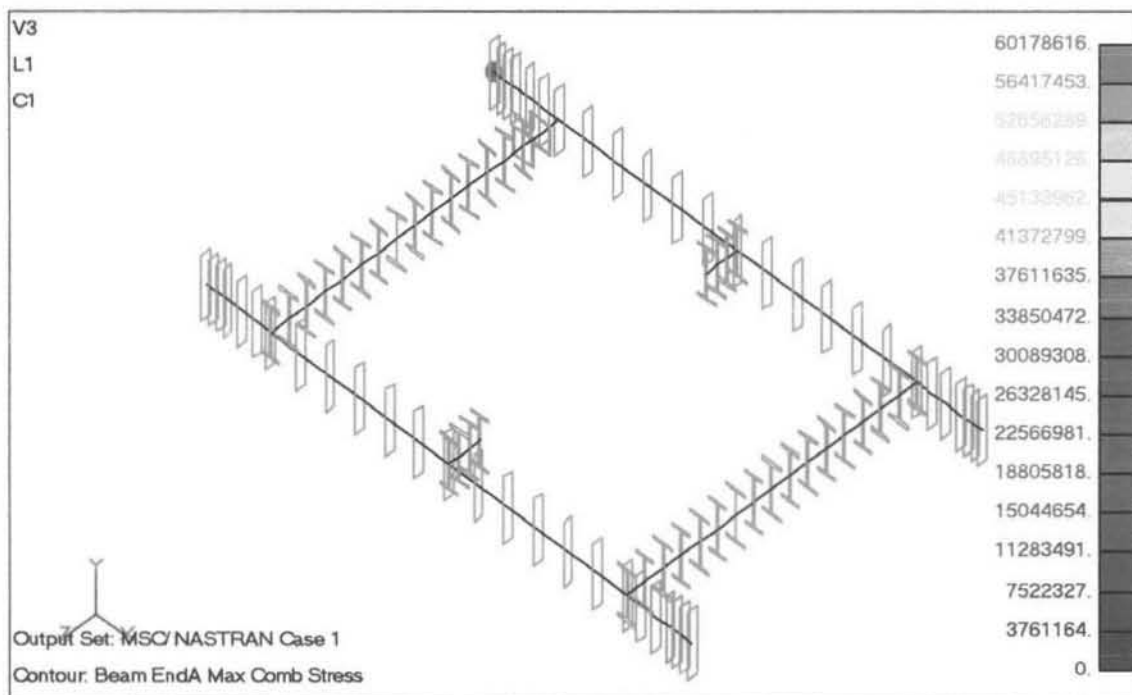


Figure 5.4.6. The structural cross section for option 2 (unit : Pascals)

Figure 5.4.6. depicts the structural make up of the chassis for option two. The dimensions, in millimeters, for the different sections are listed below.

Member 1	:	rectangular	:	height 125 width 25
Member 2 and 3	:	I section	:	height 100 width 50 web and flange 5

The analysis produced the result that was a reduction in the maximum stress from option 1, to a value of 60 MPa. The general stress contour was similar to that in Figure 5.4.4. with the exception that the maximum stress occurred just before the point of contact between M1 and M3. The author reserved the displaying of the stress contour plot for this option as the plate model that was employed would be used for the purpose, however, the maximum stress contour scale, is available in Figure 5.4.6. The maximum defection remained essentially the same. The option viability was however evaluated in the light of design for manufacture and assembly discussed in Chapter 6.

5.4.3.3. Option 3

Member 1	:	rectangular	:	height 125 width 25
Member 2 and 3	:	square section	:	side 50 thickness 3

The maximum stress was found to at the fixed constraint, the point of attachment to the hoisting unit, to be 133 MPa. The increase in the stress could have been

attributed to the decrease in the section modulus of M3 and M2.

5.4.3.4. Option 4

Member 1 : rectangular : height 125
width 20

Member 2 and 3 : square section : side 50
thickness 3

With a decrease in the moment of inertia of M1 as a result of the decrease in width the stress pattern remained essentially the same with a slight overall increase in the stress levels. The maximum stress was found at the end of M3 to be 137 MPa. Again the viability of the model was examined in the light of total design as ease of manufacturing and assembly were key specifications.

5.4.3.5. Option 5

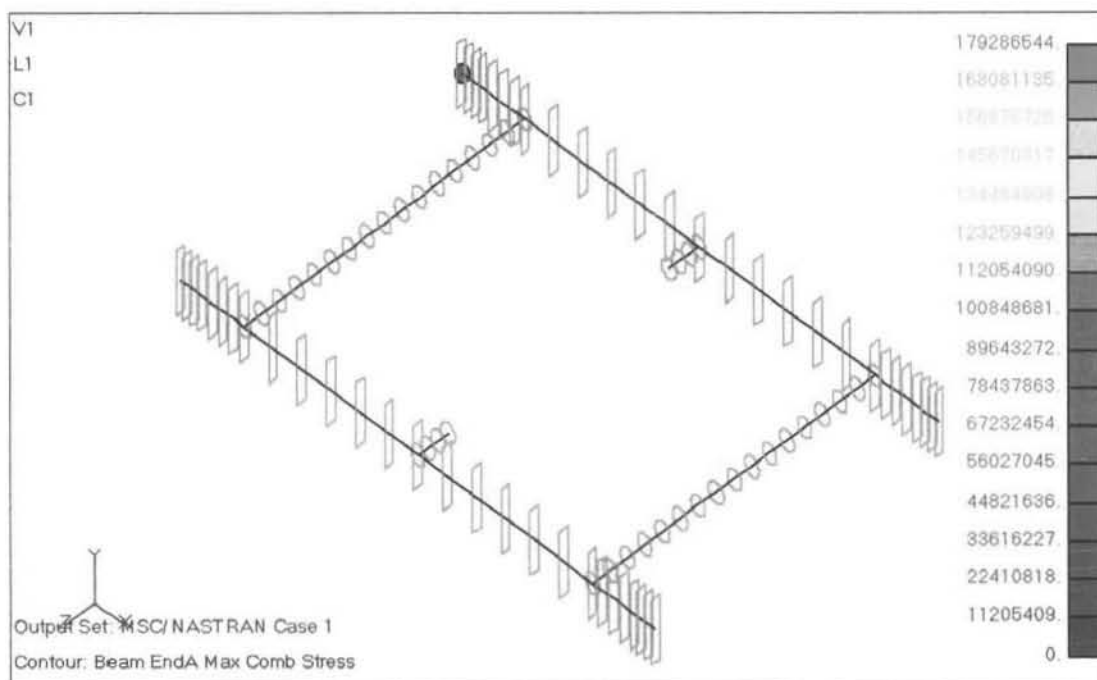


Figure 5.4.7. The structural cross section for option 5 (unit : Pascals)

Figure 5.4.7. is only a graphical description of the structural cross sections in option 5. The dimensions, in millimetres, for option 5 are listed below. The scale shows the stress values in Pascals.

Member 1 : rectangular : height 125, width 20

Member 2 and 3 : round section : diameter 40

The results of the analysis on option 5 produced results that were much higher than any of the other options. A maximum stress of 180 MPa was computed at the end of M3.

5.4.3.6. The Employment of Plate Elements for Modeling

Options 1 through 5, were verified successfully using plate elements. The author has modeled the following two options with plate elements. From the above computations, attractive aspects of the models were selected to be modeled using plate elements. As mentioned the model made use of two planes of symmetry and was modeled as a quarter of the actual chassis. The plate model had the following cross sections

Member 1 : rectangular : height 125
width 25

Member 2 : square : side 50
thickness 3

Member 3 : I section : height 100
width 50
web and flange 5

The resulting maximum von Mises stress contour plot appears in Figure 5.4.8.

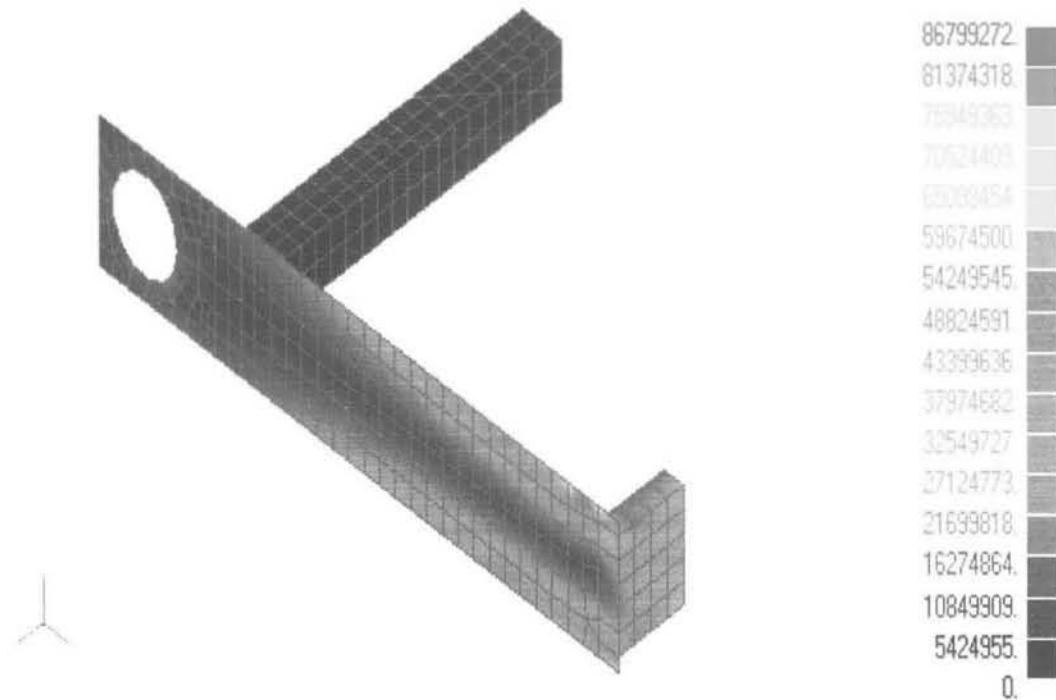


Figure 5.4.8. Maximum von Mises stress contours (unit : Pascals)

The result of the static analysis was that a maximum stress, indicated by the red contour in Figure 5.4.8 was found to be 87 MPa. This was a relatively small area and most of the material was stressed to an average value of 60 MPa. The deflection was very much the same as all of the other models (<2 mm).

The author then investigated changing the I - section at the center to a 100 nominal side square tube of 5 mm thickness, using plate elements. The computational output was a stress reduction to 65 MPa. The stresses were concentrated in the same positions as in Figure 5.4.8. However, a larger area, relative to that in Figure 5.4.8. was exposed to the stress concentration.

The designer also sought to look at the outcome of the analysis when the structural tie, was included in the design. The model whose analysis produced a maximum stress of 65 MPa, appears in Figure 5.4.9. As mentioned, the use of the ties or possibly wire rope, would act as a safety measure in the event of the joint between the chassis and pulley housing failing. This was considered to be justified by the fact that all the models tested identified this area as one of high stress concentration.

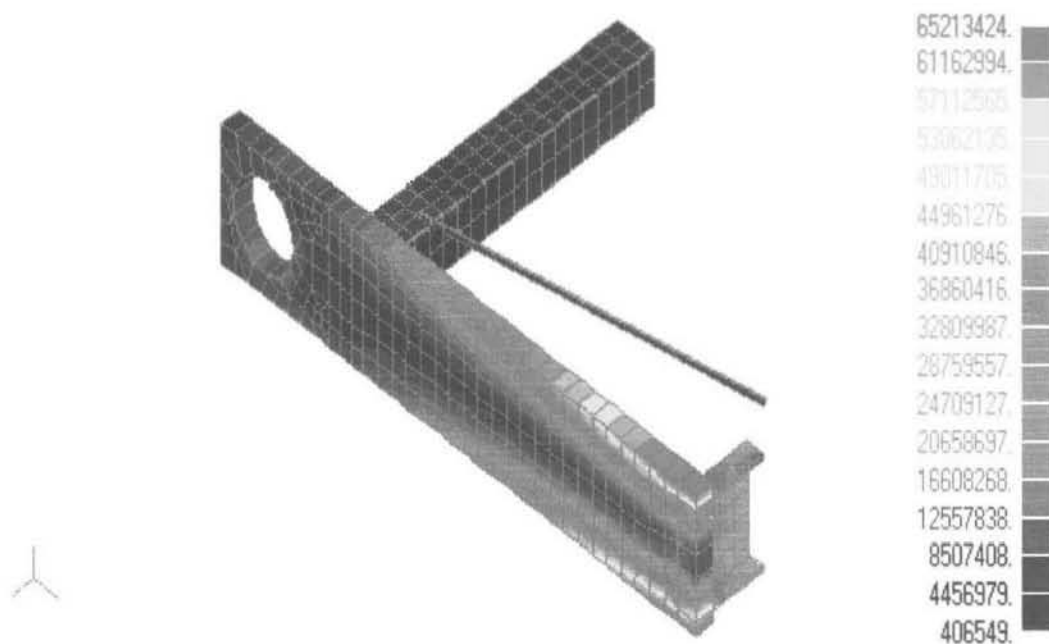


Figure 5.4.9. Plate model with tie (unit : Pascals)

5.5. Arm Actuation : design of a pneumatic motion system

5.5.1. Overview

Pneumatics can be described as the application of pressure resulting in the flow of energy. The actuation of the arms consisted of 4 actuating cylinders mounted and attached appropriately to the chassis and powered by a single pneumatic system. The system was designed to displace the arms by a predetermined angular value.

The four cylinders are connected in parallel to the pneumatic control circuit and was to be supplied with readily available compressed air (compressed air is available in an underground mining environment for pneumatic drilling equipment, and the like). Air, available at 5 bar, was to be tapped from the running pipe supply and connected to the chassis by a hose and a quick release coupling.

5.5.2. System Design Considerations

Two constraints of the mining environment dictated that two angular positions be made possible. The vertical height of the tunnel was one constraint. This was influenced by the height of the timber above the foot wall (ground) and the head room required by the hoisting unit. Therefore, in order for the gripper to have fitted within the bottom most point of the hoist and the uppermost point of the timber pack with a given clearance, the arms need to be actuated so that the overall height of the gripper (arms, shaft and chassis) would have decreased. A pictorial representation is available in Figure 5.5.1. The first angular displacement required if the arm was calculated to be 55° . This angle would result in a reduced height of 775 mm. The standard stroke length of 3 inches (76.2 mm), for a 32 mm

diameter cylinder was selected. The sections to follow outline the attainment of these values.

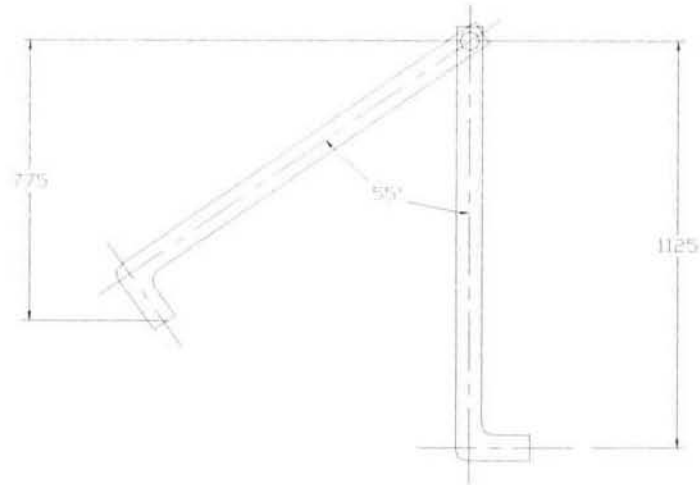


Figure 5.5.1. The arm angular displacement to reduce height



Figure 5.5.2. The displacement for the width constraint

The stacking arrangement of the timber packs in the timber bay was intended to be double row single stacked. After the location and placement of the pallet in the

bay, it was necessary to actuate (displace) the arms to the second angular value so as to effect the pallet's release and the lifting of the gripper above the timber pack. The width of the timber bay, acted as the second constraint impinging on the actuation, (depicted in Figure 5.5.2.) The suitable value of the second angle of actuation was found to lie between 15° and 17° .

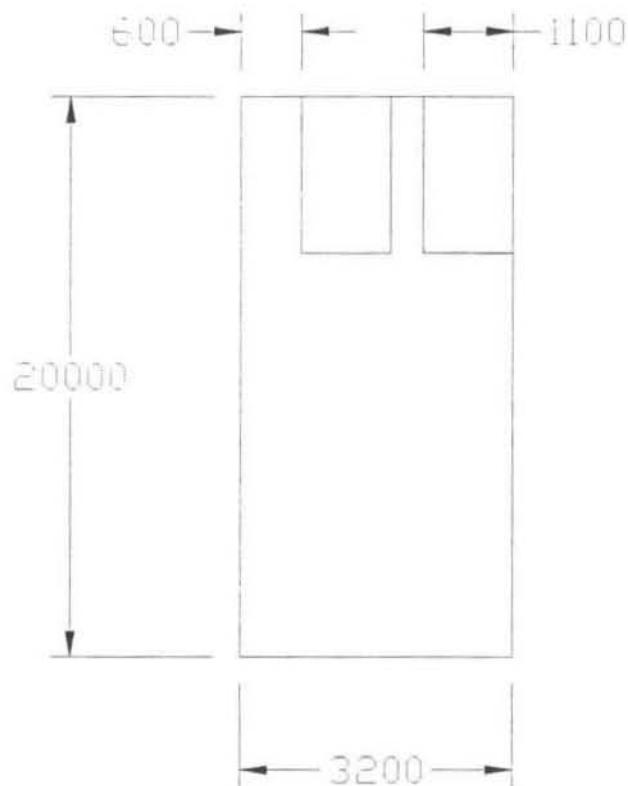


Figure 5.5.3. Timber bay showing pallet arrangement (dimension : mm)

The angle range was calculated to be those angles to allow for the effective release of the pallet and at the same time carry out this angular displacement within a constraint which dictated that the horizontal displacement of the arm should be less than 600 mm.

5.5.2.1. The Actuating Torque

In order to effect the displacement an object with mass by an angular amount, a torque must be applied to the body. The rotational analogue of Newton's Second Law relates this torque to the second time derivative of the angle. Mathematically this is expressed as :

$$T = I \frac{d^2 \theta}{dt^2} \quad (5.5.1)$$

where :

T = the torque, Nm

I = the moment of inertia about the axis of rotation, kg.m²

θ = the angle of rotation, radians

t = the time over which the torque acts, s

The solution to the equation was obtained using the technique of Laplace Transforms to solve second order differentials. The solution, which appears below, produced the following relationship with the initial condition that the initial angle is zero at time equal to zero.

$$T = \frac{2\theta I}{t^2} \quad (5.5.2)$$

Firstly rewriting the equation as

$$\frac{T}{I} - \frac{d^2 \theta}{dt^2} = 0 \quad (5.5.3)$$

And then as

$$\frac{d^2\theta}{dt^2} - k = 0 \quad (5.5.4)$$

where $k = \frac{T}{I}$

Transforming the equation into the s domain, using the Laplace transform on a second order differential equation.

$$L\left[\frac{d^2\theta}{dt^2} - k\right] = s^2\theta(s) - s\theta(0) - \frac{d\theta}{dt}\bigg|_{=0} - \frac{k}{s} = 0 \quad (5.5.5)$$

With the initial angle and angular velocity equal to zero,

$$\theta(s) = \frac{k}{s^3} \quad (5.5.6)$$

Transforming the equation back into the time domain,

$$\theta(t) = \frac{kt^2}{2} = \frac{Tt^2}{2I} \quad (5.5.7)$$

for which,

$$T = \frac{2\theta I}{t^2} \quad (5.5.8)$$

The solution could have also been attained by method of direct integration for a constant torque. The derivation presented was preferred as it is general and could be applied to the case when the torque is not a constant.

