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A Low Specific Speed, Multistage,
Turbo-Compressor for Microturbine Fuelling

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2005

Thesis submitted for the degree of Doctor of Philosophy of the
University of London and for the Diploma of Membership of Imperial
College

Acknowledgements

I would like to thank my supervisors, Dr. Keith Pullen and Dr. Shahram Etemad, for supporting and guiding me in this work. They never let me down in providing advice, ideas, resources or support and I look forward to working with them in the future. Also, my appreciation goes to Andrew Vine for his collaboration and continuation of the research.

My highest gratitude goes to Dr. Niall McGlahsan for teaching me most of what I know about practical engineering. The good doctor continues to share both his knowledge and expertise, without which this research would not have been possible.

Many thanks to Phil Wilson, Charles Milton, Steve Fay and the rest of the technicians in the Department of Mechanical Engineering for sharing both their machining skills and their practical experience.

Big thanks to my parents for their continued support of my academic pursuits.

Finally, I am thankful for all the friends I have made at Imperial. It would have been a long five years without them.

Abstract

One aspect limiting the use of microturbines in distributed power generation schemes is the lack of a suitable fuel gas compressor. Microturbines require fuel to be compressed to above the pressure of combustion, however, a low volumetric flow rate is required. Therefore, fuel gas is currently compressed by positive displacement compressors, which are large and require frequent maintenance. A more compact compressor requiring less maintenance would allow microturbines to penetrate the distributed generation and combined heat and power markets.

Gas compressor delivery requirements for a large range of turbine output powers are formulated. It is shown that gas turbines having less than 10MW output power are currently dependant on positive displacement compressors for fuel gas compression. It is established that a turbo-compressor would be more suitable for fuelling micro-turbines, and the forward-swept, centrifugal compressor is introduced as a means to achieve this.

The research evaluated this novel compressor for the purpose of multi-stage gas compression for fuelling a microturbine of approximately 100kW output power. The typical losses of windage and leakage are proportionally higher than in conventional turbomachinery, so the performance is predicted using a computer simulation specifically developed for this application. The subroutines for calculating windage losses, seal leakages, and off-design performance are formulated based on single stage data and formulations found in the open literature. The calculated effects of the various design parameters and configurations are used to assess machine robustness and the effect on system efficiency.

A dedicated, 4-stage test rig was built to experimentally validate these predictions, using air as the working fluid. The maximum single stage pressure ratio was 1.38, which was lower than the prediction of 1.58. This severely affected stage matching, resulting in a total pressure ratio of only 2.58. However, multistage simulations were validated and a greater understanding of the multistage, low specific speed gas compressor has been achieved. The design has been iterated and modifications have been recommended for improving peak performance and range.

Nomenclature

Symbol	Meaning	Units
A	Cross-Sectional Area	m^2
bhp	Power, Expressed as Horsepower	hp
C	Absolute Gas Velocity	m/s
c	Speed of Sound	m/s
C_0	Seal Pressure Recovery Coefficient	<i>none</i>
C_w	Windage Coefficient	<i>none</i>
C_m	Windage Coefficient	<i>none</i>
c_p	Specific Heat at Constant Pressure	$J/kg \cdot K$
c_v	Specific Heat at Constant Volume	$J/kg \cdot K$
D	Diameter	m or mm
f	Friction Factor	<i>none</i>
h	Specific Enthalpy, Convection Coefficient	J/kg
h_C	Enthalpy of Combustion	J/kg
H	Enthalpy	J
I	Impulse Exponent	<i>none</i>
k	Conductivity	$W/m \cdot K$
$\dot{m}$	Mass Flow (Generally Total Throughput Flow)	$kg/s, g/s$
$\dot{m}_{Xin}$	Mass Flow at inlet of X^{th} stage	$kg/s, g/s$
$\dot{m}_{Xout}$	Mass Flow at outlet of X^{th} stage	$kg/s, g/s$
Ma	Mach Number	<i>none</i>
N_s	Specific Speed	<i>none</i>
N_T	Number of Labyrinth Seals	<i>none</i>
p	Pressure	Bar, Pa
p_0	Total Pressure	Bar, Pa
p_{cr}	Critical Pressure	Bar, Pa
p_{Xin}	Pressure at Inlet of X^{th} Stage	Bar, Pa
p_{Xout}	Pressure at Outlet of X^{th} Stage	Bar, Pa
p_{atm}	Atmospheric Pressure	Bar, Pa
PR	Pressure Ratio	<i>none</i>
Q	Volumetric Flow	m^3/s
q	Heat Flow	W
R	Ideal Gas Constant	$J/kg \cdot K$
Re	Reynolds Number	<i>none</i>
Re_w	Rotational Reynolds Number	<i>none</i>
Re_D	Reynolds Number Based on Hydraulic Diameter (D)	<i>none</i>
r	Radius	m or mm
RPM	Shaft Speed expressed in Rotations Per Minute	$1/min$
s	Specific Entropy	$J/K \cdot kg$
T	Temperature	$^{\circ}K, ^{\circ}C$
T_0	Total Temperature	$^{\circ}K, ^{\circ}C$
T_{atm}	Ambient Temperature	$^{\circ}K, ^{\circ}C$
T_{cr}	Critical Temperature	$^{\circ}K, ^{\circ}C$
T_{ref}	Reference Temperature	$^{\circ}K, ^{\circ}C$
T_{Xin}	Temperature at Inlet of X^{th} Stage	$^{\circ}K, ^{\circ}C$
T_{Xout}	Temperature at Outlet of X^{th} Stage	$^{\circ}K, ^{\circ}C$

Symbol	Meaning	Units
T_{PL}	Temperature Of Pre-load Bearing	$^{\circ}C$
T_T	Temperature of Thrust Bearings	$^{\circ}C$
U	Blade Speed	m/s
u	Specific Internal Energy	J/kg
v	Specific Volume	m^3/kg
V	Gas Velocity	m/s
$\dot{V}$	Volumetric Flow	m^3/s
W	Gas Velocity Relative to Rotor	m/s
$\dot{W}$	Power	W
w	Specific Work	J/kg
w_s	Isentropic Specific Work	J/kg
γ	Specific Heat Ratio	<i>none</i>
ΔP	Pressure Loss	Pa
η_{tt}	Total-to-Total efficiency	<i>none</i>
η_0	Kinetic Energy Carryover Coefficient	<i>none</i>
η	Efficiency	<i>none</i>
η_m	Mechanical Efficiency	<i>none</i>
η_s	Isentropic Efficiency	<i>none</i>
η_p	Polytropic Efficiency	<i>none</i>
η_{IC}	Intercooler Effectiveness	<i>none</i>
μ	Dynamic Viscosity	$N \cdot s/m^3$
ν	Kinematic Viscosity	m^2/s
ρ	Density	kg/m^3
ω	Angular Velocity	rad/s
Ψ	Head Coefficient	<i>none</i>
Φ	Flow Coefficient	<i>none</i>

note: Each Section in 4.2 has its own unique nomenclature, as described.

Subscripts

Subscript	Meaning
r	Radial
s	Isentropic
0	Total
θ	Angular

Abbreviations

Abbreviation	Meaning
AC	Aftercooling
A/D	Analog to Digital
CFD	Computational Fluid Dynamics
CHP	Combined Heat and Power
CNC	Computerized Numerical Control
DG	Distributed Generation
FEA	Finite Element Analysis
GT	Gas Turbine
IC	Intercooler
ID	Inner Diameter
MT	Microturbine
OD	Outer Diameter
OP	Operating Point
PD	Positive Displacement
PRT	Platinum Resistance Thermometer
SCFM	Standard Cubic Feet Per Minute
WOT	Wide Open Throttle

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1 Introduction

This section describes the concept of distributed energy, gas turbines, their fuelling with natural gas and the reasoning behind seeking a new method of fuel compression. A number of commercially available gas turbines are cataloged in terms of output power, efficiency, and pressure ratio. This data can be used to establish a target mass flow rate and pressure ratio of a fuelling compressor for a given engine power. The theory of gas compression is introduced and a new evaluation of compressor performance is established. Finally, the objectives of the research are established.

1.1 Distributed Generation

Distributed generation (DG) is a power generation scheme in which electrical power is produced locally or on site and any difference between supply and demand is regulated by drawing power from the grid, or supplying it with power. There are multiple reasons and benefits for DG.

- Increase power reliability and quality by having a local, high-quality power source with the grid as a backup, should a disruption in supply occur.
- Reduce transmission losses by reducing the distance that the current must travel.
- Reduce the maximum output requirements of centralized power providers by reducing peak demand.
- Reduce localized chemical, particulate and thermal pollution by distributing power generation over a broader area.
- Reduce the amount of fuel used for environmental and process heating by establishing a combined heat and power (CHP) system, in which hot engine exhaust is utilized.
- Reduce the investment and maintenance cost associated with increasing grid capacity.

There are, of course, many ways of producing electrical power, and many of them are suitable for implementation in a DG system. Some possibilities are direct conversion of

solar energy to electricity, the driving of a generator from environmental energy, and fuel cells, which directly convert the chemical energy from a fuel source to electrical energy. However, the best candidate that is currently available is the most popular method under the conventional power generation scheme; that of an engine driving a generator.

1.1.1 Engines

There are two types of engines that are potential candidates for a distributed energy scheme: reciprocating and gas turbine (GT). The fundamental difference between the two is that a reciprocating engine has a cycle which is an intermittent process, while a gas turbine engine cycle is a continuous process. The primary advantage of an intermittent process is that the time-averaged temperature can be kept low while maintaining a high temperature of combustion, leading to an increased engine efficiency without overheating the materials. The primary drawback to an intermittent cycle is that it is a non-steady state process and the machine only does work for a fraction of the cycle. The result of this is that a reciprocating engine must be larger than a gas turbine for a given power output. In addition, a reciprocating engine must have pistons, crankshafts, connecting rods, valves, camshafts and other parts moving continuously for operation. The only continuously moving parts in a gas turbine are the shaft and bearings, which results in improved reliability, higher attainable speeds and higher power density.

While gas turbines cannot achieve the high temperatures that reciprocating engines can achieve, a recuperator can be fitted which transfers heat from the turbine outlet to the compressor outlet, reducing the amount of fuel required to heat the air in the combustion chamber. This detracts from the advantages of compactness, as a high-effectiveness recuperator will be large.

Additionally, the waste heat in a gas turbine engine is present in the high temperature exhaust gas. The waste heat in a reciprocating engine will be split between the exhaust and the cooling water. This adds to the benefits of using gas turbines for CHP applications.

A major drawback to gas turbine engines has been that their running speeds are higher than reciprocating engines, which often necessitates the use of a gearbox for power extraction. Such gearboxes are precision machines that often cost more than the generator itself. However, recent advances in high speed electric machine technology have produced alternators that can achieve speeds of 90,000RPM and higher, which can be directly coupled to microturbines. These generators have much higher power density than traditional generators, compounding the space saving properties of gas turbines. Furthermore, gas turbines have much lower maintenance requirements and lower particulate emissions. These properties make microturbines particularly suitable for distributed generation applications.

A recent market survey of the gas turbine engine market was performed by Beesley [1]. Although the research cataloged the power, speed and pressure ratio of 188 different gas turbines, no data was recorded regarding efficiency. Some more gas turbines are presented in Table 1, along with listed efficiency values. It can be seen that all of the microturbines utilize a recuperator to improve the efficiency. The microturbines (those with an output power of less than 300kW) are all catalogued in [2]. The mini gas turbines (300kW to 2MW output power) and small industrial gas turbines (greater than 2MW) are available from [3], [4], and [5]

1.1.2 Fuelling

Gas turbines can theoretically use any heat source that provides the energy requirements to power the engine and can be delivered to the working fluid at the turbine inlet pressure. This can be in the form of an external heat source transferring heat through a heat exchanger, but it is usually in the form of fuel introduced to the working fluid and burnt. These include, but are not limited to: kerosene, petrol, diesel, liquified petroleum gas and natural gas.

Natural gas is attractive because it already has an installed distribution network in the form of gas mainlines as well as producing low levels of CO_2 and particulates during

Engine	Power (kW)	Pressure Ratio	Efficiency (%)	Recuperator
Capstone C30	30	3.5	26	Yes
Capstone C60	60	5.1	28	Yes
Ingersoll Rand 70 SM	70	unavailable	28	Yes
Bowman TG80RC-G	80	unavailable	28	Yes
Elliott TA-100	105	4	29	Yes
Turbec T100	105	4.5	30	Yes
Ingersoll Rand 250 SM	250	unavailable	31	Yes
P+W ST5	395	unavailable	23.5	No
P+W ST5	457	unavailable	32.7	Yes
Walter	447	5	19	No
P+W ST6L-72	508	7	23.4	No
P+W ST6L-79	678	7	24.7	No
P+W ST6L-90	1175	10	28	No
P+W ST40	4039	unavailable	33.1	No
Solar Mercury 50	4200	unavailable	40	Yes
P+W FT8	25000	unavailable	38	No

Table 1: Gas Turbine Engine Characteristics

combustion, which is beneficial to the proper operation of the recuperator. However, the problem with using a gaseous fuel as opposed to a liquid one is the added complication of pressurizing the fuel to the delivery pressure. Natural gas can easily be used in spark ignition reciprocating engines because a constant fuel/air ratio is maintained and the working fluid can be mixed with the fuel before compression in the same way as liquid fuel in a carburetor or fuel injection system. However, a gas turbine must have the fuel compressed to the combustion pressure before it is introduced in the combustion chamber, as a significant proportion of the air bypasses the combustion chamber and is re-introduced before the turbine inlet nozzle. Natural gas distribution networks have delivery pressures of as low as 3psi (0.2bar) [6], well below the pressure of even the lowest pressure-ratio gas turbine engines. This establishes the need for a suitable compressor that can compress the correct amount of fuel from as low as a few millibars to the combustion pressure of a microturbine, which is generally greater than 3.5 bar absolute pressure. Larger gas turbines usually have higher pressure gas on site, but often require the use of a booster compressor to raise the gas pressure up to the pressure of combustion. [7]

1.2 Natural Gas Properties

Natural gas found in nature is variable in constitution, as shown in **Figure 2**.

Gas	Symbol	Component
Methane	CH_4	70 – 90%
Ethane	C_2H_6	0 – 20%
Propane	C_3H_8	
Butane	C_4H_{10}	
Carbon Dioxide	CO_2	0 – 8%
Oxygen	O_2	0 – 0.2%
Nitrogen	N_2	0 – 5%
Hydrogen sulphide	H_2S	0 – 5%
Rare gases	A, He, Ne, Xe	trace

Table 2: Typical Composition of Natural Gas

However, natural gas as used in domestic and small industrial applications is almost entirely methane [6]. The properties of methane are shown along with air in **Table 3**.

