REDESIGN AND MANUFACTURE OF A CENTRIFUGAL PUMP

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APPROVAL

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By

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A Dissertation Submitted to the University of Zambia in Partial Fulfilment of the Requirements of the Degree of Master of Engineering in Production Engineering and Management

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2001

Declaration

I, PAULOS NYIRENDA, do hereby declare that this dissertation represents my own work and that, to the best of my knowledge, it has not been previously submitted for the award of a degree at this or any other university.

Signed 22/10/01

Date ....

Dedication

To my late parents, my wife Annie and son James.

Abstract

Pumps, of varying designs, have historically been vital in fluid flow. In Zambia, typical applications could be found in domestic water supplies and industrial applications. Complete pumping units have been designed and manufactured elsewhere. The manufacture of pumps in Zambia has generally involved the production of simpler parts such as volutes and covers while complex parts such as the impeller and diffuser casings are imported. This research work was initiated in the School of Engineering at the University of Zambia (UNZA) to carry out product re-engineering on a widely used pump in Zambia. The objective was to re-design the centrifugal pump, submersible type, which could be produced locally while maintaining its current performance characteristics. Concurrent engineering and CAD-CAM software (MasterCam and SolidWorks) were used in the design process. The final pump design had fewer parts and the production process has been optimised. The Computer Numerical Control (CNC) technology was used to produce permanent moulds for casting the diffuser casings, impeller and bearing housing.

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Not forgetting the 'home front'; I thank my family for their support, confidence in me without ever ceasing to believing in a successful outcome. My beloved wife Annie and son James, you create my stability. You know what you can do for me. Thanks to all your patience

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Nomenclature

α = Inlet angle to the channel (°)
β_0 = Flow angle (°)
β_1 = Impeller blade inlet angle (°)
β_2 = Impeller blade outlet angle (°)
δ_{tot} = Total elongation (mm)
δ_{weigh} = Elongation due to weight (mm)
δ_{ht} = Elongation due to hydraulic thrust (mm)
δ_{temp} = Elongation due to temperature (mm)
σ = Thoma Cavitation Factor (dimensionless)
η = Overall pump efficiency (%)
η = Pump hydraulic efficiency (%)
ψ = Head coefficient (dimensionless)
μ = Slip factor (dimensionless)

A_I = Impeller inlet area between vanes (mm^2)
A_{II} = Impeller outlet area between vanes (mm^2)
A_{eye} = Impeller eye area (mm^2)
b_1 = Impeller inlet width (mm)
b_2 = Impeller outlet width (mm)
b_3 = Diffuser casing inlet width (mm)
C = Coefficient of load distribution (dimensionless)
C_{m3} = Meridian flow velocity (m/s)
C_{thr} = Average casing throat velocity (m/s)
d_r = Required diameter (mm)
d_{shaft} = Shaft diameter (mm)
D_1 = Impeller inlet diameter (mm)
D_2 = Impeller outlet diameter (mm)
D_3 = Diffuser casing inlet diameter (mm)
D_4 = Diffuser casing outlet (discharge) diameter (mm)
D_s = Impeller suction diameter (mm)
E = Modulus of Elasticity (N/m^2)
f = Deflection (mm)
F_{radial} = Radial thrust on impeller (N)
F_u = Up-thrust on impeller (N)
h = Ratio of keyway depth to shaft diameter
H = Head (m)
I = Moment of inertia (m^4)
K = Arbitrary constant
L = Free span (m)
M = Mass (kg/m)
N = Shaft rotational speed (rpm)
N_{psh} = Net Positive Suction Head (m)
N_{sm} = Discharge specific speed (dimensionless)
N_{c1} = First critical speed (rpm)
P_s = Shaft power (kW)
Q = Pump discharge (m^3/h)
r = Impeller radius (mm)
s_1 = Impeller vane thickness at inlet (mm)

s_2 = Impeller vane thickness at outlet (mm)
S = Tensile or compressive stress (N/m^2)
S_s = Shear stress (N/m^2)
S_{max} = Maximum tensile or compressive stress (N/m^2)
S_{smax} = Maximum shear stress (N/m^2)
S_{sy} = Yeild strength in shear (N/m^2)
t = Radial clearance between impeller and casing (mm)
u_2 = Impeller peripheral velocity (m/s)
V_{eye} = impeller eye flow velocity (m/s)
w = Ratio of keyway width to shaft diameter
W = Weight of the rotating element (N)
z = Number of impeller vanes
Z = Number of casing diffuser vanes

Chapter 1

Introduction

1.1 Introduction

Pumps, of varying designs, have historically been vital in fluid flow. Two main categories are realised, namely, dynamic and rotor-dynamic pumps. This project was focused on the redesign and manufacture of a rotor-dynamic pump - the centrifugal pump. The pump derived its name from its working principle-use of centrifugal force to boost pressure between the source and the ends of a pipe network. The type considered in this research was the vertical submersible pump with vane diffuser employing an open, radial impeller. This type of centrifugal pump has many uses ranging from domestic to industrial applications. Examples range from de-watering systems such as in underground mining or in domestic clean water supplies, stretching to sewer or paper mills. They are also used under stringent acidic conditions of slurries in industrial applications.

Since the Great Exhibition of 1851 [1], [2] the centrifugal pump has developed into a high efficiency machine, which can be adapted to suit almost any working condition. Nonetheless, a lot of work to improve centrifugal pump performance is still going on. In these undertakings, designs of varying complexity are made to achieve certain targets in performance. However, simpler designs do perform similar to, or even better than, the complex designs in certain applications. In these cases, the trade-off may be attributed to design and product quality, cost of production, availability of materials and ability to optimise the utilisation of the available resources for production. This thesis outlines the steps in the redesign and manufacture of a simpler design of a selected centrifugal pump by taking into account the factors above.

Similar work on centrifugal pump manufacture was carried out before in the Department of

Mechanical Engineering at The University of Zambia (UNZA) [3], [4], [5]. The main differences were in the type of pump, the methods used for production, which were all conventional as apposed to the technology in this project in which the development process was adapted to Computer Numeric Control technology. In addition, this project was a redesign problem in which project re-engineering was carried out in an effort to come up with a simpler, cheaper but still efficient design that could be adapted to the available technology for manufacture. This research was unlike the earlier designs where the process development aimed at the reproduction of an existing design. The type of pump on which this earlier work was based on was the horizontal centrifugal pump for domestic applications. These were simpler designs in that fewer considerations were made in the development process for their manufacture compared to the current design in this project. In vertical pumps covered in this project, sealing methods from the outside media were the key distinction from the past work. A vertically oriented centrifugal pump has, in addition, forces that cause residual forces resulting from the weights of the, rotating elements in it. Their treatment called for additional considerations when designing to take care of their effects on 1he pump's stability during operation. The earlier design did not concentrate much on critical component strength calculations but on production methods and tolerances to ensure assembly as well as manufacture. In this project, methods were developed through three major areas of product development: market survey, conceptual and detailed design, and production. Computer Numerical Control (CNC) Technology was incorporated to ensure good quality in the product with consistency in the output. This technology was also used in developing the processes for tooling manufacture, via permanent moulds, for production of the pump parts such as the impeller.

1.2 The Importance of this project

Zambia is a good importer of pumps as well as other small metallic products. There are some local manufacturing firms that produce small metallic products such as bolts and desks for the local furniture industry. Unfortunately, none of these companies manufacture complete centrifugal pump units. Zambia's companies known to be selling centrifugal pumps are mainly trading, maintenance and distribution points. Only a few companies produce simpler parts such as end covers, simple motor and single-volute casings but not complex parts like the impellers, diffuser casings and intricate bearing housings locally. This project was carried out with the aim of manufacturing centrifugal pumps locally using available technology at lower costs and of very good quality (efficiencies of 55-65%, nominal range) with a view to achieving a competitive selling price. This meant adapting the manufacture of the most widely used pump design to the available technology in the country: With the ever-growing demand for centrifugal pumps (outlined in Chapter 2), this would serve the Zambian market, which is currently solely dependent on imported pumps. Thus, high levels of production to meet the needs of the local as well as external market needs, were noted, hence the adaptation to the CNC Technology through product re-engineering.

Increased productivity is generally the justification for using the CAD/CAM systems [6], [7], [8]. Productivity increases with faster turn around, better quality and more accuracy. These systems allow for rapid development and editing of designs and documentation. When a 3-D geometry model is produced in the design process, it then becomes a common element for engineering analysis, machining process planning (including CNC part programming), documentation (including engineering drawings), quality control, and so on. The tight coupling of CAD and CAM considerably shorten the time it takes to bring anew product to the market. This country has not exploited the use of CNC Technology. The type of CNC machine in the University of Zambia is a Machining Centre: vertical milling machine. Th1s is new technology to the country that needs to be incorporated into its Production Technology. To enhance competition globally, Zambia could only join the race in the market niche by coping with the dynamic nature of technology in the area of manufacturing. The incorporation of CAD/CAM in this project therefore, justified the importance of the project. With the manufacture of the pump locally, this would be a source of foreign exchange for a large numbers of export as well as a positive contribution to the needs of the local market. An average Zambian would afford to buy a cheaper product at a competitive price. This would be a major economic benefit of manufacturing this pump locally.

1.3 Project Objectives

This project was focused on the redesign and manufacture of a centrifugal pump in Zambia with the view to developing a globally competitive product. Zambia lacked the technological capability to produce these pumps locally, thus this project was meant to test the ability to produce using this new technology. It encompasses redesign of an existing submersible pumps as well as development of unique manufacturing methods of the different parts to adapt their production to the available technology. The objectives of this research and development work were to:

- Carry out a literature survey of available pump designs,
- Determine the most suitable pump designs that could be competitively produced in Zambia,
- Re-design the pump,
- Develop processes for production, including tooling.

1.4 Methodology of the Research Work

The following steps were undertaken in the redesign and production process of the pump.

Step 1: Market Survey

A market survey on the pump types available on the Zambian market (Section 2.3.2) was carried out from which the most commonly used type was selected for design. The companies from which this information was obtained were Bestobell Company (Z) limited, AFE (z) limited and Flygt (Swedish) Company. The market information used was based on five years period up to 1998. Its specifications were then considered the 'customer requirements'.

Step 2: Conceptual Design

The existing design was re-engineered. This stage involved detailing the design while simultaneously developing production capability, field-support capability and quality. It involved the integration of product design and process planning into one common activity.

Step 3: Pump Design

The established 'customer requirements' were translated into design specifications, which were then used to estimate the overall expected efficiency. This was followed by design of the parts in the following order: Open Radial ***Impeller*** design, ***Shaft*** design on the basis of effects of static and dynamic forces, including critical speeds and the ***Vane Diffuser Casing*** design to size it for the contemplated application. The application consideration in this design was relatively clean water with solids in suspension up to 18mm diameter. This included dimensions for the suction and delivery nozzles. ***Bearing Selection*** followed force analysis on the shaft. The ***Bearing Housing*** (also the upper diffuser) was sized with respect to the shaft and diffuser casing dimensions. The ***Motor Housing*** and ***Pump Housing*** designs were then carried out. For the ***Other Parts,*** no complex calculations were used. They were simply sized to achieve optimum use.

Step 4: Manufacturing

Prior to production, the designed parts were tested for assembly using production solid modelling software called ***SolidWorks.*** A few adjustments were made to dimensions to accommodate standard parts such as the bearings and mechanical seals. Production process development started with the permanent moulds that were to be used for the production of the Diffuser Casing and the Impeller. The shaft would be turned on the lathe while the pump casing; motor casing, end-cover and the bearing housing would be sand-cast.

Chapter 2

Pump Design Process

2.1 Introduction

In this chapter the design process of the pump is highlighted. In the process of pump design, it was firstly found of prime significance to establish the most widely used design, which would be used in the process development in this project. In this research, the most popular rotor-dynamic pump on the market was selected. The importance of the market survey was to determine the most widely used pump in the country so that its process of manufacture could be adapted to the available technology in the country. The overall objective was to develop the process for manufacture of this popular design of the product in order to produce it locally in an effort to serving the Zambian market with a cheaper product embodying good quality. This meant carrying out product re-engineering so as to adapt this design to the available technology. Steps undertaken in determining this pump are outlined in the following paragraphs. In this chapter, market survey results are reduced to the final pump choice based on the popularity of a given type. The production process development considerations are also, highlighted.

2.2 Definitions and Terminology

2.2.1 Definitions

The principle underlying the performance of a centrifugal pump to raise liquids or compress gases is the rotating motion of the impeller relative to the stationary pump casing (see Figs. 2.1 and 2.2). The name of this pump was derived from its principle of operation. One of the forces is accountable: the centrifugal force. The fluid enters the pump in axial direction through the suction nozzle into the eye of the impeller or rotor. The pump's orientation can be either horizontal or vertical of course with varying design considerations for identical products. Rotordynamic pumps are essentially high-speed machines containing a rotating element carrying one or several rotors -impellers (Part 2 in Fig. 2.1 and Table 2.1). The rotating element is surrounded by a casing, (Part 1, Figs 2.1), that is equipped with inlet and outlet branches. The casing and its various parts serve to convert as efficiently as possible, velocity energy of the liquid leaving the circumference of the rotating impeller(s) into pressure energy. The impeller has vanes (blades) shaped in such a way as to guide the liquid, and to impart energy during the guidance.

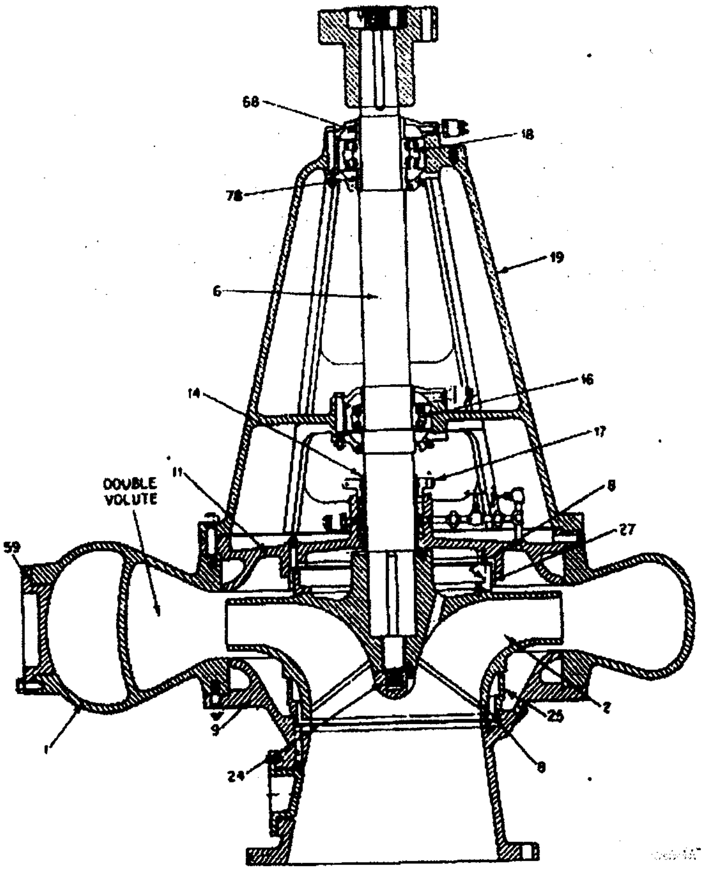


Figure 2.1 a: Terms used in centrifugal pump *(Source: Warman Pump Co., 1996)*

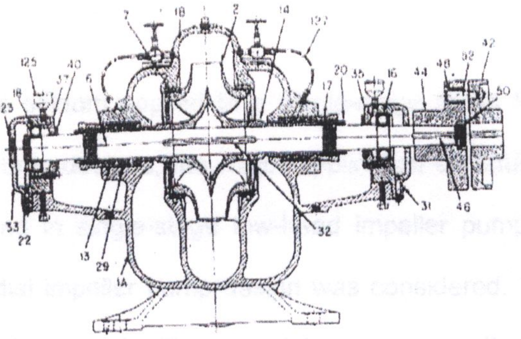


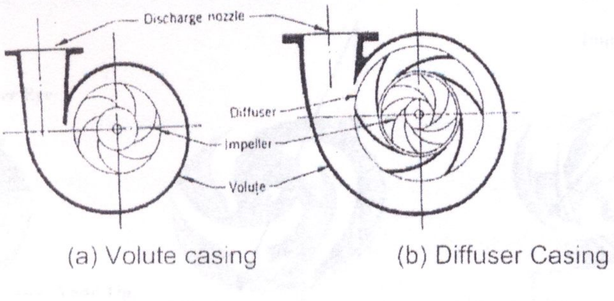
Figure 2,1 b: Terms used in centrifugal pumps *(Source Warman Pump Co" 1996)*

The following Table 2.1 gives the names of the individual parts in Figure 2.1:

TABLE 2.1: Parts Listing

Item No.	Name of Part	Item No.	Name of Part
1	Casing	25	Suction head ring
1A	Casing (lower half)	27	Stuffing box ring
1B	Casing (upper half)	29	Lantern ring
2	Impeller	31	Bearing housing
5	Pump shaft	32	Impeller key
6	Pump shaft	33	Bearing housing
7	Casing ring	35	Bearing cover
8	Impeller ring	37	Bearing cover
9	Suction cover	40	Deflector
11	Stuffing box cover	42	Coupling half
13	Packing	44	Coupling half
14	Shaft sleeve	46	Coupling key
16	Bearing (inboard)	48	Coupling bushing
17	Gland	50	Lock nut
18	Bearing (outboard)	52	Coupling pin
19	Frame	68	Shaft collar
20	Shaft sleeve nut	78	Bearing spacer
22	Bearing lock nut	123	Bearing end cover
24	Impeller nut	127	Seal pipe (tubing)

The liquid is subsequently accelerated into a circular motion by the rotating impeller blades and collected in the volute or diffuser. Fig. 22 shows the Volute and Diffuser Castings.



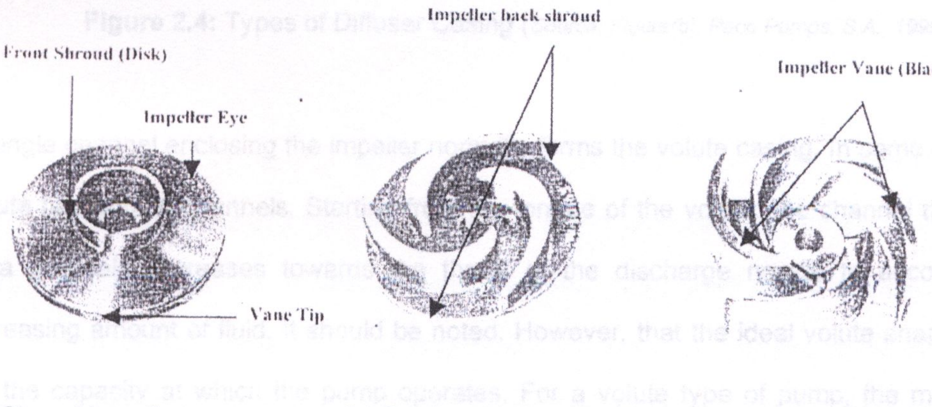
(a) Volute casing (b) Diffuser Casing

Figure 2.2: Main Casing Types

A centrifugal pump volute increases in area from its initial point until it encompasses the full 360° around the impeller and then flares out to the final discharge opening. The wall dividing the initial section and the discharge nozzle portion of the casing is called the tongue of the volute. The diffusion vanes and concentric casing of a diffuser pump fulfil the same function as the Volute casing in energy conversion.

The diffuser (Figure 2.4) is seldom applied to a single-stage radial flow pump. For certain high-pressure multistage pump designs, the major application of diffusion vane pumps is in vertical diffusion pumps and in single-stage low-head impeller pumps. In this project, the diffusion vane, vertical, radial impeller pump design was considered. The fluid finally leaves the pump through the discharge nozzle. The inlet of the pump as well as the volute (or diffuser region) is part of the pump casing. The choice of the type of pump and impeller design is justified in Section 2.3.2 of this report.

The impeller is mounted on the shaft (usually in light push fit), which is connected to the driver. Drivers in popular use are Electro-motors or small engines. Seals (Part 89, Fig. 2.1) or wearing rings are fitted on the impeller and casing to restrict flow under high pressure from leaking back to the impeller inlet or out of the pump casing. The impeller comprises a disc on which several blades or vanes are mounted. The vanes can either have discs both sides or on the back only. In the earlier case, the type of impeller is called double-shrouded or closed impellers whereas the latter are called open impellers. Within the class of open impellers are two types: fully open or semi-open impellers (Figure 2.3):

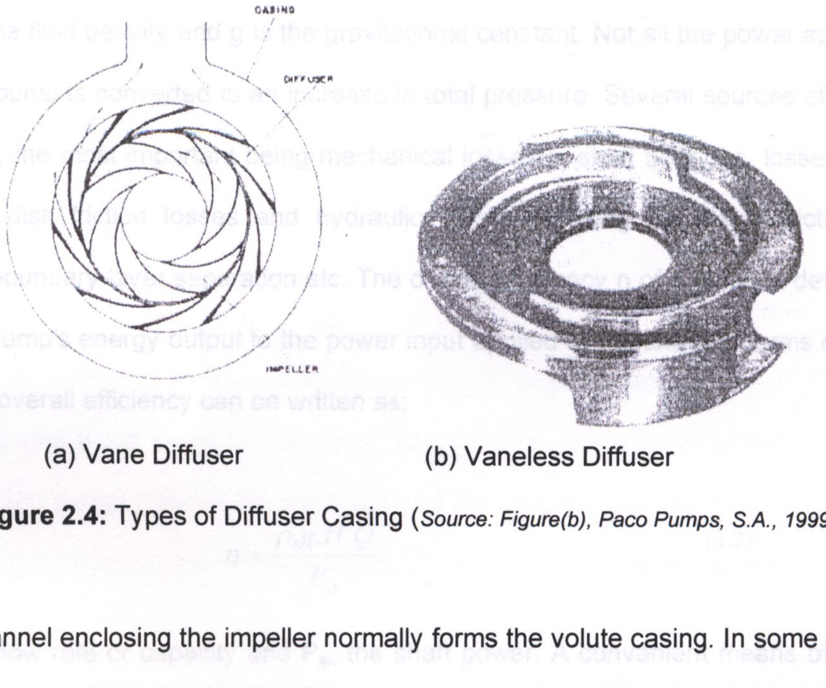


(a) Closed Impeller (b) Semi-open Impeller (c) Fully open Impeller

Figure 2.3: Impeller types *(Source: American Worthington Pump Co., 1997)*

Impellers contain blades or vanes (from 1 to 12 for most applications) which are either curved backwards or straight. They may be made of single or double curvature (twisted at the suction ends). The blades are fitted to the hub (or back shroud) of the impeller.

The discharge region or the pump collects the fluid as it leaves the impeller. This is generally called the recuperator: volute, circular or diffuser casing. The shape of this region is such that the high velocity of the fluid is partly converted to static pressure by gradual expansion. Of the three types two are popular: volute and diffuser casing (vane and vaneless diffuser, Fig 24).



(a) Vane Diffuser (b) Vaneless Diffuser

Figure 2.4: Types of Diffuser Casing (*Source: Figure(b), Paco Pumps, S.A., 1999*)

A single channel enclosing the impeller normally forms the volute casing. In some designs the volute has double channels. Starting from the tongue of the volute, the channel through-flow area gradually increases towards the throat of the discharge nozzle to account for the increasing amount of fluid. It should be noted. However, that the ideal volute shape depends Oil the capacity at which the pump operates. For a volute type of pump, the major part of conversion from kinetic energy to pressure occurs in the discharge nozzle. In a vaned diffuser pump, the impeller discharges into a channel equipped with stationary vanes and the energy conversion predominantly takes place between the diffusion vanes. However, for the vaneless diffuser, energy conversion depends on the change in the widths between the impeller discharge (casing inlet) and the diffuser outlet. Normally the width increases towards the discharge so the kinetic energy available at its inlet is converted into pressure at discharge.

2.2.2 Terminology

The head H, of the pump is defined as the height (in meters) against which the pump can lift a fluid. It is related to the increase in total pressure ΔP_o of the fluid by

$$H = \frac{\Delta P_0}{\rho . g} \qquad (2.1)$$

where ρ is the fluid density and g is the gravitational constant. Not all the power applied at the shaft of the pump is converted to an increase in total pressure. Several sources of losses can be identified; the most important being mechanical losses in shaft bearings, losses attributed to leakage, disk friction losses and hydraulic losses resulting from skin friction, mixing processes, boundary layer separation etc. The overall efficiency η of a pump is defined as the ratio of the pump's energy output to the power input applied at the shaft. In terms of delivered head H, the overall efficiency can be written as:

$$\eta = \frac{\rho . g . H . Q}{P_{sh}} \qquad (2.2)$$

With Q the flow rate or capacity and P_{sh} the shaft power. A convenient means of presenting the overall performance of a specific pump is the use of characteristic curves, which show the relation between flow rate, rotational speed, delivered head, shaft power and fluid properties. The standard for centrifugal pumps is to plot the head, the efficiency and the shaft power as a function of flow rate, for a specific pump (Figure 2.5). Efficiency, for a given specific speed, can be estimated from this figure. For example, for N_{sm} = 30 at Q = 200GPM, efficiency η = 74%. The flow rate for which the efficiency is largest is called the Best Efficiency Point (BEP).

In pump design, the BEP is called the design point as the associated values of H, Q, and N (shaft rotational speed) are the design specifications or what would be termed the 'ideal' (targeted) parameters.

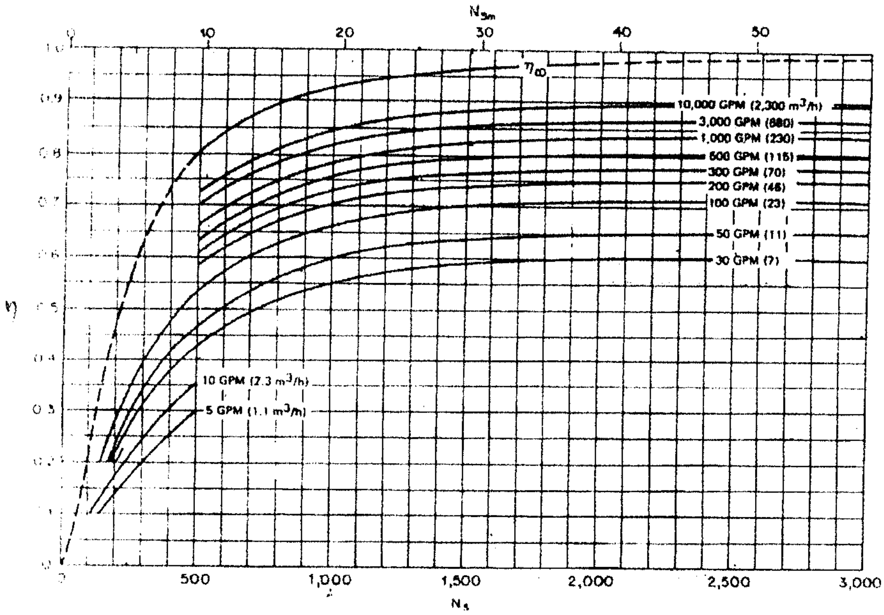


Figure 2.5: Centrifugal Pump Efficiency Plots *(Source' Worthington Pumps Co. 1995)*

A dimensional analysis reveals the existence of several scaling laws for grouping geometrically similar machines [9]. However, specific parameters were utilised in this project namely: the Discharge Specific Speed (N_{sm}) and the Suction Specific Speed (S_m) given by equation (2.3). Generally, the Discharge specific speed is referred to as simply the Specific Speed.

$$N_{sm} = \frac{N\sqrt{Q}}{H^{0.75}} \tag{2.3a}$$

$$S_m = \frac{N.\sqrt{Q}}{N_{psh}^{0.75}} \tag{2.3b}$$

where N is the shaft speed rotational speed and N_{psh} is the Net Positive Suction Head at the pump suction.

It has been proved that geometrically similar machines with similar internal flow conditions have the same specific speed value [4]. It is used as a type number to classify pumps of different designs irrespective of the actual diameter. These values of specific speed refer to a pump operating at BEP. Figure 2.6 shows a well known 'efficiency chart' in which, the maximum efficiency of numerous pumps is plotted against specific speed and lines of constant flow rate are drawn.

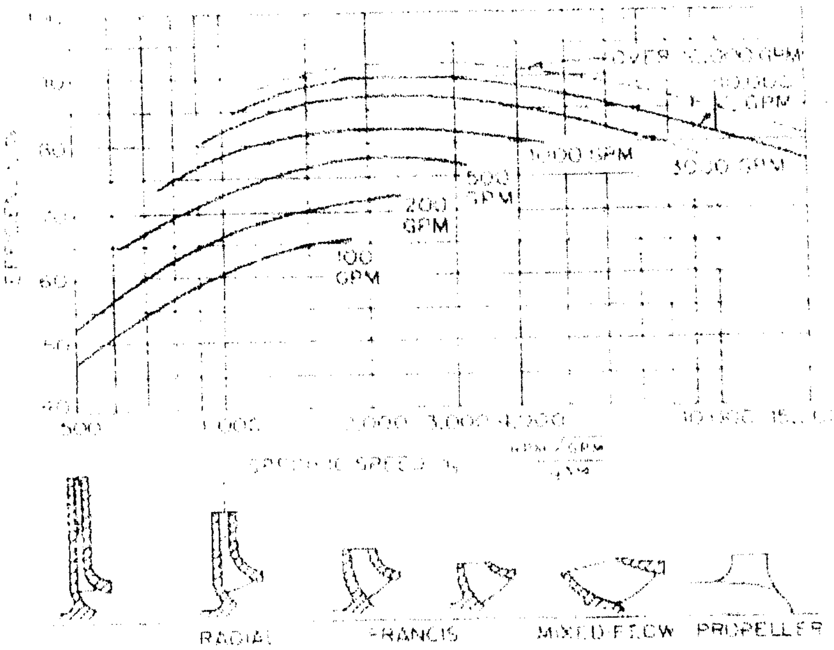


Figure 2.6: Efficiency (BEP) versus Specific Speed (*Source: Tsurumi Pumps Go. Japan, 1986*)

Note: The impellers coincide with the ranges of n_s e.g. radial impeller falls between $500 \leq n_s \leq 1500$ and Francis impellers $1500 \leq n_s \leq 4000$

Figure 2.6 shows a clear dependency on specific speed of the maximum attainable efficiency. There is also a correlation between the specific speed and the width-diameter ratio of the impellers. Radial impellers are found to have low specific speed values [500 $\leq n_s \leq$1500 in Fig.2.6] (low capacity, high head) whereas axial impellers are associated with large specific speeds. Mixed flow impellers form an intermediate class in terms of geometrical design and their intermediate values of specific speeds [4000 $\leq n_s \leq$ 15,000]. The change in maximum efficiency, with flow rate for a given class of similar pumps (at constant discharge), is directly related to the varying size of pumps within the class. For example, a Radial, Francis, Mixed and Axial (propeller) pump for discharge of 10,000GPM will have respective maximum efficiencies of 84, 90, 87, and 830% (Section 2.4.2). These values may not be exact but somewhat on the lower ends for secondary causes such as surface roughness, the varying mechanical losses with impeller diameter and the fact that seal clearances are generally not scaled according to impeller diameter.