5.5.2.2. The Moment of Inertia

The moment of inertia of the arm was an output available from the finite element package after the geometry had been served as an input. The author has, however,

sought to verify this result. The calculation made use of the geometrical simplification as depicted in Figure 5.5.4. The moment of inertia of a rectangular prism about an axis through the center of gravity and perpendicular to a face whose dimensions are a and b is given by

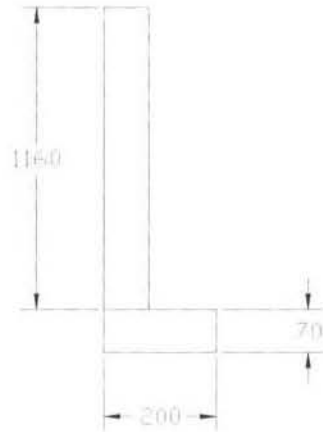


Figure 5.5.4. The simplified arm model for the moment of inertia calculation

$$I = \frac{M}{12}(a^2 + b^2) \quad (5.5.9)$$

where

I = the moment of inertia, kg.m²

M = the mass of the prism, kg

a = the length, m

b = the breadth, m

The moment of inertia about the axis of rotation was obtained from the parallel axis theorem and was expressed mathematically as

$$I_0 = \frac{M}{12}(a^2 + b^2) + Md^2 \quad (5.5.10)$$

where

I_0 = the moment of inertia about the axis of rotation, kg.m^2

D = the distance between the mass center and the axis of rotation, m

The simplified model was split into two rectangular prisms with a depth of 25 mm. The material used was steel with a density of 7849 kg/m^3 . The relevant values for the calculation appear in Table 5.5.1.

Table 5.5.1. Data for the moment of inertia calculation

	Length (m)	Breadth (m)	Mass (kg)	Inertia about mass center (kg.m^2)	Inertia about axis of rotation (kg.m^2)
Prism 1	1.090	0.070	14.972	1.488	5.935
Prism 2	0.200	0.070	2.747	1.028E-02	3.498

The effective moment of inertia of both prisms about the axis of rotation is therefore 9.433 kg.m^2 .

5.5.3. Application of a Pneumatic System to the Gripper

The author explored many options for the actuating mechanism. All of the alternatives presented in the morphological chart in Chapter 3 were given the required attention. The application of a pneumatic system had within itself a host of options. The author has presented options that were considered to have the appropriate merit. The first option employed a dedicated electronic controller to control the pneumatic circuit. The following sub section outlines the methodology and the calculations that were used to arrive at the first option.

5.5.3.1. Option 1 : The employment of an electronic controller in a pneumatic circuit

As mentioned the values quoted above bore testimony to much iteration through calculation. ISO standard pneumatic cylinders available commercially were found to have bores with diameters of ranging from 32 mm to 125. After the initial calculations the 32 mm diameter piston was found to be suitable.

The useful piston area was calculated

$$\begin{aligned} A &= \frac{\pi}{4} d^2 = \frac{\pi}{4} (32 \times 10^{-3})^2 \\ &= 8.0427 \times 10^{-4} \text{ m}^2 \end{aligned} \quad (5.5.11)$$

The theoretical force available was

$$\begin{aligned} F_{th} &= P \times A \\ &= 5 \times 10^5 \times 8.0427 \times 10^{-4} \\ &= 402.123 \text{ N} \end{aligned} \quad (5.5.12)$$

The effective force, taking into account the friction, was

$$\begin{aligned} F_e &= P \times A - 0.1 \times (P \times A) \\ &= 361.911 \text{ N} \end{aligned} \quad (5.5.13)$$

This value was verified using the Figure B2 in Appendix B. This force was calculated , for the standard 3 inch stroke and 55° angle, to act at 82.512 mm (the torque arm) from the axis of rotation, and therefore the torque was calculated as

$$\begin{aligned}
T &= Fd \\
&= 361.911 \times 82.512 \times 10^{-3} \\
&= 29.86 \text{ Nm}
\end{aligned}
\tag{5.5.14}$$

From equation (5.5.8) the time of over which the torque acted was calculated.

$$\begin{aligned}
t^2 &= \frac{2\theta I}{T} \\
t &= 0.8 \text{ s}
\end{aligned}
\tag{5.5.15}$$

This value could be altered through the use of control valves. To increase the speed quick exhaust valves were available. To decrease the speed, undersized valves could have been used. Stroke time control could have also been instituted by employing a pneumatic circuit. The author has shown how this can be done in Figure 5.5.11.

Four cylinders would be used each with a stroke of 76.2 mm and a bore diameter of 32 mm. The piston rod diameter was also given attention. The allowable stroke on a piston rod is limited by the buckling load. The load must not exceed certain limits depending on the stroke and the rod diameter. The buckling load diagram is available in Appendix C. This diagram was used to extract a value of 6 mm for the rod diameter that would be more than sufficient to prevent the rod buckling under a load of 360 N with the aforementioned stroke.

The cylinder was to be a purchased component. The author found that there is an immeasurable reserve of information available in vendor catalogues. Not only for the pneumatics but also for other aspects of design technology.

With the information specified for the cylinder as

Bore	:	32 mm (1 ¼ in)
Stroke	:	76.2 mm (3 in)
Piston rod diameter	:	6 mm
Cylinder Speed	:	< 1 m/s
A tolerance of	:	< 2 mm is usual for a cylinder of this diameter and stroke

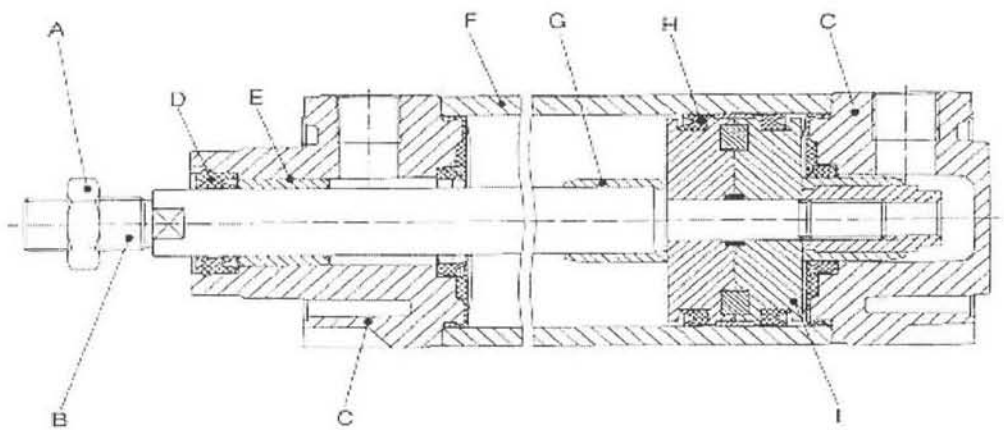


Figure 5.5.5. The cylinder components

The cylinder in Figure 5.5.5. had the following makeup. A piston rod nut (A); piston rod (B); bearing and end caps (C); piston rod seal (D); piston rod bearing (E); cylinder barrel (F); cushioning piston (G); piston seal (H) and piston (I).

As mentioned speed control of pneumatic cylinders was of as much concern to the author as to designers throughout the world. The problem may be alleviated if not eliminated by the employment of electro-pneumatic valve systems. With the speed limited to a maximum of 1 m/s the air did not have to be lubricated. A maximum speed of 1 m/s was sufficient for the purpose of design. The author has shown how speed control can be instituted with option 2.

To facilitate the actuation of the arms the cylinder was designed (selected) to have a rod end piece that would allow for the attachment to the arms. The end piece would be of the rod clevis type with modification that a rod run through both arms with a rod end pieces at the each end of the clevis rod. An important design consideration was to minimize side loading on the cylinder. Side loading on the cylinder results in premature wear on the cylinder seals. To cater for the rotational effect of the pneumatic cylinders the mounting to the chassis was designed (selected) to be of the swivel flange type. The cylinder chosen for the application together with the rod end attachment and the mounting is shown in Figures 5.5.5 through 5.5.8. The author verified that the components selected for purchasing were in fact readily available.

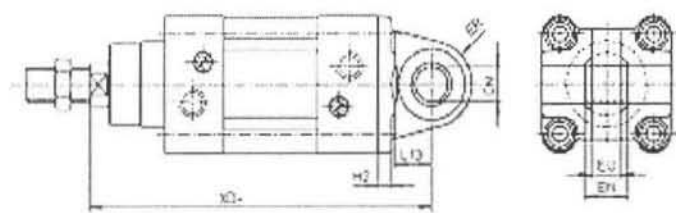


Figure 5.5.6. The cylinder layout

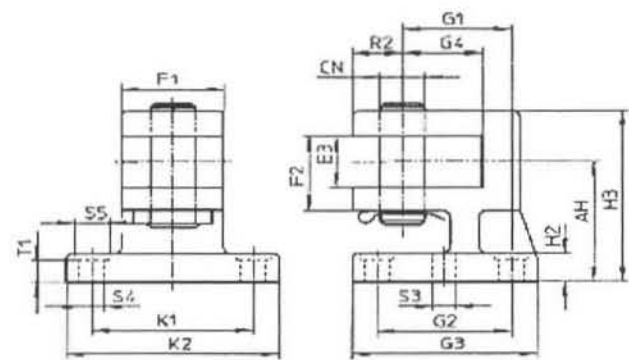


Figure 5.5.7. The clevis mounting attachment (dimensions in Table 5.5.2)

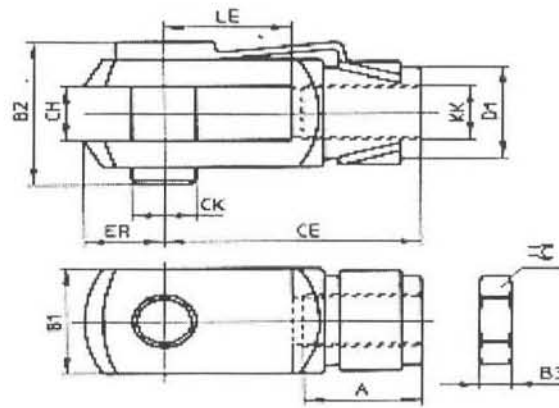


Figure 5.5.8. The piston rod end piece (dimensions in Table 5.5.2)

As discussed above, it was necessary to institute position control into the pneumatic system. Traditionally, pneumatic systems were not used with any reliability for this purpose due to system compliance. However, modern valve technology and control systems do permit the implementation of position control provided that the actuator is not required to hold a position against alternating loads.

The pneumatic circuit had to firstly carry out a full stroke linear displacement and then hold another position away from the mechanical end stops of the cylinder. To design the control system the following was borne in mind. For the first option, the initial idea of the design employed of two cascaded circuits, each of which were responsible a single position. The design approach adopted by the author was to design the two circuits independently and then to integrate them by using the appropriate valves and sequencing. The first circuit for this initial design of option one made use of two valves for the full stroke length, V1 and V2, to drive the double acting cylinder such that the piston moved out when valve V1 was actuated and remained stationary at the forward end position until the reverse signal for the return movement was applied via valve V2.

The circuit is represented by Figure 5.5.10. The operation was as follows. When valve V1 was actuated, the 4/2 way valve, V3, was switched over due to the signal at P1. The piston moves forward. It would remain there until valve V3 was piloted by valve, V2, for the return stroke.

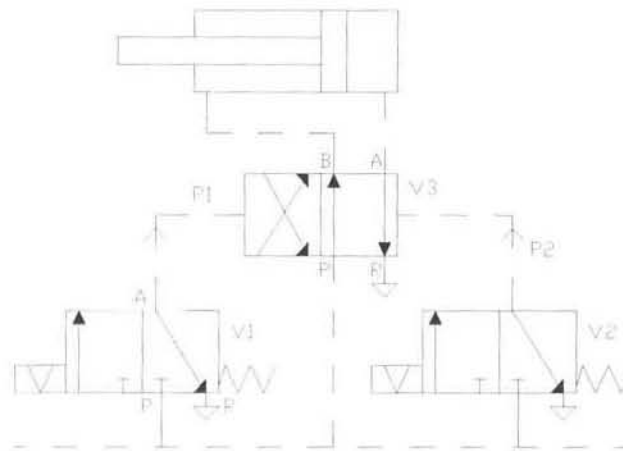


Figure 5.5.10. Circuit to drive the full stroke of actuation

The second circuit was similar to the one in Figure 5.5.10. With the following exception. Valve V2 would be actuated by a signal produced by a proximity switch located on the cylinder. The sequencing would have been made possible by the use of 'or gates' in pneumatics which are commonly known as shuttle valves.

Another circuit that the author investigated, which potentially could have replaced that in Figure 5.5.10. was one which made use of a pneumatic timing unit. This circuit would provide for the actuation of the cylinder resulting in the forward movement of piston. The piston would return after a preset time, as dictated by a throttle relief valve and a piloted 3/2 valve. Figure 5.5.11. is a manifestation of this description. However, since this was essentially an open loop circuit its validity for application was questionable.

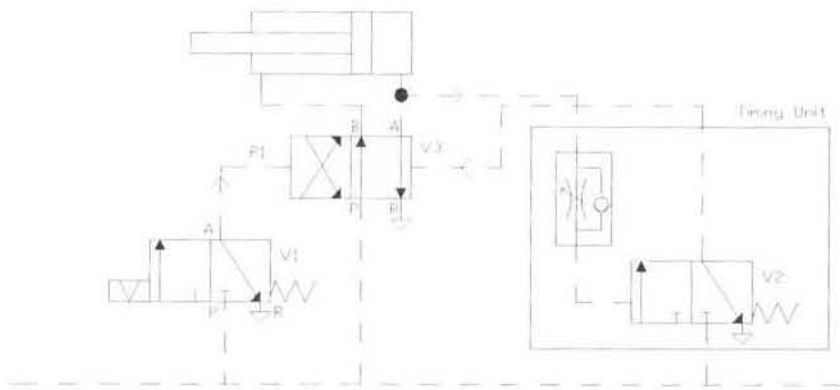


Figure 5.5.11. A timing control circuit

It became apparent that the employment of a dedicated controller circuit was a lucrative design, primarily because of the reduction in the pneumatic network by replacing two circuits with one. The circuit required a control loop incorporating position sensing and a rapid acting valve. The circuit is available in Figure 5.5.12. It depicts the circuit for one cylinder. As mentioned the four cylinders were designed to be connected in parallel and therefore the circuit would have also applied to the 4 cylinder combination. If an increase in the externally applied load occurred, then the valve would act to permit an additional mass of air to flow into the cylinder. In doing so, the pressure is increased at constant volume and therefore the position is maintained.

Single stage spool valves with direct acting solenoids are available commercially which can respond within a few milliseconds. They were to be placed as close as possible to the actuator avoid the problem of compliance in the piping network mentioned above.

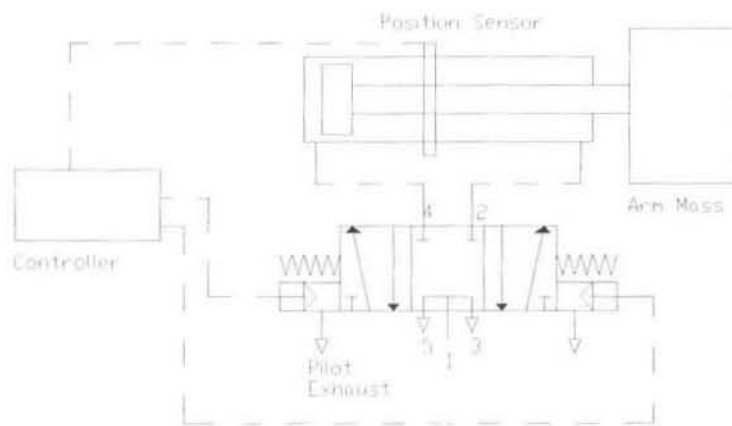


Figure 5.512. Position control circuit with dedicated controller

The valve is a 5/3, 5 ported 4 way 3 position double solenoid piloted valve. The operation required was that of a closed center valve. This valve stops the double acting cylinder by blocking the exhaust air from both ends of the cylinder, and so holding the cylinder under pressure.

The orifices are numbered one to five, with the supply port (1= P), the outlet ports (4,2 = A,B) and the exhaust ports (5,3 = R, T).

A proximity sensor was desired for non-contact position indication. It fits directly onto a cylinder which has a sensor groove. Both the sensor and the cylinder of this design are readily available commercially. The principle of operation of the sensing unit, was that when the proximity sensor enters a magnetic field (possibly a permanent magnet on the piston of a cylinder), it outputs an electrical signal. The status of the sensor is often indicated by a LED (light emitting diode). When the sensor is actuated, the LED is on. The sensors of this type are protected against reverse polarity.

5.5.3.2. Option 2: **Bi-positional pneumatic system without a dedicated controller**

The author verified through consultation with pneumatic vendors that the circuit in Figure 5.5.12. was a plausible circuit. However the specifications for minimum system complexity would be violated somewhat by the employment of such a circuit in a mining environment. The specification for a system that was rugged, easy to operate and still provided the required functionality were addressed. This manifested itself in the following design.

The author desired to place two cylinders back to back. Each cylinder would be driven by its own valve so that actuation of two separate positions would be achieved as simply and as functionally as possible. The idea is depicted in Figure 5.5.13.

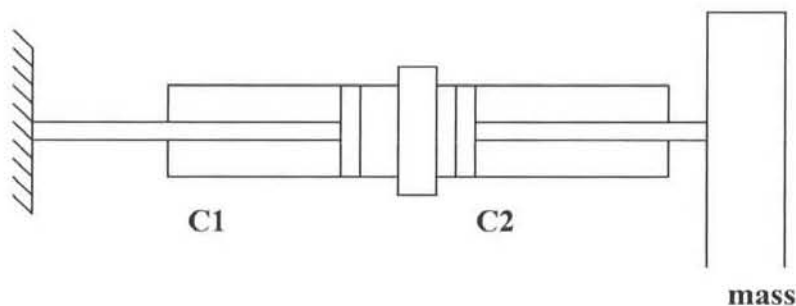


Figure 5.5.13. The back to back cylinder configuration

The figure shows two cylinders, when the one on the left exerts a forward force on a fixed support it results in a displacement in the opposite direction. This was the desired solution as it fulfilled the simplicity and functionality requirement more than adequately. The arrangement was in fact capable of 4 positions.

Assuming that

- Cylinder C2 was shown to require a stroke of 25.4 mm (1 inch) to displace the arm by 17°.