Property at 20°C and 1Bar _{abs}	Symbol	Units	Air	Methane
Density	ρ	kg/m^3	1.23	.667
Gas Constant	R	$J/kg \cdot K$	297	518
Specific Heat	c_p	$J/kg \cdot K$	2253	1005
Ratio of Specific Heats	γ		1.4	1.3
dynamic viscosity	μ	$N \cdot s/m^2$	1.79×10^{-5}	1.10×10^{-5}
kinematic viscosity	ν	m^2/s	1.65×10^{-5}	1.46×10^{-5}
Heat of Combustion	h_C	MJ/kg	n/a	54

Table 3: Properties of Air and Methane

1.3 Microturbine Fuelling Requirements

Using the heat of combustion for methane, it is possible to determine the fuel mass flow rate for an engine based on only the output power and efficiency. This is expressed in **Figure 1**, which is true for all engines using natural gas of the specified properties and air as the working fluid. **Figure 2** shows the maximum efficiency as a function of gas turbine output power, based on a fit of the recuperated engines from **Table 1**. This can

be directly used to determine the fuel mass flow as a function of gas turbine power, as shown in **Figure 3**. This is a more difficult target than that of fuelling a low efficiency machine, as it is more difficult to design a turbomachine with a high pressure ratio and low mass flow, than vice-versa.

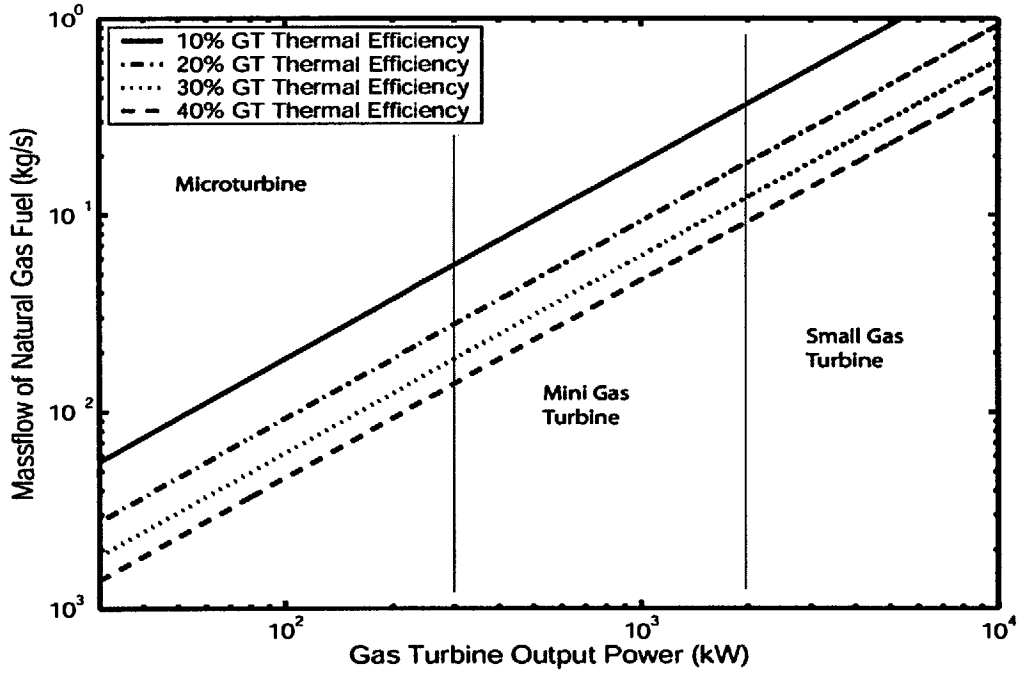


Figure 1: Natural Gas Mass Flow vs. Engine Output Power

The pressure ratio is a design choice for each gas turbine, but as the most efficient gas turbines tend to have the highest pressure ratios, it is prudent to use the highest pressure ratios as a target. **Figure 4** shows the results of [1] along with the gas turbines in **Table 1**. A curve has been fitted, which will be the gas compressor's target pressure ratio for a given engine power. This generally agrees with the pressure ratios of the most efficient gas turbine engines for a given power. Optimum PR for recuperated gas turbines is much lower than for non-recuperated ones and is typically around 4:1 for a high-effectiveness recuperator [8].

Figure 3 and **Figure 4** provide an output power-based specification envelope for fuel

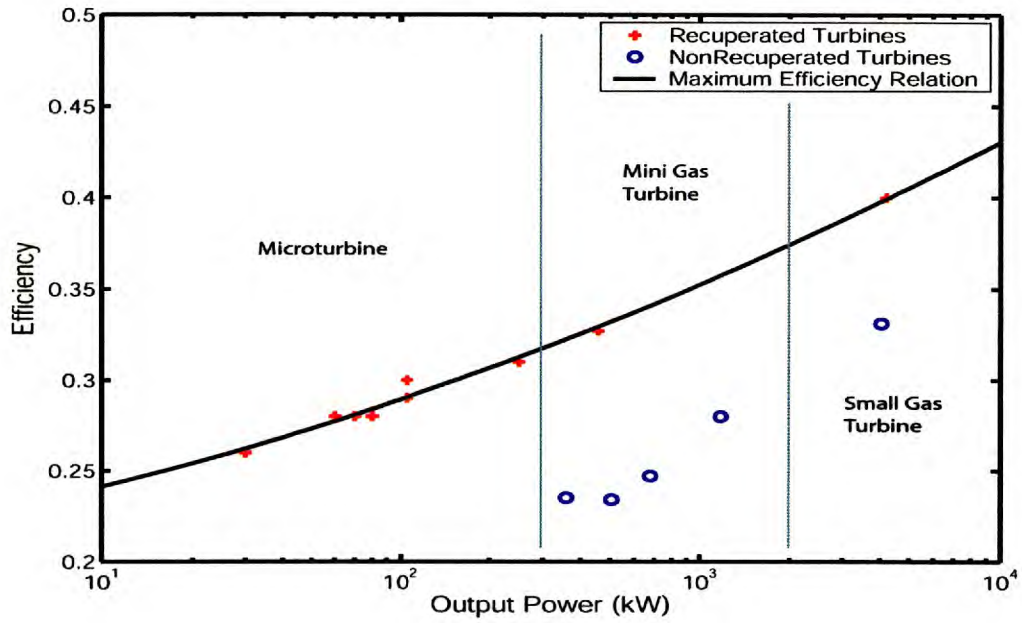


Figure 2: Efficiency vs. Gas Turbine Output Power

gas compressors in applications with low gas delivery pressures. These will be used later to show the difficulties in using conventional turbomachinery for fuel gas compression, the suitability of positive displacement compressors, and the potential of utilizing a multistage, forward-swept, turbo-compressor for natural gas fuelling in DG applications.

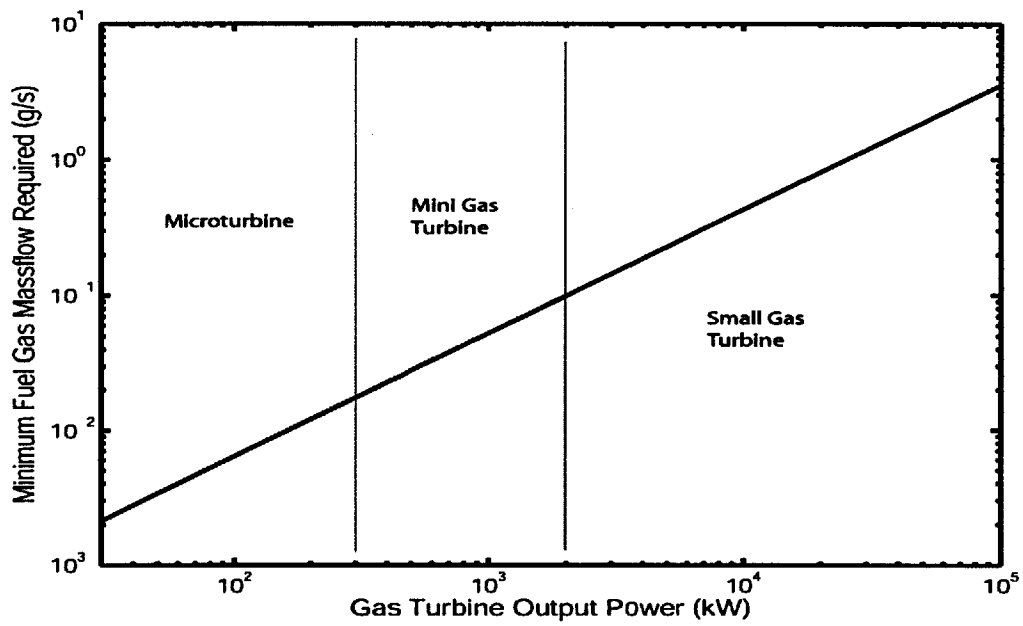


Figure 3: Minimum Fuel Mass Flow vs. Gas Turbine Output Power

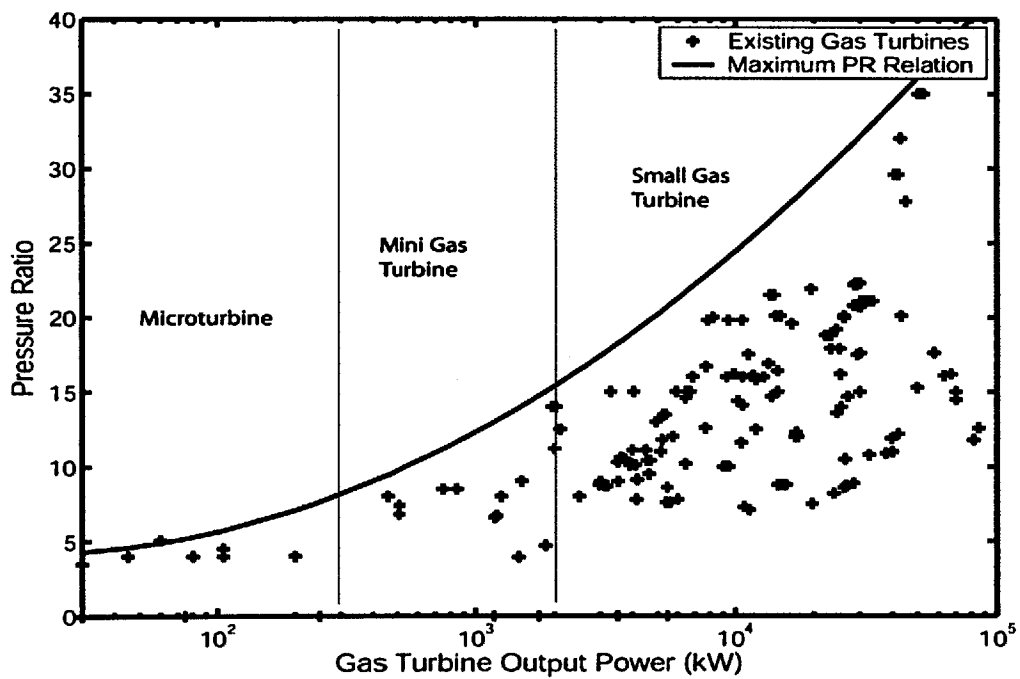


Figure 4: Pressure Ratio vs. Gas Turbine Output Power

1.4 Research Objectives

The motivation for the research completed was to provide a method of fuel gas compression for microturbines which is compact and reliable. Chapter 2 describes the currently available compressor types with the potential to meet the requirements set out in section 1.3. Existing solutions are found to have limitations that are detrimental to the application of microturbines in DG applications.

Section 2.6 introduces the forward-swept compressor, which can efficiently operate at the low-flow conditions required for the gaseous fuelling of microturbines. High blade loading, low diameter and narrow passages are incorporated into a shrouded, 2D compressor design. However, the limited pressure ratio of a single impeller requires a multistage arrangement to be implemented. The focus of the research is on the following two objectives:

- To accurately model a multistage implementation of the forward-swept compressor, generating design tools applicable to gas compressors for the range of microturbine operating powers.
- To develop a gas compressor which serves as both a prototype gas compressor for fuelling a microturbine of 100kW output power and as a test rig for simulation model validation.

From **Figure 3** and **Figure 4**, a gas turbine of 100kW would require approximately 6.5g/s (21SCFM) of fuel gas to be delivered at a pressure of > 4 Bar. This can be confirmed by examining the specifications of the Elliott Model TA-100 Microturbine Package, which runs at a pressure ratio of 4:1 and a quoted efficiency of 29%. This requires a flow of 22SCFM of natural gas for a generator output power of 105kW. The total package has an envelope of approximately 850mm x 3050mm x 2100mm including a separate fuel gas compressor. The microturbine has a single shaft running at 68000RPM. If the gas compressor were to run at the same speed as the microturbine, it could potentially be directly coupled, which would further aid implementation.

2 Compression Methods

The most common types of positive displacement and turbo compressors are shown on Figure 5.

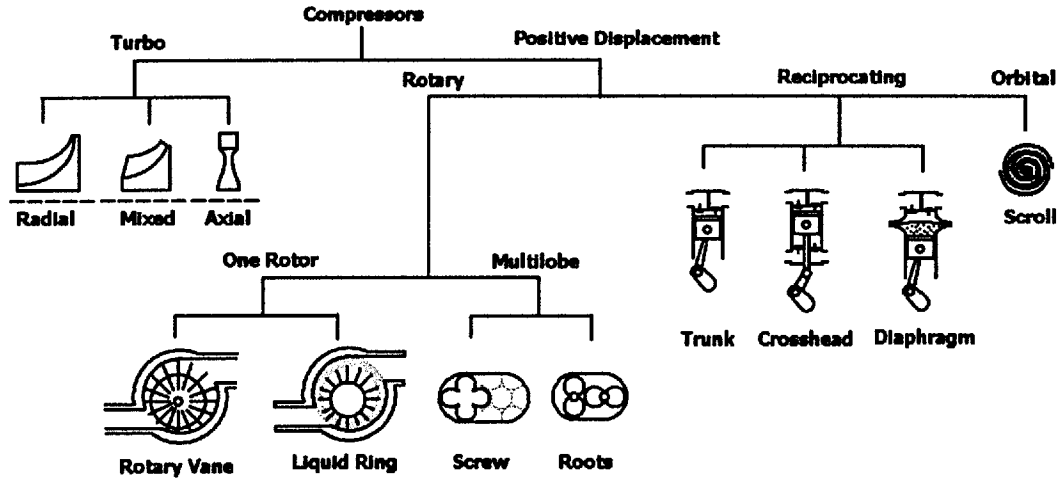


Figure 5: Compressor Types

The use of non-dimensional specific speed as a guide for selecting compressor type is investigated in this chapter. Along with the fuelling compressor targets established in the previous chapter, specific speed analysis shows positive displacement (PD) compressors to be the most suitable for microturbine fuelling. The various classes of PD compressor are cataloged and available examples are evaluated. The shortcomings of each class of positive displacement machine are identified. Conventional turbomachinery is ruled out as a feasible design option, and a new class of turbo-compressor is introduced, which will be used as the basis of a new multi-stage gas compressor.