2.3 Pump Classification and Market Survey

2.3.1 Pump Classification

In general all pumps may be classified on the basis of applications they serve, the materials from which they are constructed, the liquids they handle, and even their orientation in space. All such classifications, however, are limited in scope and tend to substantially overlap each other. A more basic system of classification first defines the principle by which energy is added to the fluid, goes on to identify the means by which this principle is implemented, and finally delineates specific geometry commonly employed. This system is therefore related to the pump itself and is unrelated to any consideration external to the pump or even to the materials from which it may be constructed. In the design process both external factors as well as the materials were considered in the redesign process.

Under this system, all pumps may be divided into two major categories: ***dynamic*** and ***displacement pumps***. Dynamic types are those in which energy is continuously added to increase the fluid velocities within the machine to values in excess of those occurring at the discharge. Thus, the subsequent velocity reduction within or beyond the pump produces a pressure increase. Displacement pumps are those in which energy is periodically added by application of force to one or more movable boundaries of any desired number of enclosed, fluid-containing volumes. The result is a direct increase in pressure up to the value required to move the fluid through valves or ports into the discharge lines.

In this research work, the type of pump under consideration was the dynamic pump centrifugal. These may be further subdivided into a variety, including other special-effect pumps. Figure 2.7 presents, in outline form, a summary of the significant classifications and sub-classifications within this category [9]. Given the variety of pump classes from the foregoing, it was difficult to exactly choose the type of design to take on. In this work, the choice reduced to only the most commonly used type of dynamic pump on the Zambian market for design and manufacture. This was a single-stage, vertical submersible pump with an open, radial flow impeller.

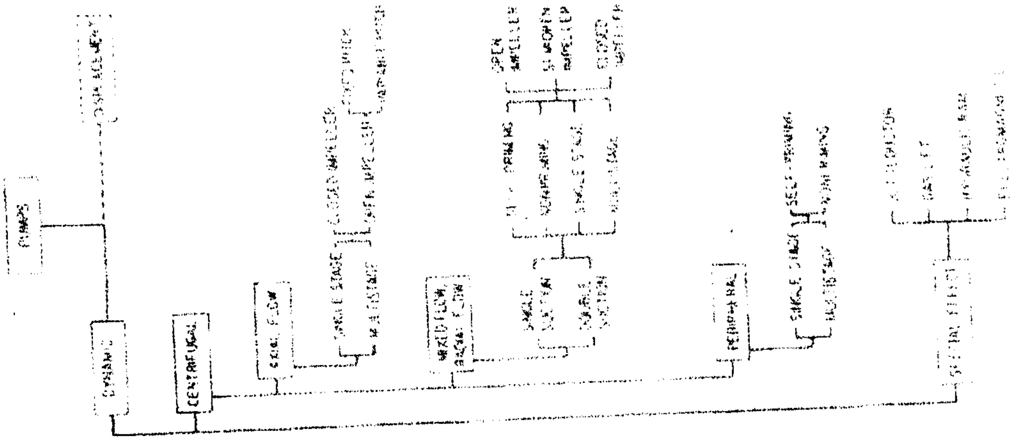


Figure 2.7: Classification of Dynamic Pumps

2.3.2 Market Survey

From the study of the Zambian market on pump sales over a period of five (5) years up to 1998, Figure 2.8 shows the trend in sales. The survey involved direct contact with the pump selling companies either through questionnaire but mostly by directly contacting to sales departments in those companies. Questionnaires proved the least effective and so sales data was utilized. Respondents were AFE (Z) Limited, Flygt Pumps, Bestobell (Z) limited and KSB Pumps.

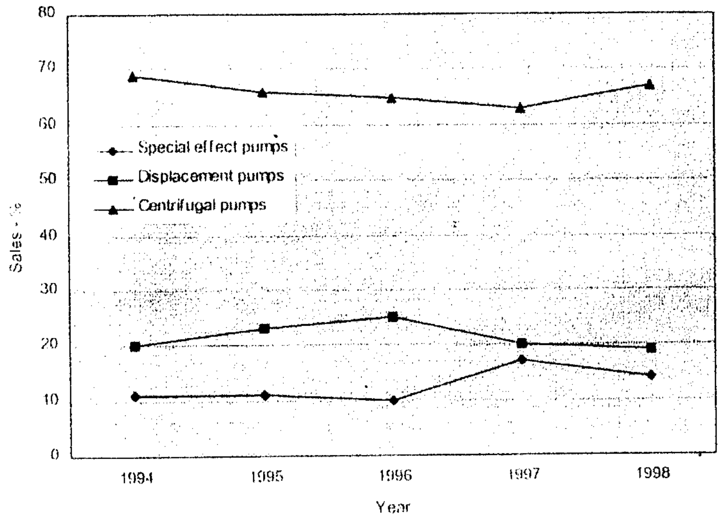


Figure 2.8 Market Survey on Dynamic Pumps over a five-year period

The following were the average percent sales as of December 1998:

Centrifugal Pumps	66%
Displacement Pumps	21%
Special Effect Pumps	13%

The centrifugal pumps were further broken down to:

Axial Flow Pumps	15.2%
Mixed Flow Pumps	33.3%
Radial Flow Pumps	42.4%
Peripheral Pumps	9.10%

Of these, the most popular design was considered further; the radial flow pumps with the following percentage weighting:

Radial Flow Pump

Open Impeller	57.1%
Closed Impeller	42.9%

The Open impeller, radial flow, single-stage centrifugal pump was selected. However, there are two main categories of single-stage pumps: horizontal and vertical (submersible). The following were their respective average demand levels over the 5 years with a positive trend:

Horizontal Pumps	48%
Vertical, Submersible	52%

Therefore, the most popular of the single-stage pumps was selected which was the

Vertical, Submersible Pump with Open Impeller Design

Further, the survey revealed that the most popular specifications of single-stage, submersible pumps fell in the range giving average values of:

Discharge	**$Q = 32\ m^3/hr$**
Head	**$H = 40m$**
Speed	**$N = 2890\ rpm$**

Classification of this pump was carried out using the specific speed method. From the equations 2.1 it was found that the (discharge) specific speed was approximately 13 (non-dimensional), which matched the radial impeller type. In addition, (Figure 2.6), this range falls in the overlap range for Radial and Mixed Flow Pumps [10], that is $25m \leq H \leq 60m$. In conclusion the complete description was

Vertical, Submersible Pump with Open Fully Radial Flow Impeller

The selling price for this pump type quoted as ranging between $ 1,680 and $ 1,750. One of the objectives was to design a product cheaper than the quoted market prices in all effort to compete favourably. This was the pump redesigned. Materials used for different parts of the pump are given in Chapter 3, in which chapter are the materials recommended for this project.

2.4 The Pump Detailed Analysis

In determining the pump dimensions, the order of design was dictated by the degree (or extent) of interdependency of the various parts. More related parts were given first preference in the design sequences: What determined which part to start with in the whole design was the working principle of the pump. Water flows from the suction to the discharge nozzles between which the impeller is the first component. The impeller is supported by the shaft and enclosed in a casing. Relating the casing and shaft respectively to the impeller, the shaft is given preference because the casing directs liquid after impeller's discharge. However, it might be argued that the impeller depends on the sealing from the casing. This, as far as design is concerned, is a secondary functional aspect. It was left to the end of the design after the casing primary dimensions were determined.

2.4.1 Specifications

From the market survey, the following were the derived parameters on which the design was based:

- $Q = 32m^3/hr$
- $H = 40m$
- $N = 2890$ rpm

2.4.2 Specific Speed

This dimensionless characteristic parameter was evaluated using equations 2.3. The suction specific speed was used to determine the suction conditions of the pump; the most important parameter being the effective head associated with it known as the Net Positive Suction Head (N_{psh}). It is the head required at the pump suction during operation to prevent the phenomenon of *cavitation.* The following empirical relation applies for a single-stage centrifugal pump

$$N_{psh} = H.f(N_{sm}) \qquad (2.3c)$$

Where $f(N_{sm})$ denotes a constant that is a function of the (discharge) specific speed.

2.4.3 Evaluating Efficiency

Pump performance prediction has largely been based (and still is, to a large extent) on experience [11]. Characteristic curves are generally obtained from theoretical head estimations, which are subsequently corrected for viscous losses. The often-unsatisfying results are only partly caused by the inadequacy of the viscous loss models used. An important contribution to the deviations should be attributed to the methods of determining the theoretical head values. Until now, these have been predicted by means of one-dimensional or quasi-two-dimensional methods. Methods to predict this efficiency on truly three-dimensional computations would improve results substantially. Unfortunately, the present-day solvers are far from adequate for reasons of making assumptions that eventually oversimplify the problem with assumptions that leave out the 'real world'. Some of these are turbulence, which is not fully understood. Standard turbulence models appear to fail for flows in a rapidly rotating framework. Obviously, an unsteady analysis of a complete rotor-stator configuration to account for the interaction at off-design conditions is way out of reach at this moment [12]. For most pumps operating near the design conditions, the influence of viscosity is restricted to thin boundary layers and wakes. The main flow can be predicted fairly accurately with 3-D methods [12]. The simple formulations in this project however offers a practical possibility of evaluating pump efficiency because it is purely based on empirical test data [8] of many pumps, from which relations were derived.

2.5 Impeller Design

2.5.1 Impeller Inlet Diameter

To prevent the phenomena of circulation and cavitation, the value of the flow angle was fixed first. This angle is the mean flow direction, which the fluid takes between the impeller eye and closest point of the blade or vane. In real applications, the shaft at impeller boss is considered to obstruct a clear inlet diameter. This problem was compensated for by, proportionately increasing the impeller inlet diameter and making impeller widths larger. The earlier applied to this project for the open impeller since the latter especially applies to double-shrouded impellers. Generally, it follows that the higher the hub ratio, the larger the number of blades used. Thus, the impeller suction diameter is proportional to discharge and flow-angles in the following manner:

$$D_s = (const1).f(Q^{1/3}, \tan^{1/3}\beta_0) \qquad (2.4)$$

where, D_s is the impeller suction diameter and β_o is the mean flow angle fixed at 17° for this type of impeller. The *(const1)* takes care of the *hub ratio* that is a measure of the extent of inlet flow blockage to the pump. This value is empirical and depends on the type of pump [9].

2.5.2 The Head Coefficient

Having determined and fixed the specific speed, the flow ratio was determined and impeller outlet angle β_2 evaluated with empirical *const2* obtained from [11]. The relation used was

$$\beta_2 = const2.f(N_{sm}) \qquad (2.5)$$

Since the vane discharge angle is not the flow angle, it was necessary to know the ***Slip Factor*** that takes care of losses as the fluid flows through the diffusion system. This was given in an implicit manner by supplying the usual number of vanes z, which also depends upon the specific speed of the pump. The most realistic expression was that which took care of the type of casing used. The relation is

$$\mu = \frac{1}{f(a, z, \beta_2, r)} \tag{2.6}$$

where μ is the slip factor, and *r* is the radius ratio of the impeller. The value of constant a, depends on the type of casing used. In this design the value of 0.6 was taken for the vane diffuser casing system. This relation has proved to be in good agreement with experimental results [13].

The ***Head Coefficient*** is a dimensionless parameter given by

$$\psi = 2.\mu.\eta_h.f(\phi, \beta_2) \tag{2.7}$$

where, η_h is the hydraulic efficiency of the recuperation system and *Φ,* the flow ratio.

Thus, the necessary impeller peripheral velocity was given by

$$u_2 = \sqrt{\frac{2.g.H}{\psi}} \tag{2.8}$$

2.5.3 Impeller Inlet and Discharge Dimensions

The impeller outlet diameter was calculated from

$$D_2 = \frac{u_2}{\pi.N} \tag{2.9}$$

The impeller outlet width was given by

$$b_2 = \frac{Q}{\pi.D_2.C_{m2}} \tag{2.10}$$

where C_{m3} is the meridian flow velocity. The impeller inlet diameter for the open impeller design was then evaluated from

$$b_1 = (const3).D_s.\frac{D_s}{D_1} \qquad (2.11)$$

where *(const3)* takes care of the hub ratio and the fraction *(D_s/D_1)* the ratio of diameters takes account of the effect of flow with and without the shroud in the impeller.

2.5.4 Vane Inlet Angle

The impeller inlet vane thickness was chosen with respect to the type of metal used as well as the size of the impeller. The impeller inlet angle was obtained through some iteration process as its formula has the inlet angle β_1 on both sides:

$$\tan\beta_1 = \frac{Const4}{f(z, s_1, D_1, \beta_1)} \qquad (2.12)$$

where *'Const4'* is an empirically obtained constant, z is the usual number of impeller blades and S_1 the impeller blade thickness at its inlet.

2.5.5 Flow Areas between Vanes

For this impeller, the inlet area between vanes was approximated by

$$A_I \approx \pi.r_1 Sin\beta_1 \qquad (2.13)$$

For water pumping, the valid formula for the outlet area between its vanes was computed using the formula

$$A_{II} \approx b_2(\pi.D_2.Sin\beta_2 - z.s_2) \qquad (2.14)$$

where s_2 is the impeller blade thickness at the outlet.

2.6 Thrust Analysis

2.6.1 Radial Thrust

The radial thrust was given by the formula

$$F_{radial} = Const5.H.D_2.b_2 \qquad (2.15)$$

where the *Const5* is dependent upon the casing configuration and the pump specific speed. The total radial force includes the force due to shaft power in rotation. This project was a case of thrust combined with torque. Thus, **tension** and **shear** can be evaluated using the ***Maximum Shear Stress (Guest's) Theory*** [14]. The maximum resultant stress encountered is then given by

$$S_{s\,max} = \frac{1}{2}\sqrt{S^2 + 4S_s^2} \qquad (2.16)$$

$$S_{max} = \frac{1}{2}S + S_{s\,max} \qquad (2.17)$$

in which S is the tensile or compressive stress due to axial load and S_s is the shearing stress due to torsion. Equations 2.16 and 2.17 are used in the design of shafts to determine the maximum stresses (see pages 63-65). Therefore, since the design process requires these computations, they are included for use as they become relevant.

2.6.2 Axial Thrusts

In this design (vertical pump), axial thrust is caused by

(i) the weight of the rotating parts (independent of pump Discharge Q, and Head H),

(ii) dynamic forces (change in momentum) caused by change in flow direction through the impeller.

2.6.2.1 Dynamic Forces

Under normal operating conditions, upthrust caused by the change of momentum is hardly significant in comparison with the downward thrust caused by the unbalanced pressures acting on a single-suction impeller [10]. Upthrust was approximated by

$$F_u = \frac{k.A_{eye}.V_{eye}}{g} \qquad (2.18)$$

where k varies depending upon the type of impeller used. A_{eye}, V_{eye} and g are respectively impeller effective eye area, eye velocity and gravitation constant *(g = 9.81m/s^2)*. Neglecting the effect of pressure distribution on the shrouds of the open impeller, the downward thrust was approximated by

$$Force = \gamma.H.A_{eye} \qquad (2.19)$$

where γ is the specific weight of water.

2.6.2.2 Static Forces

The static forces were made up of the following elements

(i) impeller weight,

(ii) rotor (motor core) weight ,

(iii) shaft weight.

Normally element (ii) and (iii) form up the rotor. However, in this report the term rotor refers to the (motor) core without a shaft in it. The selected material for the pump impeller was aluminium. Considering the impeller as a disk, the standard steel SI formula was used to determine its weight. However, in this case the metals' relative density was considered to correct the value for aluminium. The selected rotor for use was weighed. This was the rotor weight without the shaft in it. The shaft diameter was firstly estimated from

$$d_{shaft} = k\sqrt[3]{\frac{P_s}{S_{sy}.N}} \qquad (2.20)$$

where S_{sy} is Shear stress at yield k is empirical constant [11] and P_s the shaft power is given by

$$P_s = \frac{9.8.H.Q.\gamma}{\eta} \qquad (2.21)$$

in which γ is the specific gravity of the liquid being pumped. EN8 Steel was recommended for the shaft design for this clean water pumping application. By *Distortion Energy Theory [15]*, the yield shear stress in the shaft was determine from

$$S_{sy} = 0.577.S_y \qquad (2.22)$$

For standard steel material, from ISO Standards of solid steel bar [11], the weight per unit length is

$$M = 0.00616.d_{shaft}^2 (kg/m) \qquad (2.23)$$

where the shaft diameter is in millimetres (mm). The weight was approximated from multiplying the result from this equation with the shaft length.

2.6.3 Allowance for Keyways

Because of the weakening effect of a keyway, an increase in the shaft diameter was made. Morlay [16] suggests that the required diameter *(d_r)* allowing for keyway be obtained from the calculated diameter *(d_{shaft})* as follows:

$$d_r = \frac{d_{shaft}}{(1-0.2.w-1.1.h)} \qquad (2.24)$$

In this formula *w* is the ratio of the keyway width to shaft diameter and *h* is the ratio of the keyway depth to shaft diameter. This formula takes care of fatigue but yields the minimum safe shaft diameter. The dimensions of the keyway and its key were obtained from fasteners engineering handbook with standard tables.

2.6.4 Shaft Elongation

In vertical pumps, shaft elongation is caused by two separate phenomena:

(i) tensile stress due to the hydraulic thrust arising from
  (a) shaft weight and those of the impellers mounted thereon,
  (b) the hydraulic thrust itself
(ii) thermal expansion

The total elongation was calculated from

$$\delta_{tot} = \delta_{weig} + \delta_{ht} + \delta_{temp} \tag{2.25}$$

in which the first, second and third terms respectively represent elongation due to weight, hydraulic thrust and temperature, The computed resultant elongation was compensated for in assembling to avoid interference contact between stationary and rotating parts. The expansion caused by temperature was assumed to be independent of elongation of the shaft caused by the downward hydraulic thrust. This design was recommended for water up to 40°C as it was meant for cold water supply.

2. 7 Other Preventive Measures

2.7.1 Reducing Impeller Axial Thrust

There are a number of methods employed to reduce the influence of axial thrusts in impellers of centrifugal pumps. This project design procedure took care of this influence at impeller and shaft design stages. This is a critical problem, in vertical pumps, that calls for a lot of attention.

Detailed description of the other methods will not be given in this report. It will however suffice to simply point them out qualitatively. In most applications, especially for small and medium-sized pumps, balance holes are employed on the impeller back shroud. These 'bleed' thrusts into the stuffing box to allow for pressure balance between the front and back of the impeller. The other method employs ribs. These are vanes put at the back of the impeller in order to reduce axial thrusts. Most books [9], [10], [11], [17] explain in detail this method including quantitative treatments. The other method, normally used in multistage pumps, is the balance drum method. This method employs disks for hydraulic balance at every stage of the pump. Double suction pumps are given a somewhat different treatment due to their impeller configurations. In these pumps, the axial forces approximately balance each other and cancel out, though not completely. Reasons vi/ere not a subject of this research. This is the reason why most pump designers prefer two single-suction impellers mounted on the same shaft in a 'back-to-back' or 'front-to-front' arrangement in respective casings. However, the method employed in this project was the ribs method for the reason of simplicity.

2.7.2 Critical Speed of the Shaft

The major problems that result in these vertical arrangements are *torsion* and *lateral vibration* condition over speeds in the critical speed range. System mass elastic characteristics were adjusted to control critical speed location. In cases where systems employ couplings, the coupling weight can be adjusted, as well as shaft stiffness and diameter [9]. A monoblock pump was the design for this project so that couplings were not employed. Stepanoff [10] observed and concluded that to calculate the critical speed, only part (50-60%) of the moment of inertia of sections of impeller hubs or sleeves should be added to the shaft moment of inertia to account for the stiffness effect of all parts mounted on the shaft. This, it was felt, is a sufficient margin to avoid any danger caused by operating the pump close to the critical speed. Normally, the first critical speed is set between 25 to 40% below the operating speed of the pump. A 35% margin was considered reasonable in this design. This was given by a quick approximate expression that uses an experimental factor for a case where the method of support, load distribution and the shaft diameter varies little as in a multistage pump. The expression is

$$N_{c1} = \frac{Const7}{\sqrt{f(mm)}} \qquad (2.26)$$

$$f = {WL^3}/{CEI}$$

in which

Nc1: 1st critical speed (rpm)

E: Modulus of Elasticity (N/m^2)

I: Moment of Inertia *(m^4)*

F: Deflection *(m)*

L: Free span *(m)*

W: Weight of the rotating element (N)

C: Coefficient depending on shaft-support and load distribution.

Const7: Constant evaluated empirically [10] from type of loading.

To check how much the operating speed was above or below the 1st Critical Speed, the following calculation was performed:

$$\%\Delta = \frac{|Operatingspeed - Criticalspeed\ |}{operatingspeed}.100$$

Theoretically, the shaft deflection should always be less than the radial clearance between the closely fitted parts of the rotating element and the stationary casing parts. In practice however, owing to the inevitable eccentricity of the several running fits inside the pump, this condition is never fulfilled despite some of the closely fitted parts serving as internal bearings. At the critical speed, the shaft vibrations are synchronised with the natural period of vibration of the shaft If the lower side of the critical speed is desired then reduce the pump speed to at least 25% below critical speed Stepanoff [10] concluded that:

(i) **Rule 1**: The operating peed should not be an even fraction (1/2, 1/3, multiple (2,3,4,) of the first critical speed,

(ii) **Rule 2**: The operating speed should not be so close to the critical speeds, first or second, but $N < 0.8N_{c1}$ or $1.3\ N_{c1} < N < 0.7\ N_{c2}$ where N is the operating speed, N_{c1} is the first critical speed and N_{c2} is the second critical speed,

(iii) **Rule 3**: Satisfactory operation above the second critical speed is impossible.

2.8 The Casing Design

The pump for this project was a single-stage design. Therefore, the analysis of return passages was beyond the scope of this research need as that only applied to multistage pumps. However, some old designs such as of Worthington Pumps had this feature but it simply increased design costs and overall performance reduction [18]. In this research only the diffuser casing was relevant and was considered. However, the actual design of this type of casing was more involving than what theory can give. Only key considerations are highlighted in the following sections.

2.8.1 Clearance between Diffuser Vane and Impeller

The empirical constant K_3 was firstly determined from equation (2.27),

$$K_3 = \frac{(438 - N_{sm})}{820} \tag{2.27}$$

The average casing throat velocity (past channel of fluid) was then

$$C_{thr} = K_3\sqrt{2gH} \tag{2.28}$$

The diffuser vane distance from the impeller outer diameter was calculated from

$$t = \rho . r_2 \qquad (2.29)$$

in which

$$\rho = \frac{3.39 + N_{sm}}{Const8} \qquad (2.30)$$

where the *const8* depends on the type of casing design and is purely empirical [10].

2.8.2 Channel proportions

Considering the most popular and practical type of channel design, rectangular, the inlet width was obtained from

$$b_3 = 1.1 b_2 \qquad (2.31)$$

and the radial distance from the centre (radius from which it starts) was

$$r_3 = t + r_2 \qquad (2.32)$$

Therefore, diffuser inner diameter $D_3 = 2^* r_3$.

2.8.3 Diffuser Vane Angles

The inlet angle to the channel was determined from

$$\alpha = \frac{(20 + Nsm)}{Const9} \qquad (2.33)$$

where the constant is an empirically obtained value [11]. The total divergence per channel was 11° per channel [10] so that for each side it made 5.5°. The area expansion that corresponded to 8.5° for parallel-walled diffuser was 1.6. This was the channel base slope when halved. Figure 3.4 shows the diffuser casing.

2.8.4 Others Points

To avoid flow re-circulation both at impeller eye as well as outlet, pumps should not be operated below about 55% of the BEP [9]. Circulation results in severe hydraulic instabilities as a result of internal re-circulation. This is directly related to the required Net Positive Suction Head, and to some extent, the efficiency of the pump. This phenomenon generally occurs during operation below the BEP flow rate but frequently occurs when the flow rate exceeds BEP. On one hand, flow re-circulation is generally found during off-design flows. This mostly damages the impeller eye or the inlet area of the casing. On the other hand, flow re-circulation at vane tips damages impeller outside diameter. This flow, in addition, results from improper tip clearance or alignment of the impeller with the casing and not to mention the states of the surface roughness. The pulsating hydraulic forces acting on the impeller can be reduced by 80-85% through increasing the radial gap between the impeller and casing, with experiments [11]. However, there is no loss in overall efficiency when the diffuser or volute inlet tips are recessed within practical range [17].

The vane-tip clearance controls the strength and amplitude of the hydraulic shock created at vane passing frequencies. The impeller shroud clearance (nominally 50 mils) controls the severity of pressure pulsation behind the impeller hub and shroud, which give rise to high axial dynamic forces with low frequencies. The phenomenon of recirculation at the discharge of an impeller can, under certain conditions, trigger recirculation at the suction. In addition to increasing the radial gap, more effective and reliable impellers can be achieved by several design changes. These are considerations beyond the scope of this project work treatment. However, other points worth considering in the final analysis for assembly were the allowable "play" in the shaft to maintain correct Casing-Impeller relationship. Recommended running tolerances are given hereunder [9]:

◆ **At mechanical seal:** Radial play, ± 0. 1 mm
Axial play, ± 0.05mm

◆ **At bearings:** Radial play, ± 0.05mm
Axial play, ± 0.05mm

For the other parts of the pump, direct stress calculations based on strength considerations were performed. The other dimensions were added for the purpose of shape as well as ease of design for manufacture. This included adjustment of shapes to enhance casting and machining in very short work-piece turn-around times.

2.9 Production Process Development

This undertaking seeks to adapt the production of pumps to the available technology in the University of Zambia's Department of Mechanical Engineering, which includes, in addition to the conventional machinery, a CNC Machining Centre (Vertical Milling Machine). The major objective was to adapt this design of the pump to the advanced CNC technology in an effort to enhance accuracy, reduce production cost and develop tooling for production. In essence, this project research work was meant to improve pump production and tooling thereof using CNC Technology. Pump production is currently not there in Zambia; only simple parts such as end-covers and simple casing designs. Complex parts such as the impellers and diffuser casings are not manufactured.

In developing the processes of production in this project, based on the assumption of small and medium sized organisations (30-120 employees), three ideas were kept in mind, casting philosophical foundations, namely:

- Although design theories and concepts and the language used to describe them may be universal, implementation in the redesign of products cannot be simply the downsized version of a large existing product.

- A great production system will not itself make an organisation great, but an inadequate and/or misleading system will keep one from becoming great or, worse, cause it to fail,
- It is better to be approximately correct in redesign processes than be precisely wrong. In other words, accuracy is preferable to precision. With this project, the design was carried out with a view to achieving targeted performance requirements (Head H, and Discharge Q for. a contemplated speed N). We can define these as "ideal" performance parameters. When the product is tested, they are called the "actual" performance parameters or simply result for analysis in this report. It is not expected that these two would be exactly the same due to a number of factors of which inaccuracies in production processes used and losses due to application and installation methods are some of the explanations.

This project was concerned with how pumps could be produced in Zambia using the available machinery within the University. For this reason, it would somehow be argued that this report is a normative theory because it sets out to establish a standard or a norm for the country. This undertaking cannot hope to succeed in its task if it is developed in isolation from what actually does happen in practice. The project also examined how best CNC Technology could be exploited and utilised for production. With that vision in focus, design software called Mastercam, a CAD/CAM tool, had to be used in the implementation of design for production. In essence, all the decisions in the development of the parameters used were based on the comparison of alternatives, and it was in that sense that the developed theory really had its roots in the valuation processes. Therefore, although it can be said that all types of pumps produced involve the same fundamental processes, each is given its own unique characteristics by the evaluation basis and production methods combined which it employs. This implies that depending upon the approach an individual and/or a firm takes, the same product with the same number of parts will have varying production times as well as quality and cost of production to mention but key factors.

It should be emphasised that the development process was based on the assumption of a small or medium-sized organisation. For the purpose of this project, a small or medium-sized organisation is an autonomous business unit that has employees ranging from 2 to 100, and needs improved cost information but believes that thousands of hours of employee and/or consulting time for development and untold additional hours for maintenance is too great a price to pay for that improvement. In small and medium sized organisations, the investment required to implement time-based cost systems ranges from 80 hours for a small commercial printer to 500 hours for a large mining company with poor historical and financial records [7]. This contrasts with larger organisations such as a complete mine that encompasses operations from mining to refinery (about 3000 employees for ZCCM Mufulira division, 1997), which may require an interdisciplinary task force full time for many months, and incurring consulting fees reaching into six figures.