For the first position (55°), cylinder one would be actuated via a 5/2 solenoid valve, to release the position back to the initial position one would de-energise the valve. For the second position cylinder two would be operated in the a similar manner.

Each of the solenoid valves operate its respective cylinder. Each system was basically a mirror image of the other with the exception that the cylinder on the

right has the outlet ports connected to opposite cylinder ends. This was designed so that energizing the right solenoid on both valves resulted in displacement toward the right. A pair of one way flow regulators were included to control the speed of the piston rod movement. The author has adopted the conventional manner of speed control which is to restrict the exhausting air. This would allow for full power on the driving end which would work against the back pressure.

Cylinder C1 was designed to be attached to the chassis using the mounting in Figure 5.5.7. and would employ the rod end piece, see Figure 5.5.15. that would mate with the mounting in Figure 5.5.7. The double cylinder arrangement is shown in Figure 5.5.16. Cylinder C2 would employ the rod end piece in Figure 5.5.7 and attachment to the arms would be as described.

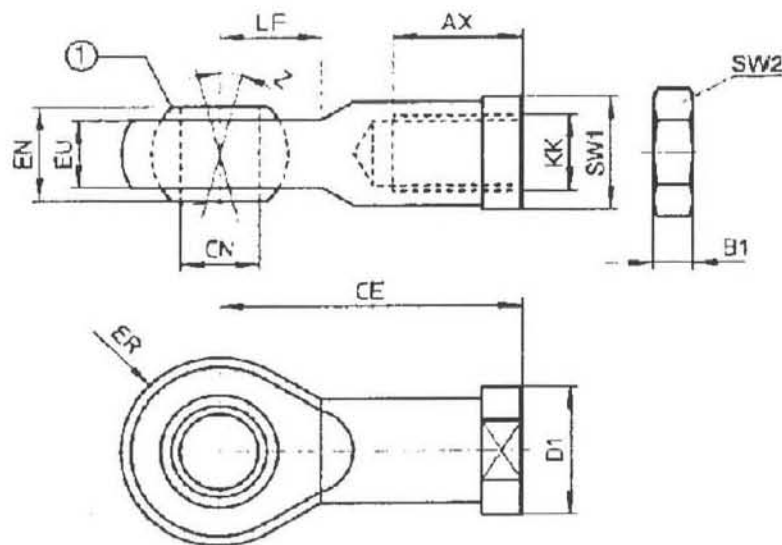


Figure 5.5.15. The rod eye end piece for cylinder C1 (dimensions in Table 5.5.2)

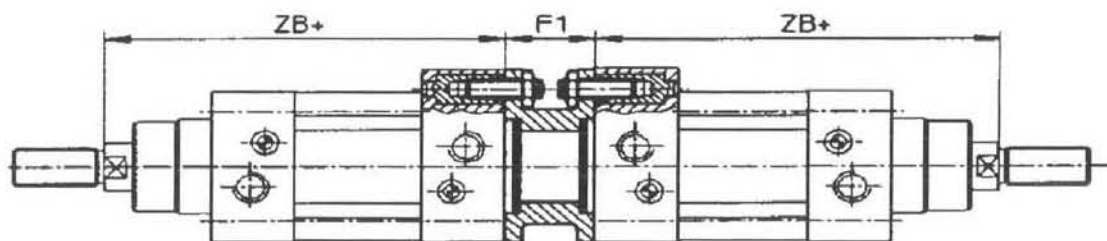


Figure 5.5.16. The double cylinder showing connection adapter

Table 5.5.2. Table of dimensions for cylinder, rod attachments, and clevis mount (dimension in mm)

(a) with reference to Figure 5.5.16

ZB	F1	Stroke 1	Stroke 2
120.00	27.00	76.20	25.40

Note $ZB + = (ZB + \text{stroke})$

(b) with reference to Figure 5.5.15.

AX	B1	CE	CN	D1	EU	EN	ER	KK	LF	SW1
20	5	43	10	19	11	14	14	M10	15	17

(c) with reference to Figure 5.5.8

A	B1	B2	B3	CE	CM	D1	ER	LE
20	20	26	5	40	10	18	12	20

(d) with reference to Figure 5.5.7.

AH	CN	E3	F1	F2	G1	G2	G3	G4	H2	H3
32	10	14.2	25	22.1	22	29	45	18	10	45
K1	K2	R2	S3	S4	S5	T1				
40	56	12.5	4.8	6.6	11	9				

5.6. The Structural Design of the Gantry

5.6.1. Introduction

Seeing as how the load on the system was under three tons a single girder was considered to be sufficient. The member AB as the girder was identified as that member that would support the moving hoist and gripper. The author had opted proceed with the preliminary design of member AB (the girder) and then to the consideration of the other structural members.

5.6.2. The constraint

As it would have been in poor design judgment to produce the concept as a monolithic structure, the use of joints was considered to be appropriate. Common joints available to the designer are welding, riveting and bolting. The joints between the girder and the gantry headgear were designed to rigidly attach these members.

Prior to the modeling of the member AB, the following argument was noted. If the joints formed by welding, riveting or bolting were ideally stiff, then when the structural constituents of system were loaded and deformed, the angles between the members at the joint would not change. In practice this is usually false, as due to the elasticity of the system there are some change in these angles. However , it was assumed for the analysis to follow that the joint was rigid and able to transmit a couple.

5.6.3. The loading on the gantry

The crane girder has to support a load that moves. In this case from a design point of view it was important, to calculate the maximum stress resultants at a given

section in a member arising from all possible load positions on the girder. The designer desired to represent this information in the form of a graph, with the value of the bending moment at a *given section* on the y-axis and the corresponding position of the load on the x-axis.

Consider the following :

At any given instant let a load, **P**, be a distance, **a**, from the left hand support. The length of the beam was designated as **L**.

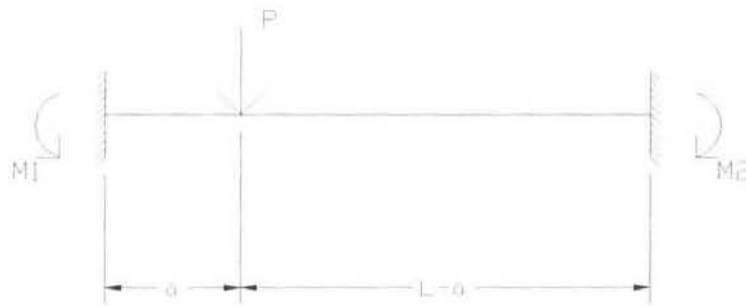


Figure 5.6.1. The girder model

Let the fixed end moments at the left and right hand supports be M_1 and M_2 respectively. Using standard beam diagrams and the method of superposition, the moments were calculated using the assumption that the rotation at the supports were zero. The result after releasing certain constraints of the model in Figure 5.6.1. was a simply supported beam. Table 5.6.1 is a summary of the information that was obtained from the standard cases for simply supported beams.

Let $b = L - a$,

Table 5.6.1. The Rotation at the ends of a simply supported beam

Load	Slope at 1	Slope at 2
P	$-Pab(L+b)/6L$	$Pab(L+a)/6L$
M1	$M1L/3$	$-M2L/6$
M2	$M2L/6$	$-M2L/3$

With the rotation at the supports equal to zero, i.e. The sum of columns one and two equated to zero and the pair of equations solved simultaneously,

$$M1 = \frac{Pab^2}{L^2}, M2 = \frac{Pa^2b}{L^2} \quad (5.6.1)$$

The moment at a distance x from the left hand support was

$$\begin{aligned} M_x &= -M1 + R1x - P\langle x - a \rangle \\ &= -\frac{Pab^2}{L^2} + \frac{Pb}{L}x - P\langle x - a \rangle \end{aligned} \quad (5.6.2)$$

where

$R1x$ = is the reaction at the point of application of $M1$ in the x direction

The use of Macaulay's bracket dictated that

$$\langle x - a \rangle = 0 \text{ if } x \leq a \quad (5.6.3a)$$

$$\langle x - a \rangle = (x - a) \text{ if } x > a \quad (5.6.3b)$$

Therefore for $x \leq a$

$$M_x = -\frac{Pab^2}{L^2} + \frac{Pb}{L}x \quad (5.6.4)$$

And the second case for $x > a$

$$M_x = -\frac{Pab^2}{L^2} + \frac{Pb}{L}x - P(x-a) \quad (5.6.5)$$

The reader should appreciate that the analysis is not only a standard bending moment calculation but attempts to calculate the moment at particular section when the load is not at one set position but moving. The author felt that the effort spent invested in the aforementioned analysis was worth while, as the result was used for the fatigue analysis. The sections of interest were at $x = 2.5$ m, 3.5 m and 4.5 m from the left hand support. A spreadsheet was used to calculate the moment at these sections as the load moved across the beam from $a = 0$ to 7 m; $P=24525$ N.

Table 5.6.2. Moment distribution at a section due to a moving load

x=2.5			x=3.5			x=4.5		
a	Mx		a	Mx		a	Mx	
0		0	0		0	0		0
0.25	-1759.61		0.25	-2635.5		0.25	-3511.39	
0.5	-2690.24		0.5	-4442.03		0.5	-6193.81	
0.75	-2838.83		0.75	-5466.51		0.75	-8094.19	
1	-2252.3		1	-5755.87		1	-9259.44	
1.25	-977.556		1.25	-5357.02		1.25	-9736.49	
1.5	938.457		1.5	-4316.9		1.5	-9572.26	
1.75	3448.828		1.75	-2682.42		1.75	-8813.67	
2	6506.63		2	-500.51		2	-7507.65	
2.25	10064.95		2.25	2181.912		2.25	-5701.12	
2.5	14076.85		2.5	5317.921		2.5	-3441.01	
2.75	12364.17		2.75	8860.595		2.75	-774.227	
3	11011.22		3	12768.01		3	2252.296	
3.25	9971.102		3.25	16978.24		3.25	5591.637	
3.5	9196.875		3.5	21459.38		3.5	9196.875	
3.75	8641.621		3.75	20028.23		3.75	13021.09	
4	8258.418		4	18769.13		4	17017.35	
4.25	8000.343		4.25	17635.16		4.25	21138.74	
4.5	7820.472		4.5	16579.4		4.5	25338.33	
4.75	7671.883		4.75	15554.92		4.75	23437.95	
5	7507.653		5	14514.8		5	21521.94	
5.25	7280.859		5.25	13412.11		5.25	19543.36	
5.5	6944.579		5.5	12199.94		5.5	17455.29	
5.75	6451.889		5.75	10831.35		5.75	15210.82	
6	5755.867		6	9159.439		6	12763.01	
6.25	4809.59		6.25	7437.269		6.25	10064.95	
6.5	3566.135		6.5	5317.921		6.5	7069.707	
6.75	1978.579		6.75	2854.472		6.75	3730.365	
7		0	7		0	7		0

The results in Table 5.6.2. were plotted and the following graphs obtained.

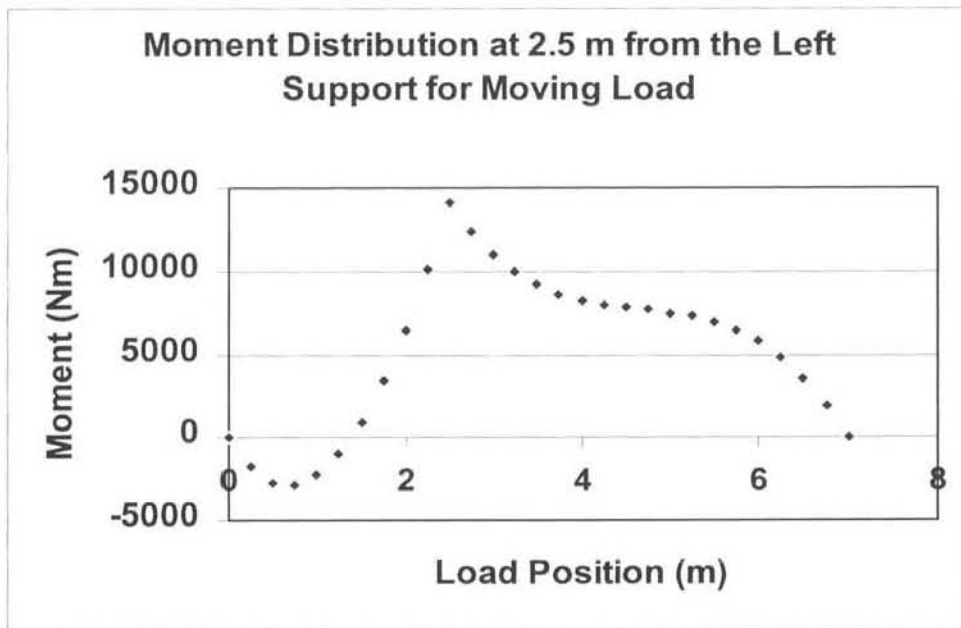


Figure 5.6.2. Moment distribution curve at 2.5 m

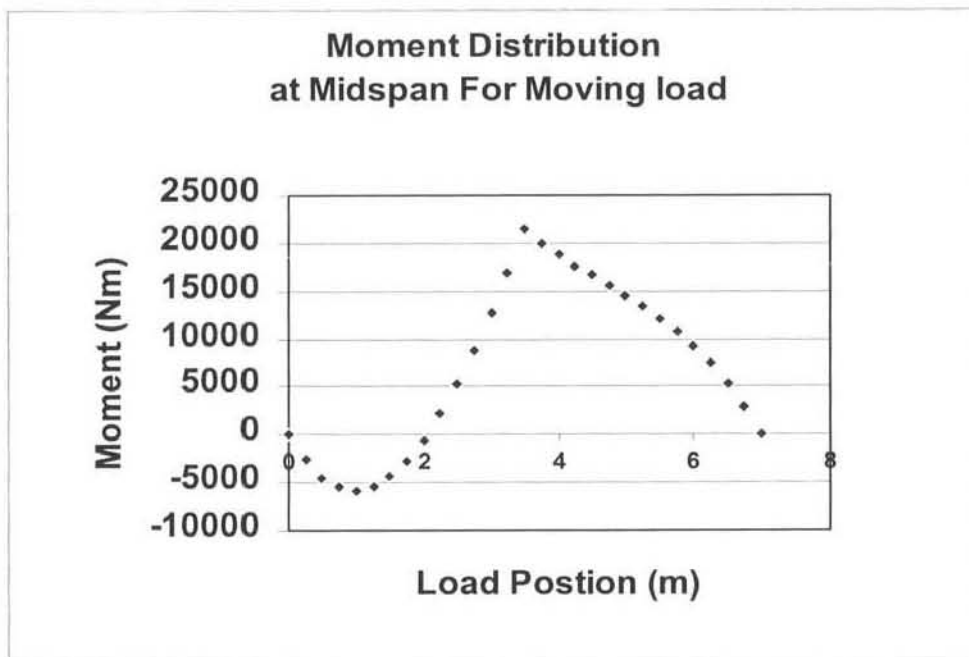


Figure 5.6.3. Moment distribution curve at 3.5 m

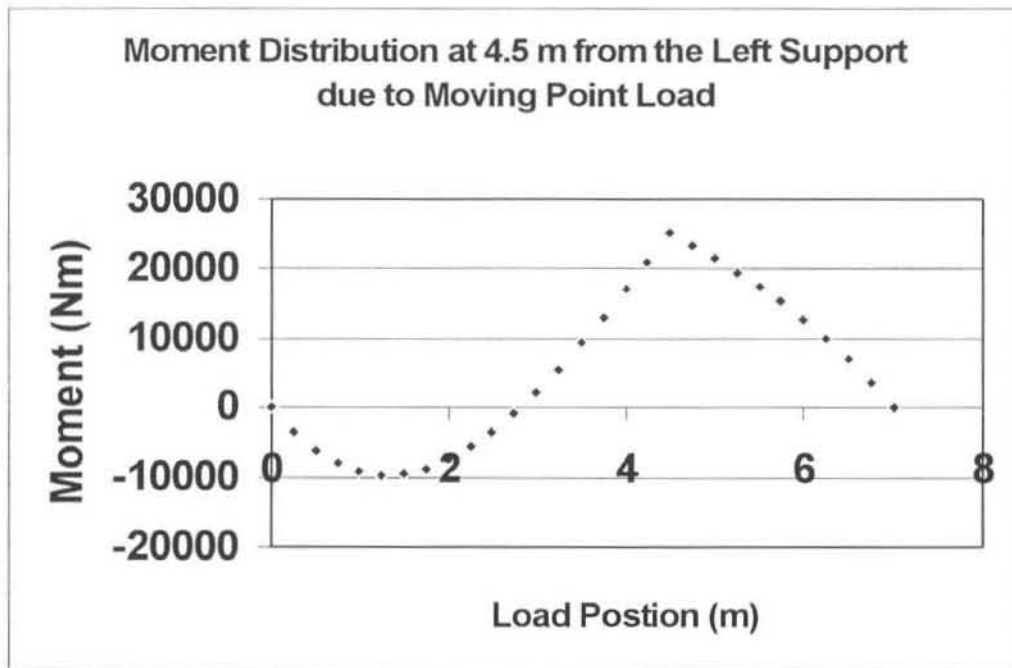


Figure 5.6.4. Moment distribution curve at 4.5 m

From the graphs it was concluded that the curves have the same basic shape and that the maximum moment occurs where $a = x$. The later of these conclusions led to the formulation of the following equation

$$M_x = -\frac{Px(L-x)^2}{L^2} + \frac{P(L-x)}{L}x \quad (5.6.5)$$

To find the value of x that produced the maximum moment the equation was differentiated and the resulting equation was equated to zero. The differentiated equation was

$$\frac{dM_x}{dx} = -\frac{P}{L^2}[L^2 - 4Lx + 3x^2] + \frac{P}{L}[-2x + L] \quad (5.6.6)$$

The solution to $\frac{dM_x}{dx} = 0$, yielded a value of $x = \frac{2}{3}L$.

Therefore the critical point along the girder occurred at $x = 4.67$ m from the left hand support when the position of the load coincided with this value. Applying the

the moment equation to this value of a yield the worst case load distribution on the girder. The data appears in Table 5.6.3 and the graph in Figure 5.6.5.