2.1 Gas Compression Theory

The theory and terminology of gas compression is introduced here in order to extract suitable methods of compressor assessment. The behavior of gases at points well above their condensation temperature and well below their condensation pressure can be described by

the ideal gas law with reasonable accuracy.

$$Pv = RT \quad (1)$$

The gas constant, R , is specific to a particular gas mixture that satisfies the ideal gas criteria. It is known for all common gases and can be calculated based on the molar ratio for any gas mixture. The property of specific enthalpy, h is a measure of the available energy of a gas and is defined as:

$$h = u + Pv \quad (2)$$

where u is the internal energy of the gas. If equation 2 is differentiated with respect to temperature, the resulting equation is:

$$\frac{\delta h}{\delta T} = \frac{\delta u}{\delta T} + R \quad (3)$$

The coefficients c_v and c_p can be used to simplify this.

$$c_v = \frac{\delta u}{\delta T} \quad (4)$$

$$c_p = \frac{\delta h}{\delta T} \quad (5)$$

c_v and c_p vary with respect to temperature, although at temperatures well away from the point of condensation, they can be approximated as constant across a relatively small temperature range. c_v and c_p represent the energy required to raise a unit quantity of a gas one degree at constant volume and pressure, respectively. Combining **Equations 3, 4 & 5** produces:

$$c_p = c_v + R \quad (6)$$

γ , the ratio of the specific heats, can be interpreted as a measure of the ease with which a gas can be compressed.

$$\gamma = \frac{c_p}{c_v} \quad (7)$$

Because a gas heated at constant pressure expands, doing work on its surroundings, more energy is required to raise the temperature by a given amount compared to a gas at constant volume. Therefore c_v will always be less than c_p , and γ will always be 1 or greater. The definition of entropy, s , is given as:

$$\delta s = \frac{\delta q}{T} \quad (8)$$

Isentropic compression is reversible compression with no heat transfer to or from the gas. q is the heat transfer into the gas. Equation 2, can be differentiated to obtain:

$$\delta h = \delta u + P\delta v + v\delta P \quad (9)$$

The change in internal energy is equal to the heat transfer into the gas, minus the work done by the gas.

$$\delta u = \delta q - \delta w \quad (10)$$

For a steady-flow process, the work done by the gas is the product of the volume and the change in pressure:

$$\delta w = v\delta P \quad (11)$$

Using equations 2, 10 and 11 give the results:

$$\delta s = \frac{\delta h}{T} - \frac{P\delta v}{T} \quad (12)$$

Using equations 12 and 9 gives

$$\delta s = \frac{\delta u}{T} + \frac{v\delta P}{T} \quad (13)$$

Equations 12 and 13 can be integrated between two points, having properties P_1, v_1, T_1, S_1 and P_2, v_2, T_2, S_2 . Assuming constant specific heats, this gives

$$S_2 - S_1 = \int_{T_1}^{T_2} c_P \frac{dT}{T} - \int_{P_1}^{P_2} R \frac{dP}{P} \quad (14)$$

and

$$S_2 - S_1 = \int_{T_1}^{T_2} c_v \frac{dT}{T} + \int_{v_1}^{v_2} R \frac{dv}{v} \quad (15)$$

If the process is isentropic and $S_2 = S_1$, these can be resolved to equations 16,17, and 18. The starting point will still be designated by the subscript 1 but the final point will be designated 2s as it is only for an isentropic process:

$$\frac{P_{2s}}{P_1} = \left(\frac{v_1}{v_{2s}} \right)^\gamma \quad (16)$$

$$\frac{T_{2s}}{T_1} = \left(\frac{P_{2s}}{P_1} \right)^{\frac{\gamma-1}{\gamma}} \quad (17)$$

$$\frac{T_{2s}}{T_1} = \left(\frac{v_1}{v_{2s}} \right)^{\gamma-1} \quad (18)$$

From these constants, the energy required to isentropically compress a gas can be calculated.

$$w_s = \frac{\gamma R T_1}{\gamma - 1} \left[\left(\frac{P_{2s}}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] = \frac{\gamma R (T_{2s} - T_1)}{\gamma - 1} \quad (19)$$

Efficiency is a comparison of the process's energy consumption to an idealized process. As isentropic processes are without loss, the isentropic efficiency is used to compare the work input to an actual compression process (with the final state denoted by a subscript 2) to a theoretical, isentropic process with the same final pressure. :

$$\eta_s = \frac{w_s}{w} \quad (20)$$

If the actual compression is adiabatic, then the work input can be calculated from the inlet and outlet temperatures using **Equation 5**, assuming a constant c_p value between T_1 and T_2 :

$$w = c_p (T_2 - T_1) \quad (21)$$

Any evaluation of the isentropic efficiency of a non-adiabatic compression using **Equation 20** does not make fundamental sense, as it is possible to have an isentropic efficiency of over 100% for an intercooled compressor. Isentropic efficiency is, however, a useful evaluation of the energy required to compress gas to a given pressure ratio, as the isentropic efficiency will be inversely proportional to the power consumed for a given mass flow and pressure ratio. The application of equation 21 to any non-adiabatic flow is erroneous, as the equation is based on the assumption of adiabatic compression, and any heat lost to the surroundings will cause a falsely higher efficiency calculation.

The ideal compression, in terms of minimal energy input for a given pressure ratio, is isothermal compression. Although not physically possible, comparison to an isothermal process can be used to evaluate compression with intercooling. However, isothermal efficiency and isentropic efficiency are not comparable, and it is better to use only one efficiency definition when comparing machines to avoid confusion. As isentropic efficiency is generally used in the field of turbomachinery, and the heat of compression is useful to a gas turbine, isentropic efficiency will be used to assess the multistage forward-swept compressor.

In order to compare efficiencies between compressors of differing pressure ratios, it is useful to use the polytropic efficiency, sometimes called the 'small-stage' efficiency. It accounts for the fact that losses occur throughout the compression process, and applies the efficiency to the incremental process rather than the overall result. For an ideal gas of

constant heat capacity, it is defined by **Equation 22**

$$dh = \frac{1}{\eta_p} dh_s \quad (22)$$

Where dh_s is the incremental change in enthalpy for an isentropic process of equal incremental pressure change to the process with change in enthalpy dh . If it is assumed to be constant, it can be used to determine the efficiency of an incremental change in pressure, for a given overall pressure ratio and temperature rise.

$$\eta_p = \frac{\gamma - 1}{\gamma} \frac{\ln(p_2/p_1)}{\ln(T_2/T_1)} \quad (23)$$

A given temperature rise and pressure ratio corresponds to a given isentropic efficiency, and isentropic efficiency can be determined as a function of polytropic efficiency, pressure ratio and specific heat ratio.

$$\eta_s = \frac{PR^{(\gamma-1)/\gamma} - 1}{PR^{(\gamma-1)/\eta_p \gamma} - 1} \quad (24)$$

This relation is plotted for air in **Figure 6**. This is useful for comparing efficiency between compressors of different pressure ratio, or comparing multistage compressors to single stage compressors.

2.2 Specific Speed and Compressor Type Selection

Specific speed, N_s , is a dimensionless parameter that can be used as a guideline for selecting the type of compressor that is appropriate for a given application. It combines the effect of rotational speed, volume flow and specific work into a single 'shape factor'. It is introduced by Shepherd as part of a detailed introduction to dimensional analysis of turbomachines [10]. Specific speed is defined for compressors in **Equation 25**:

$$N_s = \omega \frac{\sqrt{Q}}{\Delta h_{0s}^{3/4}} \quad (25)$$

Or, for an ideal gas at constant c_p , it can be expressed using **Equations 26 & 27**

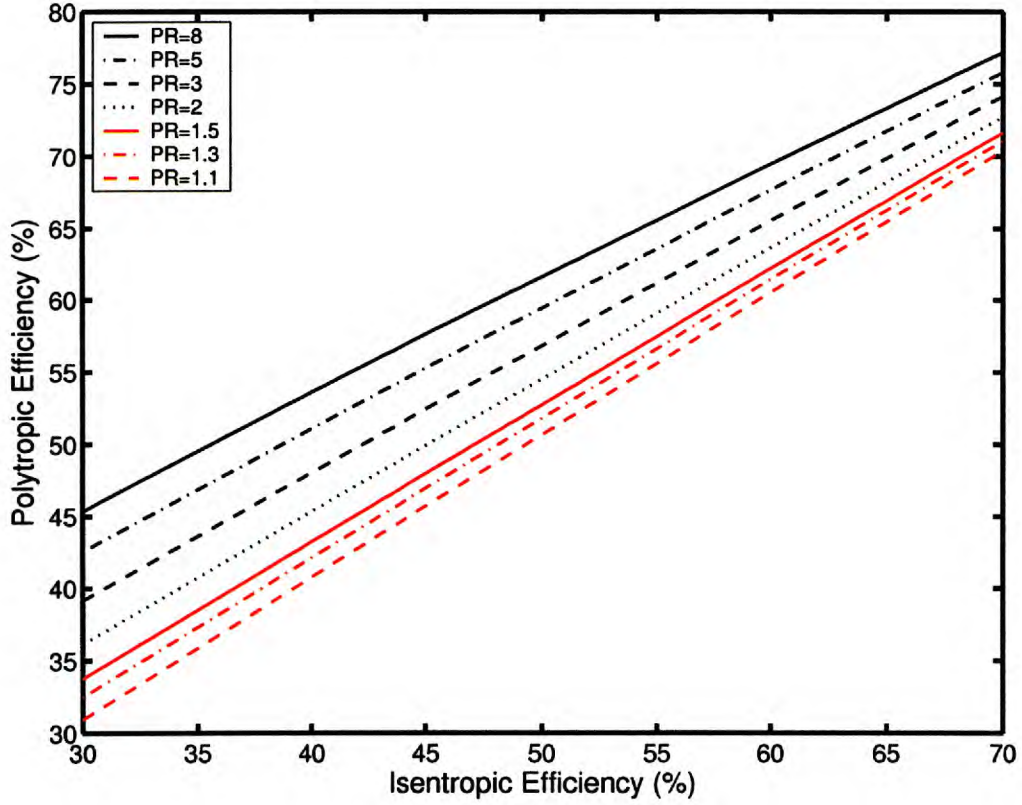


Figure 6: Polytropic Efficiency vs Isentropic Efficiency for Ideal Air ($\gamma = 1.4$)

$$N_s = \frac{\omega \sqrt{\frac{\dot{m}}{\rho_{inlet}}}}{\left(\frac{\gamma RT_1}{\gamma - 1} \left[PR^{\frac{\gamma-1}{\gamma}} - 1 \right] \right)^{.75}} \quad (26)$$

$$N_s = \omega \left(\frac{\dot{m}}{P_1} \right)^{\frac{1}{2}} \cdot (RT_1)^{-\frac{1}{4}} \cdot \left(\frac{\gamma}{\gamma - 1} \left[PR^{\frac{\gamma-1}{\gamma}} \right] \right)^{-\frac{3}{4}} \quad (27)$$

Figure 7, taken from [11] shows a graph of efficiencies of various compressors as a function of specific speed. This data is empirical and based on efficiencies achieved in practice. Similar graphs show performance for other Reynolds numbers, with efficiencies increasing with date of publication.

From the pressure ratio and mass flow rate targets shown in **Figure 3** and **Figure 4**, and **Equation 26**, the specific speed of the fuelling compressor can be determined at

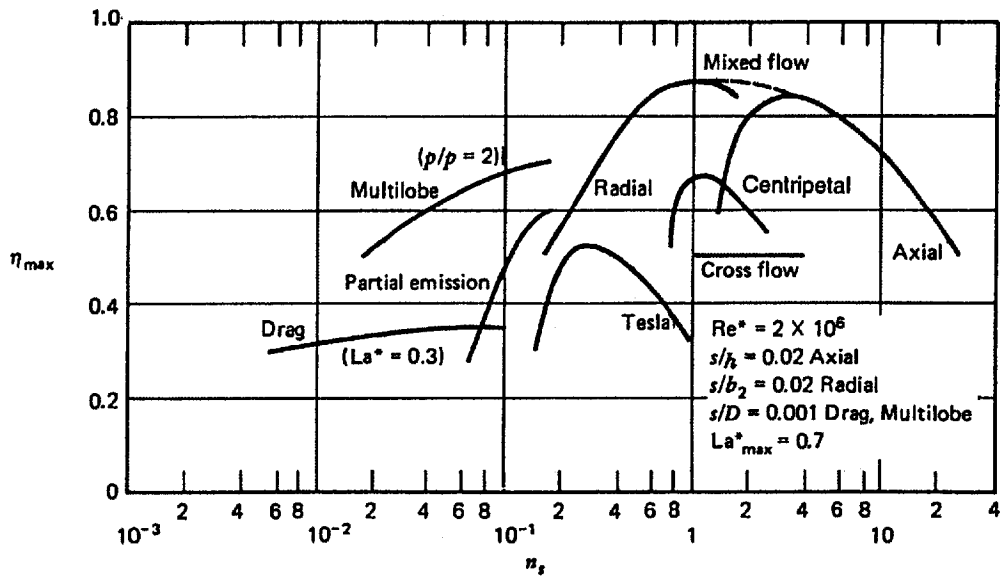


Figure 7: Compressor Efficiency vs. Specific Speed [11]

different shaft speeds. This is shown in **Figure 8**. The horizontal line represents the speed at which the optimum efficiency switches between positive displacement compressors and centrifugal compressors.

The minimum speed line (3000RPM) represents 2-pole synchronous speed motors. The maximum speed line (90,000RPM) represents the maximum speed of currently available microturbine generators. It is shown that a motor-driven single stage conventional turbomachine is not practical for fuelling a gas turbine engine of less than 10MW, when calculated from specific speed data, as $Ns = 0.2$ is considered the lowest feasible specific speed for conventional turbocompressors. Even so, a 10 MW engine would have such a high pressure ratio that a single stage would not stand the temperature generated from compression. It is therefore not surprising that all of the methods of fuel gas compression currently available are of the PD type.

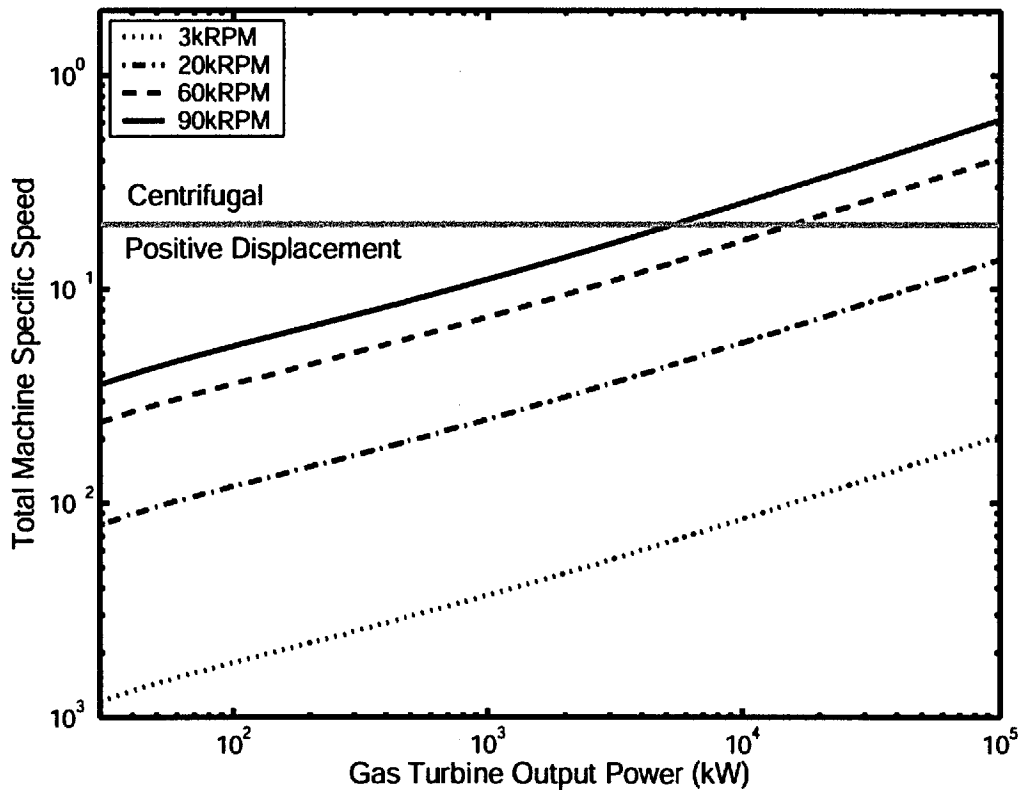


Figure 8: Gas Compressor Specific Speed at Various Shaft Speeds and Engine Outputs

2.3 Positive Displacement Compressors

For most positive displacement machines, flow is proportional to shaft speed across the range of efficient operation. However, as mass flow is reduced, leakage and mechanical losses increasingly affect efficiency and pressure ratio. A built-in pressure ratio of positive displacement machines is determined by machine geometry. Deviation from the built-in pressure ratio will affect efficiency, however, many types of machines have methods for varying the built-in pressure ratio.