Thus, the assumption made is relying on the reason that most current approaches to large organisations are based on concepts developed by the Computer Aided Manufacturing (CAM) sponsored by very big organisations. Nevertheless, these smaller organisations need to understand their production processes as well as process costs eventually.

Often, it is desirable to use low quantity methods for initial production of a new product. This minimises the lead times that would be required for procurement of high-production tooling and equipment, enabling the product to be brought to the market sooner. Later, if the product is successful, higher-production tooling can be used, with worthwhile saving in manufacturing cost.

Most of the process planning function was pre-empted by decisions made in product design. Decisions on material, part geometry, tolerances, surface finish, grouping of parts into subassemblies and assembly techniques limited the number of manufacturing processes used to make the different parts. The target was a product design that was functionally superior and at the same time that could be produced at minimum cost in order to hold great promise of success in the marketplace.

Design for manufacturing and assembly was the approach taken in the design of this product. It included considerations of manufacture and assembly in the design. Benefits were typically

- Shorter time to bring the product to market,
- Smoother transition into market,
- Fewer components in the final product,
- Easier assembly,
- Lower costs of production,
- Higher product quality, and
- Greater customer satisfaction.

The resulting sized parts were then tested for assembly. With the tolerances specified, the designs were assembled in using Production Solid Modelling software called **SolidWorks** to check if it were possible to assemble the product. The dimensions were adjusted to ensure a good design consistent with requirements of design for assembly. Materials properties were also checked. In addition, **Mastercam** 7.1 was used to develop the tool-paths, which were verified using its functions, *"Back-*

plot" and *"N-See2000"*. These were further verified using another simulation software known as Virtual **CNC, Predator,** which checks the NC (post-processed) file for errors - this being the code file that the CNC machine uses. Some parts were made on the machines using the developed programs. These software packages are explained in Appendix A.

Chapter 3

The Pump Re-design

3.1 Introduction

Design is a process that involves identifying a need, generating possible solutions to meet that need, evaluating each solution to determine its merits (engineering analysis), identifying the solution to be developed based on merit and then developing a detailed model so that it can be built. These, being the key steps undertaken, are developed upon step-by-step in the context of this report. In this project CAD/CAM principles were utilized for the process development. On this course, the major focus for manufacturing were:

- Prospects of increased productivity, and
- Creation of a database for manufacturing.

It is advantageous to incorporate CAD in design because it helps the designer or draftsman to visualize his design on the computer screen. Changes to a an existing design can be made interactively while the designer gets almost immediate visual feedback on the changes on a computer screen. Current CAD software, such as SolidWorks and Mastercam also allows for analysis and testing of components before manufacture and the presentation of the finished product. Many products, notably equipment, machines, tooling, and devices for industrial use, products for niche markets, and pilot lots, are inherently limited in production volumes.

The designer's role in this activity is to be an intermediary between scientific knowledge and production side. He has a very responsible task of satisfying in the best manner possible the conditions laid down in the customer's order, and thereby providing the essential foundation for economic manufacture. In this chapter both conceptual as well as the detailed design considerations are brought to the fore. The economic considerations involved redesign to adapt the new design to the available technology, cost reduction in the tooling to produce parts for the pump as well as the pump itself. This meant product re-engineering, to ensure the available resources were exploited to produce a complete centrifugal pump in Zambia. In this concurrent engineering all tasks were taken as elements in the integrated design. The control of cost lay in the first stage of design and this had a lot of influence on a number of features on the parts. The designs were detailed while simultaneously developing production capability, field capability and quality. Complete design and sequence for the pump was established.

3.2 Conceptual Design

When production quantities are not large, the designer cannot afford to use components or configurations that require high-cost tooling or extensive process or product development. He or she must concentrate on simpler components or those that are already in production and available elsewhere - from other products already in production in his or her firm or from commercially stocked components available from vendors [7]. This project concentrated mostly on the simplification of the existing design (Figure 3.1), reducing as much as possible the cost of production through the use of simple geometry of parts, use of reasonable geometric tolerances on most surfaces. Standard parts were recommended such as bearings, mechanical seals, bolts, nuts, to name a few. An example of the component available elsewhere is the motor. The rotor had anew shaft design to suit the current use. Therefore, only critical parts to the problem of redesign of the pump were concentrated on. This meant that the problem solving focused on design change rather than just production and resizing of the whole pump.

The existing pump was conceptualised and optimised to ensure functionality, producibility and cost constraints were taken into account. This was an iterative process in making changes to the parts. The performance requirements for the new design were utilised in making choices from many alternative processes. During these changes it was always kept in mind that the main objective was to develop a process for production with a view of adapting the developed methods to the available technology, Another interesting point is that design, for speed-to-market and design for low-quantity production have much in common. Considerations were also made in this respect.

3.3 Redesign versus the Existing Design

A number of changes were made to the existing design (Figure 3.2), the major one being the reduction of the number of parts and simplification of the design complexity in the most economical way. In this chapter, the two centrifugal, submersible pumps; the existing and proposed (Figure 3.3) design; are compared with changes justified.

The Casing

Comparing the types of casing, both were diffuser vane type. This design of casing is used mostly in vertical pumps to provide extra guidance to the liquid flowing from the impeller. However, profiles on them differ depending on how efficient the pump should be as well as how supportive it should be to the impeller. The type of impeller and application, in addition, determines its proportion, For this application, the casing was designed to accommodate open impellers of different types in the limits of radial and mixed flow in a single casing. The existing design had, in addition to the diffuser casing Part 57 -57 A, Figure 3.1), a lower diffuser (Part 58A-8) that was used to adapt different impellers in the same operation range as the ones this project targeted to accommodate. Therefore, the existing design casing was made up of parts 7B and 7C (O' ring rubber seals), 57 – 57A (diffuse main), and 58A – B (lower diffuser). In addition, it had a more complex shape than the new design. However, it had the advantage of diverse application. This was made possible with the rubber lining (in black) which in addition gave a flexible adjustment to impeller/casing clearance. Since this project was a process development, the rubber lining could always be added later, which would give more advantages than the existing one. In general, the existing design was more expensive both to maintain as well as produce because of the extra parts it had. In most cases, efficiencies of these pumps quickly deteriorate with the failing of the rubber seal[1S1,[17], which gives serious maintenance problems. The new design had worked towards mating of parts in critical areas such that no seals were necessary. An example was the inclusion of a spigot on the casing base. This further enforced with good surface finish, gives better sealing.

The Bearing Housing

This, in the existing model (Figure 3.1), was not called by this name but called the upper diffuser (Part Number 52). The bearing housing (54) was a separate unit altogether. This bearing housing in the current design was extended to combine parts 2M, 6, 7E, 12, 52 and 54. This change, it may be argued, may have some maintenance problems as a worn out part means overhaul. The only parts that are subject to wear are 2F, 5L, 5F, 59, (from impeller, though very rare [20]). Parts 12 made up the mechanical seal set (both tungsten carbide) which were also in the new design. These seals were exposed to water in both designs to allow for cooling during operation. In this project, however, a new feature was incorporated that gave the pump extra life by isolating the seals from abrasive materials. This was accomplished through the impeller design of the back shroud, with a smooth slope towards the hub. In addition, this project design had a bearing life calculation based on 18000 hours of operation as opposed to the 15000 hours for the existing design. Also, during the bearing selection, the selected size matched 1.2 bearings for the shaft. This was rounded off to the nearest upper ten (10) because of the unpredictable nature of the application environment the product would be subjected to as well as the unreliability of most estimations made in pump designs [10].

The recommended material for the pump was cast iron. In this project aluminium is used for the reason of lack of facilities to smelt ferrous metal in the University of Zambia. The same material was used in the existing design.

The Impeller

This was a single unit as in the existing design. Added features included the smooth or uniform back shroud to reduce turbulence as much as possible. In addition, the region near the hub was slightly tapered to keep abrasive materials away as much as possible. This would in effect improve the life of mechanical seals, which were a major factor in high maintenance costs in the existing design. Both impellers were open types but the one shown in the existing pump was mixed flow type whereas this project shows a radial type. However, both designs accommodated mixed as well as radial type impellers. The other difference between their designs lay in the profiles of their blades or vanes.

The recommended material for this design was cast iron, bronze or brass. The old design employed hardened high chrome iron. This could not be recommended for this design, as it required additional facilities for alloying as well as hardening.

However, there were other cases where stainless steel was used. This mainly applied to slurry pumping applications due the ability of this type of steel to withstand the aggressiveness of the medium [17]. For water pumping, cast iron sufficed; bronze or brass is normally good for chemical plant applications due to their stable behaviours. In this project cast iron was resistant enough to withstand erosion in water pumping applications.

The existing design was made of numerous parts to list. However, only major parts will be listed. Refer to the attached drawing for the other parts. The pump casing was unitary; made of 4, 5, 57, and 58. This project was made of a single casing so that a lot of parts in the original design were eliminated. Just to point it out again, the O' ring on 58 was therefore not necessary. The discharge nozzle, part of 55 in existing design, was simplified in the new design, thus eliminating a number of parts: 2, 5, 1 0, 18, 19, 23, 60, 61, 64, 67, 68 and 69. Therefore, extra machining operations were eliminated, resulting in a simpler design of the junction box. Therefore, extra sealing was eliminated. Refer to the exploded views (Figs. 3.1 and 3.2) for details.

Material used for both designs was aluminium. The difference was in the grades; in this project AZ92A (S_{ut} = 620Mpa, 14-16% elongation) was recommended where as in the existing design AZ81C (S_{ut} = 495-530Mpa, 8-14% elongation) was employed. Therefore, the new design wall thickness was made 2 mm more throughout for reason of strength.

For this application, either steel EN57 or EN8 could be used. The latter was selected as for water pumping applications as well as strength reasons. The steel for the existing design was stainless steel. However, due to the reduced number of attached parts to the shaft, the length of the shaft was reduced substantially from about 760 mm to 490 mm, therefore reducing the overall height of the pump from 1050 to 620 mm. This is approximately half the original length for the same characteristics in operation, i.e. head and discharge. This undertaking offered a number of advantages of which portability was not an exception, The short shaft has less vibration problems than a long shaft, Thus critical speed problems are easier to handle, and the minimum manufacturing expertise was much lower here than the original design.

Mechanical Seals

Only one set of mechanical seal was included in the new design. This was equivalent to part 12A in the existing design. The original design had another mechanical seal (Part 12) of carbon-carbide, which was oil-cooled since it was weaker than the tungsten-tungsten mechanical seal. Depending on design considerations one can design a pump with only a stuffing box for cotton packing that performs just as one with mechanical seals [17]. This project used a single tungsten-carbide design of mechanical sealing. The original design used what is known as a balanced design. This is a type of arrangement where two pairs, instead of single pair of mechanical seals, are used. Balanced and unbalanced sealing (arrangements are names to distinguish between the different configurations. The project design eliminated the oil chamber design (part of 54). The other common cause of problems in the original pump was the seepage of water through the oil cap (2M, 6, 7E, 12) [9]. This project's design eliminated all these maintenance problems. In addition, the overall machining processes were reduced substantially.

Grease

The recommended type for this pump was the same as for the existing pump, the Zennex lithium H.T.3 with a working temperature of 150°C.

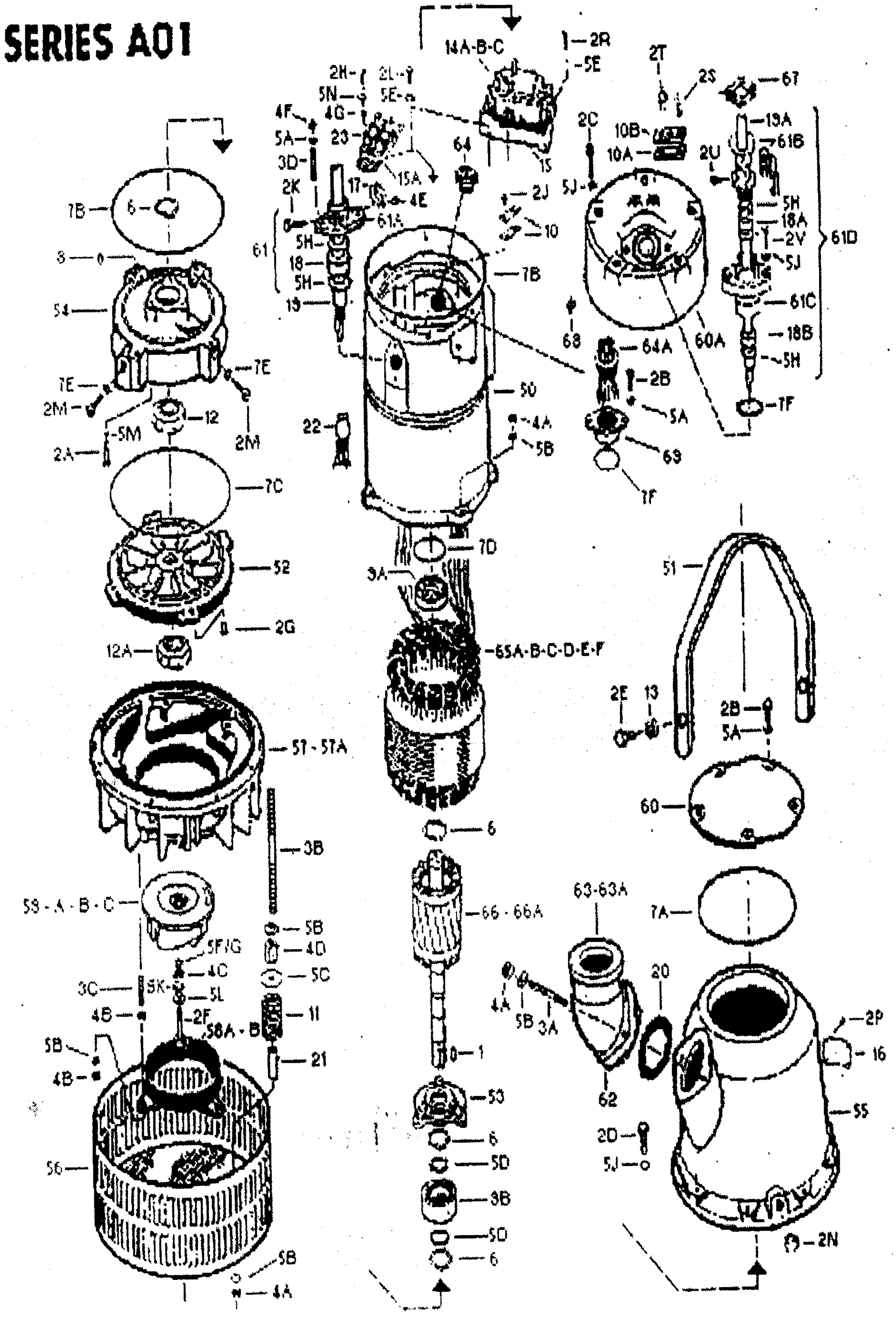


Figure 3.1: The Existing Submersible Pump Design(**source:** *Flygt Pump(Z) ltd*)

3.4 Economic Considerations in Concept Development

Before any detailed design work, a number of concepts were brought to the fore. An elimination method was employed to come up with the best of the available concepts. In certain encounters, some designs were combined while others were modified in a completely new way to suit the available technology, minimise material wastage, reduce tooling costs and ensure an easy-to-maintain product. The following guidelines were followed [7]:

(a) An economic analysis of design and manufacturing alternatives were carried out in order to permit rational choices among the design alternatives. Product design, including assemblies and production processes and selling prices were worked out together. The current selling price of about $ 1,700 and production target of $ 800 were known before the design details were developed.

(b) An important rule for design and manufacturing managers is, "Standardise, standardise, standardise!" This is even more important for low-quantity production than it is for mass production because, in the former there is not the opportunity to amortise the tooling and developmental cost of special items.

(c) "Never design a part that you can take out of a catalogue". This is because catalogue-standard parts often have advantages beyond that of company standard parts in that the supplier makes them in large quantities with efficient production methods, providing favourable costs. In addition, quality and reliability are already established and field service, if ever needed, is facilitated.

(d) Design for manufacturing processes that are suitable for low production levels, i.e. those with low tooling costs. For machined parts, if the company possesses CNC machine tools, it is desirable to design parts so that they can be processed on such equipment rather than on production equipment that requires special cutting tools and holding fixtures. Machining centres with tool changers call perform a whole series of operations in one set-up. Such equipment is particularly advantageous when it is part of a CAD/CAM system.

(e) It may prove advantageous to "hog" the part solely from solid stock material rather than to produce a sand-mould casting or other casting and finish machine it particularly when a CNC machine is available.

(f) If machined components are used, which is more likely at low production levels, utilise materials with good machinability ratings.

(g) Use stock material shapes as much as possible to avoid machining, This is a good design For Manufacture rule for all levels of production, but it may be particularly beneficial if production runs are short because in such cases set-up times are *a larger* portion of the total manufacturing cycle.

(h) Identifying product concepts that are inherently easy to manufacture and focusing on component design for manufacture and assembly.

(i) To integrate the design and manufacturing process to match the needs and requirements.

(j) Reduction in the number of parts and use the existing processes and facilities so that product yields are high.

(k) Determination of the character of the product; the design and production.

The cost saving, priority of resources in the design of a product in low qualltities were:

1. Tooling costs,
2. Other overhead costs such as power due to reduced run times,

3. Labour costs, and
4. Materials costs.

Normally, for high-production manufacture, direct labour and materials costs are more important, and it is justifiable to expend a greater investment in tooling and engineering in order to reduce them. This, however, normally will not be true at low production levels because the higher investment cannot be amortised [7].

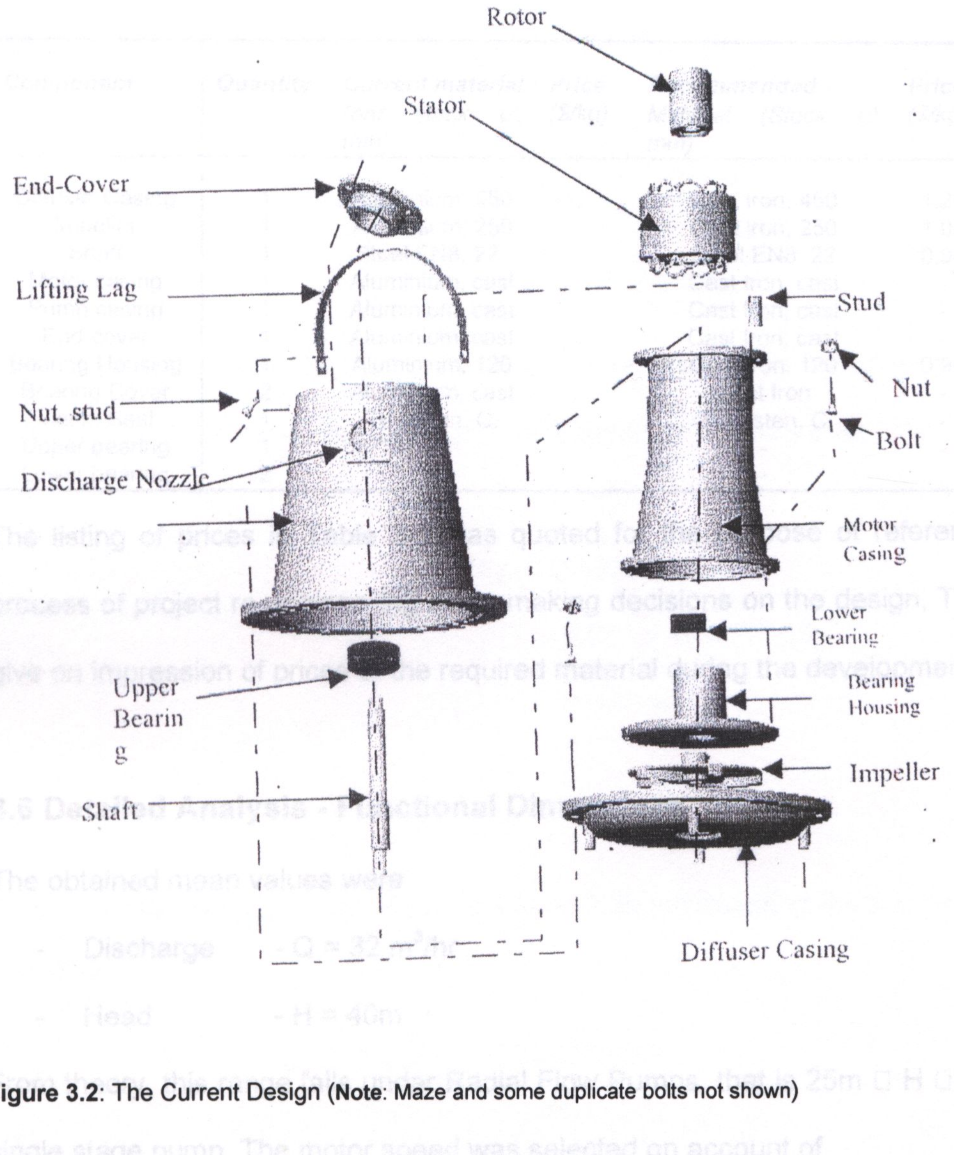


Figure 3.2: The Current Design (**Note**: Maze and some duplicate bolts not shown)

3.5 Materials

Table 3,1 lists the recommended materials for the current design. In addition the table lists the materials that are currently specified due to limits in melting facilities in the university.

TABLE 3.1: Materials for the Project and their Prices per kilogram of standard bar-stock (AISI 304)

Component	*Quantity*	*Current material (bar stock ø, mm)*	*Price ($/kg)*	*Recommended Material (Stock ø mm)*	*Price ($/kg)*
Diffuser Casing	1	Aluminium, 450		Cast Iron, 450	1.20
Impeller	1	Aluminium, 250		Cast Iron, 250	1.08
Shaft	1	Steel EN8, 22		Steel EN8, 22	0.98
Motor casing	1	Aluminium, cast		Cast Iron, cast	-
Pump casing	1	Aluminium, cast		Cast Iron, cast	-
End-cover	1	Aluminium, cast		Cast Iron, cast	-
Bearing Housing	1	Aluminium, 120		Cast Iron, 120	0.99
Bearing Cover	2	Aluminium, cast		Cast Iron	-
Mech. Seal	1	Tungsten, C.		Tungsten, C	-
Upper bearing	-			-	-
Lower bearing	2	-		-	-

The listing of prices in Table 3.1 was quoted for the purpose of reference in the process of project re-engineering when making decisions on the design, This was to give on impression of prices of the required material during the development process

3.6 Detailed Analysis - Functional Dimensions

The obtained mean values were

- Discharge - Q = 32 m^3/hr
- Head - H = 40m

From theory, this range falls under Radial Flow Pumps, that is 25m □ H □ 60m for a single stage pump. The motor speed was selected on account of

(i) achieving a smaller sized pump but of reasonable efficiency,

(ii) what is readily available on the market to provide the necessary shaft power.

3.6.1 Conditions of Service

The driver speed was 2890 rpm and

$H = 40m$

$Q = 32\ m^3/hr\ (8.89 \times 10^{-3}\ m^3/s)$;

$N = 2890$ rpm.

3.6.2 Expected Pump Efficiency

The Discharge Specific Speed [11] from equation (1.1) was

$$N_{sm} = \frac{N\sqrt{Q}}{H^{0.75}} = \frac{2890\sqrt{8.89x10^{-3}}}{40^{0.75}} = 17.13$$

This value falls below 38, the pump may be characterised by either a Volute or Diffuser (Vaned or Vaneless) Casing. The required NPSH was then [11]

$$N_{psh} = 0.001(H)\ N_{sm}^{1.36}$$

$$0.001 \times 40 \times (17.13)^{1.36}$$

$$= \mathbf{1.91m}$$

Thus, the required N_{pshr} 2.0m. From equation 2.3b the resulting Suction Specific Speed is

$$S_m = \frac{N\sqrt{Q}}{N_{psh}} = \frac{2890\sqrt{8.89x10^{-3}}}{2^{075}}$$

$S_m = 162$

From Figure 2.5, the expected Pump efficiency was

$$\eta = 66\%$$

The impeller and its definition is shown in Figure 3.3.

3.7 Impeller Design

3.7.1 Impeller Inlet Diameter

A nominal flow angle of 17° was selected. The following were the assumptions in the desi

- axial inlet of fluid flow
- the hub being small enough to obstruct inlet fluid flow

Let $D_h/D_s = 0.3$, therefore, for equation (2.11)[9],

$$Const3 = 1 - (0.3)^2$$

$$= \underline{\mathbf{0.91}}$$

The suction diameter was obtained from equation 2.4 [11],

$$D_s = \frac{2897Q^{\frac{1}{3}}}{(Const3.N\tan\beta_o)^{\frac{1}{3}}} = \frac{2897(8.89x10^{-3})^{\frac{1}{3}}}{(0.91x2890\tan 17^o)^{\frac{1}{3}}}$$

$$D_s = 64.5\text{mm}$$

3.7.2 Slip Factors and Head Coefficient

With N_{sm} = 17.13, the flow ratio C_{m3}/U_2 and the impeller outlet angle β_2 were [17],

$$\underline{\therefore \beta_2 = \mathbf{28^\circ}}$$

$$C_{m3} / U_2 = 0.0514$$

The slip factor is given in an implicit manner by supplying the number of vanes, Z, which is 4 to 8 for specific speeds up to $N_{sm} = 77$ [10]. Equation (2.6) for the slip factor gave a better approximation for $20° \le \beta_2 \le 30°$ and since we have $\beta_2 = 28°$, this formula was valid

$$\mu = 1 - \frac{\pi \mathrm{Sin} 28°}{5(1 - 0.0514 \mathrm{Cot} 28°)} = 0.673$$

Pfleiderer [10], a Physicist found a relationship, that was used here as a substitute for equation 2.5. This was quoted as

$$\mu = \frac{1}{1 + (\frac{a}{z})[1 + \frac{\beta_2}{60}]2/\{1 - (\frac{r_1}{r_2})^2\}}$$

In this design, a **Vane diffuser casing** was considered, resulting in

$$a = 0.6$$

$$r_1/r_2 = 0.304$$

Therefore, the slip factor is

$$\mu = \frac{1}{1 + \frac{0.6}{5}.[1 + \frac{28}{60}].\frac{2}{(1 - 0.304^2)}} = 0.721$$

The lowest value of the slip factors is chosen, therefore

$$\mu = 0.673$$

The hydraulic efficiency is [10]

$$\eta_h = 1 - \frac{0.071}{Q^{0.25}} = \frac{0.071}{(8.89x10^{-3})^{0.25}} = 0.769$$

$\underline{\eta_h = 77\%}$

The resulting head coefficient ψ from equation (2.7) is thus[11]

$$\psi = 2\mu\eta_h[1 - \frac{C_{m3}}{U_2}.Cot\beta_2]$$

$$\psi = 2(0.673)(0.769)(1 - 0.0514.Cot28^o)$$

$$\psi = 0.935$$

Thus, the necessary impeller peripheral velocity is

$$U_2 = \sqrt{\frac{2gH}{\psi}} = \sqrt{\frac{2(9.81)(40)}{0.935}}$$

$$U_2 = 28.97m/s$$

Therefore, the meridional velocity right after the impeller discharge is

$$C_{m3} = \frac{C_{m3}}{U_2}.U_2 = 0.0514(28.97)$$

$$C_{m3} = 1.49m/s$$

The peripheral component of the absolute impeller discharge velocity without slip is calculated with the geometric relationship as

$$C_{u3} = U_3 - C_{m3}Cot\beta_2 = 28.97 - 1.49Cot28^o$$

$$C_{U3} = 26.2m/s$$

The actual peripheral component is affected by the slip that takes place against the impeller. Its value is

$$C_{u3} = \mu C_{u3} = 26.2(0.673)$$

$$\mathbf{C_{u3} = 17.63 m/s}$$

3.7.3 Impeller Discharge Dimensions

The impeller outlet diameter was

$$D_2 = \frac{60U_2 x10^3}{\pi N} = \frac{60(28.97x10^3)}{2890\pi}$$

$$D_2 = 190mm$$

3.7.4 Impeller Inlet and Discharge Widths

Since this is an open impeller, let the ratio D_s/D_1 = 1.0. The formula is divided by an empirically obtained factor of 4 to take care of blockage by vanes to flow [10]. Therefore,

$$b_1 = \frac{K}{4}.D_s.\frac{D_s}{D_1} = \frac{0.91(64.5)}{4}$$

$$b_1 = 16mm$$

The impeller outlet width is

$$b_2 = \frac{Qx10^6}{\pi.D_2.C_{m3}} = \frac{8.89x10^{-3}x10^6}{191.4\pi 1.49}$$

$$b_2 = 10mm$$

Take on this value for b_1 as reasonable to ascertain uniform flow.