Table 5.6.3. Moment distribution at $x = 4.67$

Load Position	Moment	Load Position	Moment
0.00	0.00	3.50	7112.25
0.25	-3660.29	3.75	10787.56
0.50	-6491.62	4.00	14634.92
0.75	-8540.89	4.25	18607.40
1.00	-9855.05	4.67	25433.29
1.25	-10481.00	4.75	24778.07
1.50	-10465.67	5.00	22713.15
1.75	-9855.98	5.25	20585.67
2.00	-8698.87	5.50	18348.70
2.25	-7041.24	5.75	15955.32
2.50	-4930.02	6.00	13358.62
2.75	-2412.15	6.25	10511.65
3.00	465.47	6.50	7367.51
3.25	3655.91	6.75	3879.27

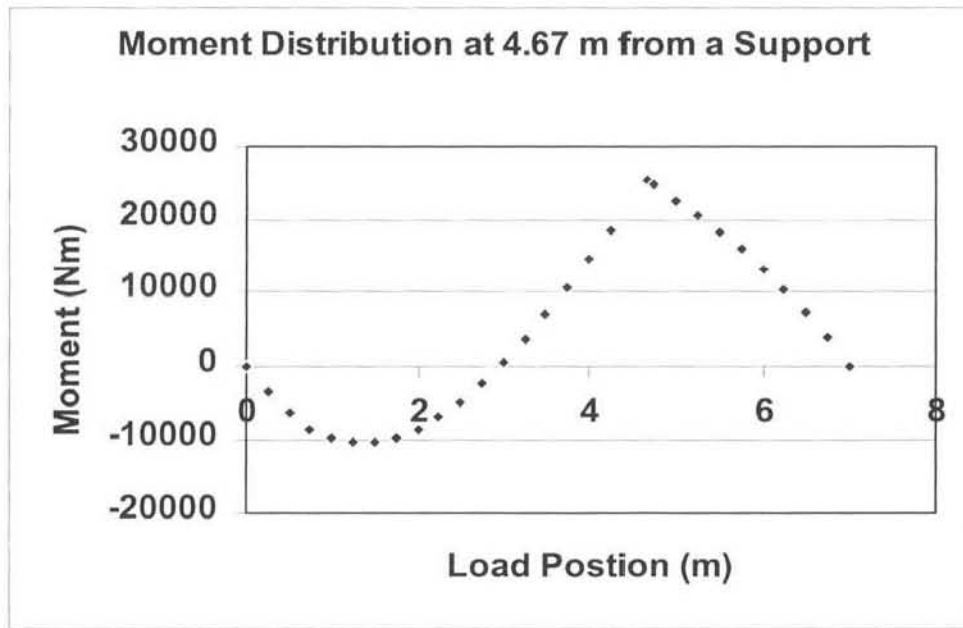


Figure 5.6.5. The moment distribution at 4.67m from a support

5.6.4. The Design of the Beam

The author decided that the constituent structural members would be standardized structural sections made of commercial mild steel. This material yields at 250 MPa, and has a ultimate tensile strength of 400 MPa. To find an allowable stress for the material a safety factor, typically used in building design according to Timoshenko is taken as 1.67. The writer desired to use a factor of 3 seeing as how the design was intended for a mining environment (for want of a greater value) and the result would be verified using computational software .

$$\begin{aligned}
 \sigma_{allow} &= \frac{\sigma_y}{n} \\
 &= \frac{250}{3} \\
 &= 83.33 \text{ MPa}
 \end{aligned}
 \tag{5.6.7}$$

In order to select a beam the section modulus, Z , was desired and was calculated as follows :

$$\begin{aligned}
Z &= \frac{M_{\max}}{\sigma_{allow}} \\
&= \frac{25433}{83.33 \times 10^6} \\
&= 306.4 \text{ cm}^3
\end{aligned} \tag{5.6.8}$$

The length of the beam, L, was designed to extend 7m into the tunnel from the timber bay . The load, P, consists of the timber pack, the hoist and the gripper. I sections and C sections are often used for members that are loaded in bending. The use of an I section was justified by the requirement placed on the member to support the traveling crane. The author analytically found that the I-configuration was the optimal cross sectional shape in terms of the strength to weight ratio, according to the pertinent theory discussed in chapter 4. The I shape puts most of the material at the outer fibers where the bending stress is maximum. As the shear is maximum at the center (neutral axis) the web serves to resist the shear.

After consulting the Table available in appendix A, a RSJ 254×102×28 was chosen. This section had a section modulus of 308 cm³. With this value the stress in the beam was recalculated and found to be **82.6 MPa**.

5.6.5. Computational Software Approach

The Nastran computer modeling software was employed once again to compute the stresses and displacements for the gantry structure. The approach adopted in this instance was to utilize the software's ability to analyze structures with beam elements. The computational analysis of this type was employed because the solution time is fast and the number of nodes and elements can be kept to a minimum as beam elements are two node elements.

The first item that was on the agenda was to verify the result calculated in the previous section. The member AB was modeled using beam elements with the RSJ cross section calculated above. The load was placed at 4.67 m from one support and fixed constraints were placed at the ends. The model submitted for analysis, appears in Figure 5.6.6. which depicts the cross section of the member in question The maximum stress was found to be 83.6 MPa and was remarkably close to that calculated above. The beam produced a deflection of under 3 mm. The author found it prudent to simulate the case when the load acted at the midspan, the result of which produced a stress of 75 MPa.

The next task was to choose prospective structural sections and associated geometry and for each of the load carrying members of the structure. This was done by considering the dimensions, materials and cost of various structural sections available commercially. Two of the many structural arrangements that were investigated are presented here. The geometry chosen was modeled on the software. The dimensions and properties of the structural members investigated appear in Table 5.6.4.

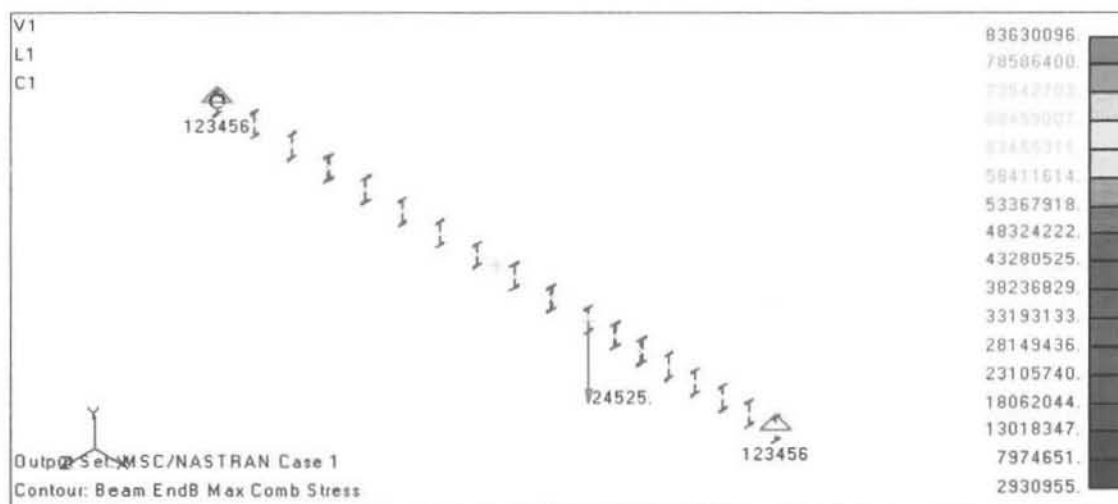


Figure 5.6.6. The I Beam model submitted for analysis (unit : Pascals)

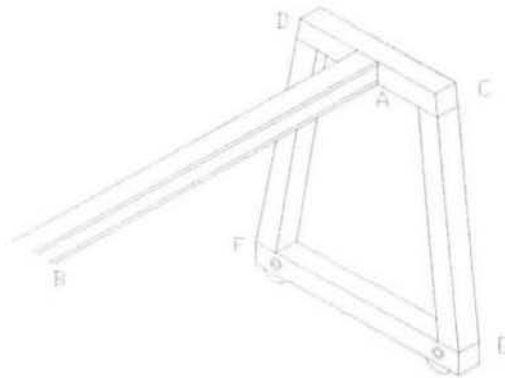


Figure 3.12 Revisited

Table 5.6.4. The dimensions for structural arrangement one

Member	Section	Detail (dimension in mm)
AB	I-Beam	Height 260.4, Width 102.1, Flange 10, Web 6.4
CD	Rectangular	Height 300, Width 100, Wall 5
EF	Rectangular	Height 300, Width 150, Wall 5
CE and DF	Square	Side 100, Wall 5

The rectangular section used for member EF was dictated by the dimensions of the traveling unit in Figure 3.10. This component was intended to be one of the ‘off the shelf ‘ components purchased for the design. Member CD was modeled as a rectangular section. The stress contour on the frame structure is depicted in Figure 5.6.7. The analysis resulted in a maximum stress of 94 MPa, which occurred at the point of load application (4.67 m from one support).

Whilst the result of the analysis was acceptable, the discrepancy may explained by the initial assumption that the joints were 100 % rigid. The FEM static analysis provided for the displacement of the frame model. The square section has used for the uprights has less resistance to bending as compared to I sections. If the girder

had been modeled as being simply supported, allowing for rotation at the support and the inability of the joint to transmit a couple, the maximum stress calculated at the midspan would be twice the stress obtained with the assumption of a completely rigid joint.

According to Benmann and Crawford, there are, in real situations, some degree of rotation between members at a joint owing to system elasticity. With this in mind the stress analysis was reasonable, and the stresses obtained for the other structural members were accepted as being viable. The maximum stresses in the upright members of the side supports was shown to be 60 MPa. This was also acceptable with a safety factor of 3 applied to the yield strength of structural steel.

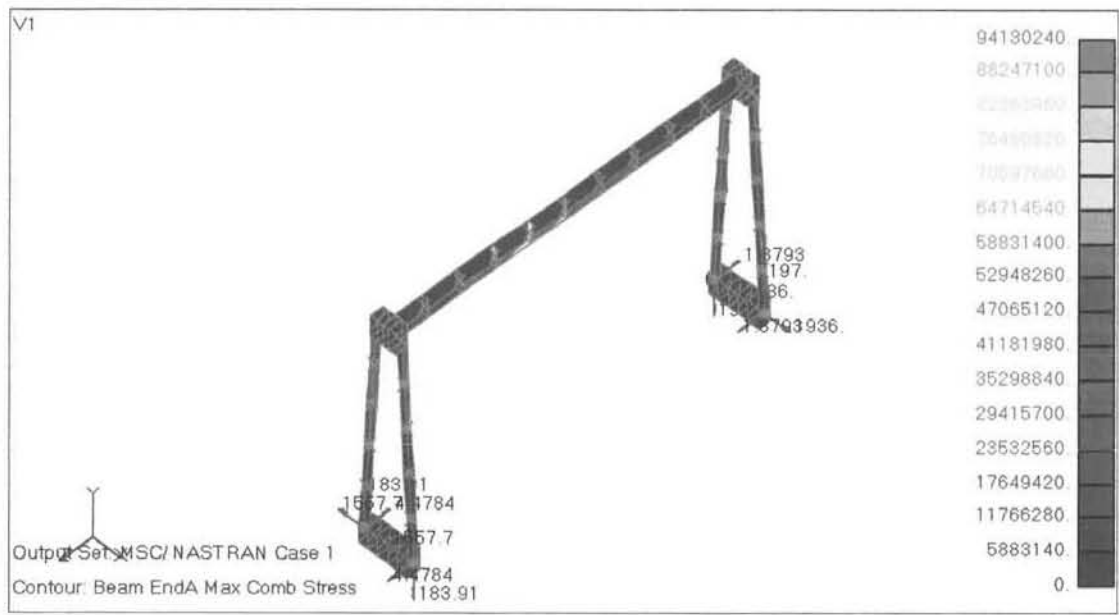


Figure 5.6.7. The gantry stress contour (units : Pascals)

The second structural arrangement for the gantry is listed in Table 5.6.5. Members CE and DF were changed to structural I section columns and were arranged as vertical members as opposed to the angled arrangement of the columns mentioned

in Table 5.6.4. Needless to say different sizes were experimented with and the dimension that appears in the table iteratively satisfied the requirements of strength, buckling (section 5.6.8) , fabrication and assembly (Chapter 6).

Table 5.6.5. The dimensions for structural arrangement two

Member	Section	Detail (dimension in mm)
AB	I-Beam	Height 260.4, Width 102.1, Flange 10, Web 6.4
CD	Rectangular	Height 300, Width 250, Wall 5
EF	Rectangular	Height 300, Width 250, Wall 5
CE and DF	I- Section	Height 203.2, Width 203.2, Flange 11, Web 7.3

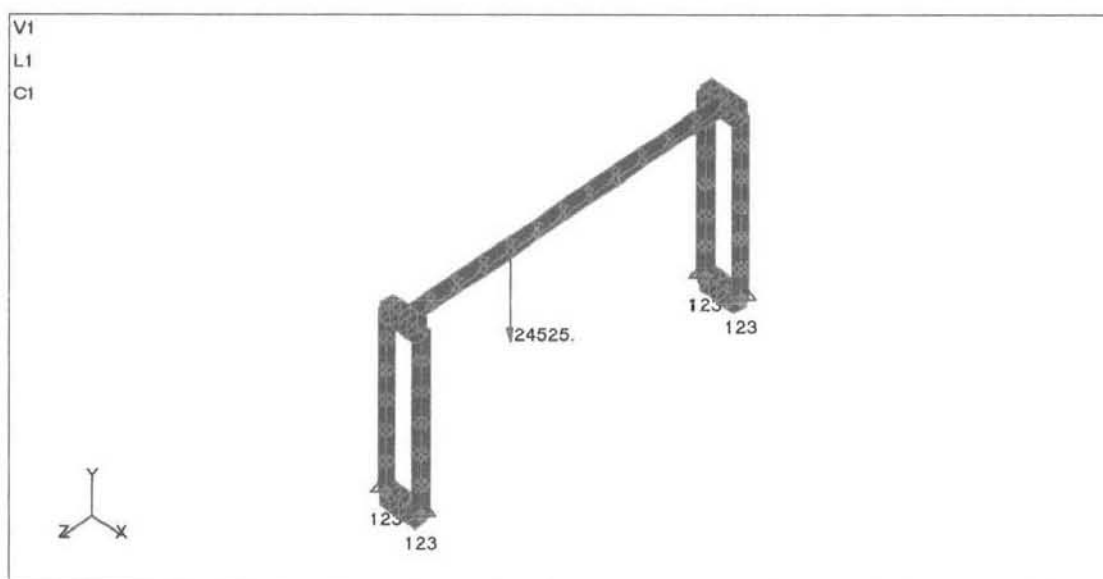


Figure 5.6.8. Structural arrangement two submitted for analysis

The model in Figure 5.6.8. produced a maximum stress of 76 MPa (within 10 % of the expected 83 MPa) at the beam center. The stress in the columns were found to be 75 MPa from the computer analysis. The author verified this value for the column stress and Table 5.6.6. is a summary of the calculation using the axis (x-y-z) shown in Figure 5.6.8.

Table 5.6.6. Summary of the Column Stress Calculation

M_x -Nm	M_z -Nm	F_y -N	I_x m^4	I_z m^4	A m^2
2.5E+04	9.2E+03	1.6E+04	4.5E-05	1.5E-05	5.8E-03

σ_x -MPa	σ_z -MPa	σ_y -MPa	σ_R -MPa
57.5	60.0	3.0	83.0

where

M_x = Moment about x-axis

M_z = Moment about z axis

F_y = The vertical force on the column

I_x = Section Moment of Inertia about x

I_z = Section Moment of Inertia about z

A = The cross sectional area

σ = The stress in the designated direction

The author proceeded with the design of structural arrangement two.

5.7. Selection Techniques for the Hoist and Travel Unit

The selection of the electric hoisting system was based primarily on two factors :

- the type of duty
- the class of operation = the time of use (hours)

The selection of the M series electric hoist was conducted according to the FEM group, in accordance with rule 9.511 for series production hoists.

The type of duty is generally categorized as being either light (L1), moderate (L2), heavy (L3) and very heavy (L4). The following graphs in Figure 5.7.1. show the relationship between % load and % operating time for each of the categories listed above.

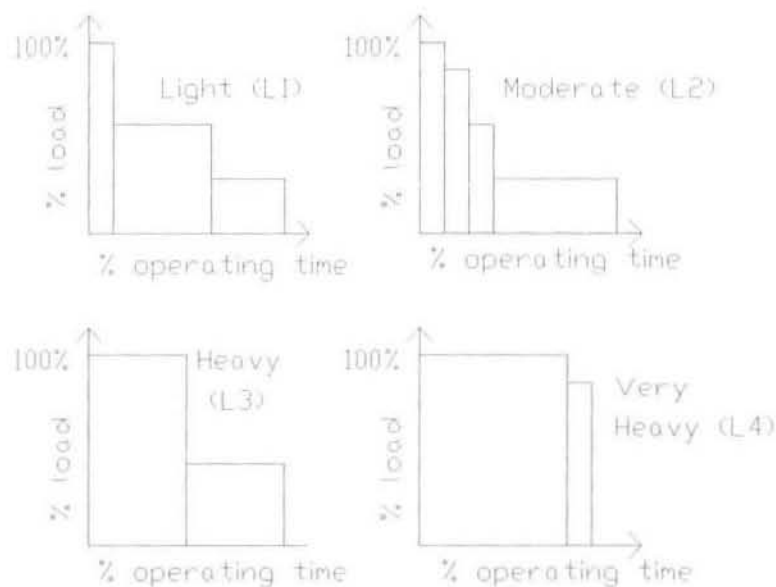


Figure 5.7.1. Hoist duty classification graphs

The following Table was compiled for the purpose of selecting the hoist system.

Table 5.7.1 Hoist selection chart

Type of Duty		Time in Use	
L1	Light	12500	25000
L2	Moderate	6300	12500
L3	Heavy	3200	6300
L4	Very Heavy	1600	3200
FEM GROUP		2m	3m
Number of ropes		Hoist Size	
2/1	4/1		
Capacities (Kg)			
800	1600	-	A
1000	2000	A	-
1250	2500	-	-
1600	3200	-	B
2000	4000	B	-
2500	5000	-	C
3200	6300	C	-
4000	8000	-	D
5000	10000	D	-
6300	12500	-	E
8000	16000	E	F
10000	20000	F	-

Note : the letters A; B; C; D; E; F are used to indicate a load class

An operation class of 10800 hours was based on the unit being used in service 3 hours daily , over a five day week, for a 15 year period. A moderate duty (L2) was

selected. Since the capacity dictated by the chart fell over into that of 3200 kg from 2500 kg, this type of duty was acceptable. The FEM group was determined to be 3m and the type of fall desired was that of 4/1. The hoist type was therefore MB. The characteristics of such a system is listed in Table 5.7.2.

Table 5.7.2. The Hoist System Characteristics

Capacity	3200.0 kg
M Hoist	B 32 S4
Lifting Speed (normal)	4 m/min
Rope diameter	10 mm
Motor (available)	220-380 V @ 50Hz
Girder Travel Speed	16 m/min
Mass (low headroom version)	405 Kg

Table 5.7.3. must be read in conjunction with Figure 5.7.2.