Most empirical equations and simulation tools are only applicable to machines over 100hp (74kW), so any quantification of performance must be based on actual machines. When oil lubrication or ingestion is an option, both oiled and oil-free machines have been studied. The market survey applies to natural gas compression unless otherwise stated. For machine types that are not in natural gas applications at the relevant mass-flow and

pressure ratio, information on air compressors with similar mass-flow and pressure ratings have been used.

Because PD compressors undergo a significant amount of cooling during compression, it is not possible to determine the isentropic efficiency from the inlet and outlet temperatures. The isentropic efficiency is calculated from the power consumed during operation, or, if power consumption data is not available, it is calculated from the power rating of the motor, however this will provide an efficiency lower than the actual value.

2.3.1 Reciprocating Compressors

Reciprocating compressors work in the same way as the intake and compression stroke of a 4-stroke reciprocating engine. The valves are typically pressure activated, which requires no crank-driven timing mechanism. This action of suction and compression is shown in **Figure 9**.

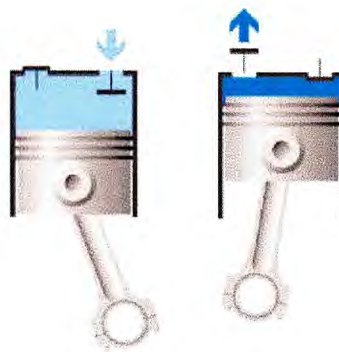


Figure 9: Piston Compressor Suction and Compression: Atlas Copco [12]

The compression that occurs in the piston is very efficient from a fluid standpoint, but inefficiency comes from the mechanical losses of the crankshaft friction and in the sliding of the piston, and from the aerodynamic losses in the necessary inlet and outlet valves. A 5% reduction in efficiency is typical for a non-lubricated compressor using PTFE or carbon rings, however, this difference can be acceptable because the non-lubricated compressor

does not require oil separation. While reciprocating piston compressors deliver a low-cost and efficient design, the nature of the design leads to large size and high maintenance, they typically have three times the maintenance costs of centrifugal compressors, and one can assume a proportional amount of down-time [13].

There are 3 categories of reciprocating piston compressor shown in **Figure 5**. Trunk compressors are similar to the pistons in ordinary automobile engines, as the piston walls guide the piston in a linear direction. They are simple and low cost, but if a large stroke to bore ratio is present, the connecting rod can hit the piston walls. The addition of a cross-head solves this problem, by having the linear guide farther away from the piston walls. A crosshead also provides the opportunity to implement a double-acting arrangement, which compresses working fluid on both sides of the piston, potentially doubling the output of the compressor. With the low flow requirements of interest, only single acting pistons will be investigated. Diaphragm compressors isolate the working fluid from the compressor with a flexible membrane. A liquid, usually oil or water, fills the space between the membrane and the piston. Diaphragm compressors are used to compress gases that are especially volatile, corrosive, dangerous, or of an application where oil contamination and piston blow-by must both be eliminated, however, this is not necessary for natural gas compression. In many piston compressors, the pressure ratio is adjustable by introducing a variable volume clearance pocket as shown in **Figure 10**.

Ancillary Equipment: Attenuation equipment is used to reduce the amplitude of compression pulses, which are present due to the transient nature of the compression, especially with only one cylinder. While initially claimed to improve the economy of operation of a reciprocating compressor, the opposite is usually true. However, when fuelling a steady state machine, it is necessary to reduce cyclic pressure fluctuations as much as possible. The piping used in a piston compressor is usually 0.4-0.6 times the diameter of the piston diameter and 8-10 diameters in length, causing some attenuation at a typical loss of 0.2% of the total system pressure. The surge bottle, which is attached to this provides additional attenuation and pressure drop. The ratio of its volume to the

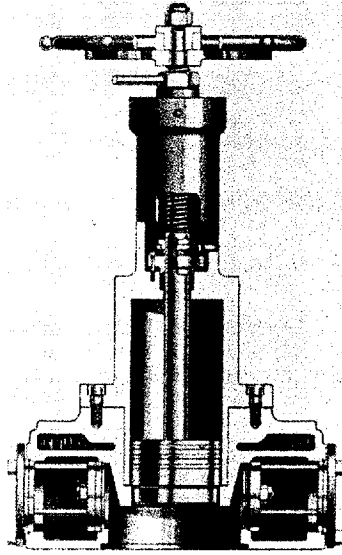


Figure 10: Adjustable Piston Clearance, Scheel, 1972 [13]

volume of a single acting compressor displacement is known as the attenuation quotient. A more proprietary system of pulse-damper can involve two or more surge bottles in series, with an increased pressure loss. It can be shown from **Figure 11**, taken from Scheel [13] that as more attenuation is required, more power is required by the compressor. This attenuation would likely be required in gas turbine fuelling in order to achieve constant fuel flow.

Analysis: Ladommatos [14] performed a very detailed investigation of the efficiency of small piston compressors using air or refrigerant. His findings are that small air compressors have mechanical efficiencies in the range of 86-92% and isentropic efficiencies of compression in the range of 72-85% for air compressors in the range of 0.18 kW to 100 kW. This corresponds to a total efficiency of 62-78%.

Scheel [13] has a complete method for the modelling of piston compressors, but it has been applied mostly to larger compressors than those of interest. Scheel states that the mechanical efficiency can be determined from the cylinder horsepower using **Equation 28**.

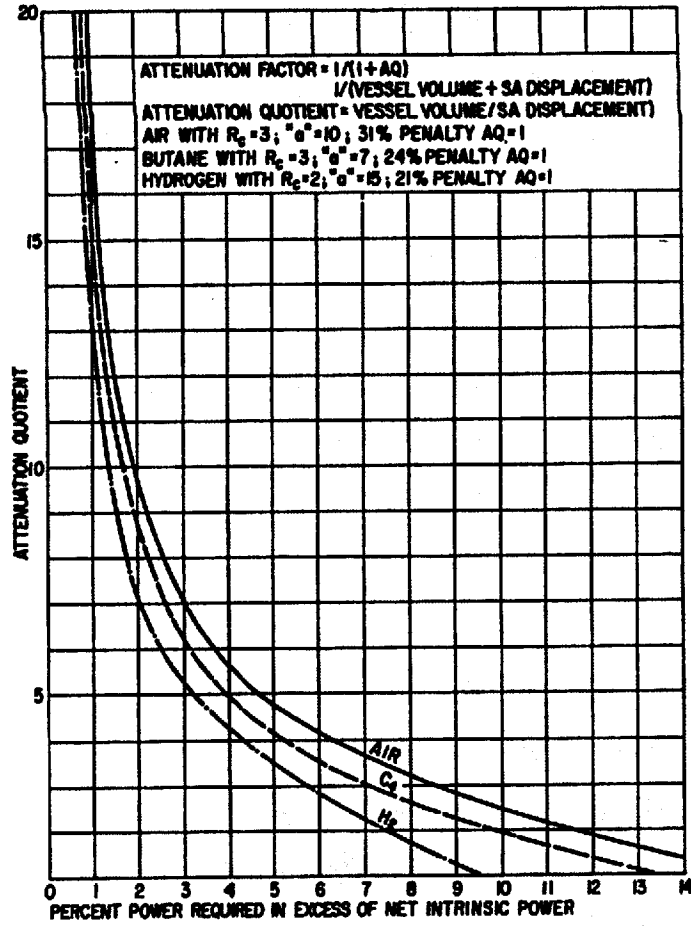


Figure 11: Power Consumption required for Attenuation, Scheel, 1972 [13]

$$\eta_m = 1 - \sqrt{\frac{1.341 kW}{\dot{W}}} \quad (28)$$

While this is a fairly simple equation, it is totally empirical with no fundamental basis. It's validity has been confirmed in the 134kW to 1340kW range, but it suggests that the efficiency will drop to zero as the power approaches 1.34kW. This is clearly inaccurate, as Davis[15] observed a 7 kW air compressor with a mechanical efficiency of 90%, which is the same efficiency given by equation 28 at 134kW, the bottom end of it's known validity. The isentropic efficiency of the compression is heavily dependent on the type and quality of valve and drive used, so it is therefore more prudent to rely on the findings of Ladammatos, which are from known machines of the range of interest.

Market: No oiled reciprocating natural gas compressors that produce pressure and flow in the region of interest exist, as most are of much higher pressure and flow. However, reciprocating air compressors for use as shop air supplies are widely available at the pressure and mass flow required. These machines are designed to be inexpensive and intermittently run, so the length of their service life is questionable and efficiency is generally not quoted, as it is of lower importance than initial cost. Relevant air compressors produced by Curtis Toledo are presented in Table 4. Efficiency calculations are performed based on the motor rating and will be lower than those achieved in practice.

Flowrate	Rated Power	PR	Isentropic efficiency	Lubrication
11g/s	3.8kW	8.5	.72	Oiled
21g/s	7.5kW	11.9	.85	Oiled
15g/s	5.7kW	6.8	.56	Non-Oiled
19g/s	7.5kW	6.8	.54	Non-Oiled

Table 4: Curtis-Toledo Reciprocating Air Compressors

2.3.2 Helical Screw Compressors

Screw compressors are a type of positive displacement machine that use two counter rotating meshing helical rotors, which trap the air in a passage and then 'screw' it into a smaller one. Figure 12 shows the process of suction (A), isolation (B), compression (C) and discharge (D).

occurs when the passage is open to the inlet port and the screw passage is at it's greatest volume. As the rotors mesh on the inlet side of the compressor, they isolate the passage from the inlet port. As the volume decreases, compression occurs, until the passage is exposed to the outlet port. A built-in pressure ratio is determined by the geometry of the rotors as well as the inlet and outlet ports, which are shown as cross-sections superimposed on each other in Figure 13. The larger the outlet port, the sooner the passage will be discharged and the lower the pressure ratio will be.

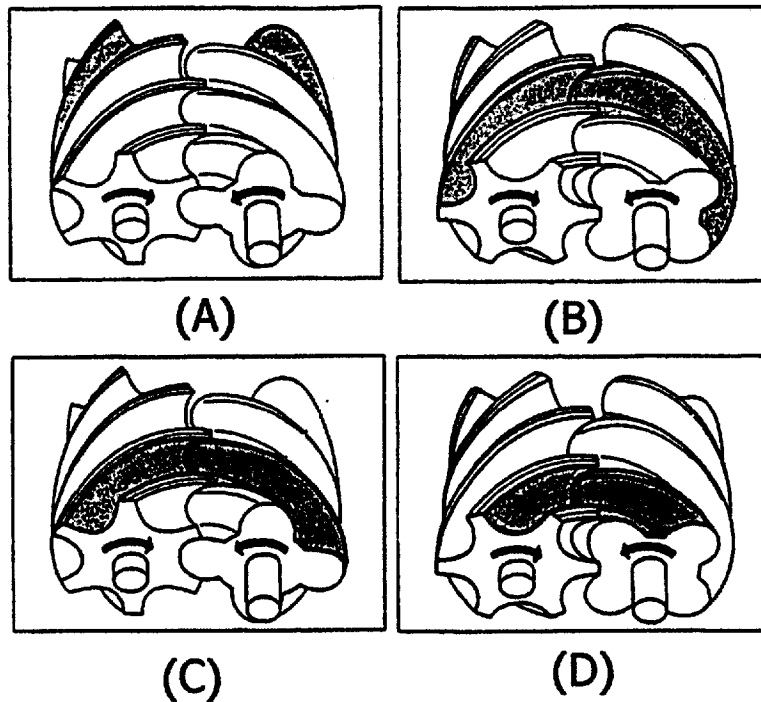


Figure 12: Screw Compressor Process [16]

Screw compressors can have the highest tip speed of all of the positive displacement machines, giving them a high power density. Furthermore, to directly quote [13]:

The helical screw compressor can develop up to 7.3 times the head of a centrifugal compressor operating at the same speed. This low specific speed characteristic is highly desirable for compressing small volumes of gas to high pressures. The helical screw compressor has performance characteristics and flexibilities that approach the piston compressor.

Oil-Flooding: Screw compressors are able to tolerate a much larger liquid content in the fluid than most other compressors. This allows oil to be sprayed into the compression chamber to aid in cooling and lubrication. As a side note, this property also allows refrigerant to be compressed without problems being caused by the flashing of liquid refrigerant from it's gaseous state upon compression. Both oiled and oil-free compressors are used depending upon the application.

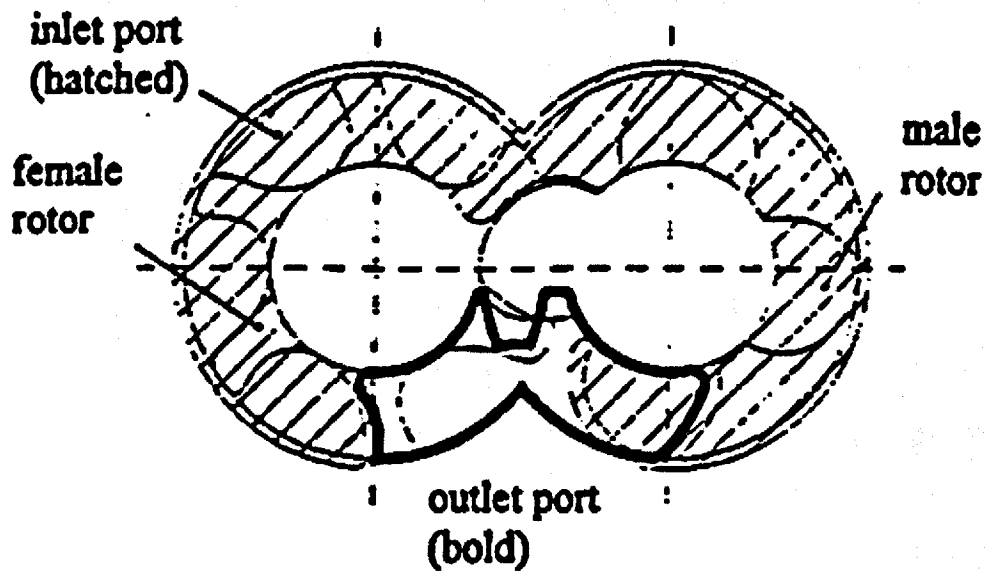


Figure 13: Screw Compressor Inlet and Outlet [16]

Oil-free compressors require a timing gear to keep the rotors from touching. This still requires lubrication oil, but the bearings, timing gear and any other sources of oil contamination are kept outside of the compression region. The compression area still requires cooling, usually in the form of a water jacket around the casing. If the discharge temperature is above 150°C , then liquid-cooling of the rotor through the bore is advisable, and achieved through the use of oil in natural gas applications.