3.7.5 Inlet Vane Angle

The standard inlet vane thickness was taken to be 6.0mm for strength with respect to the pump size. This specification is a normal practical value in this respect [11]. The vane angle at the outer radius of the inlet is calculated from equation (2.12)[17] as

$$\beta_1 = 23^\circ$$

3.7.6 Verification of the Number of Vanes

The following relation applied [13],

$$\begin{aligned} \beta_m &= 0.5\,(\beta_1 + \beta_2) \\ &= 0.5\,(23 + 28) \\ &= 25.5^\circ \end{aligned}$$

and the number of vanes,

$$Z = \frac{6.5(D_1 - D_2)\mathrm{Sin}\beta_m}{(D_2 - D_1)} = \frac{6.5(64.5 + 191.4)\mathrm{Sin}25.5^\circ}{(191.4 - 64.5)}$$

$$Z = 5.6\,\mathrm{Vanes}$$

Since the pump was designed to have a vane diffuser an *odd* number of vanes on impeller was used. An even number for diffuser vanes was taken [9]. On this basis, let impeller vane number be 5 and reserve 6 for diffuser vanes. Thus,

z = 5 vanes

This implies, for Z = 5.6, the nearest odd number is 5 and number of diffuser vanes is

$$Z_2 = (Z+1)$$

$$= 6, \text{ the nearest even number}$$

$$\underline{\mathbf{Z_2 = 6 \text{ vanes}}}$$

3.7.7 Flow Areas between Vanes

The inlet area between vanes from equation (2.13) is approximately,

$$\mathbf{A_I} \quad \doteqdot \pi\, \mathbf{r_1^2 \,Sin}\, \beta_1$$

$$= \pi\, (65/2)^2 \,\text{Sin } 23^\circ$$

$$= \underline{1296.6 \text{ mm}^2}$$

From equation (2.14) the outlet area between vanes is

$$\mathbf{A_{II} = b_2\, (\pi\, D_2\, Sin\, \beta_2 - ZS_2)}$$

The ratio $(A_{II}/A_I) = A_r$ needs to be in the range $0.95 < A_r < 1.3$ [11]. This ratio was used to determine the vane thickness at the outlet, as follows

$A_r = 1.125$ - the mean of both extremes

Thus,

$$A_{II} = 1.125 A_I$$

$$= 1.125\,(1296.6)$$

$$= \underline{1458.68 \text{ mm}^2}$$

$$\mathbf{S_2 = 27mm}$$

3.7.8 Suction Velocity

V_s should lie between 1.5 and 2.5 m/s and the suction diameter should be sized accordingly [9],

Water inlet area required = 1296.6 mm^2

The boss diameter was calculated using the following relation [6] therefore,

Boss diameter = 2t + (1/7 to 1/6) D_2

Let the ratio be 1/7 for minimum possible blockage to flow at impeller inlet

Boss diameter = 2t -1 (1/7) D_2 = 2(3) + 191.4/7 = 33.3

<u>Therefore, Boss diameter = 32 mm</u>

This value was subject to correction upon completion of the shaft design. Assuming the boss as small enough to obstruct liquid flow we take the effective inlet area of the suction mouth For the impeller eye diameter of 65 mm,

A_{eye} = п (65/2) 2 = 3318.3 mm^2

$$\text{FlowVelocity} = \frac{8.89\text{x}10^{-3}}{3318.3\text{x}10^{-6}} = 2.7\text{m/s}$$

This V_s, is greater than 2.5 m/s. Considering the required range, the inlet area was adjusted. Let the suction flow velocity V_s = 2 m/s (mean), so that

$$2 = \frac{8.89\text{x}10^{-3}}{A_{eye}}$$

$$\Rightarrow A_{eye} = 0.004445\ m^2$$

Thus,

$$D_{eye} = \sqrt{\frac{4}{\pi}.A_{eye}}$$

Therefore, the adjusted impeller inlet width was

$$D_1 = \sqrt{\frac{4}{\pi}(0.00445)}$$

$$\underline{D_1 = 75.0 \text{ mm}}$$

Thus, the suction diameter is 75 mm to achieve this velocity. Discharge velocity will be considered during volute casing design.

3.7.9 Correcting the impeller outlet width

For uniform flow of liquid in the impeller from fluid mechanics laws of continuity [16]

$$b_1 = \frac{r_2}{r_1}.b_1$$

$$= \frac{190}{75}(10)$$

$$b_1 = 25\text{mm}$$

Summary of Impeller Dimensions (Figure 3.3)

D_1 = 75.0 mm b_1 = 25 mm β_1 = 23° S_1 = 6.0 mm Z= 5 vanes D_2 = 190 mm

b_2 = 1 0 mm β_2 = 28° S_2 = 27 mm

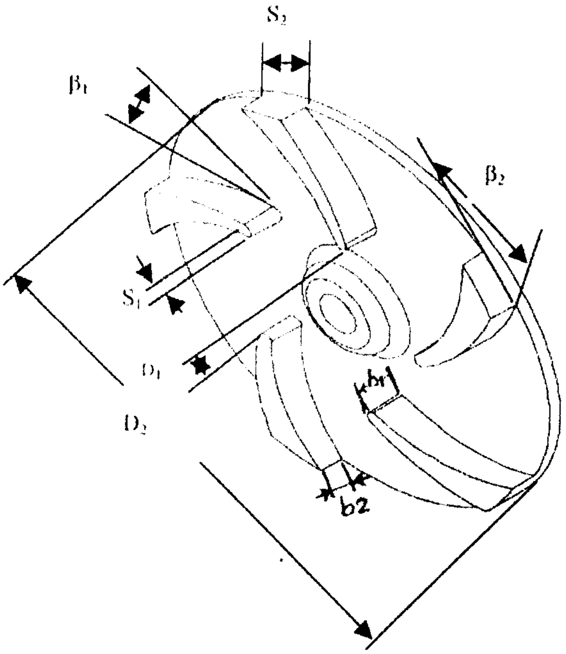


Figure 3.3: Open Impeller

3.8 Thrust Analysis on the Pump

3.8.1 Axial Thrust

This in vertical pumps is caused by

(i) Weight of rotating parts (independent of Q and H),

(ii) Dynamic forces (change in momentum) caused by change in flow direction through the impeller.

3.8.2 Dynamic Forces

3.8,2.1 Downward Thrust

Net Pressure = γH

= 9.79 (40)

= $\underline{3.916\ \text{bar}\downarrow}$

$\gamma = 9.79\ kN/m^2$: Specific weight (force) for water

Unbalanced impeller eye area $= (\pi D^2)/4$

$= \pi (0.075/4)^2$

$= \underline{0.0044179\ m^2}$

The effect of pressure distribution on the shrouds of impeller was neglected (since open impeller); the downward thrust was

Force $= \text{Net Pressure} \cdot A_{eye}$

$= (3.916 \times 10^5)\ 0.0044178$

$= \underline{1730N\downarrow}$

Force = $\underline{1730N\downarrow}$

3.8.2.2 Upward Thrust

Assuming normal operating conditions, the upthrust on the single suction impeller caused by change in momentum was determined from fluid mechanics [13]. This was obtained as follows:

$V_e = Q/A_{eye}$

$= 2\ m/s$: as found in impeller design

In this case, the impeller type used is fully radial, thus k = 1 and upthrust is

$$F_u = \frac{kV_{eye}\,2.A_{eye}}{2.g(1.02)}$$

$$= \frac{1(2)(0.004418)(10{,}000)}{2(9.81)(1.02)}$$

$$= 8.83N\uparrow$$

This is negligible relative to the downward hydraulic axial thrust. Note that the factor 1,02 in the denominator is an empirically obtained value [11].

3.8.2.3 Static Forces

To evaluate total amount of force on the shaft of the pump while at rest, only the static forces are determined. These are made up of the following:

(i) Shaft weight,

(ii) Rotor weight,

(iii) Impeller weight

3.8.3 Estimate of Shaft Diameter

A preliminary estimate of the shaft was computed using equation 2.20. The power m the shaft was obtained using equation 2.21 as

$$P_s = \frac{9.8QHSp.gr}{\eta}$$

$$= \frac{9.8(8.89\times10^{-3})(40)(1)}{0.66}$$

$$P_s = 5.28kW$$

Relating to the type of application, code 080M40 Steel EN8 was recommended for the shaft. It has the following properties:

S_{ut} = 510 MPa (min); S_y = 245 MPa; BHN = 152 (min)

By Distortion Energy Theory [11] equation (2.22) gives the yield shear stress as

$$S_{sy} = 0.577\ S_y = 0.577\ (245) = \underline{141.365\ MPa}$$

Therefore, the shaft diameter was

$$d_{solid} = (5.234).\sqrt[3]{\frac{P_s}{S_{sy}.N}}$$

$$= 5.234.\sqrt[3]{\frac{5.28\text{x}10^3}{141.365\text{x}10^6\text{x}2890}}$$

$$d_s = 0.0124\text{mm}$$

With a design factor of n = 3 for most pump designs [11]

$$d_s = 0.0124\ (3)^{1/3} = 0.018$$

Shaft diameter at impeller was therefore,

$$\underline{\mathbf{d_s = 18\ mm}}$$

3.8.4 Estimate of Shaft Weight

From equation (2.23) the shaft weight is

$$\text{Mass} = 0.00616\,d^2\ (\text{kg/m})$$

With the shaft length of 800-mm (0.8 m),

$$\Rightarrow \text{Mass} = 0.8\ (12)^2 (0.00616) = 1.6\ \text{kg}$$

$$\text{Weight} = 1.6\text{x}9.81$$

$$\underline{\mathbf{Weight = 16\ N\downarrow}}$$

This length of 0.8m was determined from the total length required for a single shaft to run right from the upper bearing to the impeller, including the motor height. For computations of mass, 12mm diameter was used as an estimate because the shaft was not a single bar but had locations for bearings, notched areas for other attachments. This was justification enough to use 12mm instead of 18mm for this computation.

3.8.5 Estimate of Impeller Weight and Axial Thrust

This is made from aluminium (relative density with steel, 0.79) and relating to its dimensions it was assumed a block in form of disc of steel of dimensions:

Average thickness = 0.5 (10 + 25) = 18 mm

Diameter = D_2 = 190 mm

Mass = 0.018 (190) 2 (0.00616) = 4 kg

<u>Weight = 39.27N</u>

Since the motor rotor was supported on its bearings, it was included as part of the shaft. Therefore,

Total downward force = 39 + 16 + 1730

<u>∑ Downward Force = 1785 N↓</u>

∑ **<u>Upthrust = 8.83 N↑</u>**

The resultant axial thrust = 1785 -8.83

<u>F_a= 1780 N</u>, -neglecting tile effect of water as it flows due to change in flow direction.

3.8.6 Radial Thrust and Shaft Design

Interpolating from Tables of stress factors [15],

$$K_t' = 1 + \frac{(2-1).(17.13-180.8)}{(15.38-180.8)} = 1.989$$

From equation (2.15), the radial force is

$$F_{radial} = 9790 K_t H D_2 b_2$$

$$= 9790(1.989)(40)(0.19)(0.01)$$

$$F_r = 1480\ \text{N}$$

The radial force due to shaft power in rotation is

$$T = \frac{30.P}{\pi N}$$

$$= \frac{30 * 5.28\text{x}10^3}{2890\pi}$$

$$= 17.45\text{Nm}$$

$$F = \frac{T}{R} = \frac{17.45}{0.19} = 92\text{N}$$

Thus, the total radial force on the shaft is

$F_r = 1.48 + 0.092$

= 1.6kN

The result is

Axial Force: **F_a = 1.78 KN↓**

Radial Force: **F_r = 1.60 KN →**

This is a case of combined torsion and tension. From equation (2.19) and (2.17)

$$S_{smax} = 0.5*[S^2 + 4S_s^2]^{0.5}$$

The total twist in the shaft is

$$T \quad = F_r * r$$

$$= 1.6 \times 10^3 (9 \times 10^{-3})$$

$T = 14.4$ Nm

$$S_s = \frac{T.c}{J}$$

$$\frac{c}{J} = \frac{16}{\pi d^3}$$

$$S_s = \frac{14.4(16)}{\pi(18x10^{-3})^2}$$

$S_s = 12.6$MPa

$$S = \frac{F_a}{A} = \frac{1.78x10^3(4)}{\pi(18x10^{-3})^2} = 7\text{MPa}$$

$$S_{smax} \quad = 0.5*[S^2 + 4S_s^2]^{0.5}$$

$$= 0.5 * [7^2 + 4*12.6^2]^{0.5}$$

$S_{smax} = 13$Mpa

$$S_{max} = 0.5\ S + S_{smax} = 0.5*(7) + 13$$

$S_{max} = 16.5$Mpa

3.8.7 Allowance for Keyways

To take care of the keyway, the following relation was used to determine the required diameter [12]. For general design of steel shafts

$$d_r = 1.05d. = 1{,}05\ (18)$$

$$\underline{\mathbf{d_{shaft} = 19.0\ mm}}$$

The value takes care of fatigue as the formula is based on the results of fatigue experiments. This is the minimum shaft diameter, which was the size at the impeller attachment point.

3.9 Other Shaft Design Considerations

For shaft design considerations [16]

$$D/d = 1.2 \Rightarrow \text{let it be } D/d = 1.1$$

Therefore,

$$\begin{aligned} D &= 1{,}1d \\ &= 1.1\ (19{,}0) \\ &= \mathbf{20.9\ mm} \end{aligned}$$

Thus, the next step on the shaft diameter is

$$\underline{\mathbf{d_{shaft} = 19.0\ mm}}$$

Check,

$$h = 0.1d = 0.1 * 21 = 2.1$$

$$h = 0.5^{*} (21-19) = 1.0 \text{ mm} < 2.1 \text{ mm}$$

$$r = 0.5h = 0.5 * 1.0 = 0.5 \text{ mm}$$

$$\underline{\mathbf{r = 0.6 \text{ mm}}}$$

3.9.1 Shaft-housing Clearance

The type of sealing mechanism was a mechanical seal and had the following dimensions:

$Ø_{min}$ = 15.0 mm on the rubber side

$Ø_{max}$ = 23.0 mm on the seal-face side

Therefore, the seal/shaft radial clearance was

$$0.5 * (23\ 21) = 1.0 \text{ mm}$$

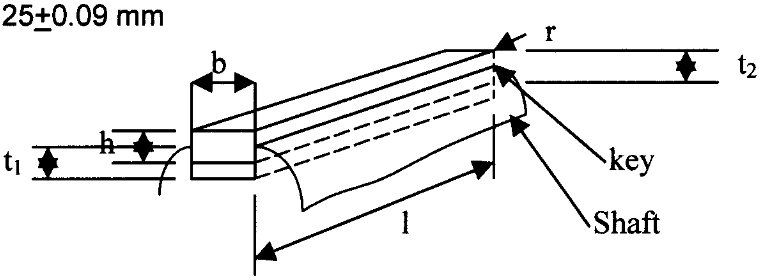
This falls within the prescribed practical range of 0.9 – 1.3 mm.

3.9.2 Keyways and Keys

From standard tables [16] on keys and shafts with d = 19 mm,

$b = 6.0$ mm $h = 6.0$ mm $t_1 = 3.5$ mm $t_2 = 2,8$ mm $l = 25.0$ mm

$r = 0.25 \pm 0.09$ mm



3.9.3 Shaft Elongation

In vertical pumps, this is caused by two separate phenomena:

(i) tensile stress due to the hydraulic thrust - arising from,

   (a) shaft weight and those of the impellers mounted thereon,

   (b) the hydraulic thrust itself.

(ii) thermal expansion.

The shaft length from the lower thrust bearing to the bottom of the impeller (overhung) was

= 162 mm

Modulus of Elasticity of the shaft, E = 207Mpa

Shaft + impeller weight

= 39-27 + 16

= 55.27 N (5.63 kg)

Shaft diameter, d_{shaft} = 19.0 mm [Cross Sectional Area =0.00028352 m^2]

Elongation caused by weight = FL/(AE) = 55.27(0.162)/(0.00028352*207x10^6)

= **0.17 mm**

Elongation caused by Hydraulic thrust is= (1730 -8.83) * 0.62_/(0.00028352*207x10^6)

= **0.0053 mm**

The resultant elongation, $\Sigma\delta$ = 0.00017 + 0.0053

$\Sigma\delta$ = 0.00547 mm

This was compensated for in assembling and setting the pump in order to avoid interference contact between stationary and rotating parts.

The expansion caused by temperature is completely independent of elongation of the shaft caused by the downward hydraulic thrust. Here, the product was recommended for fairly cold water at maximum temperature of 40°C.

3.10 Other Corrective Measures

3.10.1 Critical Speed of the Shaft

The expression for determining the critical speed was given from equation 2.26 [10]

$$f = \frac{4*5.61*9.81(0.162)^3*64}{100*207x10^9*\pi*(0.021)^4} = 0.0003048m$$

$$N_c = \frac{946}{(0.3048)^{0.5}} = 1713rpm$$

Thus, the 1st Critical Speed, **N_c = 1713 rpm**

Checking by how much the operating speed was above the 1st Critical Speed, the following computations were performed:

% = [(Operating Speed – Critical Speed)/(Operating Speed)]* 100

= [(2890 -1713)/ 2890]*100

= **40.7 %**

This is a reasonable difference when related to practical ranges stated in Chapter 2. However, the damping effects of the stuffing boxes will raise this value of the critical speed, which will further favourably narrow this gap to within the recommended range [10].

3.10.2 Checking the Rules of Critical Speed

Rule 1: 1713/2890 = 0.5927 - **Satisfied**

Rule 2: 1.3*1713 = 2226 rpm < 2890 - **Satisfied**

According to [10], $Nc_2^2 = 8*N_{c1}^2$

This gave N_{c2} = 4845 rpm

$0.7N_{c2} = 0.7*4840 = 3392$ rpm > 2890 rpm – **Satisfied**

Rule 3: Operating speed is below the 2nd Critical Speed – **Satisfied**

3.11 Casing Design

As this was a single-stage pump, no return passages were designed but only diffuser vanes. The casing is shown in Figure 3.4.

3.11.1 Vane Distance from Impeller Discharge Diameter

The constant K_3 was determined from the equation (2.27) as,

$$K_3 = (438\ N_{sm})/820$$

$$= (438 - 17.13)/820$$

$$= 0.513$$

The average casing throat velocity (past channel of fluid) was

$$C_{thr} = K_3\sqrt{(2gH)}$$

$$= 0.513\sqrt{(2*9.81*40)}$$

$$= 14{,}37 \text{ m/s}$$

and

$$\rho = (3.39 + N_{sm})/2.653$$

$$= (3.39 + 17.13)/2.653$$

$$\mathbf{= 7.73\ \%}$$

Note that in evaluating ρ *Const8* is explained in equation 2.26 with its source. The diffuser vane distance from the impeller outer diameter was

$$t = \rho\, r_2/100$$

$$= 7.73*95/100$$

$$\mathbf{= 7.3\ mm}$$

3.11.2 Channel Proportions

Taking on a rectangular channel

$$b_3 = 1.1b_2$$

$$= 1.1*10$$

$$\mathbf{= 11.0\ mm}$$

$$r_3 = t + r_2 = 7.\ 3 + 85$$

$$\mathbf{= 102.3\ mm}$$

Therefore, diffuser inner diameter was

$$\underline{\mathbf{D_3 = 205\ mm}}$$

Let diffuser outer diameter,

$$D_4 = 1.475D_2$$

$$\Rightarrow \underline{\mathbf{D_4 = 1.475*190 = 280\ mm}}$$

3.11.3 Channel Width and Divergence

Let the vane thickness be 9.8% of diffuser outer diameter [17], $S_3 = 26.0$ mm

Therefore, the space taken up by the six (6) vanes = 26.0 * 6

$$\mathbf{= 156.0mm}$$

The total circumference $= \pi 205$

= 644.0 mm

Therefore, clear length $= 644 - 156$

= 488.0 mm

Clear (unobstructed) inlet space $= 488/6$

= 81.0 mm

3.11.4 Summary of the inlet parameters

Inlet vane thickness: **S_3 = 26.0 mm**

Number of vanes: **Z_2 = 6 vanes**

Clear inlet space: **81.0 mm**

The inlet to outlet area of diffuser should be 1:6 so that for circumference

$$S_4 = S_3\sqrt{6}$$

$$= 26\sqrt{6}$$

$$= 63.0\text{mm}$$

$\underline{S_4 = 63.0\text{ mm}}$

3.11.5 Diffuser Vane Angles

The inlet angle to the channel is

$$\alpha = (20 + N_{sm})/6.23$$

$$= (20 + 17.13)/6.32$$

$$= \mathbf{6°}$$

Vane entrance angle is 36° larger than a [8]. So take 4° larger, thus

$$\theta = 4 + 6$$

$$= \mathbf{10°}$$

The total divergence per channel was 11° per channel [10] so that for each side we have 5.5°. The area expansion stated above corresponded to 8,5° for parallel-walled diffuser. This was thus taken as the 'channel base slope when halved. Thus,

Diffuser base slope = 4.2°

Total divergence, $\gamma = 11°$

SIDE DIVERGENCE

- OUTER: **10 + 5.5 = 15.5°**

- INNER: **10°**

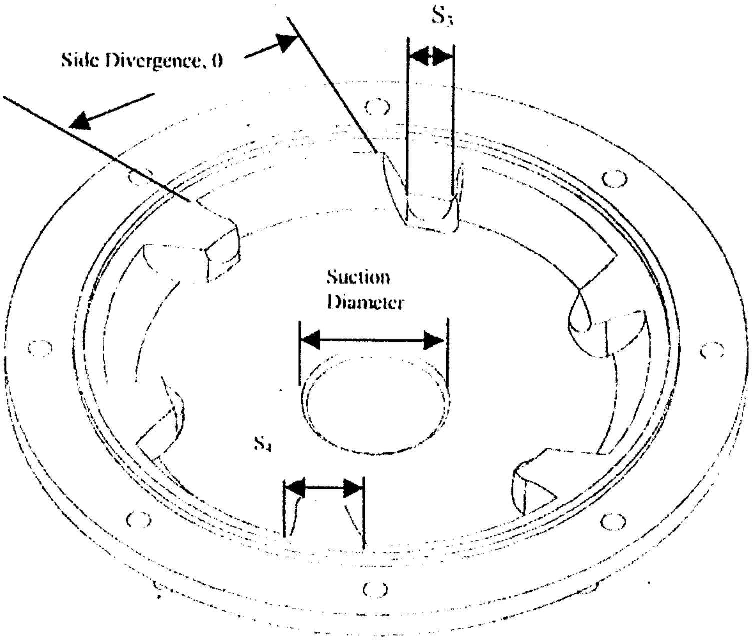


Figure 3.4: Diffuser Casing

Chapter 4
Manufacturing Considerations

4.1 Introduction

In this chapter, tooling, equipment, operation sequences and other technical means to enable production are given. This was to ensure that the designs were satisfactory from a manufacturing standpoint. The processes were also planned with a view of minimising cost. It was felt that this latter point deserved separate comment, thus it is covered in detail later in this chapter. In the author's experience, by far the most lucrative avenue is the one in which the product design is analysed for lower-cost alternatives (value analysis). This approach proved to provide a larger return (greater cost reduction) per unit of effort arid per unit of investment than other approaches, including mechanisation, automation, wage incentives, etc [7].

4.2 The ISO Tolerance Model

Tolerance requirements here follow hierarchy rules, which require the form tolerance to have the smallest value. Geometric tolerance is included in size of the part. These hierarchy rules ensure that the parallelism requirement (orientation) is larger than the flatness requirement (form) for the same surface. It must be noted that allowable deviations (tolerances) are depicted and not realised errors. Therefore, it was possible, for example, for the form tolerance to rotate and shift slightly in the drawing plane relative to other tolerances. This hierarchy rule can be described as:

Size ≥ location ≥ orientation ≥ form tolerance

From the aforementioned, it was justifiable to follow this procedure; bearing in mind that standardisation in any design is the key to international recognition. For the bearing housing hole, the geometric (size) tolerance has already been selected as ±0.07 mm (refer to Section 4.4), the location tolerance in this case relates to circularity (roundness) and is ±0.01 mm. The orientation class of geometric tolerance for this hole is perpendicularity. It should lie, for the bearing selected [19], within about 0.01 mm. What remains is then to determine the form tolerance.

4.3 Surface Texture

Surface texture consists of the repetitive and/or random deviations from the nominal surface of an object; it is defined by four elements: roughness, waviness, lay, and flaws. Roughness refers to the small; finely spaced deviations from the nominal surface and is determined by the material characteristics and the processes that formed the surface. Waviness is defined as the deviations of much larger spacing; they occur due to work deflection, vibration, heat treatment, and similar factors. Roughness is superimposed on waviness. Lay is the predominant direction or pattern of the surface texture. It is determined by the manufacturing method used to create the surface, usually from the action of a cutting tool. A majority of operations on the milling CNC machine were dominated by lay circular relative to the centre of the surfaces, This, in Mastercam tool path programming is called spiral machining. However, since most of the parts were not spirals, a type of spiral method called parallel spiral was employed. Amongst many causes of good surface finish is the tool path propagation. In areas where a lot of material was to be removed, the tool path was programmed to cut from inside to outside, to reduce on tool load. Finally, flaws are irregularities that occur occasionally on the surface; these include cracks, scratches, inclusions, and similar defects in the surface integrity.

Surface roughness and finish are two terms included within the scope of surface texture. Surface roughness is a measurable characteristic based on the roughness deviations. Surface finish is a more subjective term denoting smoothness and general quality of a surface. In popular usage, surface finish is often used as a synonym for surface roughness. The most commonly used measure of surface texture is surface roughness. Surface roughness can be defined as the average of the vertical deviations from the nominal surface over a specified surface length. An arithmetic average (AA) was generally used, based on the absolute values of the deviations, and this value is average roughness. In equation form [7],

$$R_a = \int_0^{L_m} \frac{|y|}{L_m} \qquad (4.1)$$

where R_a is the arithmetic mean of roughness (m); *y* is the vertical deviation from nominal surface (converted to absolute value), m; and L_m is the specified distance over which the surface deviations are measured. An approximate equation, easier to comprehend is given by [7]

$$R_a = \sum_{i=1}^{n} \frac{|y_i|}{n} \qquad (4.2)$$

where y_i = vertical deviations (converted to absolute value) identified by the subscript i, in m; and n = the number of deviations included in L_m. Since these parameters are usually small values, the most commonly used units are micrometers and/or millimetres. The general values for the surface finish used were 200μm for milling, turning and drilling and 600μm for sand casting.

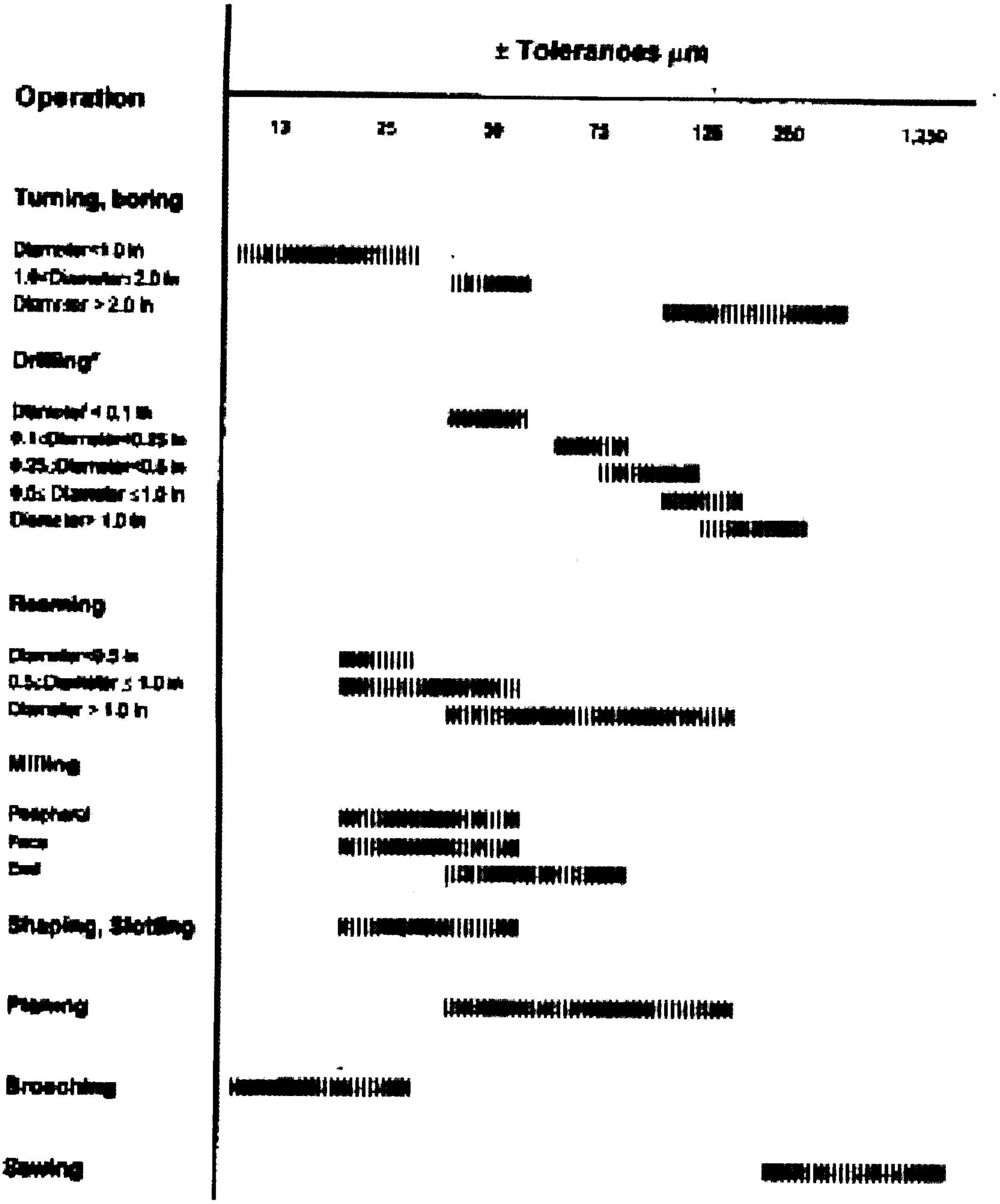
4.4 Surface Finish and Geometric Tolerances

Machining operations are used to produce parts with geometry, tolerances and surface finish specified by the product designer. In this report, issues related to the creation of the parts by machining are discussed. The following were critical:

1. The most liberal surface finish and dimensional tolerances possible, consistent with the functions of the surfaces were specified. This was to simplify the prime machining operations and to avoid costly secondary operations like grinding, lapping, reaming, etc.
2. The parts were designed for easy fixturing and secure holding during machining operations. It is the reason why no special fixtures were designed for this project despite having complex shapes such as the diffuser casing. To ensure this was achievable, large solid mounting surfaces (such as steel blank from bar stock) with parallel clamping surfaces were provided for to assure a secure set-up.