Table 5.7.3. Dimensions For the FEM GROUP 3m, MB compact monorail trolley (all dimensions in mm)

C	610
DR	125
H1	145
H2	290
H3	80
H4	365
H5	310
H6	55
L1	500

L2	400
L3	380
L4	160
LT max	300
LT min	100
IR	441
A	1095
Am	625
Ar	470
L	720
K1	35
K2	55

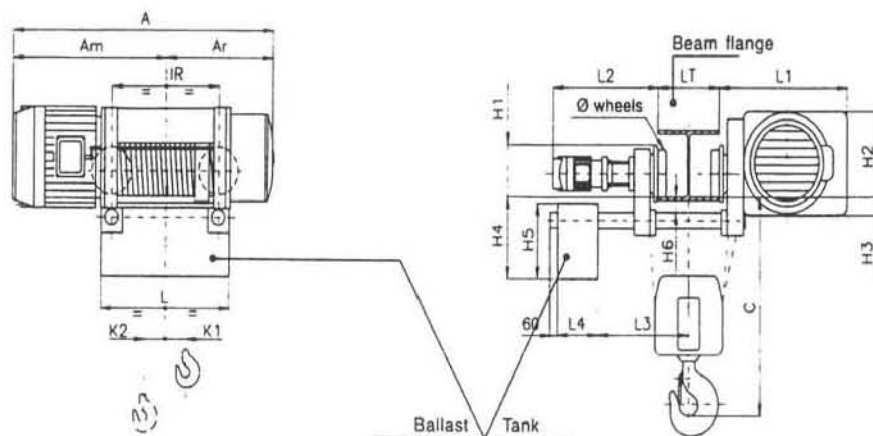


Figure 5.7.2. Low Headroom hoist selected for the system

The hoist would be modified and supplied, from the vendor, without the hook for attachment to the chassis. The hook is attached to a pulley housing that is made of a weldable material.

5.8. Fatigue Design Considerations of the Gantry

5.8.1. Application Overview

Throughout the service life of the gantry the constituent members will be subjected to cyclic loading. This was due to the fact that, the crane had a moving load. As discussed and analyzed in section 5.6.3. the critical section was identified at 4.67 m from the left hand support as the load moved from left to right. The graph of the moment variation at this section appears in Figure 5.6. 5.

It is accepted practice to assume a sinusoidal cycle having constant upper and lower stress limits throughout the service life. If one considered Figure 5.6.5 , for the case of the girder, one would be aware that this was a reasonable assumption. Very often, a component is subjected to cyclical stresses of which the magnitude of the upper and lower limit may vary considerably. Even in this case the assumption to model to cyclic load as a sinusoidal one is still viable.

5.8.2. Fatigue Loading Study on the Girder

Having utilized the assumption above, the graph in Figure 5.8.1. and the data in Table 5.6.3. the sine approximation allowed for the modeling of a wave with a maximum moment of 25433.29 Nm and a minimum value of -10481.00 Nm. These values corresponded to stresses of 83 MPa and -34 MPa respectively. The author had also calculated the effect of including the weight of the girder and also that of the bending shear stress. The bending shear stress at 4.67 m was calculated to be less than 2.5 MPa. The effect of including the uniformly distributed weight (28 kg/m) of the girder was found to change the moment by relatively insignificant amount. This lent itself to the fatigue design for a fluctuating uniaxial stress.

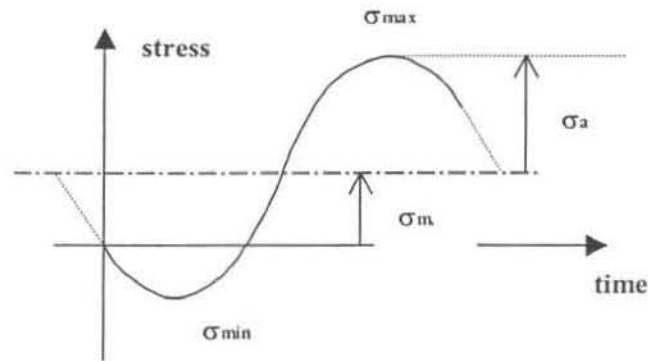


Figure 5.8.1. Load cycle at critical point along girder span

The other points that were identified as being worthy of examination were the two end points of the beam where the girder considered to be clamped. The sharp corners at these points result in stress concentration factors of 2 i.e. $K_t = 2$, as suggested by Norton, 1998, for such a clamped condition. This meant that the author had to consider a phenomenon called fretting fatigue. This would have arisen due to the slight displacement between the connecting members as the girder deflects. The mechanism of failure is explained by the breaking down of the protective coating thereby exposing the metal below to oxidation, and accelerating the fatigue-failure process. Attempts to treat the metal to prevent this are addressed in Chapter 6.

From equation (5.6.1) the moment at the end of the girder was found to be

$$M_1 = \frac{Pab^2}{L^2} = \frac{Pa(L-a)^2}{L^2} \quad (5.8.1)$$

The moment therefore changed as the position of the load varied from 0 to 7. The approach adopted was similar to that outlined above. A spread sheet was used to

calculate the moment values as a varied with steps of 0.25. The output showed that the reaction moment varied from 0 to a maximum of 25408 Nm at $a = 2.25$.

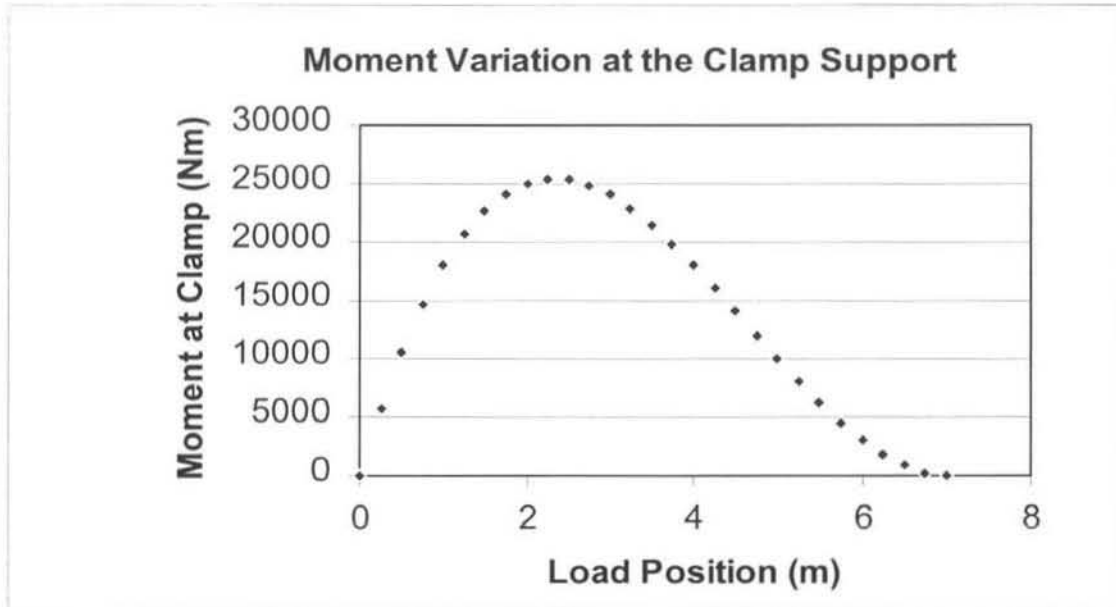


Figure 5.8.2. Moment Variation at the girder clamp support

The author also made use of an analytical approach. Equation 5.6.7. was differentiated. The resultant expression was equated to zero and the equation solved to find values of a which maximized the differentiated equation.

The differentiated equation was

$$\frac{dM}{dx} = \frac{P}{L^2}(L^2 - 4La + 3a^2) = 0 \quad (5.8.2)$$

With $L = 7$, a was found to be 2.33, which results in a maximum moment of 25433.29 Nm. The result confirmed the author's prediction, that when the load was 2.33 m from the support the moment at the support would equal the maximum moment at 4.67 m from the support. The graph showing the moment variation at the support under moving load conditions appears in Figure 5.8.2.

5.8.4. Fatigue Design at Critical Point along Span

It must be mentioned that, the calculations presented here are in fact final design calculations. They were iteratively calculated to satisfy the fatigue design criteria as well as to check the correspondence to the member geometrical selection. Therefore the reader should be aware that the discussions, calculations and conclusions in previous sections were in fact checked at every choice of parameters for fatigue as well as considerations that have already been discussed.

The structural members of the gantry system was designed for 10^6 cycles with no failure. At the critical section along the span the maximum stresses were calculated as above. Reference is made to Figure 5.8.1.

$$\sigma_{\max} = 83 \text{ MPa} \quad (5.8.3)$$

$$\sigma_{\min} = -34 \text{ MPa} \quad (5.8.4)$$

The alternating stress was calculated as follows,

$$\begin{aligned} \sigma_{alt} &= \frac{\sigma_{\max} - \sigma_{\min}}{2} \\ &= 58.5 \text{ MPa} \end{aligned} \quad (5.8.5)$$

The mean stress was found to be,

$$\begin{aligned} \sigma_m &= \frac{\sigma_{\max} + \sigma_{\min}}{2} \\ &= 24.5 \text{ MPa} \end{aligned} \quad (5.8.6)$$

The next step involved the determination of the modified value of the endurance limit. The girder comprised of a ferrous material and exhibited a general stress cycle curve as in Figure 4.3.

The endurance limit was found to be,

$$\begin{aligned} S_e &= \frac{1}{2} \sigma_{ut} \\ &= 200 \text{ MPa} \end{aligned} \quad (5.8.7)$$

The modified value of the endurance limit depended on the load factor, the size factor, the reliability factor, the surface factor and the temperature factor.

As the case in consideration is a uniaxial case of bending,

$$C_{load} = 1 \quad (5.8.9)$$

As girder manufactured by hot rolling its section, and the Brinell Hardness Number for the section was approximately 130 BHN. Figure A3 (Appendix A) was used to determine the surface factor.

$$C_{surf} = 0.72 \quad (5.8.10)$$

For parts of non-round cross section the approach adopted was to equate the part's non-round cross sectional area stresses above 95% of it's maximum stress with the similarly stressed area of a rotating beam specimen would provide a pseudo diameter.

The following formula was used to calculate the equivalent area, A_{95} , for an I-Beam as suggested by Norton , 1998.

$$\begin{aligned}
A_{95_{2-2}} &= 0.05bh \\
&= 0.05 \times 0.102 \times 0.2604 \\
&= 1.32804 \times 10^{-3} \text{ m}^2
\end{aligned} \tag{5.8.11}$$

The equivalent rotating diameter was calculated as follows,

$$\begin{aligned}
d_{eq} &= \sqrt{\frac{A_{95}}{0.0766}} \\
&= 1.317 \times 10^{-1} \text{ m}
\end{aligned} \tag{5.8.12}$$

As this pseudo diameter, converted to millimeters, was between 8 mm and 250 mm, the following formula was applied to obtain the size factor.

$$\begin{aligned}
C_{size} &= 1.189d^{-0.097} \\
&= 0.74
\end{aligned} \tag{5.8.13}$$

The reliability factor based on a 99 % reliability was,

$$C_{rel} = 0.814 \tag{5.8.14}$$

The temperature factor was taken to be 1.

The modified endurance limit was therefore calculated to be

$$\begin{aligned}
S_{em} &= C_{load}C_{surf}C_{size}C_{temp}C_{rel}S_e \\
&= 1 \times 0.72 \times 0.74 \times 1 \times 0.814 \times 200 \\
&= 86.73 \text{ MPa}
\end{aligned} \tag{5.8.15}$$

For this instance the author assumed that there was no disruption to the contours of the girder, i.e. there was no change in the cross section at the point of consideration (4.67 m from the support) and therefore sought not to modify the alternating and mean stress, for this instance, through the application of stress concentration factors.

The safety factors for any fluctuating-stress state depended on the manner in which the alternating and mean components varied with each other in service. The relationship between the alternating stress, mean stress and the safety factor, R , was taken to be

$$\begin{aligned} R &= \frac{S_{em}}{\sigma_a} \left(1 - \frac{\sigma_m}{\sigma_{ut}} \right) \\ &= \frac{86.73}{58.5} \left(1 - \frac{24.5}{400} \right) \\ &= 1.4 \end{aligned} \tag{5.8.16}$$

With this value (≥ 1) the design for fatigue on the girder was accepted. More especially since the assumption omitted the weight of the girder and the transverse shear stress which were calculated and whose magnitude was small in comparison to the bending stress magnitude.

5.8.5. Fatigue Design at the Support

The next fatigue design was applied to the joint at the girder ends. From the load analysis above carried out on the varying moment at the clamp, the sinusoidal model dictated the use of a maximum stress of 83 MPa and minimum stress of zero MPa .

With this information the alternating and mean stress were calculated.

$$\begin{aligned}\sigma_{alt} &= \frac{\sigma_{\max} - \sigma_{\min}}{2} \\ &= 41.5 \text{ MPa}\end{aligned}\tag{5.8.17}$$

$$\begin{aligned}\sigma_m &= \frac{\sigma_{\max} + \sigma_{\min}}{2} \\ &= 41.5 \text{ MPa}\end{aligned}\tag{5.8.18}$$

Now because of the geometric stress concentration factor, mentioned above, at the point of attachment the both the alternating and mean stresses had to be modified. The fatigue strength concentration factor, K_f , was obtained from the geometric stress concentration factor, K_t , and the notch sensitivity of the material.

The notch sensitivity, q , was obtained from Figure B.2. (Appendix B). With an ultimate tensile stress of 400 MPa, and a notch radius of 0, the notch sensitivity was, for the condition of bending, found to be 0.26.

Therefore the fatigue stress concentration factor was calculated as,

$$\begin{aligned}K_f &= 1 + q(K_t - 1) \\ &= 1 + 0.26(2 - 1) \\ &= 1.26\end{aligned}\tag{5.8.19}$$

The modified alternating stress was found,

$$\begin{aligned}\sigma_{a_{\max}} &= K_f \sigma_{alt} \\ &= 1.26 \times 41.5 \\ &= 52.29 \text{ MPa}\end{aligned}\tag{5.8.20}$$

$$\begin{aligned}\sigma_{alt} &= \frac{\sigma_{\max} - \sigma_{\min}}{2} \\ &= 41.5 \text{ MPa}\end{aligned}\tag{5.8.17}$$

$$\begin{aligned}\sigma_m &= \frac{\sigma_{\max} + \sigma_{\min}}{2} \\ &= 41.5 \text{ MPa}\end{aligned}\tag{5.8.18}$$

Now because of the geometric stress concentration factor, mentioned above, at the point of attachment the both the alternating and mean stresses had to be modified. The fatigue strength concentration factor, K_f , was obtained from the geometric stress concentration factor, K_t , and the notch sensitivity of the material.

The notch sensitivity, q , was obtained from Figure A4 (Appendix A). With an ultimate tensile stress of 400 MPa, the notch sensitivity was, for the condition of bending, found to be 0.26.

Therefore the fatigue stress concentration factor was calculated as,

$$\begin{aligned}K_f &= 1 + q(K_t - 1) \\ &= 1 + 0.26(2 - 1) \\ &= 1.26\end{aligned}\tag{5.8.19}$$

The modified alternating stress was found,

$$\begin{aligned}\sigma_{a_{\max}} &= K_f \sigma_{alt} \\ &= 1.26 \times 41.5 \\ &= 52.29 \text{ MPa}\end{aligned}\tag{5.8.20}$$

Since the product of the stress concentration factor applied to the alternating stress and the maximum stress in the beam was less than the material yield stress the stress concentration factor applied to the mean stress, K_{fm} , was taken to be equal to the stress concentration factor applied to the alternating stress, K_f .

Therefore,

$$\begin{aligned}\sigma_{m_{\max}} &= K_{fm} \sigma_m \\ &= 1.26 \times 41.5 \\ &= 52.29 \text{ MPa}\end{aligned}\tag{5.8.21}$$

With this value, the safety factor, R , was calculated,

$$\begin{aligned}R &= \frac{S_{em}}{\sigma_a} \left(1 - \frac{\sigma_m}{\sigma_{ut}} \right) \\ &= \frac{86.73}{52.29} \left(1 - \frac{52.29}{400} \right) \\ &= 1.44\end{aligned}\tag{5.8.22}$$

This value was acceptable and justified as in the previous case. The calculation satisfied the criteria for fatigue design.

5.9. Buckling Considerations of the Upright Members

For the purpose of this discussion, the members CE and DF were modeled as vertical members that were fixed at either end as shown in Figure 5.9.1. Long columns require the calculation of the critical load. The Euler column formula predicts this critical load. This Euler load is in fact independent of the material strength, and dependant only on the length, cross sectional area and the elastic modulus. The Euler found that the critical load was given by,

$$P_{cr} = \frac{A\pi^2 E}{S_r^2} \quad (5.9.1)$$

where

P_{cr} = the critical load

A = the cross sectional area

E = modulus of elasticity

S_r = the slenderness ratio

The slenderness ratio is given by

$$S_r = \frac{L}{k} \quad (5.9.2)$$

where k , the radius of gyration, depends on the moment of inertia and the cross sectional area by,

$$k = \sqrt{\frac{I}{A}} \quad (5.9.3)$$

Equation (5.9.1) is useful when the end conditions are both pinned. However, for the case of when the column is fixed at both ends, the length is adjusted and is

taken to be half the actual length of the column. Johnson suggested that Eulers formula is applicable only when $S_r > S$, where S is given by

$$S = \pi \sqrt{\frac{2E}{\sigma_y}} \quad (5.9.4)$$

When $S_r \leq S$, the critical load according to Johnson is,

$$P_{cr} = A \left[\sigma_y - \frac{1}{E} \left(\frac{\sigma_y S_r}{2\pi} \right)^2 \right] \quad (5.9.5)$$

With the column I section, the moment of inertia, I, and the cross sectional area were calculated to be, 4.495e-5 m⁴ and 5.793e-3 m², respectively.

The radius of gyration, k, was calculated to be 8.811e-2.

The slenderness ratio, taking the modified length of 1.5 m into account, was calculated to be 17.029 and found to be less than S (=125.6)

With the elastic modulus, E, taken to be 200 GPa, the yield strength, σ_y , taken to be 250 MPa, the critical load according to Johnson was calculated

$$\begin{aligned} P_{cr} &= 5.793e-3 \times \left[250e6 - \frac{1}{200e9} \left(\frac{250e6 \times 17.029}{2\pi} \right)^2 \right] \quad (5.9.6) \\ &= 143.5 \text{ KN} \end{aligned}$$

The design was accepted as compressive force on the girder was calculated to be 12.5 KN, for a worst case, i.e. the load position resulting in the highest vertical loading.

Chapter 6

Manufacture and Assembly of the 4 Arm Gripper

6.1. Overview of Design for Manufacture and Assembly

This chapter was the concluding chapter of the design report. It was therefore the culmination of endeavors made in preceding chapters and attempted to summarize the work outlined and specify some of the ‘nuts and bolts’ of the system. It also outlines a very important component of Total Design - that of manufacturing and assembly. As part of the Total Design Process, component and or system manufacture and assembly cannot be treated in isolation but consideration to the form and susceptibility of the subsystems to gel with other subsystems must be given attention at all times. The authors view of the Total Design approach, can be likened to that of a zoom camera. That is to say the designer must be able to zoom out to have a holistic picture of the design at all times and he must also be able to zoom in and focus on detail. This in the authors opinion is the essence of the science of design engineering.

An important characteristic of designing for manufacture as a sub process of total design is in selecting the best manufacturing process and designing the component that so that it can be manufactured accordingly. There are definite methods allied to specific processes. The knowledge of these methods are vital to the success of any design. Even if some modification to the process is needed, a firm grasp of the conventional methods is an indispensable tool. In other words it is important to know the rule before it can be broken.

Assembly requires labor and or assembly machines. The relationship between assembly cost and the number of subsystems and assembly time and the number of subsystems is almost always a proportional one. It is therefore the task of the design team to optimize the assembly process by making systems as easy to assemble as possible.