Tight clearances of $0.025 - 0.2\text{mm}$ are required for efficient operation of oil-free screw compressors. The thrust bearing is placed on the discharge end of the compressor, where the tight clearances are more critical. Only oil-free compressors produce truly oil-free air of the kind required by the food and medical industries. Some oil-free compressors are given an abradable coating which allows rubbing during the initial use of the machine, wearing away to a very close gap.

There are many reasons why oil-injection is beneficial:

- The oil cools the air as it is compressed, reducing the outlet temperature and decreasing the motor power consumption by intercooling the gas.
- The presence of oil aids in the sealing between passages improving efficiency by reducing leakage, which also reduces the high frequency bypass noise [13].
- The clearances don't have to be as low, which means that lower precision machining methods can be used.
- Oil flooding allows one rotor to drive the other, eliminating the need for a timing gear.

However, oil injection causes some complications, which include:

- Although the power requirements of the compressor are reduced, intercooling may have a negative effect on the cycle efficiency of a gas-fuelled engine, due to the lost thermal energy.
- The presence of oil increases the viscous friction of the machine, which reduces the mechanical efficiency.
- The removal of oil after compression requires a separator, which often requires aftercooling.
- Driving the one rotor with the other affects the design and requires the rotors to be more 'gear-like'

Mineral oil is the most common flooding fluid in methane compression. The dependence of achievable gas purity on equipment, taken from [17], is shown in Table 5. The cooling of gas before separation is necessary to achieve maximum purity because the oil in the vapor phase is difficult to remove. Removing oil from the fuel stream is crucial for keeping emissions down.

Method	Purity (ppm)
Gravity Separation through velocity reduction	3000
Impingement Separation	35
Coalescing Filter	5
Cooling of Gas before Filtration	0.003

Table 5: Achievable Purity vs. Filtration Method

Off-Design, Driving and Control: As mentioned before, running above or below the built-in pressure will have a detrimental effect on efficiency. **Figure 14** depicts the underlying principle behind the power loss associated with compressing gas to a pressure other than the built in compression ratio. In **Figure 14A**, the internal compression matches the delivery pressure, resulting in efficient compression. In **Figure 14B** the compression done by the rotors is insufficient and the energy in the shaded portion of the plot is wasted because the isolated gas is exposed to a higher pressure before it is finished compressing, and compression is performed at constant volume. In **Figure 14C** the overcompressed gas is expanded irreversibly down to the delivery pressure, and the shaded portion of the plot again shows the wasted energy. **Figure 15** shows the effect of these scenarios on efficiency. It is observed that small deviations in pressure ratio have a negligible effect on efficiency, but large deviations have a much larger effect, especially in the event of under-compression. It can be seen that obtaining the correct specification of screw compressor for the application is essential. Some screw compressors are fitted with a slide valve, which effectively allows control over the port area.

The two methods of driving a screw compressor are direct-drive and geared. As oil-free compressors require a timing gear, a step up gear is not a difficult component to add. Oil-lubricated designs, where one rotor drives the other, can still benefit from a step-up gear from the motor. Increasing the speed of a compressor reduces the size necessary for a given output. It is also beneficial to use a higher speed motor, as they too are more compact, although do not operate at synchronous speeds. This requires an inverter, which is necessary for variable speed, but not necessary for fixed, synchronous speed operation. Speed control allows efficient off-design operation down to 10% of the maximum flowrate.

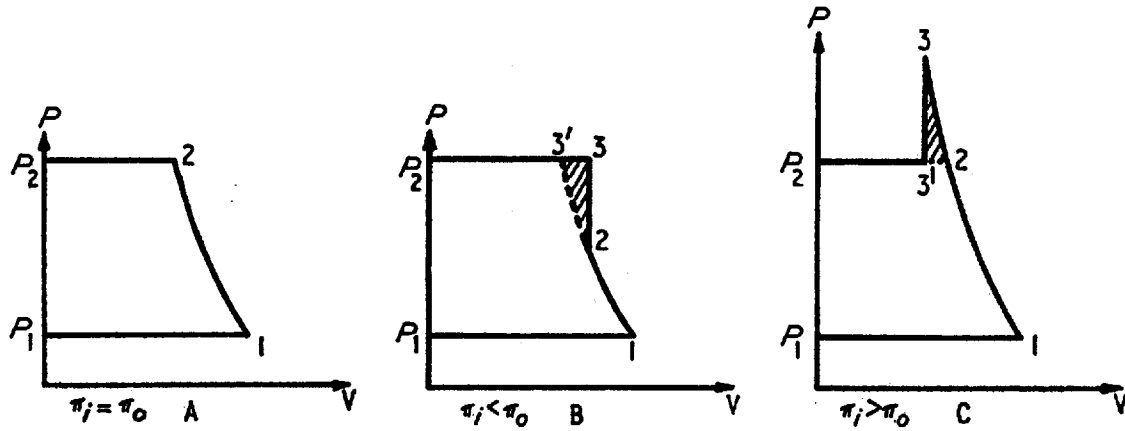


Figure 14: Effect of Built-In Compression(A), Under-Compression(B), and Over-Compression(C) [13]

Below this flowrate, leakage generally dominates the compression. If speed control and slide valve load control are not used, then mass flow and output pressure must be regulated by throttling the inlet or venting the gas to the atmosphere.

Screw Compressor Market: Screw compressors are not normally designed for gas compression at the lower end of the mass flow range of interest but many air compressors are. A 105 g/s gas compression unit is produced by Mycom Canada. It produces an adjustable pressure ratio of 2.62 to 5.8 and occupies 1220mm x 560mm x 510mm without a motor [18]. No efficiency values or power consumption data is available.

Figure 16 and Figure 17 show the efficiency and pressure ratio of three models of the KS-series of Curtis-Toledo oil-flooded screw compressors for air compression. These are designed for continuous operation and efficiency. The efficiency is calculated from the power of the motor and the quoted pressure and mass flow [19]. The dimensions of these three compressors in a packaged product with motor are shown in Table 6.

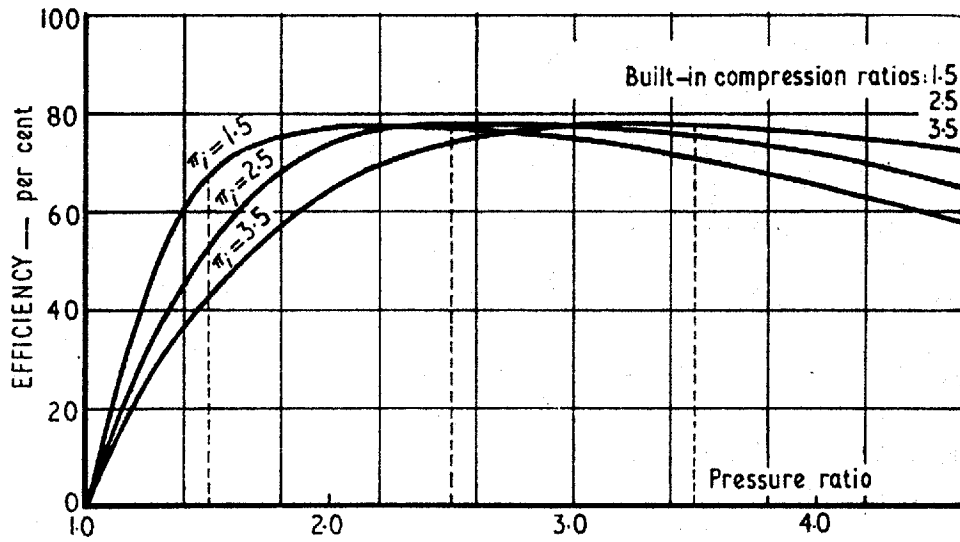


Figure 15: Effect of Built-In Compression Ratio on Efficiency, Scheel, 1972 [13]

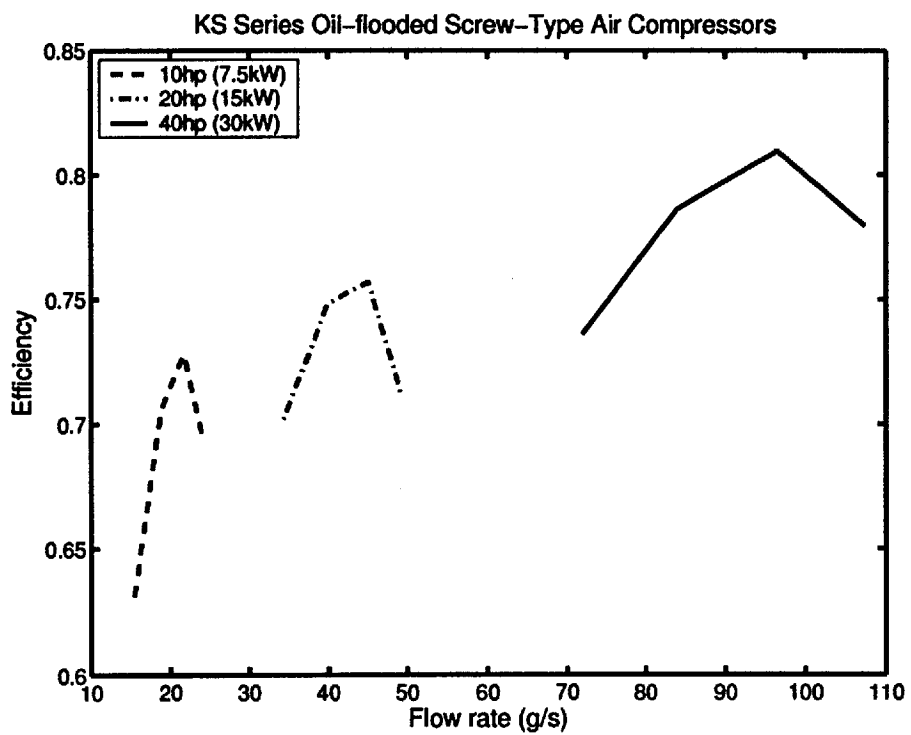


Figure 16: Curtis-Toledo Oil-Flooded Screw Compressor Isentropic Efficiency [19]

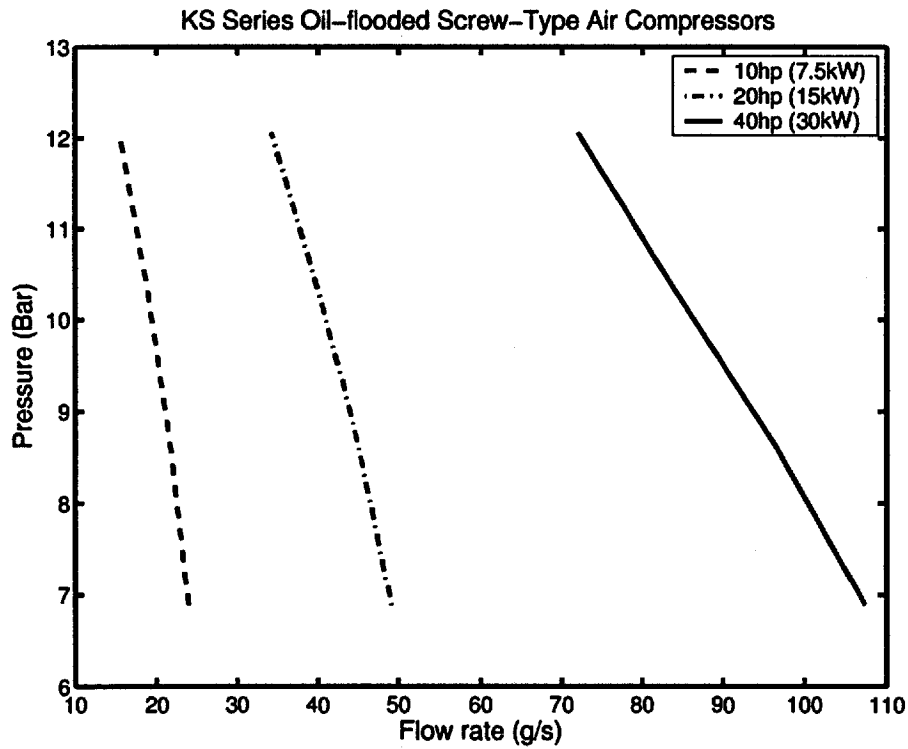


Figure 17: Curtis-Toledo Oil-Flooded Screw Compressor Outlet Pressure [19]

Model Name	Nominal Power (kW)	Mass (kg)	Dimensions (mm x mm x mm)
KS-10	7.5	159	686 x 686 x 864
KS-20	15	209	711 x 787 x 1066
KS-40	30	441	1041 x 864 x 1321

Table 6: Curtis-Toledo Oil-Flooded Screw Compressor Dimensions

2.3.3 Roots Blowers

Roots blowers use straight lobes and have a built-in volume ratio of 1:1. They achieve low pressure ratios and are inefficient because they operate using under-compression and the pressure rise comes from the resistance at the outlet. They do not produce the pressure ratio needed for most compressor applications, but are more typically used as blowers, meters, and vacuum pumps. They are mentioned here only for completeness.

2.3.4 Rotary Vane and Liquid Ring Compressors

Rotary vane compressors (also called sliding vane compressors) are positive displacement rotary machines that use a rotor mounted in an eccentric casing. Rotor-mounted vanes are free to slide radially, and are usually made from laminated phenolic resins. The sliding vanes form the walls of a control volume, while the casing holds the vanes at a specific radial position as a function of angle. The volume decreases as the gas is moved from inlet to outlet, thereby compressing the gas. The life of the vanes is heavily dependant on proper lubrication. Rotary vane compressors are generally oil-flooded due to lubrication requirements, which brings the same advantages and problems present in oil-flooded screw compressors. An illustration of a rotary vane compressor is shown in **Figure 18** [20].

A similar machine is the liquid-ring compressor. Instead of sealing between passages being done by the contact between vane and wall, a liquid ring is held against the eccentric casing by the rotor. **Figure 19** shows the operation of this principle.