Figure 4.1 [6] indicates typical tolerances that can be achieved for most machining operations examined in this project. The values in this tabulation represent ideal conditions, yet conditions that are readily achievable in a modern factory. If the machine is old and worn, process variability will likely be greater than the ideal, and these tolerances would be difficult to maintain. On the other hand, newer machine tools can achieve closer tolerances than those listed. Consider the bearing housing design in Figure 4.3 (check at the end of this chapter) for example.

These ranges were used in the design of moulds as well as functional surfaces of the pump. Taking into account the processes for the production of the bearing housing, tolerances were determined. For example, two surfaces are critical: the surface mating with the motor casing and the surface bounding the cavity in which the bearings are assembled, For the bearing/casing surface the processes in machining were generally turning followed by milling.



*Drilling tolerances typically expressed as a biased bilateral tolerance (for example, +0.005/-0.001). Values in this tabulation are expressed as closest bilateral tolerance (e.g., ±0.003).

FIGURE 4.1: Typical tolerances achievable in machining operations [6].

The last process determines the final surface finish quality. The nominal size of the hole is ∅52.09 mm. From Figure 4.1 it was justifiable to take a tolerance of ± 0.05 mm. Thus it gave ∅52.090.05 mm diameter implying the upper and lower limits are respectively ∅52.14 mm and ∅52.04mm. The bearing still fits for both the lower and upper limits. The run-out on the higher and lower sides will respectively be 0.07mm and 0.002mm. In Chapter 3 it was noted that the radial play allowable was ±0.05mm. There, ±0.07mm for a thrust bearing was considered fair. Table 4.1 gives recommendations for location tolerances [7].

TABLE 4.1: **Recommended Location Tolerances for Machined Holes**

		± Distance from true position (mm)	
		Normal tolerance	Close tolerance
1.	Using normal layout, centre punch & drill	0.50	0.25
2.	Using drill fixture with bushing; part located in fixture from existing surface	0.25	0.13
3.	Using precision milling or Numerical-Control machine; part located in fixture from existing surface	0.20	0.10
4.	Using precision milling or NC machine; part located by indicating, optical measurement, or from previous hole	0.05	0.025
5.	Jig boring machine; part located as in number 4	0.025	0.005

Tolerances taken generally for this design were for 1 and 4 in Table 4.1. For all drilling operations, a tolerance of ±0.5mm was taken whereas for CNC machining it was ±0.05mm was recommended for location tolerances. Table 4.2 is a list of the dimensional tolerances for Milling Machine operations [7].

Two options (Table 4.1, point 4) were possible:

(i) Turn the whole part except for the hole ∅52.09mm, or

(ii) Turn the whole part with roughing of ∅52.09mm hole

TABLE 4.2: Recommended Dimensional Tolerances for Milling-Machine Operations (variations from basic dimensions).

Operations	*Basic Dimension, mm*					
	To 6.3	To 12.7	To 19	To 25	To 50	To 100
Straddle Milling	± 0.5	± 0.5	± 0.5	± 0.5	± 0.5	± 0.5
Slot width	± 0.04	± 0.04	± 0.05	± 0.05	± 0.05	± 0.06
Slot width (end mill)	± 0.05	± 0.06	± 0.06	± 0.06		
Face Milling	± 0.05	± 0.06	± 0.05	± 0.05	± 0.05	± 0.05
Hollow Milling		± 1.5	± 2.0	± 2.5		

Source: General Motors Equipment Standards.

Design recommendations for NC Milling relating to this case state that:

(a) When a small, flat surface is required, as for a bearing surface or a bolt-head seat perpendicular to a hole, the product design should permit the use of spot facing, which is quicker and more economical than face milling. Relating to this case, there was no advantage to start the ∅52.09-mm hole on the milling machine but rough-turn the hole in advance and spot-mill the then contour.

(b) When spot-faces or other small milled surfaces were required for castings, a low boss for the surface to be machined was incorporated in the design. This simplified machining and paint removal and resulted in a less sharp edge.

Therefore, it was found much better to work with a contour than a pocket. This option was taken because a turned hole becomes a contour in milling finish operation, making tool life reliable.

Project case: ± 0.07-mm possible loss of location.

If followed that with this Process of production, the tolerance fell within the limits because what was needed was ±0.07 mm when the process capability was ±0.05mm.

4.5 Surfaces and Manufacturing Processes

The manufacturing process determines surface finish and surface integrity. Surface integrity is not discussed in detail here. Some processes are inherently capable of producing better surfaces than others. In general, processing cost increases with improvement of surface finish. This is because additional operations and more time are usually required to obtain increasingly better surfaces. Table 4.3 shows the processes applicable to this method together with the usual surface roughness that can be expected from them.

TABLE 4.3: Surface Roughness Produced by Various Manufacturing Processes *(source:* [6])

Process	Typical surface finish range of roughness[1], μm	
Casting:		
Die casting	Good	30-65
Investment casting	Good	50-100
Sand casting	Poor	500-1000
Machining:		
Boring	Good	15-250
Drilling	Medium	60-250
Milling	Good	30-250
Shaping	Medium	60-500
Sawing	Poor	100-1000
Turning	Good	15-250

1 subjective description and typical range of surface roughness values are given, micrometer Roughness can vat)' significantly for given process, depending on process parameters.

4.6 Geometric Factors

These were also used to determine the geometry of the surface on machined parts. They included

- type of machining operation,
- cutting tool geometry, most importantly nose radius, and

- feed-rate.

The surface geometry that would result from these factors is referred to as the ideal or theoretical surface roughness, which is the finish that would be obtained in the absence of work material, vibration, and machine tool factors. Tool geometry and feed-rate combine to form the geometry of the surface. In tool geometry, the shape of the tool point is the important factor. Figure 4.2 presents a listing of typical surface finishes that can be achieved in machining operations. Roughness values generally used in this project are included in the third column of Figure 4.2. The data in this figure represent finishes that should be readily achievable by modern, well-maintained machine tools.

Machining operation	*Surface Roughness (AA), μm*	*Values of roughness used in this project, μm*
Turning	0.81-6.3	0.86
Boring	0.4-6.3	0.6
Drilling	0.81-6.3	0.9
Reaming	0.4-3.2	1.00
Milling	0.4-3.2	0.45
Shaping	1.6-12.7	1.8
Planing	1.6-12.7	N/A
Broaching	0.2-3.2	0.5
Sawing	6.3-25.4	10.5

Figure 4.2: Surface-finish values (AA) typically achieved in various machining operations [7].

The effects of nose radius can be combined in an equation to predict the ideal arithmetic average surface roughness for a surface produced by a single-point tool [6]. The equation applies to operations such as turning, shaping, and planing:

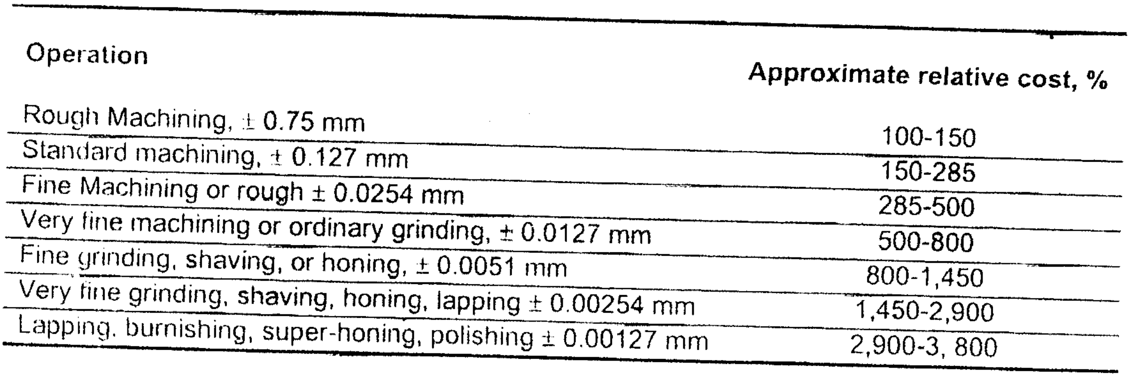
$$R_i = \frac{f^2}{32.NR} \qquad (4.3)$$

where R_i is the theoretical arithmetic average surface roughness, mm, f = feed, mm; and NR nose radius on the tool point, mm. The equation assumes the nose radius is not zero and that feed and nose radius will be the principal factors that determine the geometry of the surface. This equation was also used to estimate the ideal surface roughness in face milling with insert tooling, using f to represent the chip load (feed per tooth). However, it should be realised that both the leading and trailing edges of the rotating face milling cutter reduce feed marks on the work surface, which complicates the surface geometry.

4.7 Costs versus Tolerances

Another point worth of consideration was the cost due to dimensional tolerances. Table 4.4 [7] shows the effects.

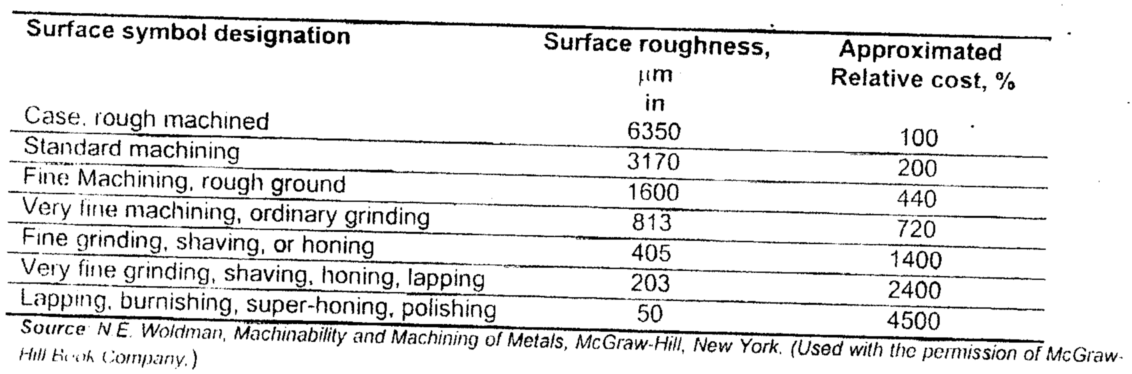
TABLE 4.4. Approximate relative cost of progressively tighter dimensional tolerances, *(From N. E. Woldllli1r1. fMahinability and Machining of tv1etals, McGrraw-Hill, New York, Used with pem1ission of McGra'N-Hill Book Company)*



Operation	Approximate relative cost, %
Rough Machining, ± 0.75 mm	100-150
Standard machining, ± 0.127 mm	150-285
Fine Machining or rough ± 0.0254 mm	285-500
Very fine machining or ordinary grinding, ± 0.0127 mm	500-800
Fine grinding, shaving, or honing, ± 0.0051 mm	800-1,450
Very fine grinding, shaving, honing, lapping ± 0.00254 mm	1,450-2,900
Lapping, burnishing, super-honing, polishing ± 0.00127 mm	2,900-3, 800

The following Table 4.5 [6] shows the cost with respect to surface roughness recommendations.

TABLE 4.5: Cost of Producing Surface Finishes



Surface symbol designation	Surface roughness, μm in	Approximated Relative cost, %
Case, rough machined	6350	100
Standard machining	3170	200
Fine Machining, rough ground	1600	440
Very fine machining, ordinary grinding	813	720
Fine grinding, shaving, or honing	405	1400
Very fine grinding, shaving, honing, lapping	203	2400
Lapping, burnishing, super-honing, polishing	50	4500

Source N.E. *Woldman, Machinability and Machining of Metals, McGraw-Hill, New York. (Used with the permission of McGraw-Hill Book Company.)*

These tables were used in most decisions from the conceptual design stage. For the Ø52.09-mm hole in the bearing housing, a tight tolerance of 0.025 (option 4, Table 4.1) would imply 285-500% relative cost. This is approximately 3-5 times the normal expectation. It is therefore justifiable to consider the tolerance of ± 0.07 mm despite having a difference of 0.02 mm from the recommended normal of ±0.05 mm. However, this deviation would reduce the bearing life with time. From Table 4.2, the compromise dimensional tolerance is ± 0.06 mm. In the choice for this project, ±0.07 mm was recommended. Thus, this was taken as the maximum deviation permissible. From the foregoing, the dimension for the hole is summarised as follows:

- Location tolerance: ± 0.1 mm,
- Dimensional tolerance: ±0.07 mm, yielding Ø52.09 ±0.07 mm.

This tolerance was more liberal than the earlier ±0.05-mm and thus fairly low cost for production.

4.7.1 Mating of the Bearing Housing with the Motor Casing

The following is an illustration of the tolerance selection process undertaken in this project. This was so with the view to using logic from Table 4.5 to ensure low cost for production. The example illustrates two mating parts: the bearing housing on its lower spigot and the motor casing.

Bearing housing spigot

This part would be produced through a turning operation, thus the tolerance was chosen as ±0.065 mm (Fig. 4.2). The dimension of spigot (outer diameter) is then

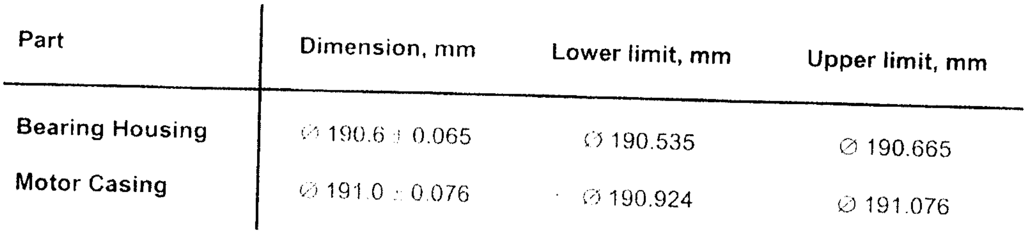
Ø190.6 ±0.065 mm

Motor Casing

This was a turned surface after a sand casting operation. The tolerance was ±0.076 mm giving a dimension of

Ø 191.0 ±0.076 mm

Check for collision



Part	Dimension, mm	Lower limit, mm	Upper limit, mm
Bearing Housing	Ø 190.6 ± 0.065	Ø 190.535	Ø 190.665
Motor Casing	Ø 191.0 ± 0.076	Ø 190.924	Ø 191.076

Thus,

The maximum clearance = $(UL)_{mc} - (LL)_{BH}$

0.541 mm

The minimum clearance = $(LL)_{mc} - (UL)_{BH}$

0.259 mm

Deduction: It was still safe to adjust dimensions to

- Spigot: ∅190.6 ± 0.08 mm
- Motor Casing: ∅191.0 ± 0.1 mm

This adjustment was still alright because it left a minimum clearance of 0.38 mm. A more liberal tolerance was specified for the Motor Casing since it was a product from the sand casting process, which required a greater amount of metal removal and has more irregularities. The bearing housing casting was then machined, hence the reason for the lower tolerance selected in specific areas: - the bearing seat and spigot. With the above method, the cost of the product reduces to reasonable levels.

4.8 Low Manufacturing Cost Approach

Methods to optimise manufacturing were considered in the conceptualisation stage of this project. During the design stage the preparation of the final drawings was focused on ensuring that the dimensional tolerances were realistic. This section covers planning of the manufacturing processes for the economic production of the pump, in addition to high quality product design considerations. The areas of concern include Process Planning, Problem Solving and Continuous Improvement, and Design for Manufacture.

4.8.1 Process Plans in this Project

Process Planning, being the principal activity of manufacturing engineering, includes:

- (i) deciding which processes and methods to use and in what sequence,
- (ii) determination of tooling requirements,
- (iii) selection of production equipment and systems, and,
- (iv) estimating costs of production for the selected processes, tooling and equipment.

The following were the outlines of the Process Plans for the Impeller, Bearing Housing and the Diffuser Casing:

The Impeller

1. Manufacture of the permanent mould of the impeller (an example program of the processes is shown in Appendix C).
2. Cast the impeller using the permanent mould.
3. Clean (debur) the product of the sprue and gates.
4. Bore the shaft hole.

The Bearing Housing

1. Sand-cast the bearing housing
2. Turn all the cylindrical parts of it to dimension but rough parts to undergo additional machining. Include the spigot. Rough-turn the bearing seat.
3. Drill all the holes to size.
4. Tap the holes on the bearing-covering end.
5. Mill, using the CNC machine, the bearing seat for accurate location of shaft, impeller and casing with respect to the bearing housing.

The Diffuser Casing

1. Use the permanent mould to cast the diffuser casing.
2. Clean the product by removing all casting attachments.
3. Drill all holes.
4. Tap holes for maze support.
5. Turn the part on a lathe machine to the right dimensions on the suction end followed by turning the bearing housing seat on the side of the diffuser vanes.

The following is an illustration of how the methods of production were selected for some parts in this project. The bearing housing, diffuser casing and impeller illustrated the approach. The material used for all of them is aluminium. (Refer to Table 4.6 and Figs. 4.4, and 4.5).

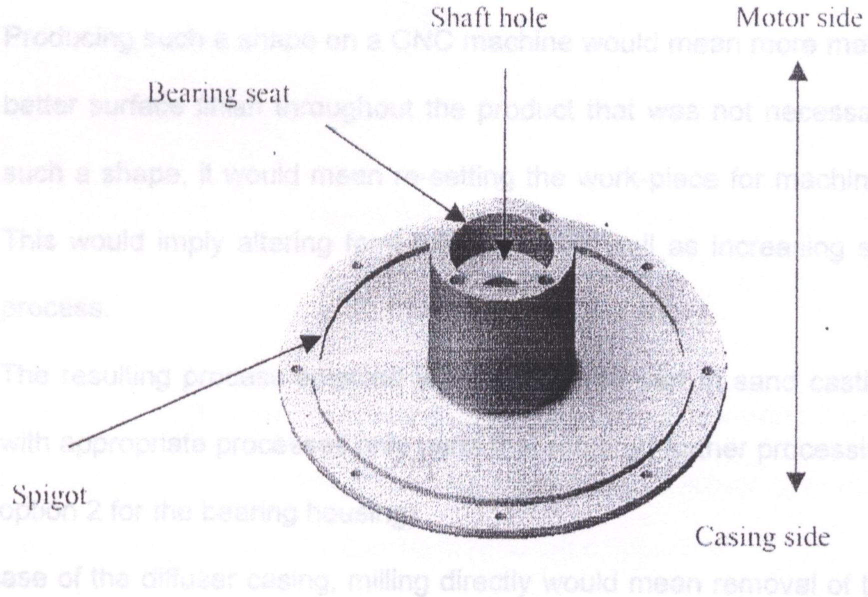


Figure 4.3: Bearing Housing

It can be noted from the bearing housing geometry, (Fig. 4.3), that it was reasonable to produce it either from a standard cylindrical blank (0250 mm) of aluminium or casting to shape followed by machining to finish. Comparing this alternative .to casting contemplated upon, it was less economical to use Permanent Mould Casting, because of tooling development. Reasons were as follows:

1. The component was cylindrical and the resulting shape symmetrical about the central axis.
2. The volume of material to machine justified the choice of turning the high volumes on a cylindrical part. However, waste is quite high so that sand casting followed by turning and milling critical parts was preferred.
3. Surface finish quality was not critical on all surfaces to machine. Only the bearing seat has the specified form tolerances and geometrical tolerances to ensure long bearing life in service.
4. Producing such a shape on a CNC machine would mean more material removal for a better surface finish throughout the product that was not necessary. In addition, for such a shape, it would mean re-setting the work-piece for machining the other side. This would imply altering form tolerances as well as increasing set-up times in the process.
5. The resulting process selected was casting the part in sand casting and machining with appropriate processes only parts that required further processing. (See Table 4.6 option 2 for the bearing housing).

In the case of the diffuser casing, milling directly would mean removal of too much material, thus promoting wastage. To economise on material, this wastage was allowed once, that is, during mould production. Thus, despite the cost of permanent mould being high, it was cheaper with increased production since the same would be used repeatedly. In other words, it is a case of "machine once to cast 1000 pieces from a single mould". The subsequent processes are economical in the way that machining tools are not used in subsequent production operations until the need to replace the old mould was found.

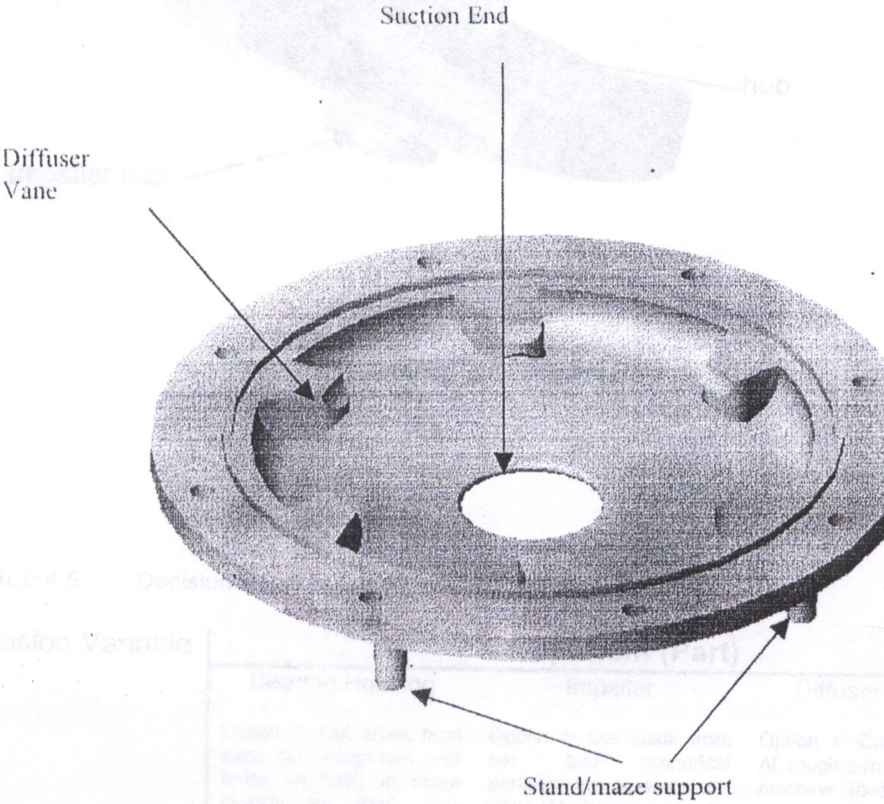


Figure 4.4: Vane Diffuser Casing.

With the impeller, the blades or vanes were difficult to generate with added drafts in Mastercam during a 'join' translation function. It was much easier to trim profiles or patches in pockets than on contours. This approach reduced design time.

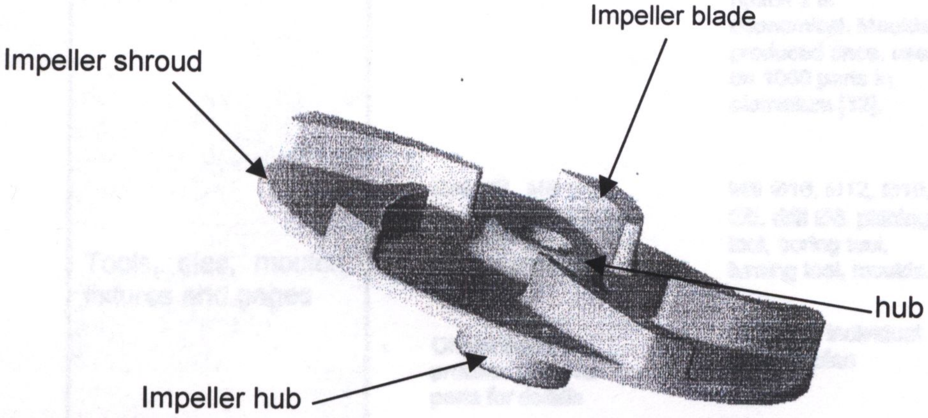


FIGURE 4.5: The Impeller

TABLE 4.6 Decision Variables used in Process Planning

Decision Variable	Component (Part)		
	Bearing Housing	Impeller	Diffuser Casing
Processes & sequence	Option 1: Cut blank from steel bar, rough-turn, mill faces on CNC to make moulds for PMC, use moulds to cast, clean casting, drill, tap, assemble. Option 2: Cutting blank from bar, turning all, drilling, tapping, Mill finish bearing seat (All-cylindrical &symmetrical) on CNC machine, assemble. Option 3: Sand cast then machine	Option 1: Cut blank from bar, turn cylindrical portions, mill to shape on CNC Machine, bore for shaft, cut key-way, assemble. Option 2: Cut steel blank from bar, turn, mill pockets to make moulds for PMC on CNC machine, use moulds to cast, clean casting, boring for shaft cut key-ways, assemble.	Option 1: Cut blank form Al, rough-turn, Mill on CNC machine (one face), drill (centre and peck), tap, Mill on CNC (other side), bore suction hole, drill stands, tap, assemble. Option 2:Cut steel blank, Rough-turn (symmetrical portions), Mill on CNC machine in one (1) setting for each half of mould, Cast using moulds, clean casting, drill, tap, assemble.
Equipment Selection	- All processes are carried out in the workshop, no new investment. - Option 2 therefore economical.	- No new investment required, same machinery. - Due to geometrical conditions and accuracy reasons,	- Same, available machinery used; no investment in new machines necessary. - Option 2 preferred to 1 for geometry as

		option 2 is economical. Moulds produced once, used on 1000 parts in aluminium [12].	well as tolerance accumulation reasons.
Tools, dies, moulds, fixtures and gages	- Drill Ø8, Mill Ø10, turning tool, cutting saw, tap M6, no special fixtures involved. - Check individual process plans for parts for details	Mill Ø16, Ø12, Ø10, Ø5, drill Ø8, planing tool, boring tool, turning tool, moulds. - Details in individual process plan	Mill Ø16, Ø12, Ø10. drill Ø8, Ø12, turning tool, moulds. Details in individual process sheets.
Cutting tools and cutting conditions for machining operations	- Coolant (flood or mist) required on conventional machines, flood required for continuous milling operations on CNC machines - Preferably 5% soluble. 10% soluble optional.	- Inevitably flood coolant for steel turning on lathe, flood coolant with lower feed rates (reasonably below industrial) [11] and speeds. Compensation in control is a must for multiple operations using same tools 5% soluble coolant	Must be flood type coolant for steel turning as well as on CNC machine at safe loads on tools. - 5% soluble coolant
Methods	Milling, turning, drilling, tapping. Mostly man co-ordinated.	Casting, turning, drilling, boring, broaching. Mostly man co-ordinated.	Casting, turning, drilling, boring. Mostly man co-ordinated.
Work standards	Work measurement techniques not automated.	CNC tool path generation, simulated to give time estimates	CNC tool path generation, simulated to give time estimates
Estimating production costs	See process individual plans	See process individual plans	See process individual plans
Material handling	98% Manual	96% Manual	96% Manual
Plant layout and facility design	No alterations in this project	No alterations in this project	No alterations in this project

4.9 Tools used in CNC Programming

After all the design on Mastercam was complete and tool paths programmed from a defined tool in question, the information was transformed into a form useful to the CNC controller. To achieve this, the post-processor was run from which an NC program results. But before this an intermediate file, the NCI file was generated before post-processing. This is the name under which the tool paths are stored while modifications and/or changes and additions are being made to the operations.

4.9.1 The Tool Library and its Creation

To do anything further than the design, the available tools must be known. A tool library was created based on the available tools. Such a practice offers a number of advantages such as

- lowered the chance of using wrong tools,
- Information such as feed, maximum depth of cut, etc was entered beforehand for a known material reference,
- tool designation, such as T1, Ø10 mm, Drill were easy to follow up, as these were the same definitions to use when mounting tools on the CNC machine, and
- one has a choice when programming tool paths whether to calculate feeds and speeds from the material or the tool.

When a tool library is made and saved, in ***Mastercam*** 7, a suffix *tl7* is put against the name. For example if the library is APN, it will be saved as APN.tl7. The following tool library was created for the work in this project.

TABLE 4.7: Tool Library for Mastercam Mill – "tueunza.tl7"

Tool Number	Type	Ø, mm	Nose radius, mm	Length, mm	Cutting length, mm	No. of flutes
T1	Probe	10	5	168	-	-
T2	Flat Mill	3	-	20	8	2
T3	Flat Mill	5	-	23	8	2
T4	Flat Mill	7	-	28	10	2
T5	Flat Mill	12	-	39	30	2
T6	Flat Mill	16	-	45	34	2
T7	Bull Mill	10	5	55	30	2
T8	Centre Drill	8	-	50	4	2
T9	Drill	5	-	62	55	2
T10	Shell	63	-	70	40	8

To create a tool library, in the main menu select "NC-Utils" □ Library. If creating a new library, then the system prompts for a name. However, if the name already exists and the library simply needs updating, make changes and "Save", The tool library for this project was given the name "tueunza" and an extension "tl7" was automatically added to have " tueunza.tl7". This library can either be filtered or not filtered. The filtering option optimises the structure of existing NCI files This function removes collinear points from a tool path that lie within a specified error tolerance and replaces the points with lines and arcs. A filtered tool path can usually be machined faster than an unfiltered tool path due to lower processing time. The tolerance must be specified.

TABLE 4.8: Results from Titex Electronic Catalogue

Tool Number	Spindle Speed, rpm	Feed rate, mm/min	Plunge Rate, mm/min	Axial depth of cut or max. depth of cut, mm	Radial depth of cut or maximum width, mm
T1	-	-	-	-	-
T2	2, 800	88	35	1	2
T3	1, 910	85	34	1.5	3.5
T4	1,364	88	35	2.3	5
T5	1,100	103	41	3.9	8.4
T6	900	90	43	5	11.2
T 7	1, 100	131	52	-	-
T8	1, 393	-	279	-	-
T9	2, 300	-	253	-	-
T10	680	10	-	20	8

The following formulae [6] apply for tool axial and radial depths (also listed in Table 48)

1. Plunge Rate (depending on tool geometry, only mills and drills, for axial feed),

$$\text{Plunge Rate} = (1/n)^{*}\ \text{Feed Rate} \qquad (4.5)$$

Where n is the number of teeth (flutes) a tool has. Most tools have 2 flutes; thus the plunge rate is half the feed rate.