6.2. Manufacturing of the 4 - Arm Gripper System

6.2.1. The modifications made during the design

The arm model presented in section 5.2. did not take into account the modifications and allowances that were instituted in sections that followed. The author opted not to present the modifications in section 5.2. as this would not serve to highlight the total design process. To summarize, the diameter at the shaft-arm interface was increased to 52 mm. The actuation of the arms required that a further 11 mm diameter hole be made at 82.5 mm from the axis of rotation and a force of 720 N (safety factor of two) acts in the direction of displacement. The 52 mm diameter hole was also loaded by the pressure of 20 MPa at the shaft arm interface.

These changes were modeled using the finite element computational approach. This resulted in a small area of stress concentration around the 11 mm hole and an increase in the value of stress around the 52 mm hole. The most important output was that the overall maximum value of the stress increased by approximately 20 % from the final model in section 5.2. This value was still below the yield strength of commercial mild steel of 250 MPa which was the material selected for the arm.

6.2.2. The material selected for the arm, shaft and chassis

The chassis structure consisted of plates and structural sections. The material chassis end plates would be the same as the commercial steel used for the arm. Since the chassis end plate thickness had to be greater than 20 mm, the author chose to make use of a 25 mm thick plate as this was the size chosen for the arm. The author decided that this would make optimum use of the plate purchased from which the arms and end plates would be cut. The chassis structural sections would also be commercial structural steel. The arm and chassis end plates would be manufactured from commercial mild steel plate for the following reasons - infinite life characteristic, low cost, availability and ease of fabrication.

The mild steel (a low carbon steel) employed was produced by hot rolling. Hot rolling is a shaping process that is carried out above the recrystallization temperature of the steel. This temperature is usually between 815° and 1260°. There are fewer steps involved in the hot working of steel as compared to cold working as it is simpler to work with the steel when it is red hot. Hot worked steels are therefore relatively cheaper. Hot rolled mild steel also has better weldability and is generally more stable during machining.

The disadvantages of hot working was the relatively poorer surface finish and the absence of adequate corrosion resistance of the material. This would be compensated for by finishing process like grinding or filing and the galvanizing the material, as discussed below. Generally low carbon steels do not harden well because of its low carbon content. However, low carbon steels can be carburized and quench hardened to obtain hardened surfaces. Mild steel required for the arm and chassis are readily available from steel merchants. The steel produced locally is designated as 300 WA.

The material used to manufacture the shaft was a hollow bar designated as SP52. The author verified that the material was readily available from more than one supplier. The shaft strength which was verified in section 5.3. was based on the dimensions of the bar available. The material had a minimum yield strength of 510 MPa. It possessed the stability required during the machining of the shaft. The surface after turning, to obtain the steps, would be finished (filed) to the required roughness. The material would also need to be protected against environmental corrosion through galvanizing.

6.2.2.1. Galvanizing

Galvanizing is protection treatment against corrosion. Zinc is deposited on the surface of steel by hot-dipping. Zinc has an extremely low corrosion rate. If one considers that a 75 μm coating of zinc may protect the underlying steel for approximately thirty years under normal atmospheric humidity conditions, one can understand that a zinc coating is certainly affective. Zinc has the added advantage of being anodic to most metals. Because the atmosphere in which the gripper must operate can be considered to be corrosive (large amounts of fissure water), the author has designed for a coating thickness of 75 μm on all components by the hot dip process. With this size, one can expect the coating to provide protection for 15 - 20 years as required by the specification without affecting the press fit were required.

6.2.3. Obtaining the profile of the arm and chassis end plates

The arm and chassis profile was to be fabricated at a professional cutting plant. The method of computer controlled profiling and cutting would be required to obtain the arm and chassis shape form with excellent accuracy. The utilization of a

professional cutting plant as opposed to carrying out the fabrication in house was justified as :

- an in house operation would require a substantial capital investment in the required machinery;
- cutting plants are specialized to carry out the task at hand therefore their scrap rate would probably be less than an in house operation;
- the cost of errors are borne by the specialist plant;

The method employed by these plants for the profile cutting of low carbon steels of 25 mm thickness or greater is that of oxyfuel cutting. The fuel used was to be acetylene.

6.2.3.1. Oxyfuel Cutting

The material is heated to its kindling point usually by a number of preheat flames. Once the material is at the required temperature, a stream of oxygen at high speed is introduced. This stream would result in the oxidation of the material. The force of the stream of oxygen literally blows the oxides away from the cutting path resulting in a clean cut. The oxidation process also serves the purpose of adding thermal energy to the material. This energy is radially conducted into the material and in doing so raises the temperature of the steel ahead of the cut and maintains the kindling temperature.

6.2.4. Obtaining the surfaces of revolution (steps) on the shaft

The shaft required two steps. One step would be required for the abutment of the arm and one for the bearing. The operation chosen for the machining of the shaft is turning. The machine employed for the task was a lathe. A lathe is essentially a

machine tool for producing and finishing surfaces of revolution. The machine is designed to hold and revolve the shaft about an axis of rotation so that it may be subjected to the action of a cutting tool moving in a horizontal plane through the axis of the work. When the cutting tool moves in a longitudinal direction or parallel to the axis, the operation is known as turning; when it moves in a transverse direction, it is known as facing.

The author verified through consultation with industrial fabricators of steel that the radii (dictated by the bearing selected and stress reduction), were attainable through the use of precision cutting tools. For each of the steps, two turning operations would be required. The first operation would be a rough cut, size of which would be determined by a machinist, with the knowledge of his machine and ability. A predetermined amount is left for the second finishing turn. The tool would then be either changed, resharpened or stoned for the second cut. A trial finishing cut would be required; this cut should be made just long enough so that the bar diameter can be measured. The final surface roughness required by the shaft would be attained through filing with a mill file after the second turning operation. It is recommended that at least $12.7\text{e-}3$ mm is left for the final filing operation.

6.3. Design of the Welded Joints

As outlined the gantry and chassis would consist by and large of structural steel sections and flat plates. With respect to the gantry, Table 5.6.4. contains the final specifications for the sections, which were modeled and analyzed using the finite element computational software. The structural sections needed only to be cut to size and galvanized. The structural sections and plates selected were available to be purchased from steel merchants. As mentioned in the preceding section all steelwork was to galvanized with a $75\text{ }\mu\text{m}$ thick layer of zinc.

After having done preliminary calculations, experimenting with various weld shapes, the author has found that the calculation to follow was most appropriate. The weld configuration selected is shown in Figure 6.1. This configuration will be used to illustrate the calculation method employed. The rest of the welding calculations data appears in Table 6.3.1. and Table 6.3.2. The calculations are based on the theory presented by Juvinall and Marshek, 1991.

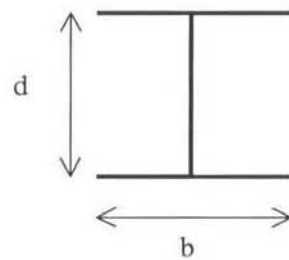


Figure 6.3.1. Weld configuration for I-Beam fixed support

The task at hand was to determine the required weld size, using a E60 welding electrode with a yield strength of 345 MPa. For the application the author chose a safety factor of 2.5. The girder would be welded at its ends with the configuration in the x-y plane shown in Figure 6.3.1. (dimension d parallel to the y axis).

As discussed in section 5.6. the moment at support 1 was given by equation (5.6.1)

$$M_1 = \left(\frac{Pa(L-a)^2}{L^2} \right) \quad (5.6.1)$$

This equation was differentiated with respect to **a**, equated to zero and solved, to find the load position from the support which maximized the equation. The analysis returned a value of 2.33 m for **a**.

This value yielded a maximum moment of 25433,29 Nm at the support. The vertical reaction force at the support for load position was calculated to be 1.63617e4 N, from the equation

$$R = \frac{P(L - a)}{L} \quad (6.3.1)$$

where

- P = the load on the gantry, N
- L = the beam length, m
- a = the load position from the support, m
- R = the reaction at the support, N

Having defined the forces, the weld moments of inertia were calculated.

The inertia of one of vertical welds was

$$\begin{aligned} I_y &= \frac{d^3 t}{12} \\ &= \frac{0.260^3 t}{12} \\ &= 1.465e - 3 \times t \end{aligned} \quad (6.3.2)$$

The inertia of one of the horizontal welds was

$$\begin{aligned} I_x &= by^2 t \\ &= 0.102 \times 0.13^2 t \\ &= 1.723e - 3 \times t \end{aligned} \quad (6.3.3)$$

The rectangular moment of inertia about the neutral bending axis consisted of the contributions made by 2 vertical welds and 2 horizontal welds.

$$\begin{aligned}
 I &= 2(I_y + I_x) \\
 &= 6.3776e-3 \times t
 \end{aligned}
 \tag{6.3.4}$$

where

- L = the beam length, m
- y = the load position from the support, m
- t = the throat thickness, m
- b = The length of the weld parallel to the neutral axis, m
- d = the length of the weld perpendicular to the neutral axis, m

The tensile bending stress was given by

$$\sigma_b = \frac{Mly}{I} = \frac{5.1842e5}{t}
 \tag{6.3.5}$$

The transverse shear stress on the weld was

$$\tau = \frac{R}{A} = \frac{2.2599e4}{t}
 \tag{6.3.6}$$

where

- A = the total length of the weld

The vector resultant of the transverse and the bending stresses was found and equated to the allowable weld stress taking the appropriate factor of safety into account.

$$\frac{0.58 \times 345e6}{2.5} = \frac{5.19e5}{t}
 \tag{6.3.7}$$

This produced a value of 6.5 mm for t , which was in the expected range and the author specified the weld with a value of 7 mm. The weld on either end of the columns were calculated following the method of calculation outlined above. The calculation summary appears in Table 6.3.1.

The column was welded using an E60 electrode with the specification mentioned above. The column was designed, in theory, to be rigidly attached at either end to the gantry headgear and the gantry travel unit. The load values quoted in the table are critical values. The appropriate weld footprint was found after some preliminary calculation to be that depicted in Figure 6.3.1. The footprint was in the x-z plane, with dimension b parallel to the x axis. The task as before was to determine the appropriate weld size. The dimensions of the column were listed in Table 5.6.5.

Table 6.3.1. Weld calculation summary for column ends

Moment along x axis (Nm)	Weld inertia parallel to bending axis	Weld inertia perpendicular to bending axis	Stress along x axis
25433	$(2.0975e-3)t$	$(6.99e-4)t$	$4.4565e5/t$
Moment along z axis (Nm)	Weld inertia parallel to bending axis	Weld inertia perpendicular to bending axis	Stress along z axis
9197	$(6.99e-4)t$	$(2.0975e-3)t$	$1.6707e5/t$

Similar to equation 6.3.7. the throat t , with a safety factor of 2.5, was found to be 6 mm. The chassis was to be welded to the hoisting unit pulley housing. Utilizing the method outlined above, with an E60 electrode and a safety factor of 2.5, the weld throat was calculated to be 5.4 mm for the 100 mm square weld footprint. The chassis construction was to be welded as well. The attachment of the 100 mm

square tube to the endplates was also calculated to be 5.4 mm. The author has recommend the following :

- a 6 mm weld at both ends of the 100 mm square tube i.e. at the end plate and at the pulley housing,
- a 3 mm weld around the 50 mm square tube joining opposite end plates.

Table 6.3.2. is a summary data used in the weld calculation for the 100 mm square weld.

Table 6.3.2. Weld calculation summary for chassis center tube

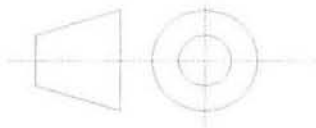
Moment along bending axis (Nm)	Weld inertia along parallel to bending axis	Weld inertia perpendicular to bending axis	Bending stress
5886	$2.5e-4t$	$8.333e-5t$	$4.4189e5/t$
Force on weld (N)	Weld area	Shear stress	Stress Resultant
7358	$0.4t$	$1.839e4/t$	$4.4227e5/t$

6.4. Drawings of the Arm, Shaft, Chassis and Gantry

Unless otherwise stated, the following notes apply to all figures to follow.

Dimension	mm
Angle	degrees
Scale	1:1

- All views were drawn in the first angle.



Machine Tolerances

over		to	tolerance
0	-	6	± 0.1
6	-	30	± 0.2
30	-	100	± 0.3
100	-	300	± 0.5
300	-	1000	± 0.8
1000	-	3000	± 1.2

The materials have been specified above. The fabrication methods have also been discussed above.

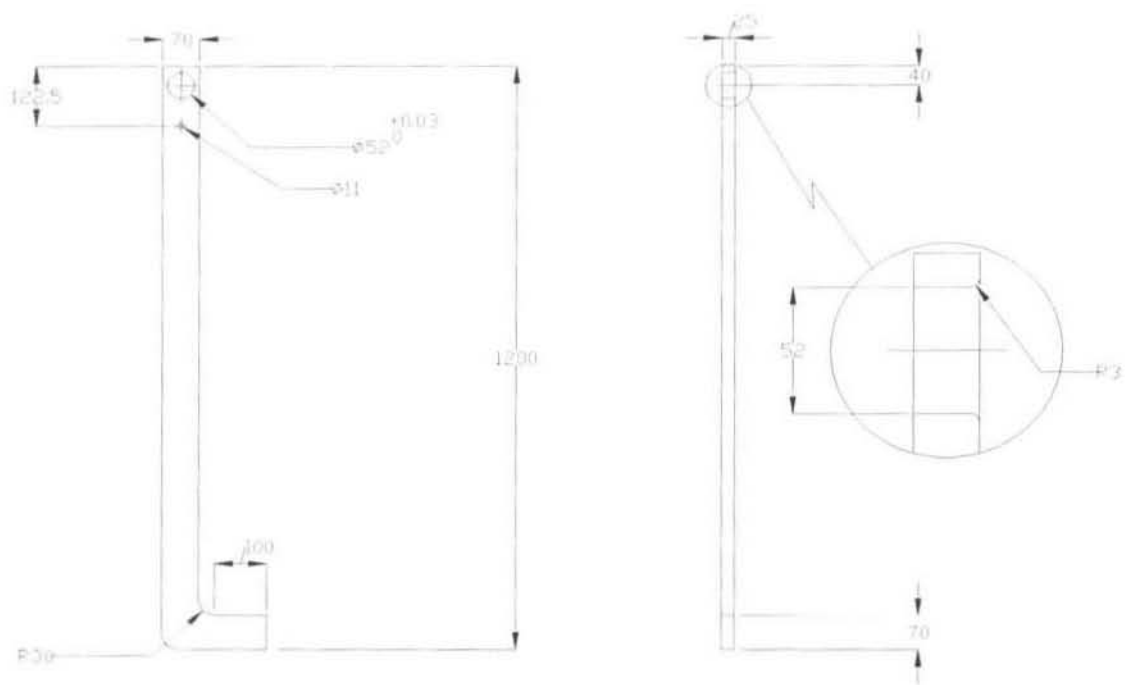


Figure 6.4.1. The arm drawing

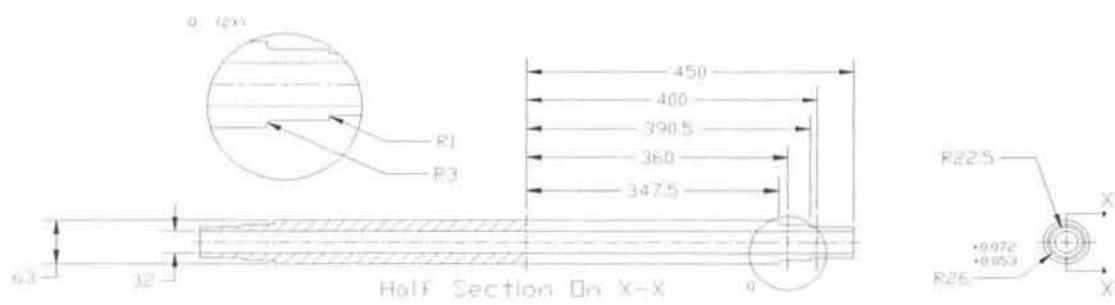


Figure 6.4.2. Shaft drawing

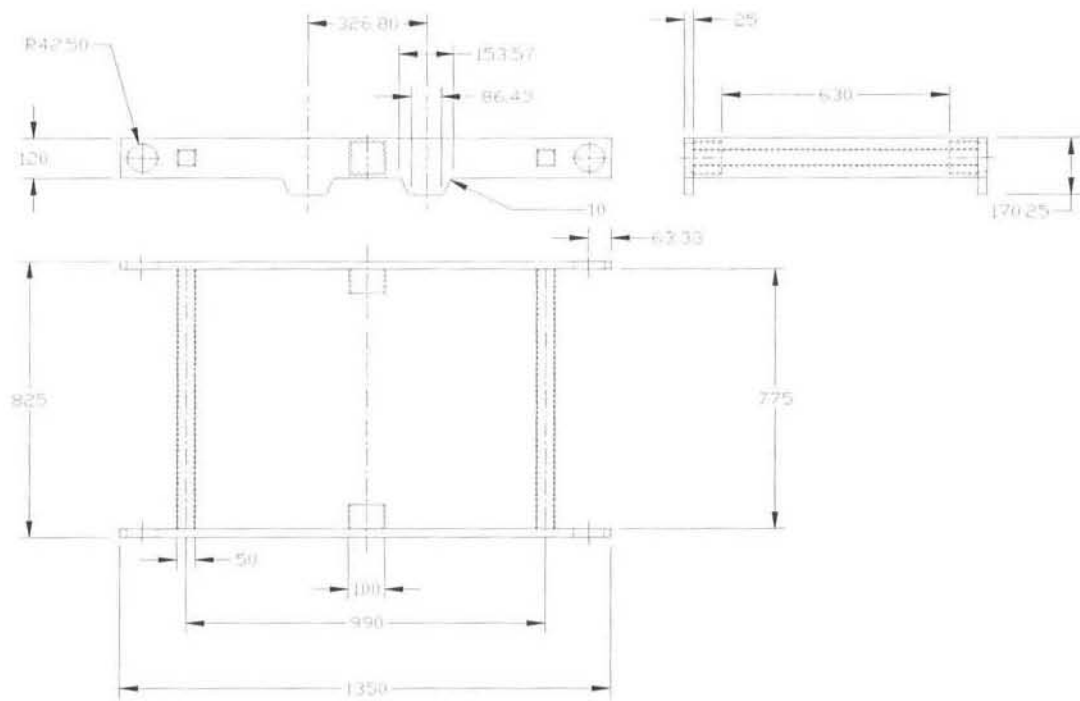


Figure 6.4.3. The chassis drawing

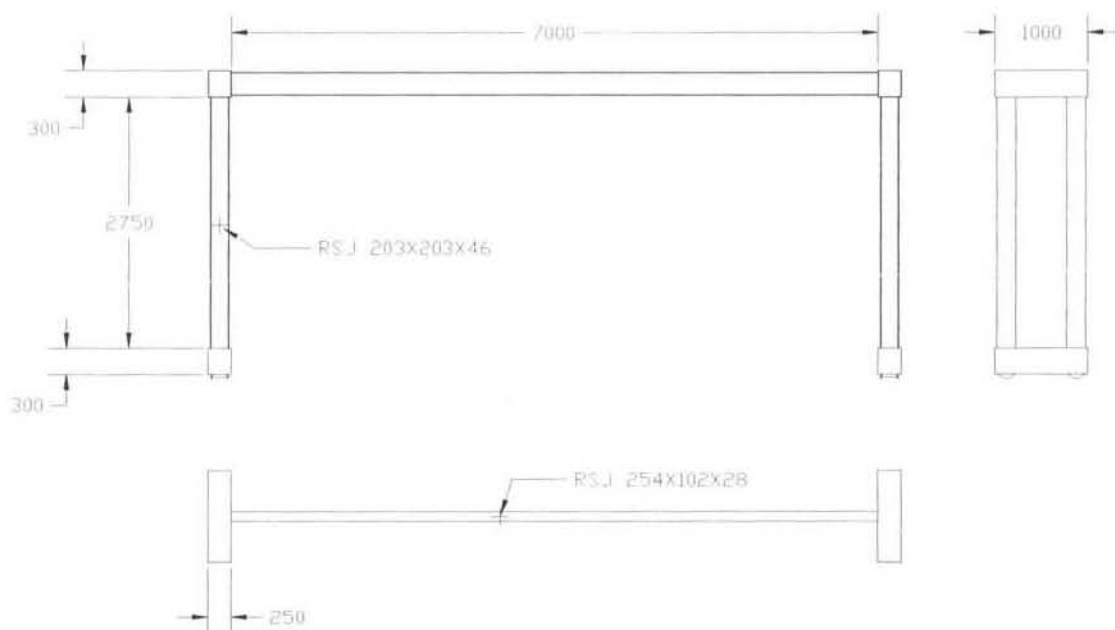


Figure 6.4.4. The gantry drawing

6.5. Conclusion

The electrical wiring diagrams for the system are available in Appendix D. After having considered the trade off between functionality and cost for the materials and components, the most appropriate current prices were obtained. As the design is for the “Future Mine”, these costs may not apply when the design is implemented, and are therefore not included. The gantry and gripper must be assembled underground, provided that the materials are provided to size and specification. Pneumatic vendors should be consulted when installing the pneumatic actuation system. One trained electrician would be required for the necessary wiring. One operator would be sufficient for the pendant control system

In the authors opinion design is a technical art. It requires the utmost from those individuals who choose to practice it in this manner. For us, as the human race, to move technically forward into the future, design must be practiced as a science. The importance of the techniques outlined in this thesis can not be ignored more espicially since the drive in industry is heavily biased toward team orientated “Total Design” efforts. There is in all certainty room for perpetual growth, in the area of Design Methodolgy. Every design team will find that the techniques available are stepping stones to the formulation of new ones. In this way the Art of Design Engineering has an insurmountable potential for growth.