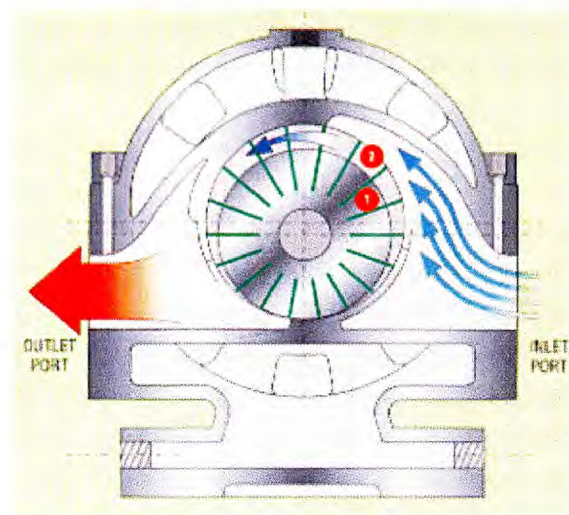


Figure 18: Rotary Vane Compressor, FLUIDAIR Sales Literature[20]

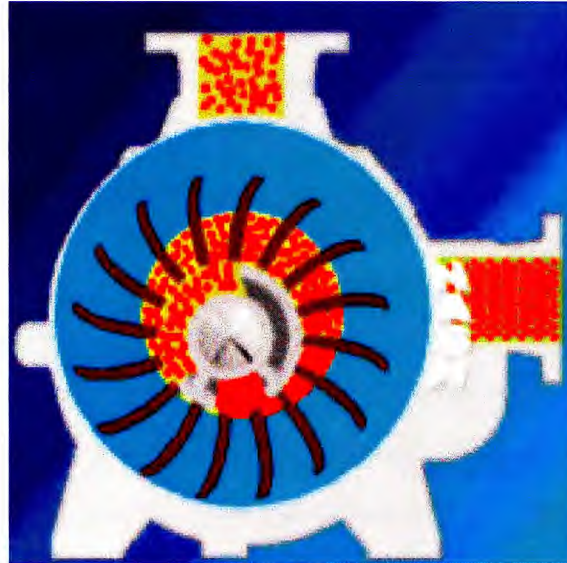


Figure 19: Liquid Ring Compressor, NASH Engineering, [21]

Rotary Vane Market: CompAir Ltd. has produced 4 models of sliding vane compressors for the purpose of micro-turbine fuel gas compression. CompAir's previous experience in sliding vane compressors is in air compression, and they have produced air compressors that have functioned for up to 100,000 hours using hydrodynamic bearings. They typically have 8000 hour service intervals, and have carried this over to their gas compressors without any problems. [22]

The specifications and performance of the machines currently produced are shown in **Table 7**. These are operating at 0.07 bar gauge inlet pressure, 6 bar outlet pressure. These are all available as fixed speed packages, and the two at 5.5kW and 7.5kW are available as variable speed packages. Delivery flow control of fixed speed operation is achieved by throttling the inlet while flow control of variable speed operation is done by varying the speed, achieving as little as 45 % of the nominal flow rate.

Liquid Ring Market: Nash Engineering produces a large line of liquid ring compressors, however all of the models with pressure ratios of interest operate at higher powers.

Power	Flow	PR	Efficiency	Mass	Approximate Cost
4kW	6.6g/s	5.6	0.53	155kg	£3000
5.5kW	10g/s	5.6	0.58	155kg	£4000
7.5kW	14g/s	5.6	0.60	155kg	£5000
18.5kW	32g/s	5.6	0.56	428kg	£6000

Table 7: Compar Sliding-Vane Natural Gas Compressor Details

Similarly all of the low-flow compressors operate at low pressure ratios. The model closest to the range of interests is one operating at 1750 RPM and at a nominal power of 38 kW. It is 430 kg with dimensions 1000x800x550mm without a motor or the required separator. **Table 8** shows it's performance with air as the working fluid. [23]

Power	Mass Flow	PR	Efficiency
32kW	58g/s	4	0.25
38kW	54g/s	5.5	0.26

Table 8: NASH Liquid Ring Compressors

2.3.5 Scroll Compressors

Scroll compressors use orbital motion rather than rotary or reciprocating action. One spiral or 'scroll' remains stationary while the other orbits. Gas is introduced in the outermost passage, and is compressed as the passage rotates inward. This is illustrated in **Figure 20**, adapted from [24]. The pink area shows the pocket of gas moving from the inlet at the periphery to the outlet at the center as the rotors orbit relative to each other in steps 1-4. Step 5 is illustrating the fact that all of the gas pockets are being compressed simultaneously.

Scroll compressors are particularly suited to low mass flows, however, the main drawback of the scroll compressor is its lack of robustness. Air easily leaks from one passage to the next unless very high tolerances are achieved. Such high tolerances are required that the rotors must be machined as a pair. A design incorporating a PTFE seal has been



Figure 20: Scroll Compressor Operation [24]

developed for vacuum applications by BOC Edwards.

Market Analysis: The scroll compressor sold by Copeland Corporation specifically for microturbine natural gas boosting runs at an inlet pressure of 1-2 bar and compresses gas to an outlet pressure of 5.1-7.8 bar. The power consumption curves from the product literature are shown in **Figure 21**. **Figure 22** shows machine efficiency based on this data and calculated isentropic power consumption, with units converted to SI.

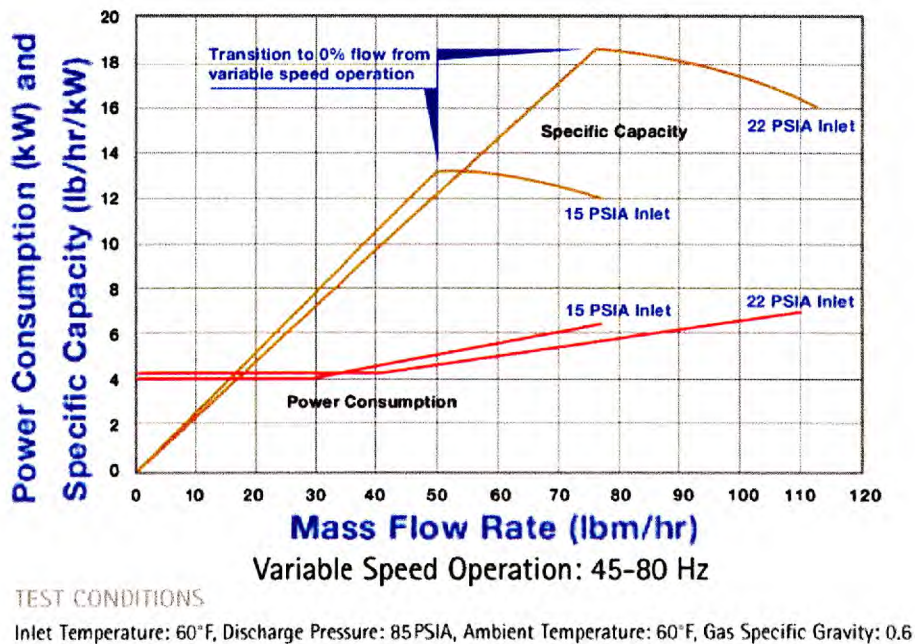


Figure 21: Scroll Compressor Power Consumption and Specific Capacity [24]

Copeland Corporation quotes a two year service life requiring a one-hour annual maintenance. The machine is similar to other positive displacement machines in terms of size

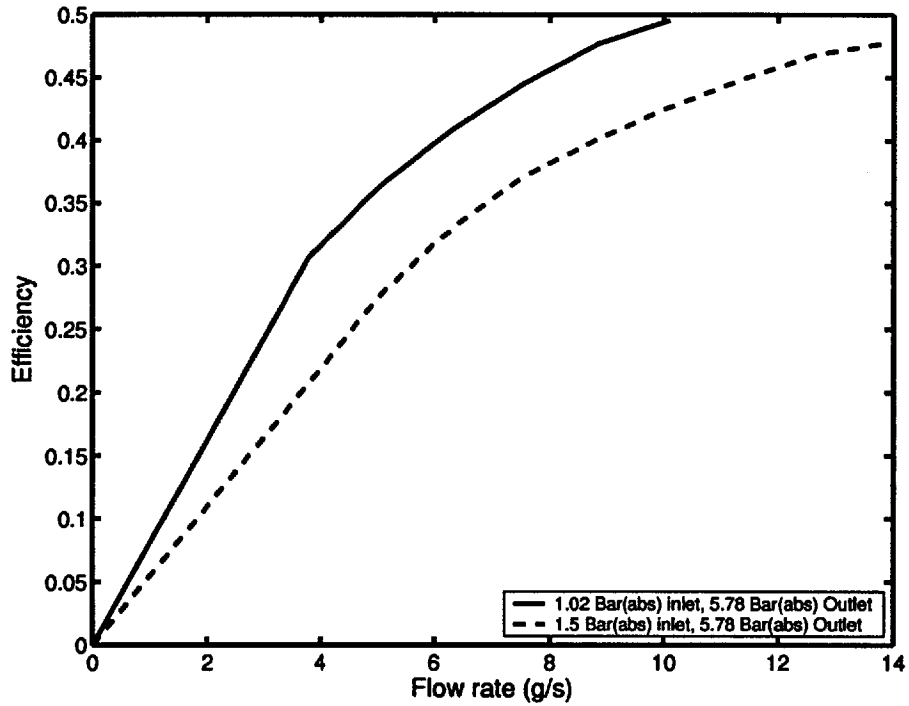


Figure 22: Scroll Compressor Efficiency, adapted from Figure 21

and weight. As only one model is produced, higher mass flow rates require that identical machines be used in parallel.

Power Consumption	Flow	Pressure Ratio	Efficiency
7.5kW	14g/s	5.7	0.50

Table 9: Copeland Scroll Compressor Performance

Mass	Size (Fully Packaged)	Cost
250kg	1m x 0.6m x 1.4m	£6800

Table 10: Copeland Scroll Compressor Specifications

2.3.6 Positive Displacement Conclusions

It has been found that the primary problem with using positive displacement machines for fuel gas boosting in microturbine applications is that their sizes approach that of the genset being fuelled, oil is often ingested, and maintenance is frequent. Frequent maintenance of the fuel compressor would negate the advantage of low maintenance in the microturbine. The most viable of methods are the screw, rotary vane, and scroll compressors. The liquid ring compressor is inferior to the similar rotary vane compressor in all aspects. The piston compressor is too large and high-maintenance to be viable and it will have the largest pulses, which will require additional equipment and may complicate the combustion process. It is therefore not surprising that screw, scroll and rotary vane compressors are the types that have been selected for use in this particular application.

Type	Flow Rate	PR	Efficiency	Size and Mass
Screw	20g/s	8.5	0.72	159kg occupying 686mm x 686mm x 864mm
Rotary Vane	10g/s	5.6	0.58	155kg
Scroll	14g/s	5.7	0.50	250kg occupying 1400mm x 1000mm x 600mm
Piston (oiled)	11g/s	8.5	0.72	259kg inc. 0.3m ³ tank
Piston (dry)	15g/s	6.8	0.56	unavailable
Liquid Ring	54g/s	5.5	0.26	430kg occupying 1000mm x 800mm x 550mm

Table 11: Positive Displacement Product Summary

2.4 Conventional Centrifugal Compressors

Despite the current use of PD technology for microturbine fuelling, a smaller compressor of similar efficiency is desirable. A gas compressor utilizing a conventional turbo compressor could potentially have high power density and reliability, but it has been shown that it would be impractical for compressing fuel gas for engines of less than 10MW output power. This section shows why a conventional centrifugal compressor is unsuitable for low specific speed operation.

2.4.1 Fundamentals of Centrifugal Compressors

The design techniques of conventional centrifugal compressors of the type shown in **Figure 23** are well established and can achieve isentropic efficiencies upwards of 86%, as was shown in **Figure 7**.

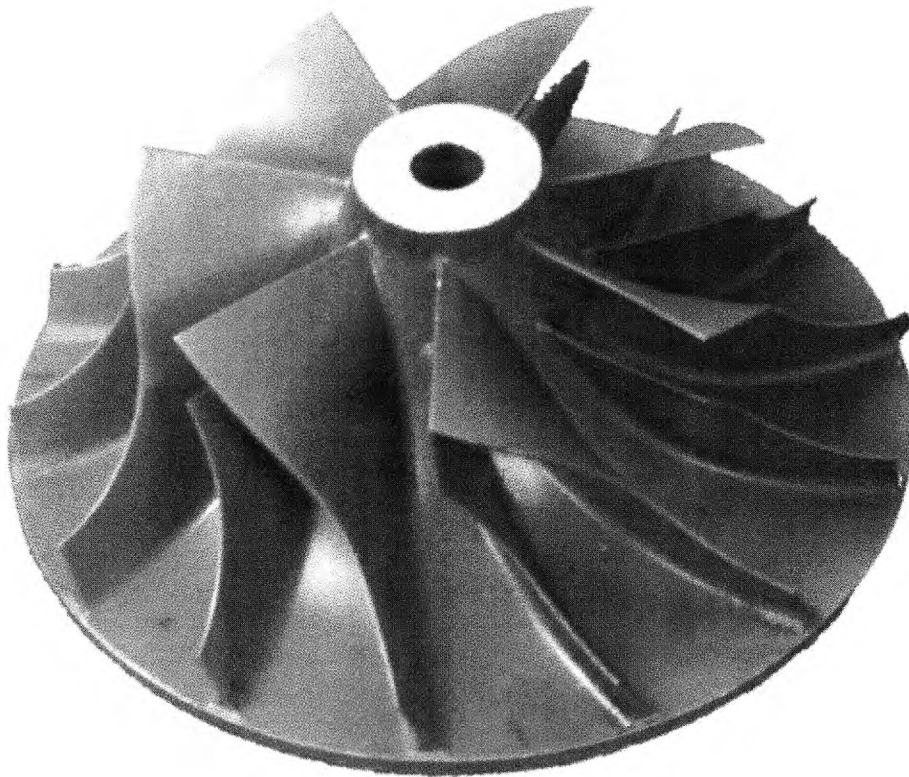


Figure 23: Conventional Centrifugal Compressor of Recent Design

In positive displacement compressors, the velocities of the gases involved are generally much lower than the sonic velocity, so the difference between static and total pressure is negligible. However, a turbo-compressor works on the principle of accelerating the gas to a high velocity and diffusing it to raise the pressure. Therefore, the gas velocities in turbomachines approach (and often exceed) sonic velocities, so it is important to quantify the kinetic contribution to total pressure. The sonic velocity, c , in a gas is determined by **Equation 29**.

$$c = \sqrt{RT\gamma} \quad (29)$$

The Mach number is the ratio of gas velocity, C , to sonic velocity.

$$Ma = \frac{|C|}{\sqrt{RT\gamma}} \quad (30)$$

It is easier to analyze temperature and pressure while including the contribution of kinetic energy to the total energy of the gas. The static pressure, P , is the pressure that would be measured while moving relative to the gas. The total pressure, P_0 , is the pressure that would be measured if the fluid was brought to rest isentropically. The total pressure is calculated by **Equation 31**

$$P_0 = P \left(1 + \frac{\gamma-1}{2} Ma^2 \right)^{\gamma/\gamma-1} \quad (31)$$

The total temperature, which is the temperature of a gas were it to be brought to rest is:

$$T_0 = T \left(1 + \frac{\gamma-1}{2} Ma^2 \right) \quad (32)$$

The impeller blade movement is represented by the vector U , the absolute gas velocity is represented by the vector C , and the velocity of the gas relative to the blade is represented by the vector W . The radial and tangential components are designated by the subscripts θ and r .

Figure 24 shows the velocity triangle of the outlet for a backswept compressor. Back-sweep is used because $C_{\theta 2}$ can be kept below the sonic velocity while increasing U_2 . This extends the range and reduces the PR for a given tip speed and rotor diameter.

The change in total enthalpy is a measure of change in the energy available, assuming a constant heat capacity over a small temperature range.

Some amount of diffusion or acceleration usually occurs in the rotor due to changing passage cross-section. This will affect the gas velocity, static pressure, and static temperature of the gas, but it will not affect the total temperature. An efficient compressor will achieve a high total pressure rise for a given work input.