2. Maximum axial depth of cut (max. depth of cut)

$$\text{axial depth of cut} = 0.33.d \qquad (4.6)$$

Where, d is the milling cutting-tool diameter.

3. Maximum radial depth of cut (max. width of cut)

$$\text{radial depth of cut} = 0.7\ .d \qquad (4.7)$$

in which d is the milling cutting-tool diameter. Normally lower values are used, typically, 0.6-0.65 (i.e. 60-65% of tool diameter).

Normally, whenever adjustments are made, the tool life is checked. This is compared with the simulated duration of operation under consideration to make sure there is no tool failure in the process of machining. Otherwise, spindle speeds have to be edited accordingly to meet the requirements.

4.10 Casting Considerations

4.10.1 Permanent Mould Casting (PMC)

Liquid metal is allowed to enter the mould by gravity or under pressure; the earlier will be applied in this project. This type of casting is particularly suitable for high volume production of small, simple castings that have fairly uniform wall thickness and no undercuts or intricate internal coring. Production quantities should be high enough to justify the cost of moulds. The following were considered in this project:

4.10.1.1 Physical Shape

Thickness

For the sake of heat transfer from molten metal to the permanent mould and the need to maintain temperature gradient for progressive solidification of the casting, a relatively uniform mould thickness throughout was taken. Wall thickness was generally 25 60 mm depending on

- mould size - Permanent mould casting generally has sizes from 0.5 kg minimum to 300 kg with typical surface finishes of 2-3µm. For the impeller mould, diameter 250 mm, the minimum wall thickness was 22 mm and the maximum was 45 mm. Minimum values were applied where there were cavities near the surfaces. These were regions such as the impeller vane pocket with maximum thickness on the sidewalls.
- Thickness of sections of the casting -minimum thickness of sections is as low as 2.0 mm to as high as 50.0 mm. The value used for all other parts of the moulds was 20 mm.

However, a majority of the wall thickness varied between 30.0-45.0 mm especially on parts such as the riser height.

Size

The sizes of the moulds for the impeller and the casing were respectively 250 mm and 450 mm diameter 50-100 mm on mould sides and somewhat less at the bottom since there are normally no gates into the bottom. For space at the top, gates and risers and the head of the molten metal were allowed for. Since the only pressure used to get molten metal into the cavity was gravity, it was important to have molten metal in the gates sufficiently higher than in the cavity to create this head.

Molten metal in gates > in cavity

Difference: 120 mm absolute.

4.10.1.2 Shrinkage Allowance (Aluminium and Steel considered)

The moulds were designed slightly larger than actual dimensions as follows [6]:

- 0.007in/in (as quoted in [6]) for the cold mould to a cold casting was allowed assuming no retardation in shrinkage by restriction in mould shape,
- Cores or other mould configurations reduce actual casting shrinkage to as little as 0.004in/in (as quoted in [6]), this amount of shrinkage occurs after the casting is removed from the mould. This project's mould design had no cores.
- A rule that sometimes applied was that dimensional tolerances should be approximately half the maximum shrinkage allowable for the particular type of metal. This rule did not hold in these large and complex castings made with such accuracy that no machining or finishing outside the foundry was required; other than just removing gates and risers. Generally, 5/32 in contraction per ft of casting for aluminium alloys was allowed for. For Steel, 5/16 is provided. This project took the average of the two metals as one was for the product and the other for the mould. This gave 15/64 in per ft (or 0.0197 mm/mm).

4.10.2 Machine Finish Allowance

Allowance for machine finish was dependent on'

(a) type of metal used, steel
(b) design of the casting
(c) size
(d) tendency to warp, and
(e) machining method.

Typical Machine Finish Allowance (in addition to shrinkage allowance) used in this project were as tabulated below:

TABLE 4.9: Casting Dimension against expected tolerances 16]

Dimension of Casting, in.	*Expected tolerances for "As Cast" Dimension, in.*
Up to 8	± 1/16
Up to 14	± 3/32
Up to 18	± 1/8
Up to 24	± 5/32
Up to 30	± 3/16
Up to 36	± ¼

Allowances for machining cylindrical parts of single bores in this project were:

Diameter, in	*Allowances, in*
4	0.12-0.2
4-8	0.12-0.24
8-12	0.2-0.34
12-20	0.25-0.40

4.10.3 Gating System

Molten metal should enter from sides:

- one side,
- greater lengths of runner system hamper high yield. Thus the runner systems were made as short as possible with the longest being 120 mm, in the diffuser casing. This was also undertaken to eliminate superheating.

Size Relationships

- Runner (down-sprue) was made large enough to
    - (a) allow flow
    - (b) fill cavity as required

- Feeder (or shrink bob) was 1 5 times the section of the casting that was being fed.
- To allow the feeder to feed the casting, each section of the in-gate was somewhere in between 70-90% of the section of the casting being fed. This was the most important consideration for design of the mould top.

4.10.4 Other Considerations

Fillet all Sharp edges

Fillets had two functional purposes

(i) reduce stress concentration in the casting in service

(ii) prevent cracks, tears and draws at re-entry angles it is recommended to make corners more mouldable.

To fulfil engineering stress requirements and reduce stress concentration, relatively large fillets were used [16]

$$R = T/3 \text{ or } T/2$$

where

R. Radius of fillet

T. Thickness of casting

Avoiding Abrupt Section Changes

The difference in the thickness of adjoining sections was a minimum and not exceeding a ratio of 2: 1. Where a greater difference was unavoidable, design with detachable parts such as the pouring cavity can be bolted.

Where the change of thickness was less than 2:1, it took the form of a fillet; where the difference was greater, the recommended form was a wedge. When the thickness of flanges differed from that of the body of the casting, the change in thickness was gradual and tapered 1:4.5.

Bosses, Lugs and Pads were not used unless absolutely necessary

Bosses and pads increase metal thickness, create hot spots and cause open grain or shrinks. Such parts were blended into casting by tapering or flattening the fillets, Bosses were not included in casting design when support for bolts could be obtained by milling or countersinking.

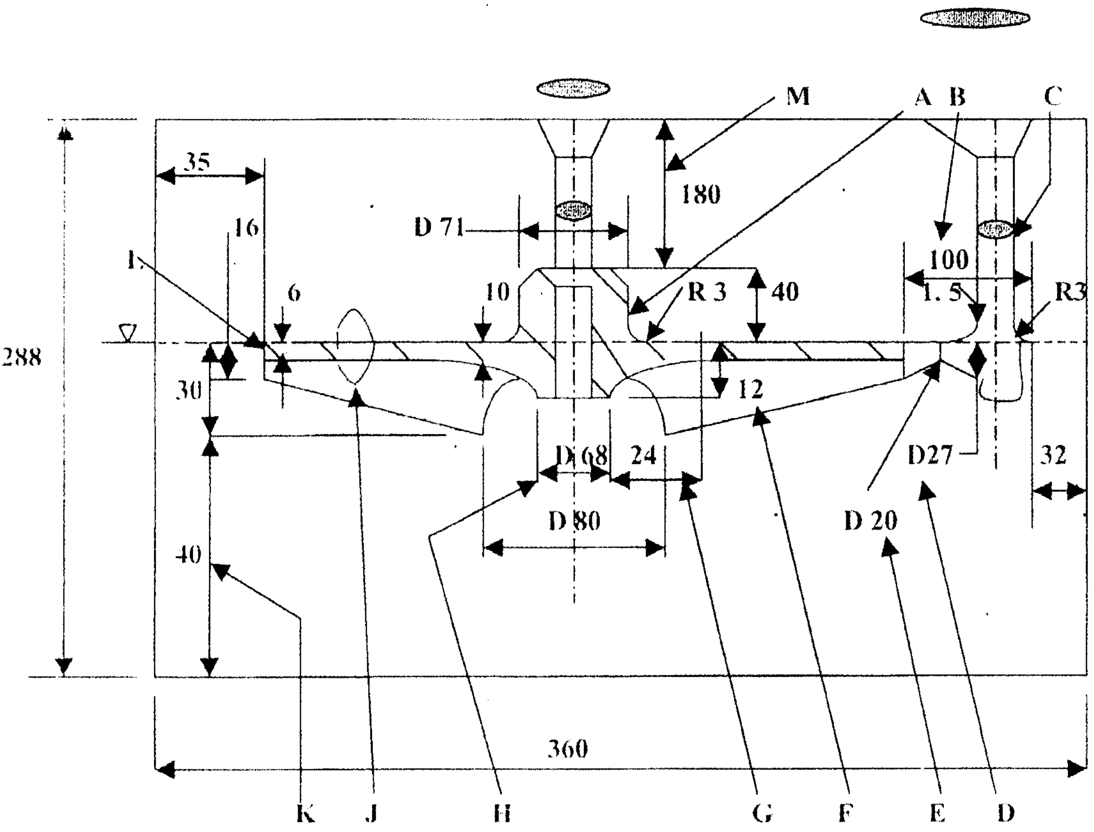
Thickness of bosses and pads was preferably made less than the thickness of the casting sections they adjoined, but thick enough to permit machining without touching the casting wall. Where the casting section was light and did not permit use of this rule, then the following minimum recommended heights served as a guide.

Approximate Casting [16]

Length (ft)	Boss height (in)
≤ 1.5	0.25
1.5 – 6.0	0.75
> 6.0	1.0

4.10.5 The Mould Design for Impeller

The following were the dimensions arrived at for the mould used in the production of the open impeller They were based Oil theoretical as well as practical analyses, Figure 4.6 shows a section of the impeller mould.



Mould Width = 35 + 35 + 190 = **260 mm**

Figure 4.6: Impeller Mould Design

Justification of Critical Dimensions

In the sketch, letters were used to label critical mould dimensions for product quality soundness. The following were the identified points:

A: In the actual design of the mould an additional 1.6 mm machining allowance to the diameter was added,

B: The total length was not more than 2/3 to 5 times a section thickness; thus, it was 20 mm*5 = 100 mm,

C: This was neither very wide nor very narrow because they should be filled adequately every time with molten metal during pouring, D: Each section of feeder system was = 70 -90% of the casting being fed. Generally, 90% was taken. For example, 90% of 30 mm = 27 mm (30 mm was the total impeller depth from the parting line)

E: The smallest section of feeder = 2*(16 6) = 20.0 mm

F: R = 2t = 2*6 = 12.0 mm

G: L = 4*(T -t) = 4*(12 6) = 24.0 mm

H: D = 0.9*75 = 67.5 = 68.0 mm -Hub and thickness proportions.

J: Concurs with casting rules on bosses. The rule gave 6.35 mm for a length of 190 mm, thus accurate enough.

K: Mould thickness at the bottom; majority is between 1.25 1.74 in. Thus, let the mould wall thickness = 40 mm,

L: Diametral shrinkage allowance for aluminium of diameter 190 mm is in the range 3mm to 6 mm. 4 mm was taken. This was included in the mould manufacture.

M: It was important to have molten metal in gates sufficiently higher than the cavity to create enough head: varying between an absolute minimum of 175 mm to as much as 180 mm. 180 mm was taken.

Chapter 5

Results and Discussion

5.1 Introduction

The pump design was altered to adapt its production to the available technology. This refers to the usual conventional machines such as lathe, milling machine and the CNC Machining Centre, vertical milling machine, in the University of Zambia's Mechanical Engineering Department. A number of considerations were made to ensure that the production processes were simplified and the number of operations reduced in an effort to reducing the overall product cost without compromising quality. A number of design options were considered at concept stage into detailed design stage. Product re-engineering was carried out through concurrent engineering during the development process. Not all parts were for manufacture; standard parts were included to enhance standardisation as well as avoiding the high costs of manufacturing such parts, which are cheaper to buy than manufacture. It is more economical to buy than manufacture most standards parts. Standardisation has also got the advantage of parts interchange ability, which adapts the product to global competition on the world market. The standard parts included the bearings, mechanical seals, power cables, bolts, nuts and washers, the motor stator and its rotor.

5.2 Results

Chapter 3 was dedicated to conceptual and detailed designs, as this was the basis of the production process development. The design was with respect to the performance characteristics of the original design from which, upon redesigning the same parameters applied to the new design Appendix C shows the major parts of the pump. The general tolerance of the parts was ±0.025 mm unless otherwise stated. The new design of the centrifugal, submersible pump is shown in Figure 3.2.

Parts designed for permanent mould casting included the impeller and the diffuser casing. The bearing housing was to be machined from a bar-stock of aluminium of diameter 220 mm. The method of manufacture for the bearing housing included cutting the bar stock, turning the cast to shape on conventional lathe and finally milling critical parts on the CNC machine. This was a more economical way to manufacture because of the amount of material to be removed. Thus, direct turning on a conventional lathe was preferred to milling a permanent mould, or milling a bar stock to finish the part on a CNC machine. The next process was to mill on the CNC Machine parts that were critical to the functioning of the whole product, that is, the bearing seat and the spigot. The motor casing, pump casing and the end-covers, were designed for sand casting as they were not critical to the efficiency of the pump compared to the bearing housing, diffuser casing and the impeller. The shaft was to be turned from steel EN8 followed by milling to cut the keyway. Chapters 3 and 4 give details of the major parts.

Redesign of the parts involved product re-engineering in which case the new and old designs were compared as the design progressed Detailed design followed, in which the parts were sized. This was a more iterative process in which the concepts and detailed designs were constantly compared to ensure the product accommodated all the parts, including standard parts. The Solid-Modelling software, SolidWorks, was utilised to test the designed parts for assembly. Mastercam, CAD-CAM software, was also utilised for the design of the parts. Then CAM was used to develop the manufacturing process tool paths. Mastercam formed the link between the computer and the CNC Machine's controller. The CAD-CAM considerations were outlined in Appendix A.

All the pump parts were designed, with the resulting number of parts reduced by 35% compared to the original design. The tooling for production was also developed along with the process for production. The development process was finally completed with permanent mould designs' for the impeller and diffuser casing. The other parts were dimensioned to a state in which the product would be possible to assemble, as verified using SolidWorks, The new design was thus cheaper and easier to produce than the original design. The final design was an optimised centrifugal pump design for pumping cold water up to a maximum temperature of 40ac and suspended solid sizes of up to 12 mm. For temperatures higher than this: the efficiency of the product would begin to drop. These design considerations were made with respect to the pump duty *at* its optimum speed in an effort to achieve the targeted efficiency.

5.3 Discussion

Work carried out encompassed a market survey on the most widely used rotodynamic pumps in Zambia. This product was redesigned to the same specifications (Head, Discharge, and Efficiency) but with a view to adapting the production process to the available technology. This meant application of product re-engineering through concurrent engineering. Tooling for the manufacture of the various parts was developed.

5.3.1 Comparison of Parts

The part numbers refer to Figures 3.1 and 3.2.

The Strainer (56)

This remained the same. It is a standard part for most submersible pumps. Theoretical considerations had the same merits.

The Lower Diffuser (58)

This was eliminated and the profile incorporated on the casing. This eased assembly and maintenance problems due to the reduced number of parts.

The Impeller (59)

The casing was designed to have a clearance from the impellers of up to 15% so that different types of impellers could be accommodated without having to change the casing. Clearances up to 15% between impeller and casing do not alter the overall pump efficiency substantially [14]. Different impellers, radial, mixed in both, low, medium as well as high head would be used without having to replace the Diffuser Casing.

Mechanical Seal (12, 12A)

Only one seal was necessary in this design. The carbon-tungsten seal 12 was eliminated and replaced with tungsten-tungsten for operating temperatures up to 150°C. The resulting number of seals was half the total in the original design.

The Diffuser casing (57, 58)

This design principle remained the same but its profile was changed to a constant flow design. A spigot was incorporated for precise location and to allow for a reduction of 0' ring seals.

Upper Diffuser (52)

The elimination of the upper mechanical seal (103) led to the design change eventually of this component. It was then redesigned to incorporate the bearing housing.

The Bearings (9A, 9B)

These remained unchanged. They were only resized for longer life of operation, that is, from 15000 to 18 000 hrs.

Lower Bearing Housing (53, 54)

This was eliminated due to the elimination of Sea112. The upper diffuser's uses were then increased to incorporate that of the bearing housing.

The Rotor (66)

This term in everyday engineering includes the motor core and its shaft. Normally, this is made up of a press-fit of the core and the shaft. In this project, the shaft was redesigned and resized to adapt to design change. This included spacing between locations for seals as well as the positions of the rotor. The span of the shaft also reduced.

The Motor casing (50)

This design was changed to incorporate circular flanges both on the lower and upper sections to simplify the design. The lower flange was then reduced in diameter to match the new bearing housing (52 and 54) so that securing was done together and not through the diffuser casing. This approach gave room for smaller diffuser casing design, as the earlier space taken up by Part 54 (lower bearing housing) was no longer included. The upper flange was adjusted to act as a barrier to prevent water from entering the motor from the upper end. The new design had a side discharge set-up instead of upper end, thus reducing the number of rubber seals. This does not affect performance but flexibility in installations, That is however dependent on the type of application. Therefore, just like the earlier design has disadvantages, this design can have as well. But for the contemplated application in this project, the design stands with greater advantages. The new design is shown in Figure 3.2.

Permanent moulds for the manufacture of the impeller and the diffuser casing were designed using Mastercam Design and Mill respectively. These were designed for CNC Milling. Parts designed for turning included the bearing housing and the pump shaft. The motor casing, pump casing, and end covers were designed for sand casting. These choices were made upon making a number of considerations some of which included adaptation of production to the available technology in the country, reduction in the product cost as well as standardisation of the product through incorporation of standard parts.

Most of the work involved the redesign, and development of the tooling for the production of the parts of the pump. However, the operations with Mastercam were more extensive than with SolidWorks as the earlier is a complete CAD-CAM package whereas the latter is only a CAD package. However, the two have differences, which form a complete CADCAM set-up when used in unison. The link between the two packages is the IGES graphics exchange (read-write) format for file transfers. A choice can be made whether to write and/or read surfaces or B-Reps (Boundary representations). A lot more time was taken on Computer Aided Manufacturing (CAM) with software known as the Virtual CNC Predator. This is a simulation program that checks Numeric Control (NC) files for validity in safe manufacture. It also ensures the tool library in use is the right one. In addition, it simulates the duration of the production process on the CNC machine. The most important feature of the Virtual Predator is that it simulates a process specific to the type or machine controller to be used for production This has a number of advantages as compared with the NSee-2000 in Mastercam. The reason is that the Nsee-2000 only simulates the NCI (intermediate) program that is not useful to the controller. The NCI is a step before post-processing in ASCII format, whereas NC is a post-processed binary file used by the controller. Therefore, it is more assuring to work with a program that simulates an NC rather than an NCI file because the post-processors are machine-specific. This is safer as it reduces on risks of accidents Safety on the CNC Machine as well as on precautions to undertake when linking the computer to the CNC Machine for DNC (Direct Numeric Control) was practiced.

Considerations to be made when machining to reduce on tool loading were also taken into account from Mastercam possibilities in machining options. The methods of finishing as well as roughing were critically considered, together with tool capabilities on given materials given therein Databases within Mastercam program were utilized in calculating feeds with respect to materials so that the determination of feeds and cutting speeds were determined automatically. On the other hand, a tool library was created to ensure every other production process used the same set of tools. The tool library involved the determination of the best safe feeds and cutting speeds for the various materials. The maximum axial as well as radial depth of cuts in slot and peripheral milling were included as well. Of great importance are the tool parameters such as the tool total length, cutting length, and diameter, number of flutes as well as the tool material. Tool life is very important as well. However, parameters such as feed rate, cutting speed and tool life depend on the manufacturers' specifications supplied with the tools at various cutting speeds. These parameters can be utilised for the determination of safe operations as regards tool safety. In this development process, TITEX Electronic Catalogue was used to develop a good tool library. The selection was made based on material type. For the permanent moulds, mild steel was the choice as such most of the operations were considered for an average between mild and stainless steel strengths for safety. The steps in production of the parts were selected in a way to minimise tool barrier as well as optimising the cutting order while taking the right tool for the process. These choices were made while considering safe feeds and depths of cut. Surface texture was also as important to the step-over distance during machining.

Chapter 6

Conclusion and Recommendations for Future Work

6.1 Conclusion

This project's objectives were achieved. It was seen that a design of a product could be modified in a number of ways, so long as its purpose is not ignored. The development of the process for manufacture of the new design was successful. Tooling design for the manufacture of the pump's impeller and diffuser casing moulds was completed including design of all the parts. Standard parts were also incorporated into this design, so it does not lose track of the need for standardisation. The resulting design was cheaper and had fewer parts. It is also seen that it is possible to design a product that is adaptable to available technology. This gives the advantage of eventual global competition as well as diversification of design to enhance favourable competition in the market niche. By this approach, advanced technologies in this ever-advancing world can easily be incorporated into any other types of products. It was also observed that with the CNC Technology, very complex parts could be manufactured accurately and within a good period of time and to high quality standards. This was evident from such complex parts as the impeller. The most important thing was the method selected for production to ensure that the part was produced as economically as is possible. That was the reason why permanent mould casting was selected for the manufacture of the impeller and the diffuser casing. However, some parts required a combination of both the conventional as well as the CNC technology as outlined in Chapter 4 for the bearing housing.

It was also observed that the use of Mastercam, in itself, for simulation of processes is not enough, but with an NC file simulator, as that is machine controller specific. There are a number of controllers, such as the FANUC (Japan) and MAHO (German), which call for a completely different set of starting blocks in the NC file to actuate the controller correctly. At the same time, within these types of controllers are those such that unique starting command blocks are required for the machine to operate correctly. An example is the vertical-milling machine running on a FANUC Controller. For example two types of machines by different manufacturers such as the FADAL (Made in America but with FANUC Controller) and SUPERMAX (Taiwan and assembled in the Netherlands with a FANUC Controller) machines have different starting and ending blocks in their programs. It was also understood that machining operations on associative tool paths (contours, pockets, drilling, tapping) are more risky than non-associative tool paths (surface rough, finish) in the z-movements. Thus, a lot of care must be taken when setting the work-piece to zero on the machine, especially with the fixture offset code allocations.

6.2 Recommendations for Future Work

1. CAD-CAM software has to be studied extensively in order to make full and correct use of it in manufacturing. It should be understood that Mastercam couldn't be used for complex shapes successfully due the complex mathematical problems in irregular geometry. Thus SolidWorks should be used and the need to use SolidWorks in place of Mastercam should be identified. The Virtual Predator should also be utilised before the parts to manufacture are certified fit for DNC.
2. All the moulds for the parts should be made from the recommended materials to ensure the right results from the design specifications.

3. The product design should be optimised through additional design changes to functional parts together with the comparison of costs resulting from these alterations. The pump cost should be evaluated with a hypothetical analysis of full-time production.
4. Carry out a survey of all the developed processes to give optimal results for manufacture using project evaluation and review techniques. This should include all activities associated with the complete production of the pump.
5. Assume a plant using the processes in step 4 above to design a simple plant layout that should be fully optimised for minimum costs of production. It is realised that this exercise of pump design for a targeted market cannot hope to succeed in its task if it is developed in isolation from what actually does happen in practice.

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Appendix A

A.1 The Computer Numerical Control (CNC) and Numerical Control (NC)

Numerical Control (NC) is a form of programmable automation in which the mechanical actions of a piece of equipment are controlled by a program containing coded alphanumeric data [6]. Computer Numerical Control (CNC) is the process of manufacturing machined parts in a production environment, as controlled and allocated by the computerised controller that uses motors to drive each axis. The controller controls the direction, speed and length of time each motor rotates. A programmed path is loaded into the machine's computer by the operator and then executed. This technology is one of manufacturing's major developments in techniques and the achievement of higher production levels. It has also helped to increase product quality and stabilise manufacturing costs. The goal behind the development of NC was to realise the following objectives [7]:

- Increase production,
- Improve the quality and accuracy of manufactured parts,
- Stabilise manufacturing costs, and
- Manufacture complex or otherwise impossible jobs.

Direct- and Distributed Numerical Control (DNC) describe communication to a machine tool from a remote computer. In Direct Numerical Control, part program instruction blocks are communicated to a machine tool as required and as fast as the machine can accept them. This rate is fixed by what is called the '***baud rate***'. This is the most popular communication method because it increases the length of programs hence allowing for more complicated designs. This was the method used in the project for larger parts such as the impeller complete NC program. In the Distributed Numerical Control method, the whole program or multiple programs are communicated to a CNC Machine tool or several CNC Machine tools. This requires increased memory capacity of CNC controllers. It is this kind of communication that the CNC Machining Centre (*Supermax* with a *Fanuc* Controller) in the University of Zambia employs. This method was only used for smaller programs such as only a selection of a huge program of the diffuser casing loaded in parts on the CNC machine until the product is complete. Machining centres, which take the place of half a dozen machines, are now capable of many operations – including milling, boring, drilling, facing, slotting, counter-boring, threading and tapping – all in one set-up. In addition, CNC is now a well-established technology and CNC machines have become far more commonplace.

A.2 Mastercam, SolidWorks and Virtual NC Predator

A.2.1 Mastercam

This is a CAD-CAM package that helps provide the CNC programs with a valuable productivity tool for both the generations of CNC part programs and process planning. It helps reduce the time it takes to generate accurate machine-ready NC programs. Mastercam CAD/CAM solutions are provided for 2-D, 21/2-D and 3-D modelling, including two-through 5-axis milling, 2- and 4-axis wire EDM, sheet metal punching and unfolding, plasma cutting and lasers.

Standard CAD features in Mastercam include: standard geometry and surface creation. The easy-to-use CAD system allows for the creation of the following entities in 2-D or 3-D: points, lines, arcs, fillets, splines, ellipses, rectangles, chamfers and letters, as well as surfaces such as Loft, Coons, Ruled, Revolved, Swept, Draft and Trimmed. Also included are information exchange systems: IGES, DXF, CADL and ASCII bi-directional data converters that allow communication with other CAD Software. Other features in Mastercam include:

- Dimensioning in any plane or view,
- Cross hatching,
- Multiple view ports,
- Dynamic rotation, panning and zooming, and
- Plotting capabilities.

CAM features in Mastercam include the following:

- Graphical tool path editing with full tool path simulation,
- Built-in tool libraries and materials files,
- Canned cycle support,
- Links to third-party applications,
- Surface machining,
- Drilling,
- Pocketing, and
- Cycle time estimation.

CAD and CAM together create a direct link between product design and manufacturing. The CAD system is used to develop a geometric model of the part. This model then is used by the CAM system to generate part programs for CNC machine tools. The computer is the common element in both procedures. Both the CAD and CAM functions may be performed either by the same system or by separate systems located in different rooms or even in different countries. The network between engineering, design and manufacturing computers becomes the critical information highway that ties the CAD and CAM functions together.

A.2.2 SolidWorks '98

This is a solid-modelling tool, which is, in addition to design, used to analyse properties of designed parts. This package was very handy in the analysis of assemblies with specified tolerances. It utilises the principle of collision detection. Unlike Mastercam, SolidWorks uses secondary operations such as extruding a cut, boss, revolve and sweep to generate solid parts. *Boolean operations make SolidWorks handy in complex operations such as filleting* and trimming of complex geometry. This makes SolidWorks more user-friendly than *Mastercam. However, this is only CAD Software and does not include CAM.*

A.2.3 Virtual NC Predator

This package was used to simulate an NC program useful to the CNC machine controller. It simulates the processes with respect to the type of controller for a given CNC machine to ensure the right ASCII code is generated for a given controller. It also ensures that the correct tool library is used for a job in question. This gave details of possible accidents with respect to tool collisions, failures and fixture offsets. It also simulates the actual expected duration of production of a given part, which makes it easy to estimate the cost of labour, giving the chance to optimise processes. It was found to be better than the Mastercam's Nsee200 NCI simulation as Vritual NC checks the file useful to the controller as opposed to the Nsee2000 which only simulates an intermediate file. Only the NC and not NCI file is useful to the CNC machine's controller.

A.2.4 The Link between Mastercam and SolidWorks

Mastercam was a very good tool for geometry and surface creation. SolidWorks was a typical solid modeller. CAD drawings made in any of these, including other CAD Software such as AutoCAD, can be written to or read from the other. This rich exchange capability enabled CAD drawings to be useful to any other CAD software. The exchange between Mastercam and SolidWorks was possible through IGES graphics exchange format, which is one of the bi-directional data converters.

Mastercam, being a CAD-CAD software, was the mostly used for designs to manufacturing. However, there are operations in Mastercam that were difficult to make than in SolidWorks. These operations included trimming and filleting complex surfaces as well as blending. In this case the design was written to SolidWorks for these complex operations that were easily handled in SolidWorks after which it was written back to Mastercam for CAM. It was however noticed that Mastercam had to be configured for SolidWorks as well as SolidWorks for Mastercam to ensure a successful exchange of data. It is difficult to write to SolidWorks from Mastercam unless Mastercam is configured to handle a greater number of points per spline, for complete information. due to precision differences between the software. SolidWorks has a precision of 10^{-11} whereas Mastercam, 10^{-17}.