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Appendix A

Figure A.1. Selected ISO fits : holes basis (after Childs, 1998)

Clearance fits										Transition fits										Interference fits									
Nominal sizes		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)		Tolerance (0.001 mm)							
Over	To	H11	e11	H9	d10	H9	e9	H8	f7	H7	f6	H7	h6	H7	k6	H7	n6	H7	p6	H7	s6								
(mm)	(mm)																												
-	3	+60	-60	+25	-20	+25	-14	+14	-6	+10	-2	+10	-6	+10	-6	+10	-6	+10	-6	+10	-6	+10	-6						
		0	-120	0	-60	0	-39	0	-16	0	-8	0	0	0	0	0	0	0	0	0	0	0							
3	6	+75	-70	+30	-20	+30	-20	+18	-10	+12	-4	+12	-8	+12	-8	+12	-8	+12	-8	+12	-8	+12	-8						
		0	-145	0	-78	0	-50	0	-22	0	-12	0	0	0	+1	0	+8	0	+12	0	+19	0							
6	10	+90	-80	+36	-25	+36	-25	+22	-13	+15	-5	+15	-9	+15	-9	+15	-9	+15	-9	+15	-9	+15	-9						
		0	-170	0	-98	0	-61	0	-28	0	-14	0	0	0	+1	0	+10	0	+15	0	+23	0							
10	18	+110	-95	+43	-30	+43	-30	+27	-16	+18	-6	+18	-11	+18	-11	+18	-11	+18	-11	+18	-11	+18	-11						
		0	-205	0	-120	0	-75	0	-34	0	-17	0	0	0	1	0	+12	0	+18	0	+28	0							
18	30	+130	-110	+52	-40	+52	-40	+33	-20	+21	-7	+21	-13	+21	-13	+21	-13	+21	-13	+21	-13	+21	-13						
		0	-240	0	-149	0	-92	0	-41	0	-20	0	0	0	+2	0	+15	0	+21	0	+35	0							
30	40	+160	-130	+62	-50	+62	-50	+39	-25	+25	-9	+25	-16	+25	-16	+25	-16	+25	-16	+25	-16	+25	-16						
		0	-280	0	-180	0	-112	0	-59	0	-25	0	0	0	+2	0	+25	0	+33	0	+43	0							
40	50	+180	-150	+74	-60	+74	-60	+46	-30	+30	-10	+30	-19	+30	-19	+30	-19	+30	-19	+30	-19	+30	-19						
		0	-290	0	-210	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
50	65	+190	-160	+87	-72	+87	-72	+54	-36	+35	-12	+35	-22	+35	-22	+35	-22	+35	-22	+35	-22	+35	-22						
		0	-310	0	-220	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
65	80	+220	-180	+100	-84	+100	-84	+63	-43	+40	-14	+40	-25	+40	-25	+40	-25	+40	-25	+40	-25	+40	-25						
		0	-340	0	-260	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
80	100	+250	-210	+115	-100	+115	-100	+72	-50	+46	-15	+46	-29	+46	-29	+46	-29	+46	-29	+46	-29	+46	-29						
		0	-390	0	-280	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
100	120	+280	-240	+130	-110	+130	-110	+81	-56	+52	-17	+52	-32	+52	-32	+52	-32	+52	-32	+52	-32	+52	-32						
		0	-400	0	-300	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
120	140	+300	-260	+145	-120	+145	-120	+96	-66	+58	-21	+58	-36	+58	-36	+58	-36	+58	-36	+58	-36	+58	-36						
		0	-450	0	-340	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
140	160	+320	-280	+160	-130	+160	-130	+108	-76	+66	-25	+66	-40	+66	-40	+66	-40	+66	-40	+66	-40	+66	-40						
		0	-460	0	-350	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
160	180	+340	-300	+175	-145	+175	-145	+120	-86	+72	-34	+72	-49	+72	-49	+72	-49	+72	-49	+72	-49	+72	-49						
		0	-480	0	-370	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
180	200	+360	-320	+190	-160	+190	-160	+135	-96	+81	-44	+81	-59	+81	-59	+81	-59	+81	-59	+81	-59	+81	-59						
		0	-500	0	-380	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
200	225	+390	-360	+210	-180	+210	-180	+150	-100	+96	-39	+96	-54	+96	-54	+96	-54	+96	-54	+96	-54	+96	-54						
		0	-510	0	-400	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
225	250	+400	-380	+230	-200	+230	-200	+165	-110	+108	-49	+108	-64	+108	-64	+108	-64	+108	-64	+108	-64	+108	-64						
		0	-520	0	-410	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
250	280	+420	-390	+250	-220	+250	-220	+180	-120	+120	-59	+120	-74	+120	-74	+120	-74	+120	-74	+120	-74	+120	-74						
		0	-530	0	-420	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
280	315	+430	-400	+270	-240	+270	-240	+200	-130	+130	-69	+130	-89	+130	-89	+130	-89	+130	-89	+130	-89	+130	-89						
		0	-540	0	-430	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
315	355	+460	-430	+290	-260	+290	-260	+220	-140	+140	-79	+140	-99	+140	-99	+140	-99	+140	-99	+140	-99	+140	-99						
		0	-550	0	-440	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
355	400	+480	-450	+310	-280	+310	-280	+240	-150	+150	-89	+150	-109	+150	-109	+150	-109	+150	-109	+150	-109	+150	-109						
		0	-560	0	-450	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
400	450	+500	-460	+330	-300	+330	-300	+260	-160	+160	-99	+160	-119	+160	-119	+160	-119	+160	-119	+160	-119	+160	-119						
		0	-570	0	-460	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							
450	500	+520	-480	+350	-320	+350	-320	+280	-170	+170	-109	+170	-129	+170	-129	+170	-129	+170	-129	+170	-129	+170	-129						
		0	-580	0	-470	0	-134	0	-60	0	-29	0	0	0	+2	0	+20	0	+31	0	+39	0							

Reproduced from BS 4500, data sheet 4500A.

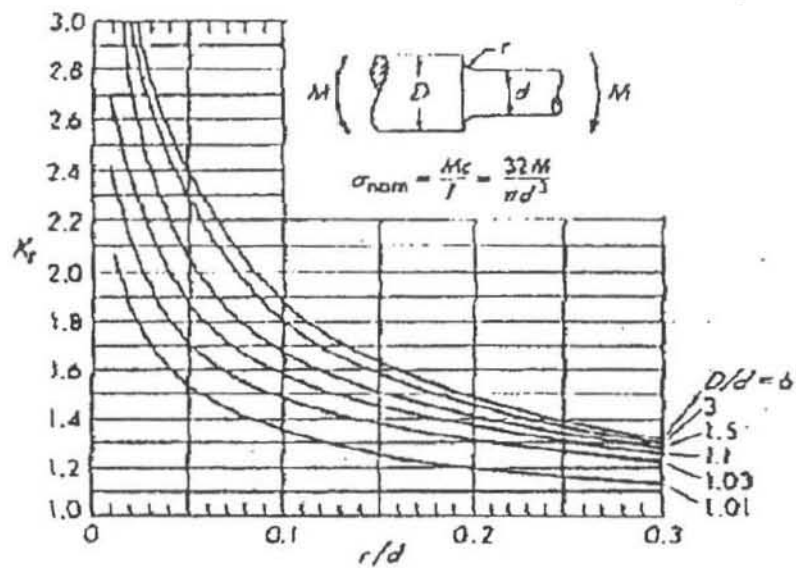
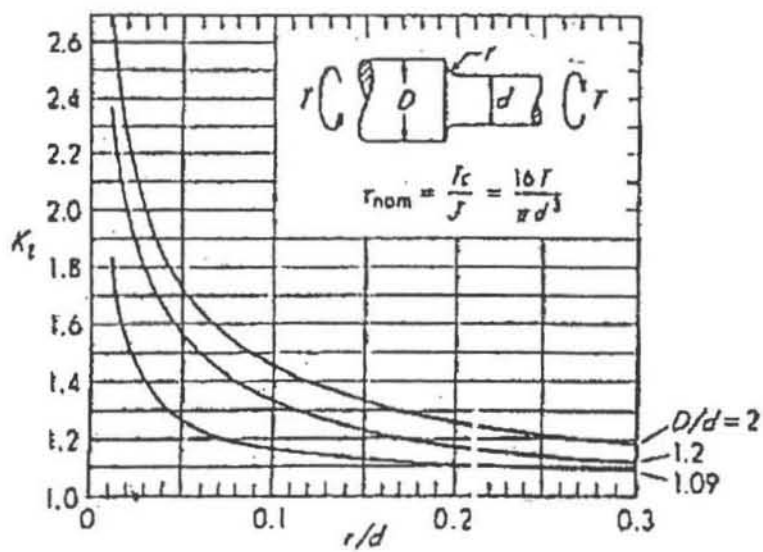


Figure A.2(a) Stress concentration factor for a shaft loaded in bending



(A) Bending; (B) Torsion

Figure A.2(b) Stress concentration factor for a shaft loaded in torsion

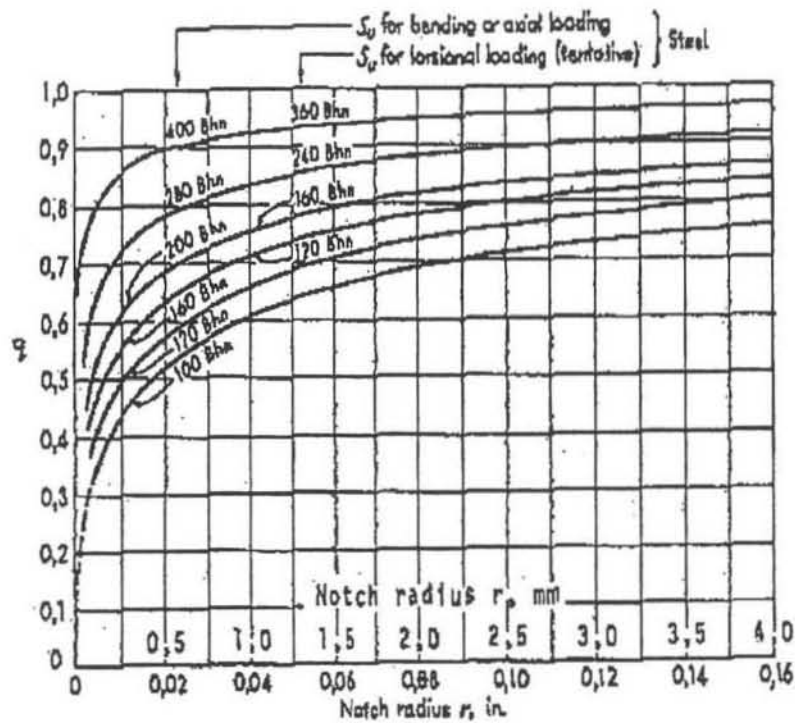


Figure A.3. Notch sensitivity factor (q)

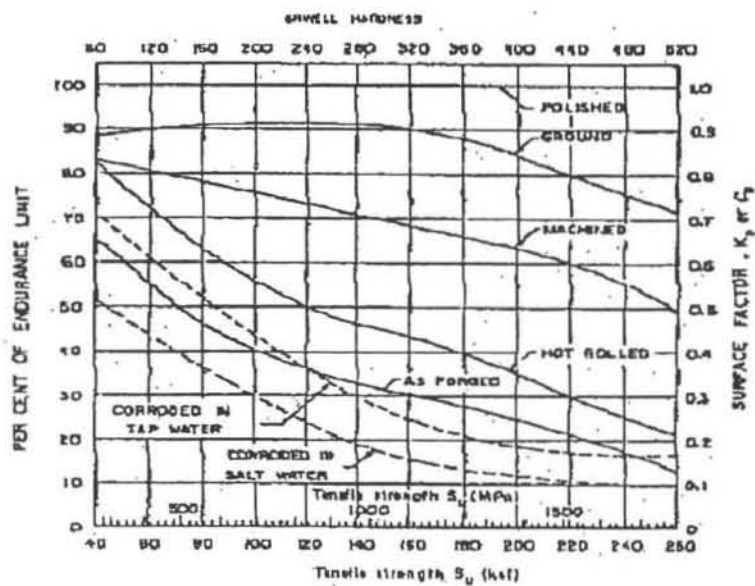


Figure A.4. Surface modification factor, C_s

Table A.1. Selected Universal Beam Sections (dimension in mm)

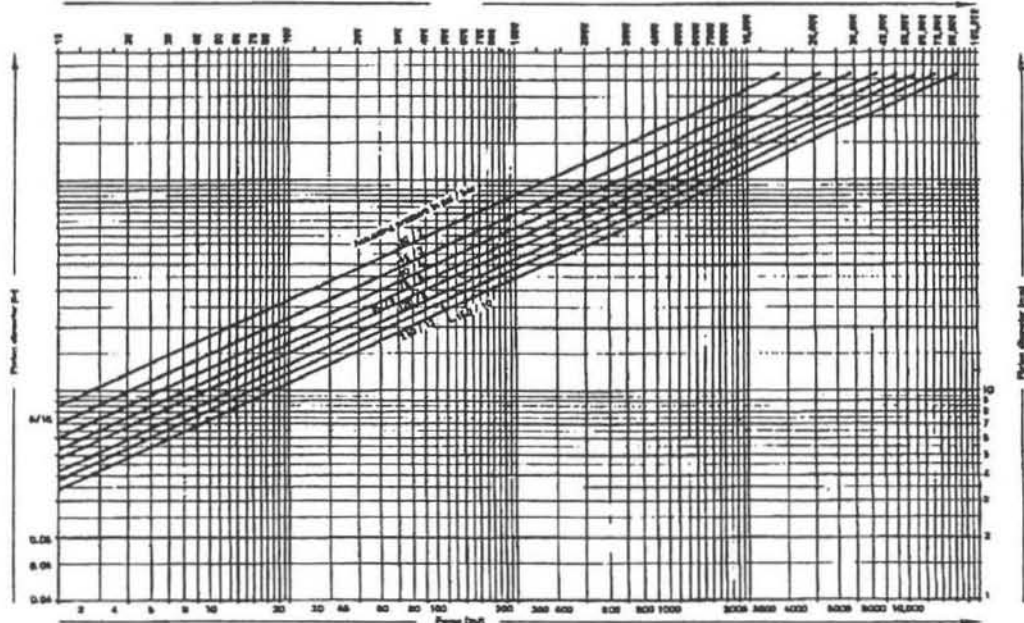
Designation	Mass	Depth	Width	Web	Flange
203 X 133	25	203.2	133.4	5.8	7.8
254 X 102	28	260.4	102.1	6.4	10
305 X 102	25	304.8	101.6	5.8	6.8
356 X 127	39	304.8	101.6	6.5	10.7
406 X 140	46	402.3	142.4	6.9	11.2
457 X 191	98	467.4	192.8	11.4	19.6

Table A.2. Selected I section Columns

Designation	Mass	Depth	Width	Web	Flange
203 X 203	46	203.2	203.2	7.3	11
203 X 203	52	206.2	203.9	8	12.5
203 X 203	59	209.6	205.2	9.3	14.4

Appendix B

Piston Force Diagram (Allows approximately 10% for frictional resistance)



Buckling Load Diagram

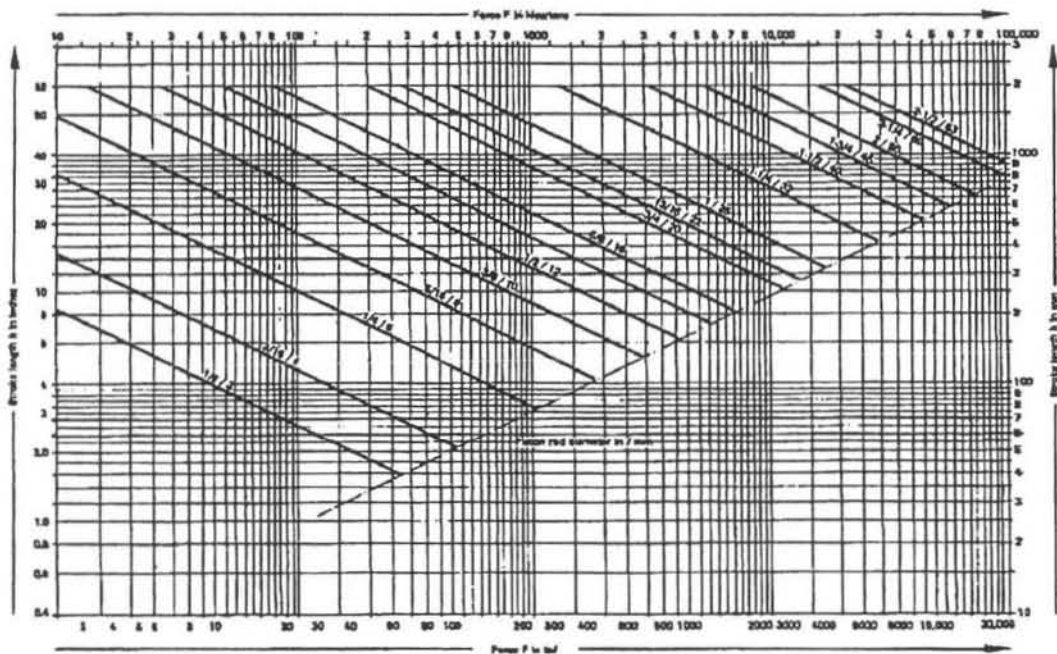


Figure B.1 and B2. Allowable piston force and buckling load on pneumatic cylinder

Appendix C

Nastran Finite Element Properties

C.1. Shear Element Properties

Shear panel properties are limited to element thickness and distributed nonstructural mass. For some analysis programs, one can also specify effectiveness factors which provide for treatment of the effective extensional area of the shear panel.