2.4.2 Low Specific Speed Conventional Turbomachines

For turbomachinery, it is often useful to use the terms of head coefficient, Ψ , and flow coefficient, Φ , described in **Equations 37 & 38**. These can be related to specific speed through **Equation 39**. The difficulty in producing a centrifugal compressor of low specific speed is actually the combination of trying to produce a high head coefficient and low flow coefficient. As the flow coefficient is reduced, the losses of windage and leakage increase, as shown in **Figure 25** [25].

$$\Psi = \frac{\Delta h_o}{U_2^2} \quad (37)$$

$$\Phi = \frac{\dot{V}_{in}}{U_2 D_2^2} \quad (38)$$

$$N_s = \frac{2\sqrt{\Phi}}{\Psi^{3/4}} \quad (39)$$

Figure 25 shows the contribution of the various loss mechanisms as flow coefficient changes. The primary loss is that of aerodynamic loss due to low blade height. As the blade height is reduced, the pipe losses through the blades increase. Additionally, aerodynamic recirculations will increase, due to a higher ratio of tip clearance to blade height.

Additional problems associated with a low specific speed machine are that the typical parasitic losses of seal leakage and viscous friction on the back face of the impeller (the phenomenon known as 'windage') are proportionally more significant than in a conventional machine. Because the power and mass flow of a low specific speed compressor are

lower, the additional losses and leakages have a greater proportional effect. For example, a typical labyrinth seal with a pressure difference of 5 bar could have a leakage as high as 10g/s. This may not affect a conventional centrifugal compressor of optimal specific speed significantly, but a leakage of this order would be similar to the required output of the fuel gas compressor.

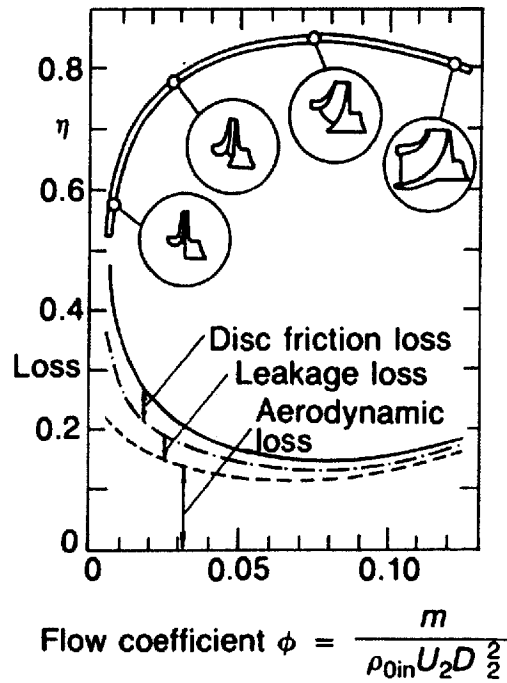


Figure 25: Effect of Flow Coefficient on Efficiency [26]

Casey et al. [27] give a comprehensive review of the design and testing of low flow coefficient, conventional, centrifugal compressors. The designs tested feature low blade heights, high back sweep, and high diameter.

2.5 Existing Low-Ns Turbomachines

There have been previous attempts at producing a turbo-compressor of low specific speed. The basic premise and theory is the same as conventional compressors, but innovative geometry and implementation have allowed a reduction in specific speed. These are cov-

ered in [29], [30], and [31], but are summarized here.

2.5.1 Regenerative Axial Compressor

The regenerative axial compressor uses an axial compressor structure with a toroidal flow path. This is done to force the air through the blades of one rotor in series, rather than in parallel, theoretically reducing throughput and increasing pressure ratio. However, this causes the compression to suffer due to leakages and poor range due to high incidence. This machine is also known as the 'Rotary Periflow Compressor', manufactured by Burton Corblin, and is shown in **Figure 26**. A peak isothermal efficiency of 58% at a PR of 1.2 and a peak PR of 2.0 at 35% efficiency is reported in [29].

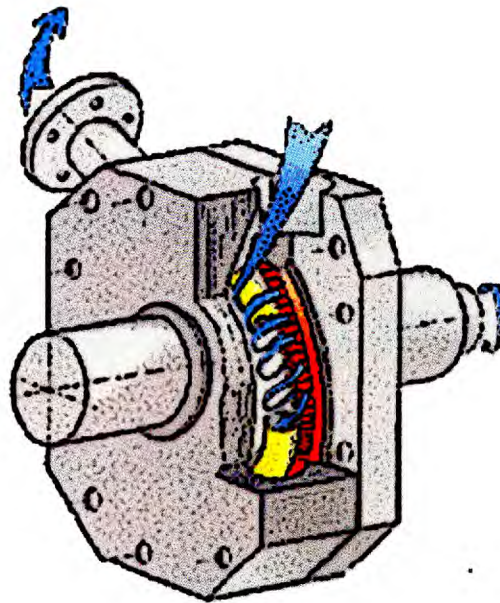


Figure 26: Periflow Compressor, [32]

2.5.2 Partial Emission Compressor

The partial emission compressor uses highly scalloped radial blades and blanking plates at the inlet and outlet, effectively allowing only a fraction of the compressor to do work at any one time. Although a pressure ratio of 2.38 was achieved, the efficiency was poor

at 34%. The specific speed was 0.15. The design is shown in **Figure 27**.

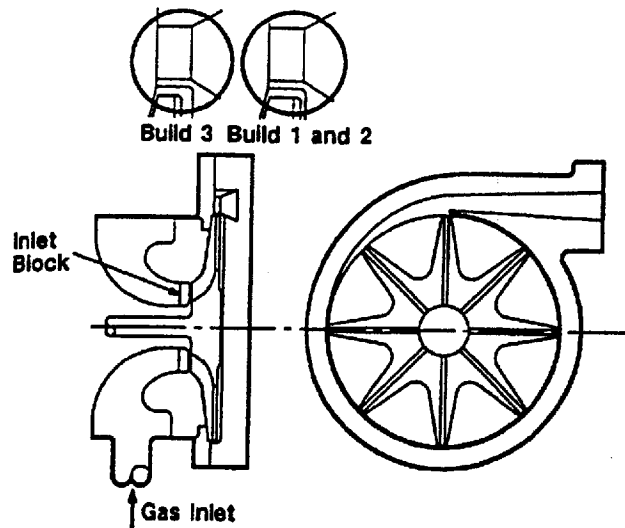


Figure 27: Partial Emission Compressor [30]

2.5.3 Wedge Compressor

A wedge compressor uses very thick blades and a large amount of backsweep. At the outlet, only 70% of the circumference consists of passage, which presumably causes large blade wakes. In addition, the high level of backsweep forces the design to be of large diameter, causing high windage losses. This design is shown in **Figure 28**. An efficiency of 60% was quoted at a specific speed of 0.29 and a pressure ratio of 1.37 [31].

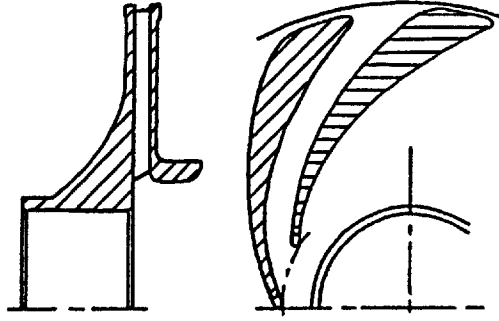


Figure 28: Wedge-Type Compressor [31]

2.6 Forward Swept Compressors

The present section introduces technology and development made by Vine in the aerodynamics of forward-swept compressors [33], [34], [35]. The blade profile, CFD, and single stage testing are all the work of Vine. The blade profile is used as a basis for designing the multistage arrangement described in Chapter 3. The CFD results and test data described in this chapter were utilized within a multistage simulation routine, as described in Chapter 4.

2.6.1 Flow Stability of Forward-Swept Compressors

Figure 29 shows how the head coefficient and flow coefficient are related at various blade angles. While increasing the backsweep of a compressor at a given tip speed reduces the head coefficient, forward sweep allows a high head coefficient at a given flow coefficient. Forward sweep is less stable than backsweep, due to a positive pressure slope, and is not used conventionally. However, at low specific speed the frictional losses will be large, and these losses will maintain a negative pressure slope, which will aide in flow stability.

2.6.2 Research Background

This compressor development began with a rotor that was essentially a shrouded, 2D radial compressor, with many of the rotor passages being 'blanked off'. Gradual development steps have led to a machine of relatively low tip speed, high blade solidity and high forward

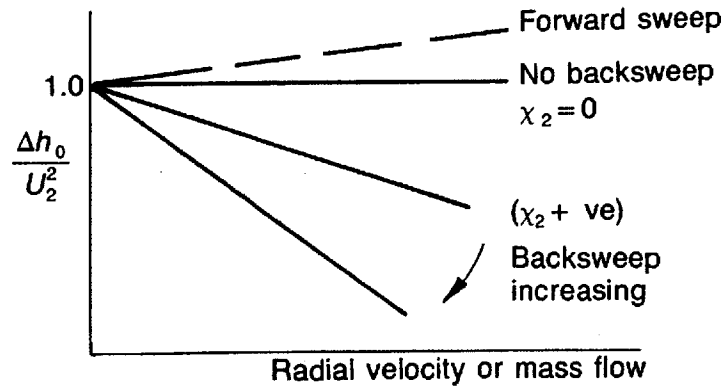


Figure 29: Effect of Blade Angle on Head Coefficient [28]

sweep, specifically designed for low specific speed applications. An inducer, as found on a conventional 3D rotor, is not needed due to the low inlet Mach number. A small vaneless space is used to achieve flow stability. The forward-swept design is shown along with the diffuser in **Figure 30**. A new compressor design methodology based on high forward sweep, high solidity, low diameter, and low vaneless space has been developed. Although a high solidity is present, the blades are thin at the rotor outlet, which prevents large jet and wake effects.

This high level of forward sweep allows more turning to occur, and therefore more specific work at a lower diameter. The forward sweep also causes the passages to be much smaller. The constant passage width reduces the likelihood of flow separation caused by the high degree of turning within the rotor. The velocity at rotor exit is close to sonic, but the mass flow rate remains low because most of the velocity is tangential rather than radial. The features of this design would not be possible without the inclusion of a large degree of forward sweep.

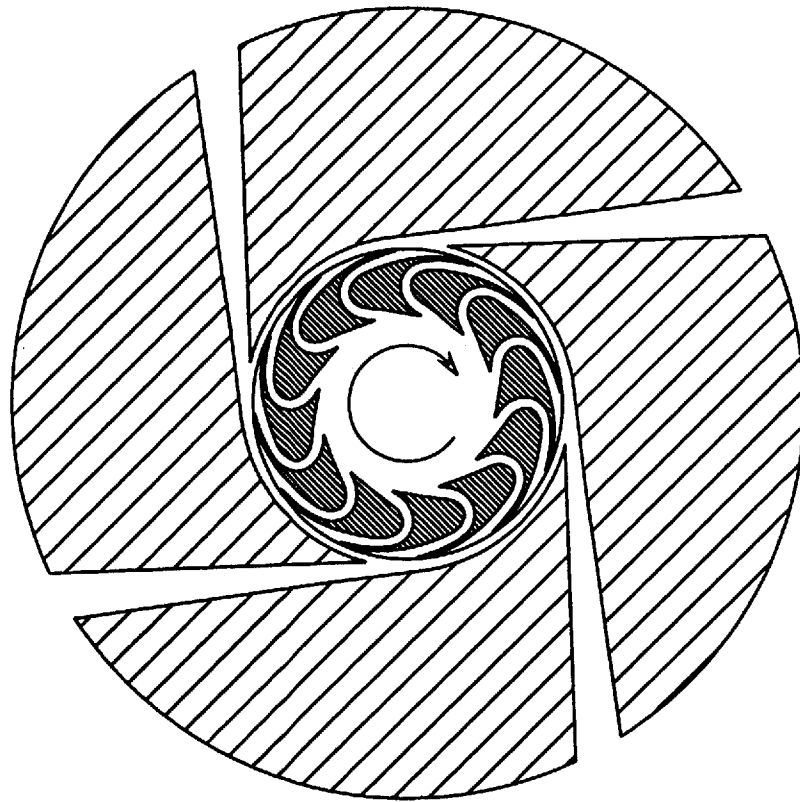


Figure 30: Partial Entry Design of Rotor and Diffuser [34]

Advantages of Forward Sweep

1. Relatively High Total Pressure Rise at Low Tip Speed

A high pressure rise reduces the number of stages required.

- A large amount of forward sweep allows more work to be done by the rotor because more turning occurs.

2. Low Static Pressure Rise Across Rotor

Because most of the total pressure at rotor outlet is present as dynamic pressure, the static pressure across the rotor is minimal. This brings about the following advantages.

- Low axial load - The pressure on the back face is very close to the pressure on the front face, resulting in low axial load, which will prolong bearing life.
- Low leakage across shroud - Because there is a relatively small pressure difference across the shroud, the leakage is minimized. This is especially important because the seals on the shroud are at a higher diameter, which causes sealing to be compromised.

3. Low Rotor Diameter

Because of forward sweep, the rotor can achieve near sonic velocities at the output with a smaller diameter than conventional 2D compressors, which has the following consequences.

- Low windage - The windage loss from a rotor increases with radius to the power of 4.75 at constant shaft speed. Although typically less efficient, smaller rotors incur lower windage losses. Because less work is done in low flow compressors, windage losses have a more significant effect on machine efficiency than in standard compressors.
- Low axial load - Less axial load is applied due to the reduced area that the pressure acts on.

- Low stresses - Lower rotor diameter results in lower stresses, allowing less exotic materials and use of lower cost manufacturing methods.
- Shrouded impeller - Because the stresses are low, the addition of an attached shroud is possible. This eliminates tip losses, improving the flow at low mass flow rate and blade height.
- Low mass - Lighter rotors keep the resonant modes of the shaft at higher frequencies.

4. Low Diffuser Incidence

- Although diffusers usually reduce the range of a compressor, the angle of incidence is affected very little by changes in mass flow, allowing for a large range in the diffuser.

5. Efficient Compression at Low Specific Speed

- Low Radial Velocity - At high levels of forward sweep, only a small component of the rotor exit velocity is in the radial direction, which keeps the mass flow low.
- High Tangential Velocity - Although mass flow is kept low, the tangential velocity can be kept high, which keeps the pressure ratio high.
- High Openness - Although there is high blade solidity as the flow is turned within the rotor, the blade is thin at the tip, keeping mixing losses to a minimum. The outlet area can be kept relatively large, which allows the blade height to be kept approximately equal to the passage width, minimizing pipe losses.

2.6.3 CFD Predictions

CFD predictions have been performed for this design, as shown in **Figure 31**, which has a design point tip-speed of 220m/s, and a blade height equal to passage width [35]. Performance has been predicted at atmospheric inlet for speeds of 20*kRPM* and 60*kRPM*.

These speeds correspond to two single stage test rigs described later. The swirl in the diffuser geometry is present due to space constraints in the corresponding test rigs. By curving the diffuser passages, a longer diffuser passage can be accommodated in a given casing diameter. This geometry is identical in rotor and stator inlet to the geometry shown in **Figure 30**. The first set of computational predictions at $60kRPM$ are expressed as a compressor map in **Figure 32**.