A.2.5 The TITEX Catalogue

Mastercam Mill requires parameters such as feed rate, speed, table feed percentage and many other primary as well as secondary machining parameters. The other parameters are the lay on the surface that is to be machined. In such a case, the "Finish Parameters" have to be specified appropriately. These include parameters such as cut tolerance, cutting methods (zigzag, constant overlap spiral etc) and number of passes to mention but a few. However, all these parameters can be input best if the designer has good information about the available tools, such as those in Table 4.8. It is recommended that catalogues are used to check details of the tools, such as the flute lengths, overall length of the tool and so on. This provides a guideline when programming the tools for operations. For example if the tool length is smaller than the pocket depth to be machined, then the tool holder will collide with the stock. Many more problems would arise with insufficient details about a tool, which would lead, to failures and unexplained accidents for those that do not appreciate tool detail description. Thus to facilitate a good program, the detailed information about the available tools should be given during the creation of the library. Catalogues are often supplied with the tool manufacturers when purchasing the tools. This catalogue can either be an electronic book or hardcopy from the manufacturers. In this project an electronic catalogue, known a Titex Catalog, was used for that purpose. Feeds, speeds, tool life were made. A number of parameters including tool effective cutting length were utilised to get the primary and secondary cutting parameters. This information was used to create the tool library. For example, the following were some parameters obtained from the catalogue for the tools listed in Table 4.8. Note that Table 4.8 was based on

- *Main material group:* Steel and Cast Steel.
- *Material Subgroup:* Steel and Cast Steel from 700 N/mm^2 up to 1000 N/mm^2.
- *Coolant:* Soluble Oil 5%.

Appendix B

B 1. Other Parts For the Pump

Introduction

The parts for the pump were as indicated in Chapter 4 of this report. The bearing housing, diffuser casing and the impeller were explained more extensively through the sequel of the report. The following were the parts for the project:

Motor Casing - Isometric View

The Bearing Housing Section

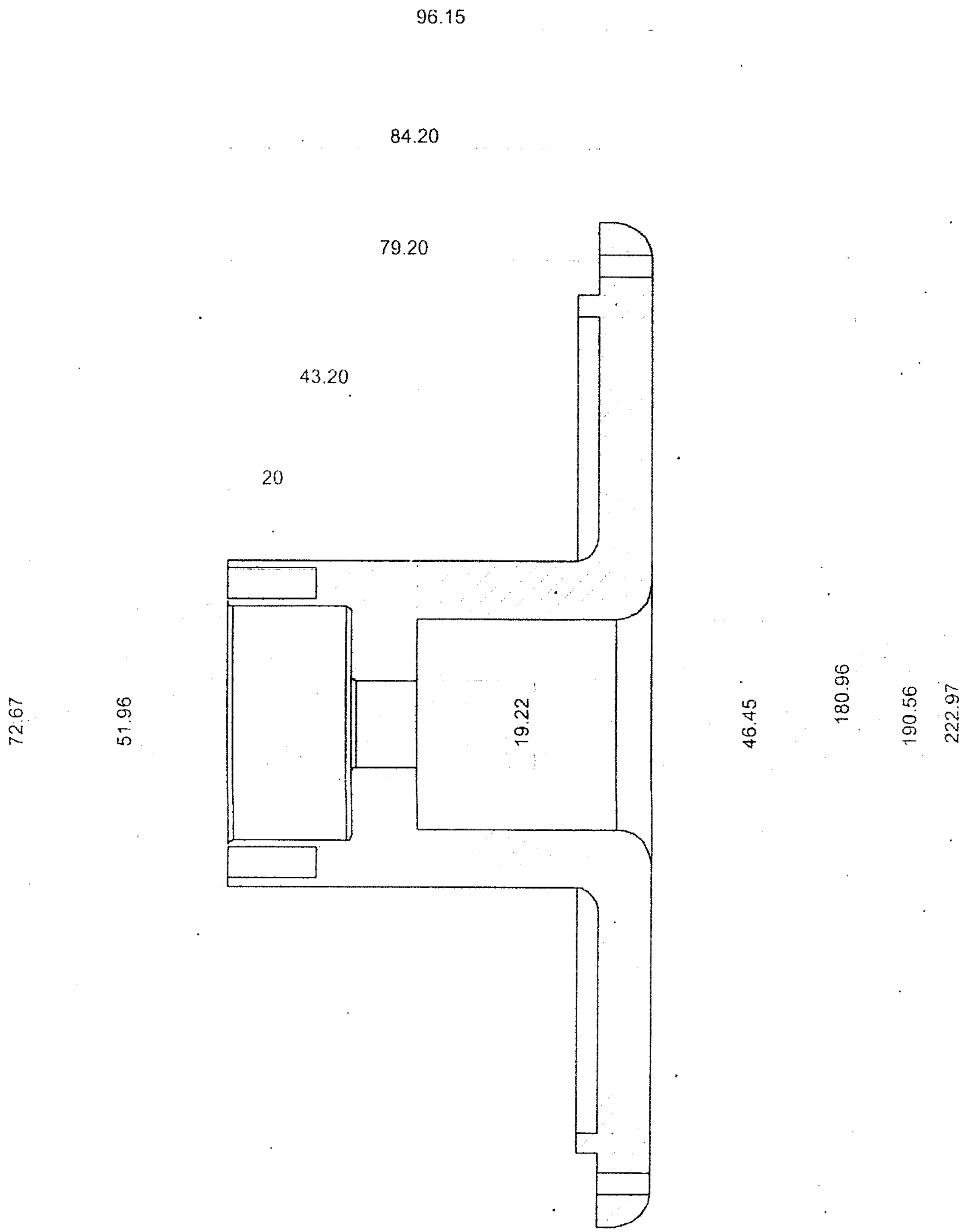


Figure B1: The Bearing Housing Section (*All dimensions in mm, All tolerances ± 0.12 mm*)

Figure B2: The Pump Shaft

(All dimensions in mm, all tolerances ±0.05mm)

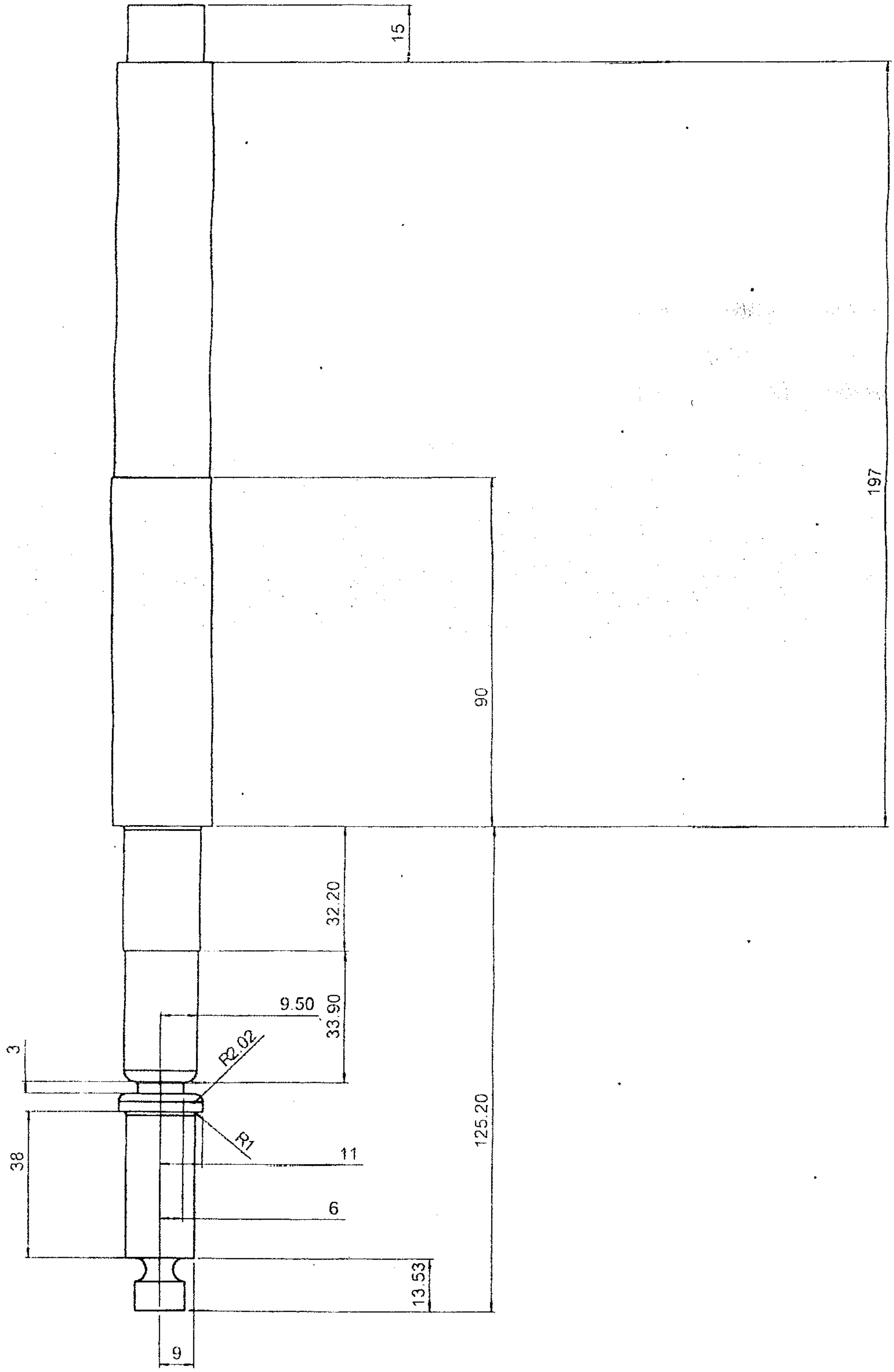
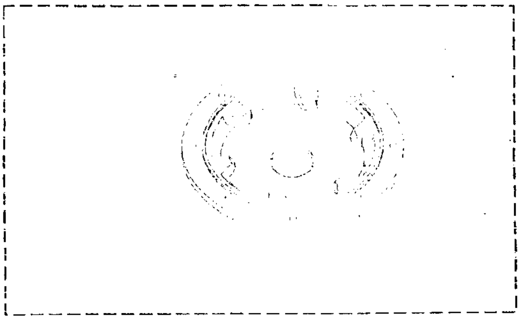
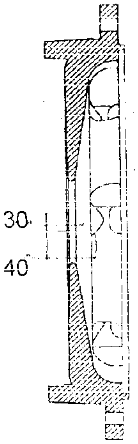


Figure B3: The Diffuser Casing

(All dimensions in mm, all tolerances ±0.06)



Isometric View



Section A-A

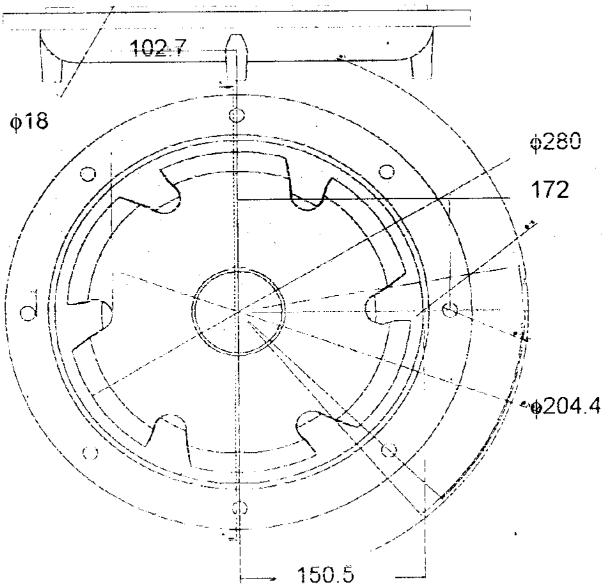


Figure B4: The Pump Casing

(All dimensions in mm, Tolerances ±0.12 mm)

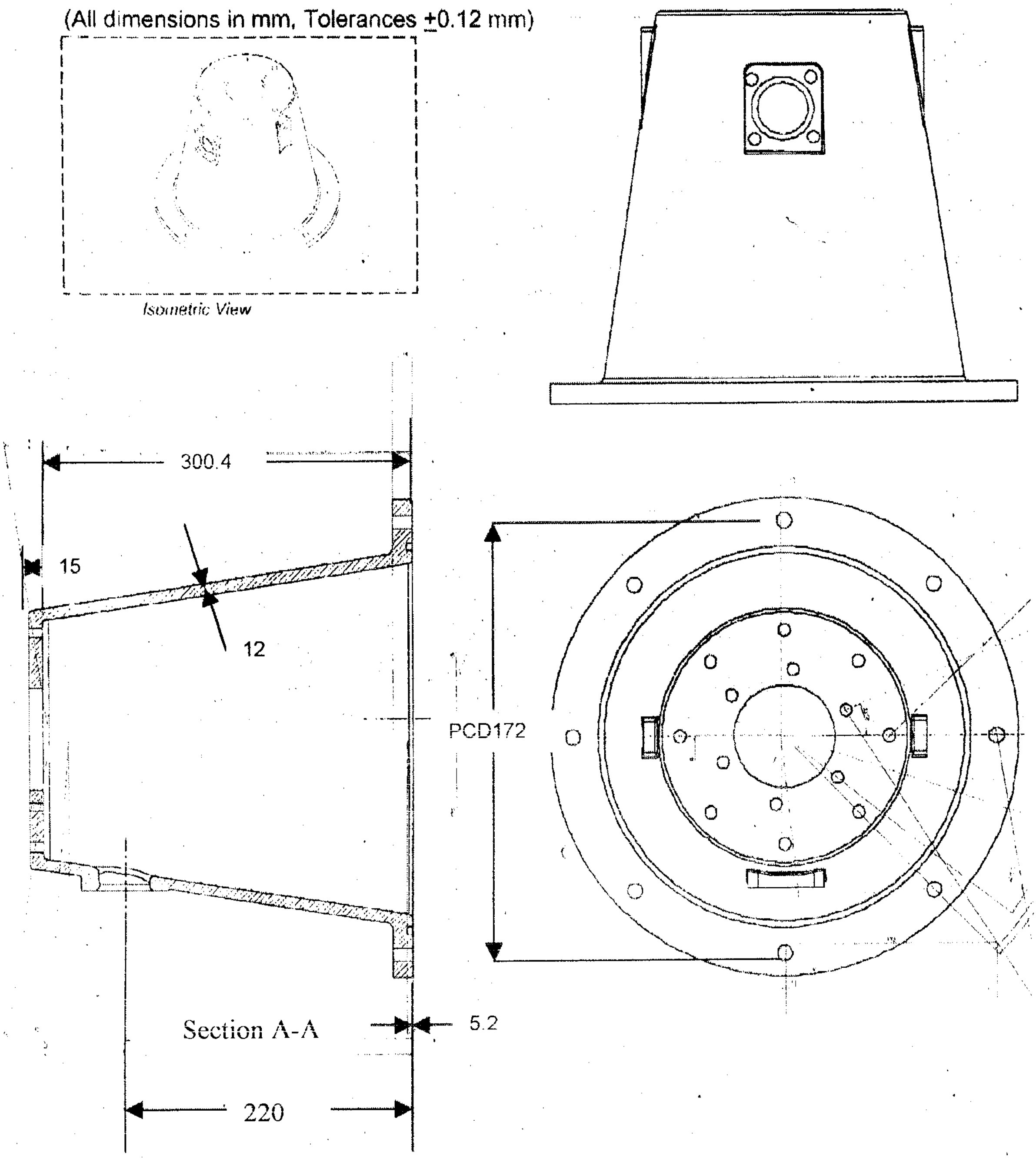
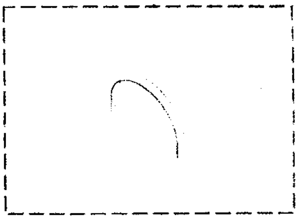


Figure B5: The Lifting Lag

(All dimensions in mm, All tolerances ±0.1mm)



Isometric View

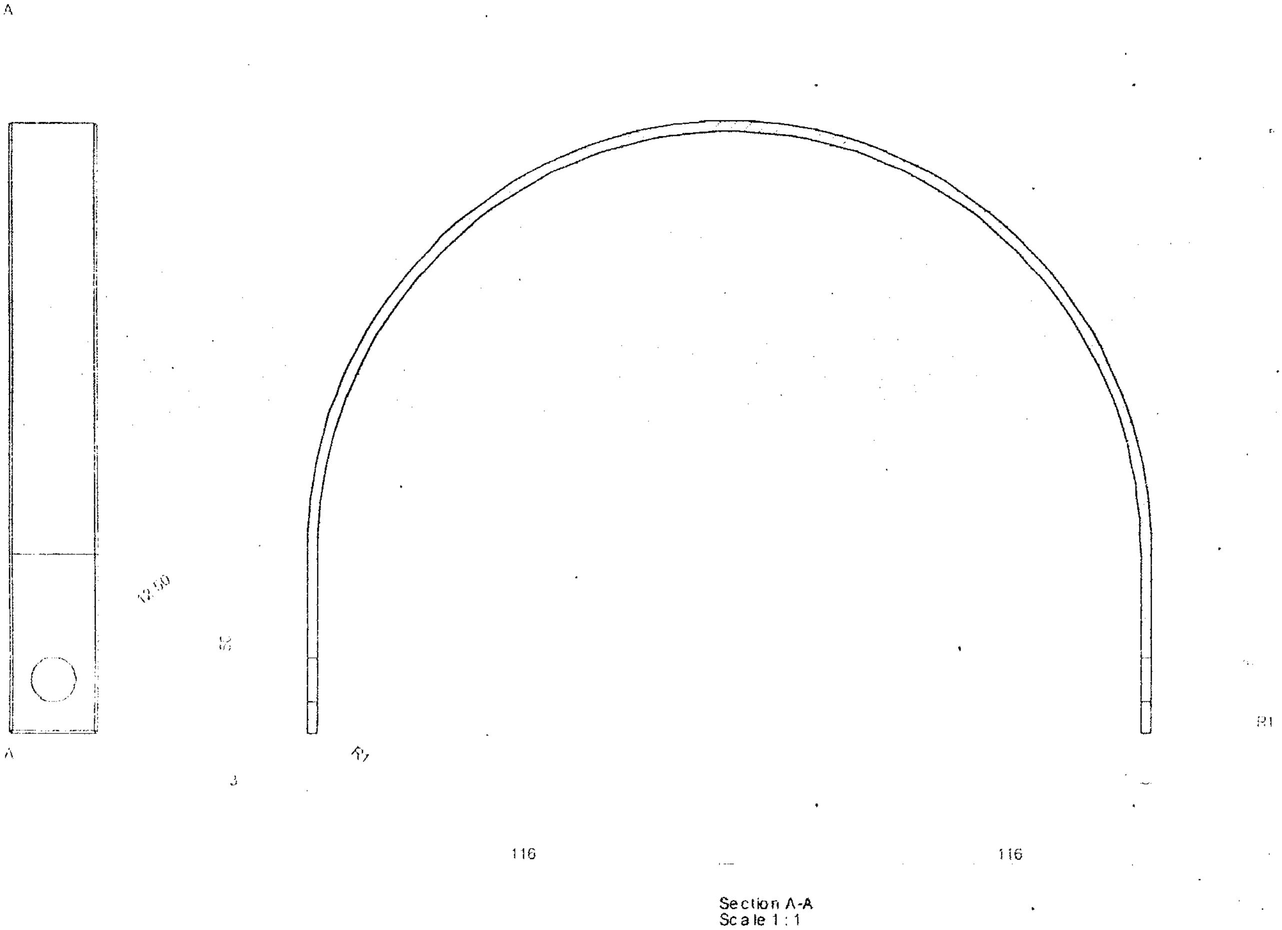
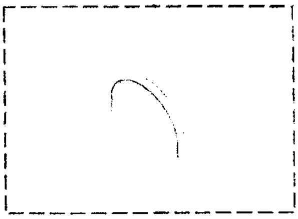


Figure B5: The Lifting Lag

(All dimensions in mm, All tolerances ±0.1mm)



Isometric View

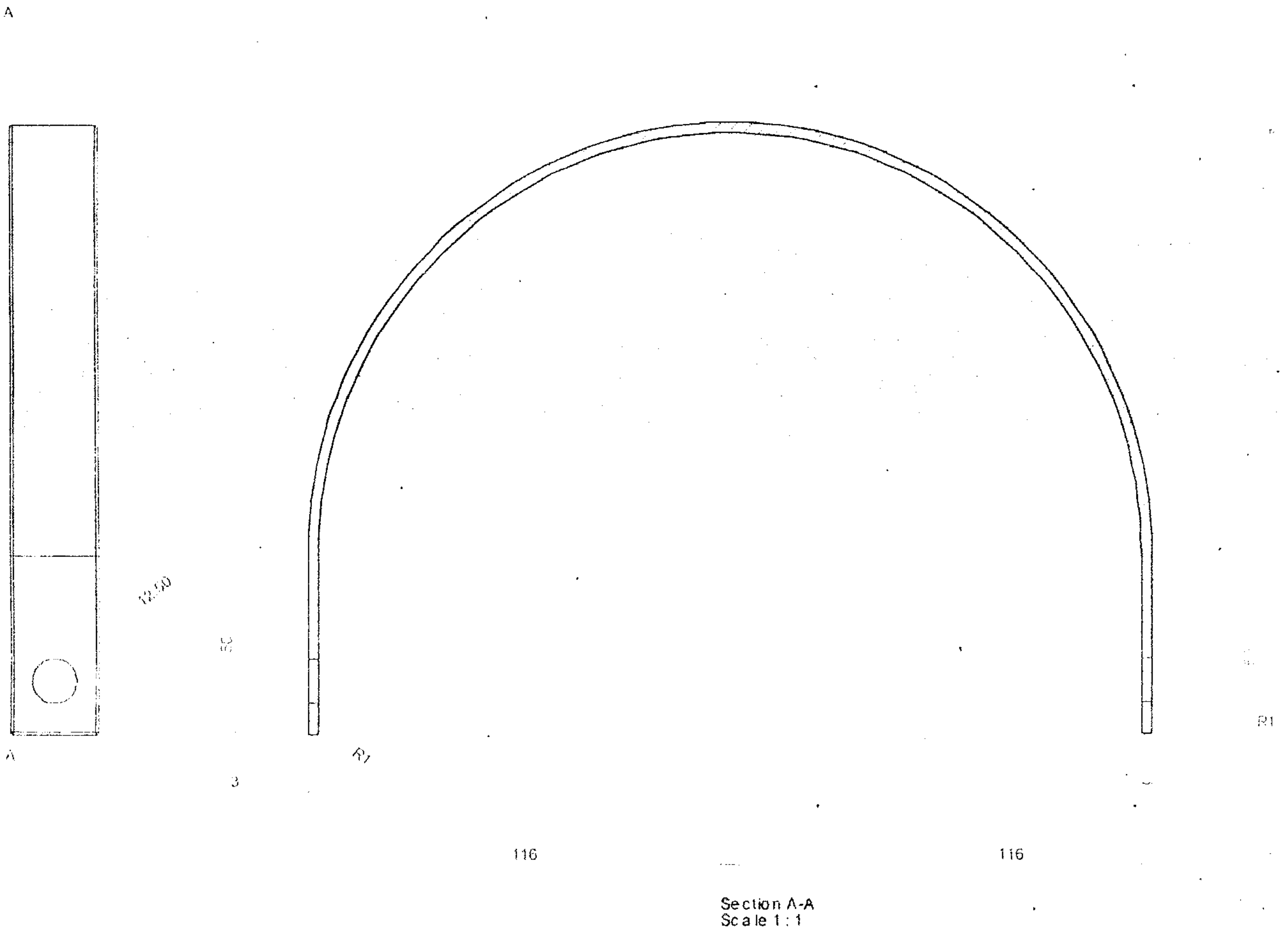
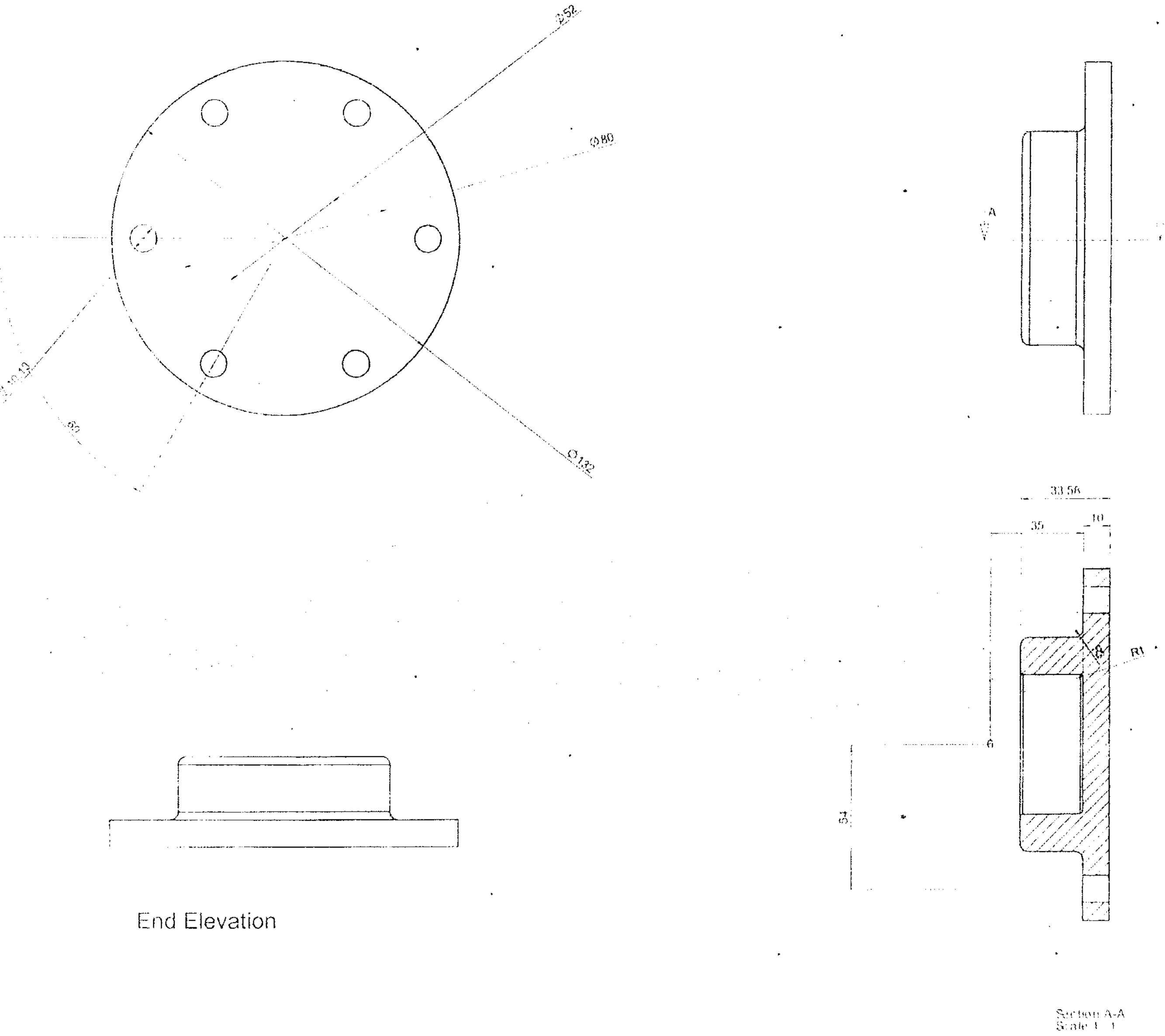


Figure B6: The End-Cover

(All dimensions in mm, All tolerances ± 0.08 mm)



Appendix C

The following is an edited version of the impeller NC Program. It only includes vanes machining in edited form for both roughing and finishing.

```
O9993 ( VANE )
(DATE : 14 10 99         TIME : 08:49 )
(TOOL=06  LENGTH OFF.=06   DIA. OFF.=56   DIA.=16.000  TOOL NAME= FLAT
ENDMILL )
(TOOL=05  LENGTH OFF.=05   DIA. OFF.=55   DIA.=12.000  TOOL NAME= FLAT
ENDMILL )
(TOOL=03  LENGTH OFF.=03   DIA. OFF.=53   DIA.=05.000  TOOL NAME= FLAT
ENDMILL )
(TOOL=02  LENGTH OFF.=02   DIA. OFF.=52   DIA.=05.000  TOOL NAME= SPHERE )
(TOOL=07   LENGTH OFF.=07   DIA. OFF.=57   DIA.=10.000  TOOL NAME= SPHERE )
```

```
N2 G40 G49 G00 G80 G90 G17
N3 G91 G28 Z0 M19

N3181 M01
N3182 G90 T2 M06  ( SPHERE )  //Start vane
rough
N3183 G17 G91
N3184 S3055 M03
N3185 X-87.761 Y41.535
N3186 G43 H2 Z56.
N3187 Z6. M8
N3188 Z-7.487
N3189 G1 Z-7.487 F122.2
N3190 X-87.873 Y42.293 Z-7.489 F244.4
N3193 X-89.219 Y44.259
N3194 X-89.924 Y44.652
N3195 X-90.699 Y44.875 Z-7.491
N3207 X-93.394 Y23.661 Z-7.465
N3208 X-92.126 Y19.923
N3209 X-91.067 Y16.238 Z-7.464
N3212 X-86.614 Y5.592
N3213 X-84.734 Y2.211 Z-7.462
N3214 X-82.772 Y-1.07
N3215 X-80.594 Y-4.243
N3216 G3 X-77.333 Y-7.819 I60.479 J43.845
N3217 G1 X-75.318 Y-10.729
N3218 X-72.648 Y-13.505 Z-7.463
N3219 X-69.843 Y-16.14
N3220 X-66.906 Y-18.627
N3221 X-63.845 Y-20.96 Z-7.464
N3222 X-60.668 Y-23.132
N3223 X-57.383 Y-25.137 Z-7.465
N3243 G2 X-29.185 Y-31.374 I14.296 J78.408
N3244 G1 X-32.834 Y-30.281 Z-7.443
N3245 X-36.427 Y-29.016
N3246 X-39.964 Y-27.58 Z-7.444
N3247 X-43.41 Y-25.978
N3248 X-46.785 Y-24.212 Z-7.445
N3274 X-88.624 Y43.712 Z-8.385
N3275 X-89.219 Y44.255
N3276 X-89.922 Y44.649
N3277 X-90.697 Y44.871 Z-8.384
N3342 X-78.564 Y8.12 Z-8.357
N3343 X-80.268 Y11.524 Z-8.359
N3344 X-81.807 Y15.006 Z-8.361
N3356 X-88.625 Y43.709 Z-9.278
N3357 X-89.219 Y44.252
N3358 X-89.921 Y44.645
N3359 X-90.694 Y44.868 Z-9.276
N3360 X-91.497 Y44.908 Z-9.275
N3372 X-92.155 Y19.941 Z-9.25
N3373 X-91.047 Y16.259
N3374 X-89.751 Y12.639 Z-9.249
N3375 X-88.271 Y9.091
N3376 X-86.61 Y5.623 Z-9.248
N3377 X-84.773 Y2.246
N3378 X-82.764 Y-1.032
N3379 X-80.59 Y-4.203
N3380 G3 X-77.299 Y-8.42 I60.494 J43.820
N3381 G1 X-74.751 Y-11.299
N3382 X-72.059 Y-14.044
N3383 X-69.229 Y-16.647
N3384 X-66.269 Y-19.1 Z-9.249
N3385 X-63.187 Y-21.398
N3386 X-59.99 Y-23.535 Z-9.25
N3405 X-19.082 Y-33.486 Z-9.229
N3406 X-22.838 Y-32.876
N3407 G2 X-30.011 Y-31.186 I14.669 J78.337
N3408 G1 X-33.645 Y-30.056
N3409 X-37.22 Y-28.754
N3410 X-40.729 Y-27.283 Z-9.23
N3411 X-44.163 Y-25.646 Z-9.231
N3412 X-47.516 Y-23.847
N3413 X-50.779 Y-21.89 Z-9.232
N3420 X-70.515 Y-4.186 Z-9.242
N3421 X-72.813 Y-1.154 Z-9.244
N3422 X-74.964 Y1.984 Z-9.246
N3423 X-76.963 Y5.222 Z-9.248
N3424 X-78.805 Y8.551 Z-9.25
N3425 X-80.486 Y11.964 Z-9.252
N3432 X-87.473 Y37.54 Z-9.27
N3433 X-87.761 Y41.382 Z-9.272
N3434 X-87.77 Y41.53
N3435 X-87.775 Y41.528 Z-10.165
N3438 X-88.626 Y43.705 Z-10.171
N3439 X-89.219 Y44.248
N3440 X-89.919 Y44.641 Z-10.17
N3444 X-93.021 Y44.439 Z-10.162
```