C.1.2. Membrane, Bending, Plane Strain and Plate Element Properties

These property types are all variations of plate element properties. They all require the thickness property, but the Plate type allows one to vary the thickness at each element corner. Be careful though, these corner thicknesses will be applied to each element that references this property. The stress recovery locations are measured from the neutral axis of the plate toward the top fiber. These are not offsets, they are simply the location where stresses are recovered.

NASTRAN Options

The bending stiffness ($12I/T^{**3}$) and transverse shear thickness/element thickness (TS/T) properties are used by nastran to simulate nonisotropic or sandwich material behavior. in addition to these options, MSC/N4W supports choosing different materials for the bending, transverse shear and membrane-bending coupling behavior. by default, the plate will use the material that one select at the top of the dialog box, however one can disable any of these properties, or select a different material simply by choosing the options in the lists.

C.2. Axisymmetric Element Properties

Actually, axisymmetric elements do not have any property values. The MSC/N4W property for these types is simply used to reference the desired material.

C.2.1. Solid Element Properties

Unlike the plane elements, which orient their material axes with using an angle on each element, solid element properties can reference a coordinate system to align the material axes. This difference is due to the fact that solid elements require orientation of all three principal directions. Plane elements always have their Z direction normal to the plane and can therefore be oriented with a single rotation angle. One can also choose to orient solid elements based on the directions defined by the element's corner nodes.

C.3. Line Elements

All Line Element types (Rod, Bar, Tube, Link, Beam, Spring, DOF Spring, Curved Beam, Gap, and Plot) connect two node points. Proper choice of the type depends upon the structural behavior that one want to represent. For all of these elements however, one will see one of two possible dialog boxes. The first, and simplest, creates all elements except the Bar, Beam and Curved Beam. In addition to the standard parameters, it just requires two nodes to define the element.

C.3.1. Beam Element Properties

Beam properties are identical to Bar Properties except that one can specify different properties at each end of the Beam, and one can define a Neutral Axis

Offset from the Shear Center. One must turn on the Tapered Beam option if one want to enter different properties at the second end of the Beam. If this option is off, the properties at the second end will be equal to the first end.

Care must be taken in properly specifying these properties with respect to the element axes. For MSC/N4W, I_1 is the moment of inertia about the elemental Z axis, which will resist bending in the outer fiber in the elemental Y direction. Some people look at this as the moment of inertia in Plane 1, the plane formed by the elemental X and Y axes. Distributed, nonstructural mass (per unit length) can also be specified.

One can specify up to four Stress Recovery locations in the plane of the element cross section. If one just specify the first location, and leave the remaining ones blank or zero, MSC/N4W will automatically assign the remaining three locations with positive and negative combinations of the location that one specified. This feature automates stress recovery for the four corners of a rectangular cross section.

The Neutral Axis Offsets should be specified in the local Beam Coordinate system, based upon the Orientation Node or Vector for the particular elements. This Offset is only used to Offset the Neutral Axis from the Shear Center. The Offset of the Shear Center (and Neutral Axis) from the vector between the two Nodes defining the Beam is input on the Beam Element command, not the Beam Property command.

A graphical cross section property generator is available for this property type (as well as Bar and Curved Beam). MSC/N4W can automatically compute the cross section properties and stress recovery locations for common or arbitrary shapes. The common shapes include rectangular, trapezoidal, circular, and hexagonal bars and tubes, and structural shapes such as I, C, L, T, Z and Hats. Required input for these standard shapes is shown below.

An arbitrary shape requires creating a surface before entering Model Property, and then selecting General Section, pushing the Surface button, and selecting the surface. Whether one select a common or arbitrary shape, one can have MSC/N4W draw the cross section by hitting Draw. An error in the input will prevent drawing of the cross section. This dialog box can also be used to define the Stress Recovery locations and orientation vector direction.

Stress Recovery and Reference Point

The Stress Recovery section of this dialog box allows the selection of stress recovery locations at standard points on the cross section. By hitting the Next button, MSC/N4W will move the location to the next standard point. Whether one specify Stress Recovery locations here or not, they still have the option to input values directly on the previous dialog box.

The Reference Point is only used when Mesh Attributes are assigned to a curve (Mesh Mesh Control Attributes Along Curve). The Reference Point provides an easy method to automatically define the Shear Center/Neutral Axis offset for beams that are automatically meshed onto a curve.

When a curve is meshed containing Mesh Attributes, and the Offsets method has been set to Location, MSC/N4W will place the Reference Point on the line joining the two nodes, and then calculate the offset of the shear center from this point. The result is stored on the element record as the Shear Center/Neutral Axis Offset.

The offset stored on the element record calculated from the Reference Point moves both the Neutral Axis and Shear Center from the line joining the two nodes of the beam. The offset stored on the property record and calculated when Compute Shear Center offset is checked offsets the Neutral Axis from the Shear Center.

The Attributes Along Curve command also has the capability to place the Reference Point at a distance from the line joining the two nodes of the beam by setting y and z values.

Orientation Direction

This section simply allows one to specify the direction of the orientation vector. This is very important since an inappropriate direction of the vector with respect to the beam mesh will result in erroneous results. The Cross Section Definition dialog box provides a visual representation of the required direction of the orientation vector for the beams.

Change Shape

This option is only available when editing a cross section for which properties have already been calculated. This option must be turned On before any properties can be changed. Once this option is selected, MSC/N4W will use the cross section generator to calculate new properties when exiting this dialog box via the OK button. If one simply want to edit stress recovery locations or orientation, MSC/N4W will use stored values to calculate any change in properties instead of creating an entire new set. This can save some time when making these simple changes.

Compute Shear Center Offset, Compute Warping Constant

These options are only available for Beam Properties. They are not available for Bar or Curved Beam properties since they are not supported by most analysis codes for these types of elements.

If Compute Shear Center Offset is On, MSC/N4W will use its cross section generator to compute the offset of the neutral axis from the shear center and store

the result on the property record. This is On by default since this offset can be important with certain cross sections and such programs as NASTRAN, ABAQUS, and ANSYS provide support for these offsets.

If Compute Warping Constant is On, MSC/N4W will calculate the warping constant for the cross section. This is Off by default since warping is often not important in beam analysis and there is limited support among the analysis programs for warping.

Appendix D

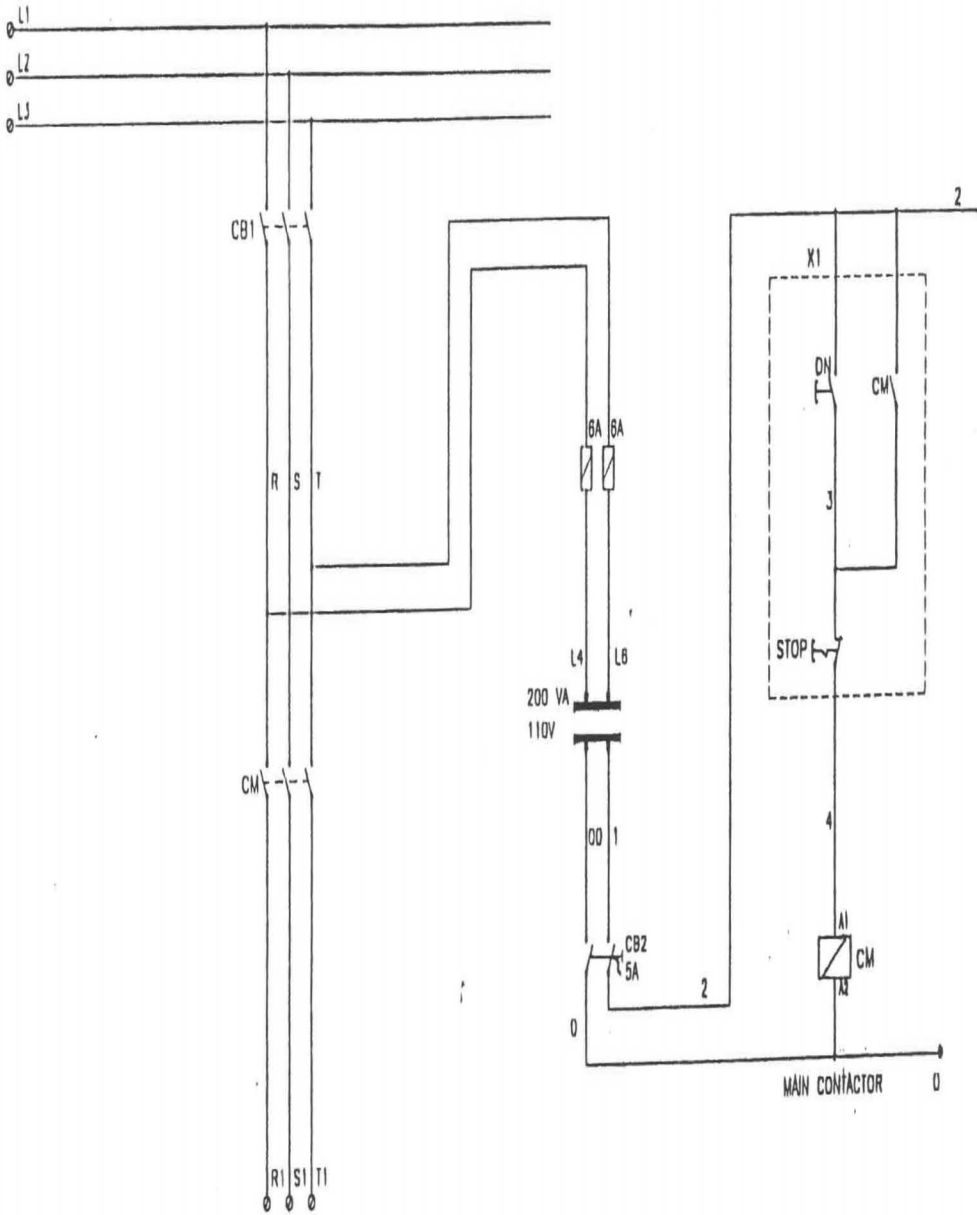


Figure D.1. Power supply and control

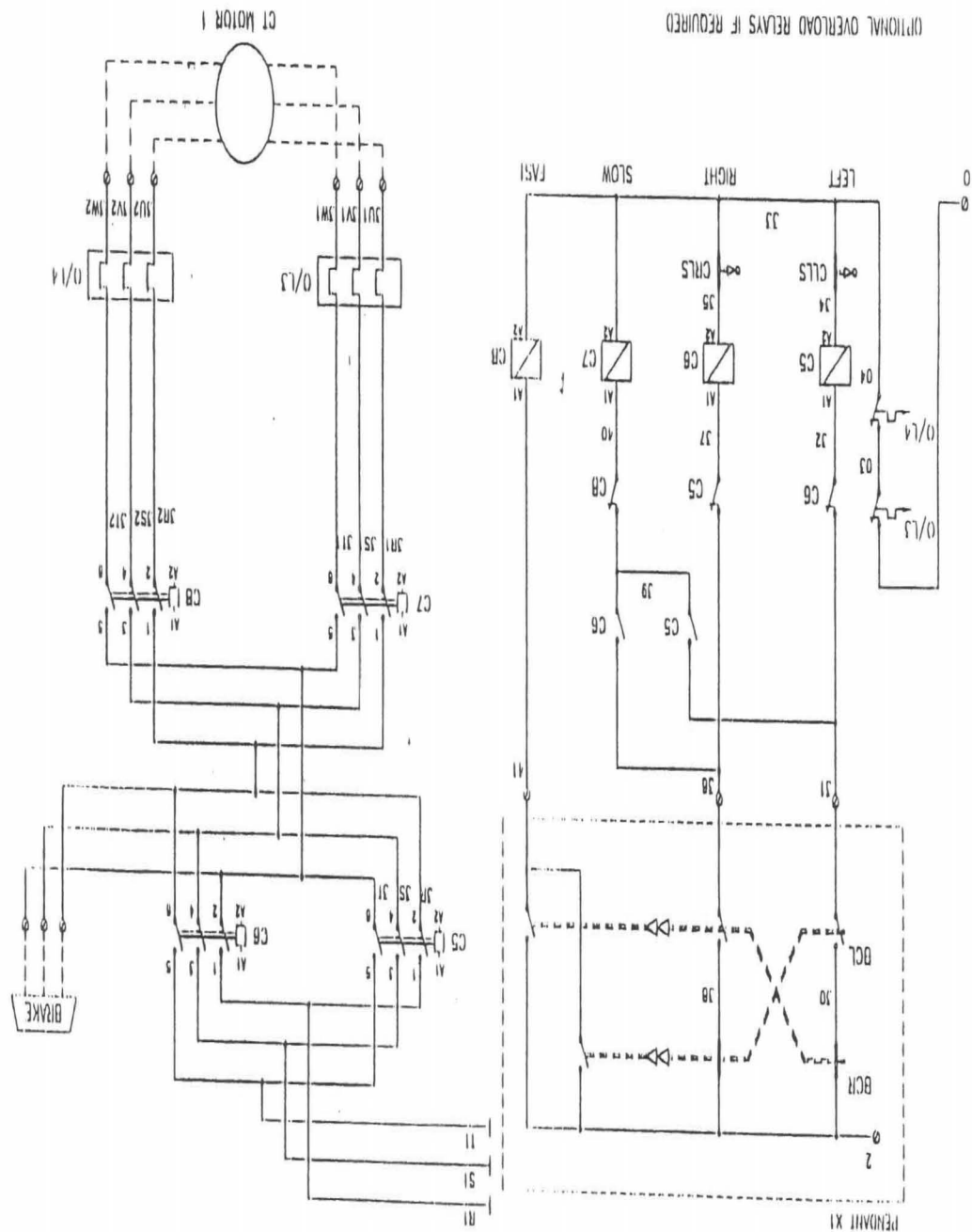
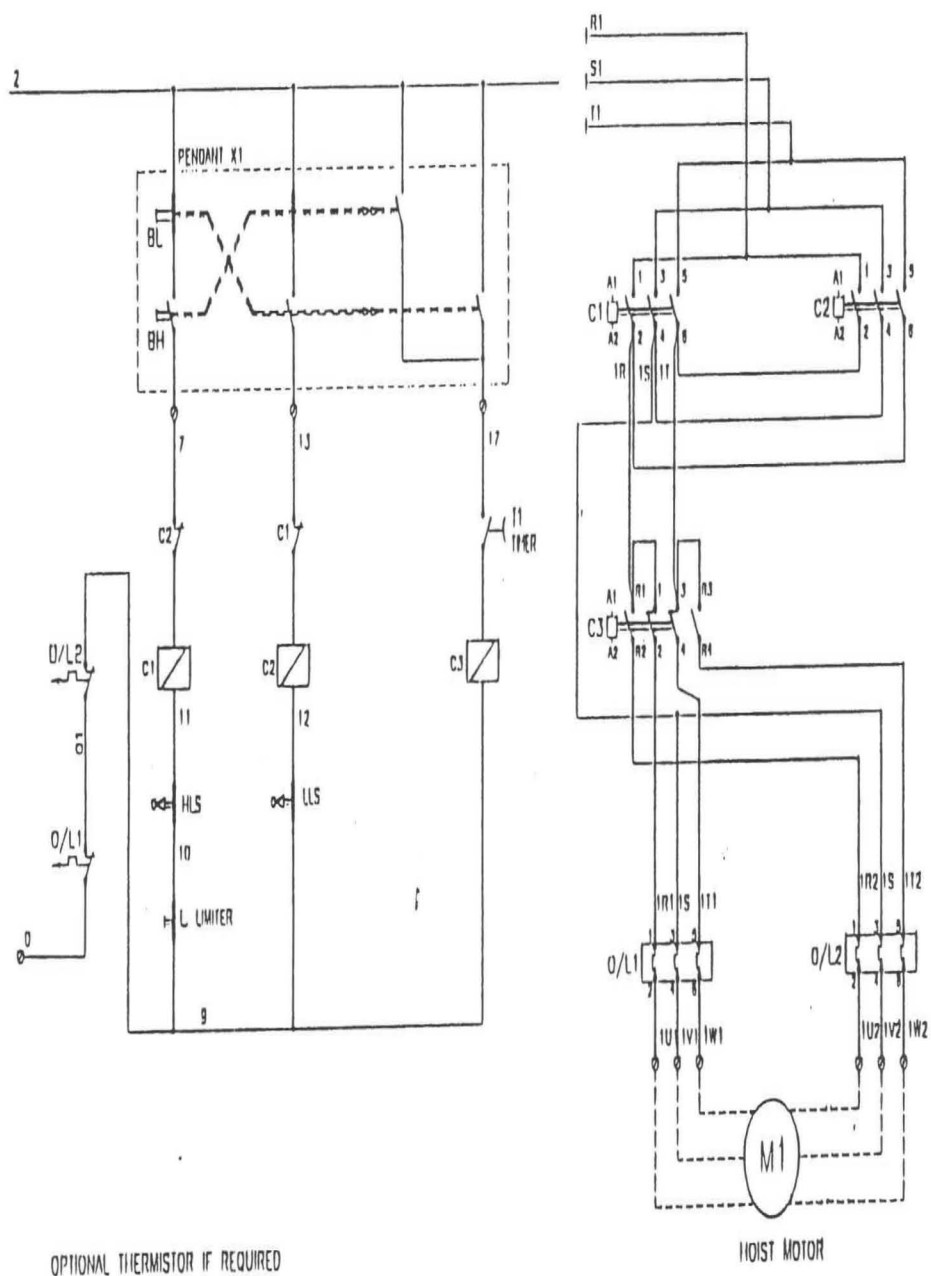


Figure D.2. Cross travel main and control
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Figure D.3. Hoist power and control

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OPTIONAL THERMISTOR IF REQUIRED

Figure D.4. Long travel power and control

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