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N3448 X-94.133 Y41.851 Z-10.15
N3449 X-94.177 Y38.763 Z-10.148
N3450 X-94.034 Y34.923 Z-10.147
N3453 X-9[illegible].025 Y23.518 Z-10.144
N3454 X-[illegible] Y19.788 Z-10.143
N3455 X-[illegible].465 Y16.111 Z-10.142
N3456 X-[illegible].682 Y12.496
N3457 X-[illegible].194 Y8.953 Z-10.141
N3458 X-[illegible].527 Y5.49
N3459 X-[illegible]4.683 Y2.119
N3460 X-[illegible].669 Y-1.154
N3461 X-[illegible].489 Y-4.318 Z-10.14
N3462 G3 X-77.084 Y-8.654 I60.405 J43.943
N3463 G1 X-74.526 Y-11.521 Z-10.141
N3464 X-71.823 Y-14.253
N3465 X-[illegible].984 Y-16.843
N3466 X-[illegible].925 Y-19.283 Z-10.142
N3467 X-[illegible] Y-21.567
N3468 X-[illegible].721 Y-23.689 Z-10.143
N3469 X-[illegible].413 Y-25.644 Z-10.144
N3470 X-[illegible].098 Y-27.426
N3471 X-[illegible].227 Y-29.03 Z-10.145
N3487 X-[illegible].159 Y-33.497 Z-10.122
N3488 X-[illegible].912 Y-32.883
N3489 X-[illegible].831 Y-32.091
N3490 X-30.409 Y-31.122
N3491 X-33.936 Y-29.979
N3492 X-37.534 Y-28.664 Z-10.123
N3493 X-41.006 Y-27.18
N3494 X-44.432 Y-25.532 Z-10.124
N3495 X-47.777 Y-23.721 Z-10.125
N3496 X-51.031 Y-21.753
N3499 X-[illegible].179 Y-14.951 Z-10.129
N3500 X-[illegible].401 Y-12.402 Z-10.13
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N3522 X-89.918 Y44.637 Z-11.063
N3523 X-90.669 Y44.86 Z-11.062
N3524 X-[illegible].49 Y44.9 Z-11.06
N3525 X-92.279 Y44.757 Z-11.058
N3534 X-[illegible].745 Y27.296 Z-11.037
N3535 X-[illegible].014 Y23.525
N3536 X-92.089 Y19.797 Z-11.036
N3537 X-90.975 Y16.122 Z-11.035
N3538 X-[illegible].677 Y12.508
N3539 X-[illegible].186 Y8.967 Z-11.034
N3540 X-[illegible] Y5.506
N3541 X-[illegible].676 Y2.136
N3542 X-[illegible].665 Y-1.135 Z-11.033
N3543 X-[illegible].487 Y-4.298
N3544 G3 X-76.868 Y-8.887 I60.413 J43.931
N3545 G1 X-74.299 Y-11.742
N3546 X-71.586 Y-14.461 Z-11.034
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N3549 X-[illegible].662 Y-21.734 Z-11.035
N3550 X-[illegible].452 Y-23.842 Z-11.036
N3551 X-[illegible] Y-25.782
N3552 X-[illegible].777 Y-27.55 Z-11.037
N3553 X-49.231 Y-29.139 Z-11.038
N3554 X-[illegible].658 Y-30.547 Z-11.039
N3555 X-[illegible].026 Y-31.769 Z-11.04
N3556 X-[illegible] Y-32.803 Z-11.041
N3569 X-19.236 Y-33.507 Z-11.015
N3570 X-[illegible].996 Y-32.89
N3571 X-26.703 Y-32.095
N3572 X-30.378 Y-31.123
N3573 X-[illegible].002 Y-29.979
N3574 X-[illegible].567 Y-28.661 Z-11.016
N3575 X-41.066 Y-27.175
N3576 X-44.49 Y-25.525 Z-11.017
N3594 X-[illegible].036 Y26.539 Z-11.048
N3595 X-86.418 Y30.252 Z-11.05
N3596 X-87.052 Y34. Z-11.053
N3597 X-87.507 Y37.773 Z-11.056
N3598 X-87.779 Y41.525 Z-11.058
N3599 X-87.784 Y41.523 Z-11.951
N3600 X-87.89 Y42.284 Z-11.953
N3617 X-93.003 Y23.532 Z-11.929
N3618 X-92.079 Y19.806
N3619 X-90.965 Y16.133 Z-11.928
N3620 X-89.663 Y12.521 Z-11.927
N3621 X-88.178 Y8.981
N3622 X-86.513 Y5.522
N3623 X-84.673 Y2.153 Z-11.926
N3624 X-82.661 Y-1.116
N3625 X-80.485 Y-4.279
N3626 G3 X-76.651 Y-9.119 I60.422 J43.919
N3627 G1 X-74.071 Y-11.961
N3628 X-71.349 Y-14.668 Z-11.927
N3629 X-68.491 Y-17.231
N3630 X-65.506 Y-19.644
N3631 X-62.4 Y-21.9 Z-11.928
N3632 X-59.182 Y-23.994
N3633 X-55.862 Y-25.92 Z-11.929
N3634 X-52.446 Y-27.672 Z-11.93
N3635 X-48.945 Y-29.247 Z-11.931
N3636 X-45.368 Y-30.64 Z-11.932
N3651 X-22.827 Y-32.941 Z-11.908
N3652 X-26.544 Y-32.154
N3653 X-30.218 Y-31.191
N3654 X-33.843 Y-30.054 Z-11.909
N3655 X-37.409 Y-28.745
N3656 X-40.909 Y-27.268
N3657 X-44.335 Y-25.626 Z-11.91
N3677 X-87.025 Y33.765 Z-11.946
N3678 X-87.49 Y37.535 Z-11.948
N3679 X-87.775 Y41.373 Z-11.951
N3680 X-87.784 Y41.523
N3681 X-87.788 Y41.521 Z-12.844
N3684 X-88.629 Y43.696 Z-12.849
N3685 X-89.218 Y44.237
N3686 X-89.915 Y44.63
N3687 X-90.683 Y44.852 Z-12.848
N3688 X-91.482 Y44.893 Z-12.846
N3702 X-89.654 Y12.534 Z-12.82
N3703 X-88.17 Y8.995
N3704 X-86.506 Y5.538 Z-12.819
N3705 X-84.667 Y2.171
N3706 X-82.657 Y-1.098
N3707 X-80.483 Y-4.259
N3708 G3 X-76.324 Y-9.475 I60.430 J43.906
N3709 G1 X-73.728 Y-12.3
N3710 X-70.989 Y-14.987
N3711 X-68.117 Y-17.531 Z-12.82
N3738 X-41.402 Y-27.067 Z-12.803
N3739 X-44.815 Y-25.403
N3740 X-48.144 Y-23.578 Z-12.804
N3741 X-51.383 Y-21.596 Z-12.805
N3781 X-92.057 Y19.824 Z-13.714
N3782 X-90.945 Y16.154
N3783 X-89.645 Y12.546 Z-13.713
N3820 X-45.082 Y-25.286
N3821 X-48.403 Y-23.45 Z-13.697
N3822 X-51.633 Y-21.458 Z-13.698
N3865 X-88.154 Y9.024 Z-14.605
N3866 X-86.493 Y5.57
N3867 X-84.656 Y2.206
N3868 X-82.65 Y-1.06
N3869 X-80.478 Y-4.219 Z-14.604
N3870 G3 X-75.775 Y-10.057 I60.446 J43.882
N3871 G1 X-73.151 Y-12.851 Z-14.605
N3872 X-70.388 Y-15.506
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N3873 X-67.491 Y-18.017
N3874 X-64.47 Y-20.375 Z-14.606
N3875 X-61.332 Y-22.576
N3876 X-58.085 Y-24.612 Z-14.607
N3877 X-54.738 Y-26.48 Z-14.608
N3878 X-51.299 Y-28.173
N3879 X-47.778 Y-29.688 Z-14.609
N3901 X-4[illegible].139 Y-25.279 Z-14.59
N3902 X-4[illegible].457 Y-23.442
N3903 X-5[illegible].684 Y-21.448 Z-14.591
N3945 X-[illegible].627 Y12.571
N3946 X-[illegible].146 Y9.038 Z-15.498
N3947 X-[illegible].486 Y5.586
N3948 X-[illegible].651 Y2.223
N3949 X-[illegible] Y-1.041 Z-15.497
N3950 X-[illegible].476 Y-4.199
N3951 G3 X-75.443 Y-10.407 160.455 J43.869
N3952 G1 X-72.802 Y-13.182
N3953 X-[illegible] Y-15.819 Z-15.498
N3954 X-[illegible] Y-18.31
N3955 X-64.078 Y-20.648 Z-15.499
N4109 X-[illegible] Y-1.113
N4110 X-[illegible].374 Y-4.293
N4111 G3 X-74.986 Y-10.977 160.374 J43.980
N4112 G1 X-72.219 Y-13.722
N4113 X-69.415 Y-16.326
N4114 X-66.481 Y-18.784 Z-17.284
N4115 X-63.425 Y-21.088
N4116 X-60.256 Y-23.232 Z-17.285
N4117 X-56.98 Y-25.211
N4157 X-85.851 Y20.191 Z-17.294
N4163 X-87.911 Y41.509 Z-17.308
N4164 G0 Z50.
N4165 G40 M9
N4166 G91 G28 Z0 M19
N4167 M01                              //End vane
roughing
N4199 G91 G28 Z0 M19
N4200 M01
N4729 G90 T3 M06  ( FLAT ENDMILL )    //Start
vane finish
N4730 [illegible] R1.5
N4731 S1[illegible] M03
N4732 X-83.967 Y14.11
N4733 G43 H3 Z10.
N4734 [illegible]
N4735 Z-2.
N4736 G1 X-80.095 Y6.036 Z-2.47 F7.5
N4737 X-75.366 Y-1.568 Z-2.939
N4742 X-41.156 Y-29.621 Z-5.287
N4743 X-20.237 Y-34.748 F30.
N4744 X-19.993
N4745 X-30.733 Y-31.299
N4746 X-42.497
N4747 G2 X-51.389 Y-27.85 I22.382 J70.901
N4760 G1 X-88.45
N4761 G2 X-[illegible].297 Y-3.709 I60.290 J52.660
N4762 G1 X-80.547
N4763 G2 X-82.876 Y-.261 I60.432 J43.312
N4791 G3 X-91.251 Y44.573 I-.978 J-3.011
N4792 G1 X-91.301
N4793 Z-3.[illegible] F200.
N4794 G0 Z10.
N4795 X-83.967 Y14.11
N4796 Z-2.287
N4797 G1 X-80.095 Y6.036 Z-2.756 F7.5
N4798 X-75.366 Y-1.568 Z-3.226
N4799 X-69.836 Y-8.611 Z-3.695
N4803 X-41.156 Y-29.621 Z-5.573
N4804 X-20.237 Y-34.748 F30.
N4805 X-19.993
N4806 X-30.733 Y-31.299
N4807 X-42.497
N4808 G2 X-51.389 Y-27.85 I22.382 J70.901
N4849 G1 X-88.09
N4850 G2 X-88.113 Y41.562 I79.930 J4.379
N4851 G3 X-90.297 Y44.418 I-3.163 J-.156
N4852 G3 X-91.251 Y44.573 I-.978 J-3.011
N4853 G1 X-91.301
N4854 Z-3.573 F200.
N4855 G0 Z10.
N4856 X-83.967 Y14.11
N4857 Z-2.573
N4858 G1 X-80.095 Y6.036 Z-3.043 F7.5
N4859 X-75.366 Y-1.568 Z-3.512
N4860 X-69.836 Y-8.611 Z-3.982
N4864 X-41.156 Y-29.621 Z-5.86
N4865 X-20.237 Y-34.748 F30.
N4866 X-19.993
N4867 X-30.733 Y-31.299
N4868 X-42.497
N4869 G2 X-51.389 Y-27.85 I22.382 J70.901
N4908 G1 X-94.44
N4909 G2 X-94.449 Y41.124 I74.325 J1.927
N4910 G1 X-88.09
N4911 G2 X-88.113 Y41.562 I79.930 J4.379
N4912 G3 X-90.297 Y44.418 I-3.163 J-.156
N4913 G3 X-91.251 Y44.573 I-.978 J-3.011
N4914 G1 X-91.301
N4915 Z-3.86 F200.
N4916 G0 Z10.
N4917 X-83.967 Y14.11
N4918 Z-2.86
N4919 G1 X-80.095 Y6.036 Z-3.33 F7.5
N4920 X-75.366 Y-1.568 Z-3.799
N4923 X-56.648 Y-20.688 Z-5.208
N4924 X-49.146 Y-25.577 Z-5.677
N4925 X-41.156 Y-29.621 Z-6.147
N4926 X-20.237 Y-34.748 F30.
N4927 X-19.993
N4928 X-30.733 Y-31.299
N4929 X-42.497
N4930 G2 X-51.389 Y-27.85 I22.382 J70.901
N4945 G1 X-80.547
N4946 G2 X-82.876 Y-.261 I60.432 J43.312
N4947 G1 X-73.839
N4948 G2 X-76.112 Y3.188 I65.679 J45.763
N4959 G1 X-84.183
N4960 G2 X-85.235 Y23.88 I76.023 J25.071
N4961 G1 X-92.784
N4962 G2 X-93.445 Y27.329 I72.669 J15.722
N4969 G1 X-94.44
N4970 G2 X-94.449 Y41.124 I74.325 J1.927
N4971 G1 X-88.09
N4972 G2 X-88.113 Y41.562 I79.930 J4.379
N4973 G3 X-90.297 Y44.418 I-3.163 J-.156
N4974 G3 X-91.251 Y44.573 I-.978 J-3.011
N4975 G1 X-91.301
N4976 Z-4.147 F200.
N4977 G0 Z10.
N4978 X-83.967 Y14.11
N4979 Z-3.147
N4980 G1 X-80.095 Y6.036 Z-3.616 F7.5
N4981 X-75.366 Y-1.568 Z-4.086
N4982 X-69.836 Y-8.611 Z-4.555
N4983 X-63.572 Y-15.01 Z-5.025
N4984 X-56.648 Y-20.688 Z-5.494
N4985 X-49.146 Y-25.577 Z-5.964
N4986 X-41.156 Y-29.621 Z-6.433
N4987 X-20.237 Y-34.748 F30.
N4988 X-19.993
N4989 X-30.733 Y-31.299
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N4990 X-42.497
N4991 G2 X-51.389 Y-27.85 I22.382 J70.901
N4992 G1 X-40.213
N4993 G2 X-47.166 Y-24.401 I32.053 J73.353
N4994 G1 X-57.949
N4995 G2 X-63.254 Y-20.953 I37.834 J64.004
N4996 G1 X-52.789
N4997 G2 X-57.538 Y-17.504 I44.628 J66.455
N4998 G1 X-67.726
N4999 G2 X-71.581 Y-14.055 I47.611 J57.106
N5000 G1 X-61.447
N5001 G2 X-65.254 Y-10.607 I53.487 J59.558
N5002 G1 X-74.451
N5032 G1 X-88.09
N5033 G2 X-88.113 Y41.562 I79.930 J4.379
N5034 G3 X-90.297 Y44.418 I-3.163 J-.156
N5035 G3 X-91.251 Y44.573 I-.978 J-3.011
N5036 G1 X-91.301
N5037 Z-4.433 F200.
N5038 G0 Z10.
N5039 X-83.967 Y14.11
N5040 Z-3.433
N5041 G1 X-80.095 Y6.036 Z-3.903 F7.5
N5042 X-75.366 Y-1.568 Z-4.372
N5043 X-[illegible] Y-8.611 Z-4.842
N5047 X-41.156 Y-29.621 Z-6.72
N5048 X-20.237 Y-34.748 F30.
N5049 X-19.993
N5050 X-30.733 Y-31.299
N5051 X-42.497
N5052 G2 X-51.389 Y-27.85 I22.382 J70.901
N5053 G1 X-40.213
N5054 G2 X-47.166 Y-24.401 I32.053 J73.353
N5093 G1 X-88.09
N5094 G2 X-88.113 Y41.562 I79.930 J4.379
N5095 G3 X-90.297 Y44.418 I-3.163 J-.156
N5096 G3 X-91.251 Y44.573 I-.978 J-3.011
N5097 G1 X-91.301
N5098 Z-4.72 F200.
N5099 G0 Z10.
N5100 X-83.967 Y14.11
N5101 Z-3.72
N5102 G1 X-80.095 Y6.036 Z-4.19 F7.5
N5103 X-75.366 Y-1.568 Z-4.659
N5108 X-41.156 Y-29.621 Z-7.007
N5109 X-20.237 Y-34.748 F30.
N5110 X-19.993
N5111 X-30.733 Y-31.299
N5112 X-42.497
N5113 G2 X-51.389 Y-27.85 I22.382 J70.901
N5150 G1 X-87.112
N5151 G2 X-87.827 Y37.675 I79.252 J11.276
N5152 G1 X-94.44
N5153 G2 X-94.449 Y41.124 I74.325 J1.927
N5154 G1 X-88.09
N5155 G2 X-88.113 Y41.562 I79.930 J4.379
N5158 G1 X-91.301
N5159 Z-5.007 F200.    //feed 200 mm/min
N5160 G0 Z10.
N5161 X-83.967 Y14.11
N5162 Z-4.007
N5163 G1 X-80.095 Y6.036 Z-4.476 F7.5
N5164 X-75.366 Y-1.568 Z-4.946
N5165 X-[illegible] Y-8.611 Z-5.415
N5169 X-41.156 Y-29.621 Z-7.293
N5170 X-20.237 Y-34.748 F30.
N5171 X-19.993
N5172 X-30.733 Y-31.299
N5173 X-42.497
N5174 G2 X-51.389 Y-27.85 I22.382 J70.901
N5175 G1 X-40.213
N5176 G2 X-47.166 Y-24.401 I32.053
J73.353
N5213 G1 X-94.44
N5214 G2 X-94.449 Y41.124 I74.325 J1.927
N5215 G1 X-88.09
N5216 G2 X-88.113 Y41.562 I79.930 J4.379
N5218 G3 X-91.251 Y44.573 I-.978 J-3.011
N5219 G1 X-91.301
N5220 Z-5.293 F200.
N5221 G0 Z10.
N5222 X-83.967 Y14.11
N5223 Z-4.293
N5224 G1 X-80.095 Y6.036 Z-4.763 F7.5
N5225 X-75.366 Y-1.568 Z-5.232
N5230 X-41.156 Y-29.621 Z-7.58
N5231 X-20.237 Y-34.748 F30.
N5232 X-19.993
N5233 X-30.733 Y-31.299
N5234 X-42.497
N5235 G2 X-51.389 Y-27.85 I22.382 J70.901
N5274 G1 X-94.44
N5275 G2 X-94.449 Y41.124 I74.325 J1.927
N5276 G1 X-88.09
N5277 G2 X-88.113 Y41.562 I79.930 J4.379
N5278 G3 X-90.297 Y44.418 I-3.163 J-.156
N5279 G3 X-91.251 Y44.573 I-.978 J-3.011
N5280 G1 X-91.301
N5281 Z-5.58 F200.
N5282 G0 Z10.
N5283 X-83.967 Y14.11
N5284 Z-4.58
N5285 G1 X-80.095 Y6.036 Z-5.05 F7.5
N5286 X-75.366 Y-1.568 Z-5.519
N5289 X-56.648 Y-20.688 Z-6.928
N5290 X-49.146 Y-25.577 Z-7.397
N5291 X-41.156 Y-29.621 Z-7.867
N5292 X-20.237 Y-34.748 F30.  //Plunge rate
30 mm/min
N5293 X-19.993
N5294 X-30.733 Y-31.299
N5295 X-42.497
N5296 G2 X-51.389 Y-27.85 I22.382 J70.901
N5297 G1 X-40.213
N5298 G2 X-47.166 Y-24.401 I32.053 J73.353
N5299 G1 X-57.949
N5300 G2 X-63.254 Y-20.953 I37.834 J64.004
N5301 G1 X-52.789
N5302 G2 X-57.538 Y-17.504 I44.628 J66.455
N5319 G1 X-88.355
N5320 G2 X-89.745 Y13.534 I68.240 J29.517
N5335 G1 X-94.44
N5336 G2 X-94.449 Y41.124 I74.325 J1.927
N5337 G1 X-88.09
N5338 G2 X-88.113 Y41.562 I79.930 J4.379
N5339 G3 X-90.297 Y44.418 I-3.163 J-.156
N5340 G3 X-91.251 Y44.573 I-.978 J-3.011
N5341 G1 X-91.301
N5342 Z-5.867 F200.
N5343 G0 Z10.
N5344 X-83.967 Y14.11
N5345 Z-4.867
N5346 G1 X-80.095 Y6.036 Z-5.336 F7.5
N5347 X-75.366 Y-1.568 Z-5.806
N5352 X-41.156 Y-29.621 Z-8.153
N5353 X-20.237 Y-34.748 F30.
N5354 X-19.993
N5355 X-30.733 Y-31.299
N5396 G1 X-94.44
N5397 G2 X-94.449 Y41.124 I74.325 J1.927
N5398 G1 X-88.09
N5399 G2 X-88.113 Y41.562 I79.930 J4.379
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N5400 G3 X-90.297 Y44.418 I-3.163 J-.156
N5401 G3 X-91.251 Y44.573 I-.978 J-3.011
N5402 G1 X-91.301
N5403 Z-6.153 F200.
N5404 G0 Z10.
N5405 X-83.967 Y14.11
N5406 Z-5.153
N5407 G1 X-80.095 Y6.036 Z-5.623 F7.5
N5411 X-[illegible] Y-20.668 Z-7.501
N5413 X-41.156 Y-29.621 Z-8.44
N5414 X-20.237 Y-34.748 F30.
N5415 X-19.993
N5416 X-30.733 Y-31.299
N5417 X-42.497
N5418 G2 X-51.389 Y-27.85 I22.382 J70.901
N5437 G1 X-84.937
N5438 G2 X-86.757 Y6.637 I64.822 J36.414
N5439 G1 X-78.142
N5440 G2 X-79.949 Y10.086 I69.982 J38.866
N5441 G1 X-88.355
N5454 G2 X-94.27 Y34.227 I73.824 J8.824
N5457 G1 X-94.44
N5458 G2 X-94.449 Y41.124 I74.325 J1.927
N5459 G1 X-88.09
N5460 G2 X-88.113 Y41.562 I79.930 J4.379
N5461 G3 X-90.297 Y44.418 I-3.163 J-.156
N5462 G3 X-91.251 Y44.573 I-.978 J-3.011
N5463 G1 X-91.301
N5464 Z-6.44 F200.
N5465 G0 Z10.
N5466 X-83.967 Y14.11
N5467 Z-5.44
N5468 G1 X-80.095 Y6.036 Z-5.91 F7.5
N5469 X-75.366 Y-1.568 Z-6.379
N5474 X-41.156 Y-29.621 Z-8.727
N5475 X-20.237 Y-34.748 F30.
N5476 X-19.993
N5477 X-30.733 Y-31.299
N5478 X-42.497
N5479 G2 X-51.389 Y-27.85 I22.382 J70.901
N5537 X-19.993
N5538 X-30.733 Y-31.299
N5539 X-42.497
N5540 G2 X-51.389 Y-27.85 I22.382 J70.901
N5541 G1 X-40.213
N5556 G2 X-82.876 Y-.261 I60.432 J43.312
N5557 G1 X-73.839
N5558 G2 X-76.112 Y3.188 I65.679 J45.763
N5577 G1 X-94.44
N5580 G2 X-94.449 Y41.124 I74.325 J1.927
N5581 G1 X-88.09
N5582 G2 X-88.113 Y41.562 I79.930 J4.379
N5583 G3 X-90.297 Y44.418 I-3.163 J-.156
N5584 G3 X-91.251 Y44.573 I-.978 J-3.011
N5585 G1 X-91.301
N5586 Z-7.013 F200.
N5587 G0 Z10.
N5588 X-83.967 Y14.11
N5589 Z-6.013
N5590 G1 X-80.095 Y6.036 Z-6.483 F7.5
N5591 X-75.366 Y-1.568 Z-6.952
N5596 X-41.156 Y-29.621 Z-9.3
N5597 X-20.237 Y-34.748 F30.
N5598 X-19.993
N5599 X-30.733 Y-31.299
N5600 X-42.497
N5601 G2 X-51.389 Y-27.85 I22.382 J70.901
N5626 G1 X-81.55
N5627 G2 X-82.957 Y16.983 I73.390 J31.968
N5642 G1 X-88.09
N5643 G2 X-88.113 Y41.562 I79.930 J4.379
N5644 G3 X-90.297 Y44.418 I-3.163 J-.156
N5645 G3 X-91.251 Y44.573 I-.978 J-3.011
N5646 G1 X-91.301
N5647 Z-7.3 F200.
N5648 G0 Z10.
N5649 X-83.967 Y14.11
N5650 Z-6.3
N5651 G1 X-80.095 Y6.036 Z-6.77 F7.5
N5657 X-41.156 Y-29.621 Z-9.587
N5658 X-20.237 Y-34.748 F30.
N5659 X-19.993
N5660 X-30.733 Y-31.299
N5661 X-42.497
N5662 G2 X-51.389 Y-27.85 I22.382 J70.901
N5663 G1 X-40.213
N5664 G2 X-47.166 Y-24.401 I32.053 J73.353
N5665 G1 X-57.949
N5666 G2 X-63.254 Y-20.953 I37.834 J64.004
N5667 G1 X-52.789
N5668 G2 X-57.538 Y-17.504 I44.628 J66.455
N5685 G1 X-88.355
N5686 G2 X-89.745 Y13.534 I68.240 J29.517
N5687 G1 X-81.55
N5688 G2 X-82.957 Y16.983 I73.390 J31.968
N5701 G1 X-94.44
N5702 G2 X-94.449 Y41.124 I74.325 J1.927
N5703 G1 X-88.09
N5704 G2 X-88.113 Y41.562 I79.930 J4.379
N5706 G3 X-91.251 Y44.573 I-.978 J-3.011
N5707 G1 X-91.301
N5708 Z-7.587 F200.
N5709 G0 Z10.
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N7188 Z6. M8
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N7283 X-94.198 Y43.348 Z-8.37
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N7285 X-94.754 Y41.852 Z-8.364
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N7287 X-94.667 Y35.088 Z-8.361
N7288 X-94.336 Y31.256 Z-8.36
N7289 X-93.807 Y27.445 Z-8.359
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N7291 X-92.166 Y19.932 Z-8.357
N7292 X-91.057 Y16.248
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N7294 X-88.279 Y9.077.
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N7296 X-84.778 Y2.229
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N7382 G1 X-74.751 Y-11.299
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N556 X-90.965 Y16.133 Z-11.928
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N562 X-80.485 Y-4.279
N563 G3 X-76.651 Y-9.119 I60.422 J43.919
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N740 X-29.565 Y-34.431 Z-13.722
N741 X-25.751 Y-34.822 Z-13.723
N742 X-21.921 Y-35.017 Z-13.725
N743 X-18.086 Y-35.015 Z-13.726
N744 X-15.121 Y-34.873 Z-13.705
N756 X-41.677 Y-26.962 Z-13.696
N1130 G0 Z5.
N1131 G40 M9 //Comp off, coolant off
N1132 G91 G28 Z0 M19   //Return reference
N1133 G90        //Absolute
N1134 M30        //End of program
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