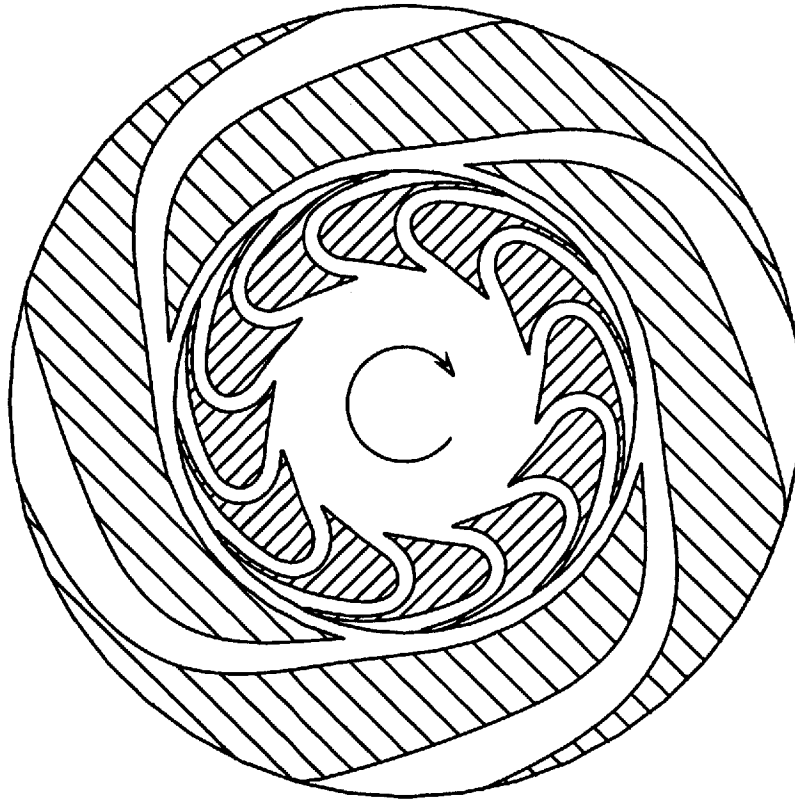


Figure 31: Diffuser Design for Low-Speed, Single-Stage Test [35]

After the first tests did not meet the computational predictions of Vine [35], the CFD techniques were re-evaluated. It was determined that the unsteady effects of mixing could not be ignored, and emphasis switched to unsteady modelling, which is constant speed, but includes mixing effects at the rotor-diffuser interface. Due to limitations in computing power, full maps can not be produced. Efforts were focused on accurate design-speed predictions. The CFD predictions are shown in **Table 12**.

Speed	Rotor $\varnothing$	Mass flow	Pressure Ratio	Efficiency	Analysis Type
20kRPM	210 mm	90 g/s	1.58	65%	Transient
60kRPM	70 mm	10 g/s	1.58	62%	Steady State

Table 12: Design Point CFD Predictions [35]

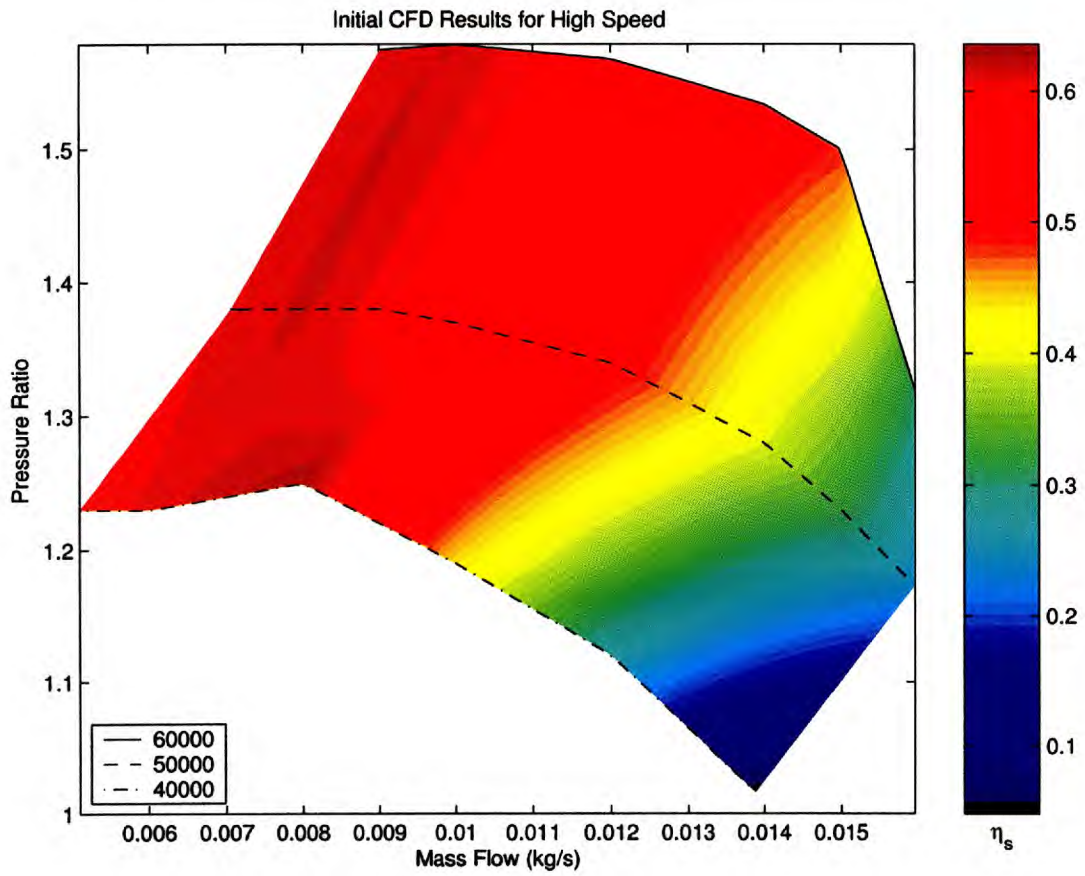


Figure 32: Initial CFD Predictions for High Speed Compressor [35]

2.6.4 Single Stage Experiments

Two sets of single stage experiments have been performed. The first test conducted was a high speed test at $60kRPM$ in order to initially assess the compressor performance and demonstrate feasibility. Later, a 3X scale rig (designed for $20kRPM$) was built in order to obtain detailed measurements. The measured maps are presented for the high speed test in **Figure 33**, and for the low speed test in **Figure 34**. The high speed test results contain only seven data points, due to the difficulty of speed control of the turbine-driven compressor. The low speed compressor, which was motor-driven, contains 30 data points. The results are interpolated using the methods described in section 4.2.2.

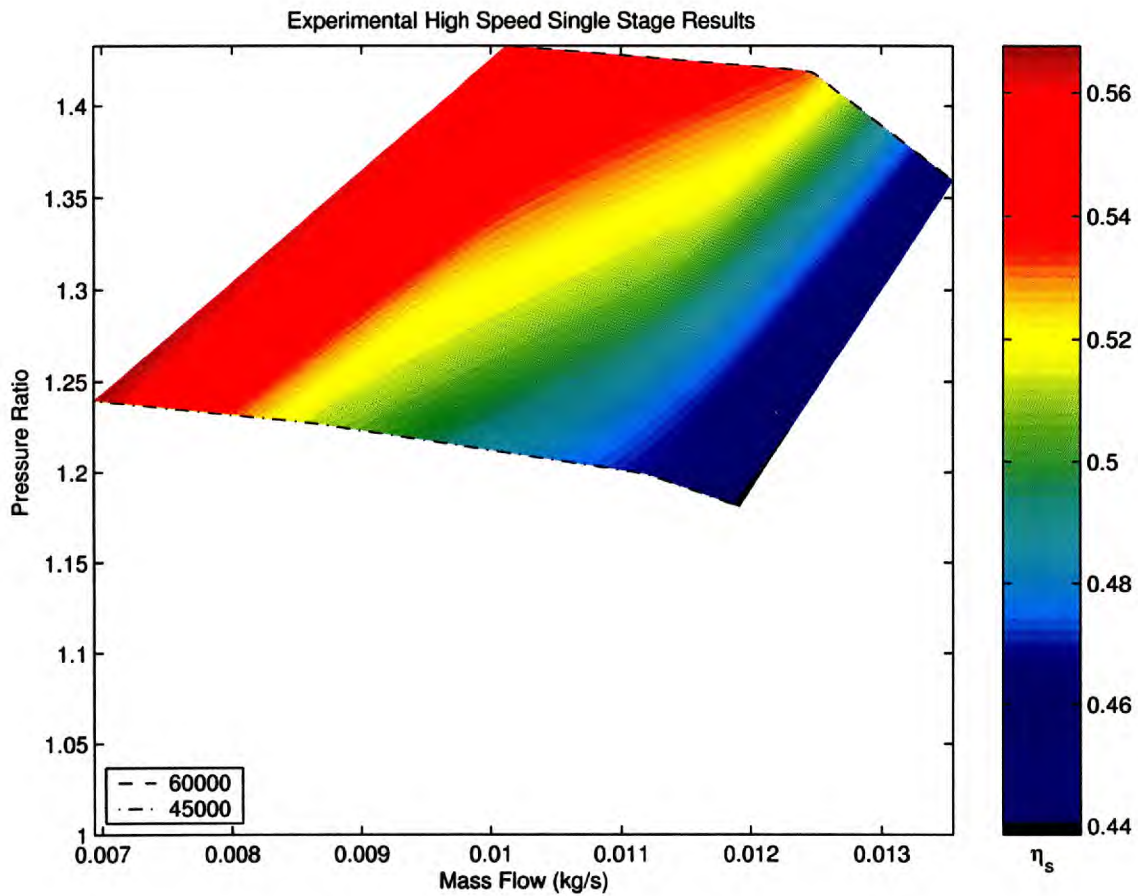


Figure 33: Experimental test map at $60kRPM$ [35]

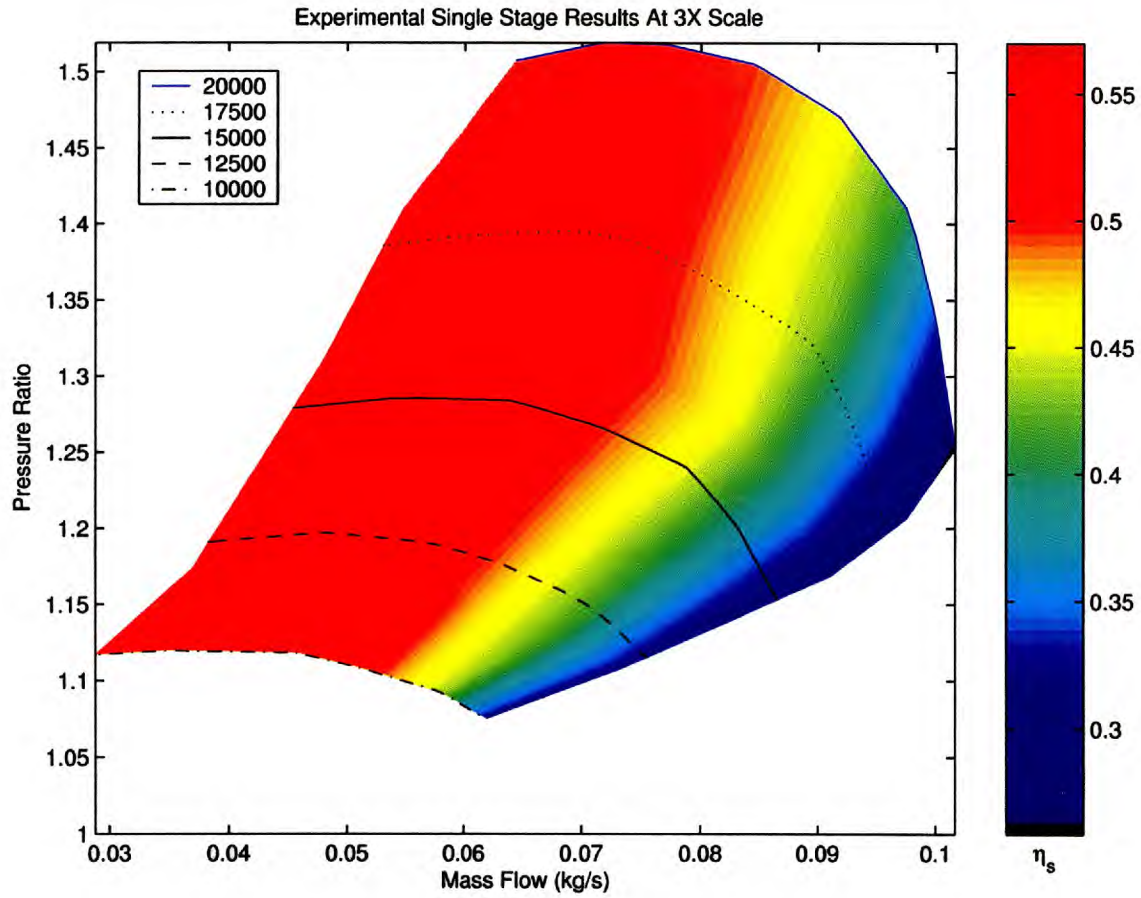


Figure 34: Experimental test map at 20kRPM [35]

At design point, the high speed single stage test fell short of the initial pressure ratio prediction of 1.58 by achieving a pressure ratio of only 1.43. Upon reevaluation of the CFD, it was determined that a steady state model was not sufficient and analysis switched to transient modelling. Attention then turned to designing the low speed test rig and CFD analysis of the 20,000RPM case was performed. For the low speed test, the pressure ratio achieved was 1.52, compared to a predicted value of 1.58. This is expressed in **Table 13**. The efficiency values should not be taken as precise, due to rig cooling effects, which are more significant than in conventional turbo compressors. Although both pressure ratios were predicted at 1.58, the low speed prediction is more conservative, as the high speed compressor is operating at lower Reynolds number and higher relative surface roughness, which will cause increased losses.

Speed	Steady State Prediction	Transient Prediction	Measured Value
20kRPM	N/A	1.58	1.52
60kRPM	1.58	N/A	1.42

Table 13: Design Point Pressure Ratio Comparisons

2.6.5 Specific Speed of the High Forward Sweep Compressor

The specific speed of both predictions and experimentations are summarized in Table 14.

	CFD Prediction	Experimental Measurement
20kRPM	0.2	0.2
60kRPM	0.2	0.25

Table 14: Specific Speed of Forward-Swept compressors

Although the data in **Figure 7** suggests that compressors of this specific speed have already been developed at an efficiency of 55%, it must be considered what type of compressor this represents. [37] shows a compressor that achieves a specific speed of 0.2 as having 55° of back sweep. Such high backsweep will result in a low head coefficient. The low losses are likely to be a result of low diameter. Efficiency may be kept high at the expense of pressure ratio.

3 The Multistage Forward-Swept Centrifugal Compressor

Both analysis and experimentation show that multiple forward-swept stages are required to meet the goals of a microturbine fuelling compressor set out in chapter 1. The forward-swept design described in section 2.6 is suitable for the first stage of such a machine. For subsequent stages, the specific speed at the design point is further reduced, and the forward-swept design will be used for these stages as well.

The present research is focussed on the design and testing of a multistage compressor utilizing the forward-swept stage design. This chapter describes the overall design of the compressor.

3.1 Machine Arrangement

There are two arrangements used in industrial, multistage, turbo-compressors. The first, an example of which is shown in **Figure 35**, fixes each stage in series to a common shaft. The gas is passed through return channels and into the next stage. The second type is known as an integrally geared compressor and an example is shown in **Figure 36**. The integrally geared compressor is driven at the central gear. The smaller pinion gears allow an increase in speed, and each compressor can operate at its optimal speed. This arrangement is too complicated for this application, whereas the single shaft arrangement could be coupled directly to the microturbine generator or driven by a high speed motor. An alternative design was investigated and is described in Appendix A.

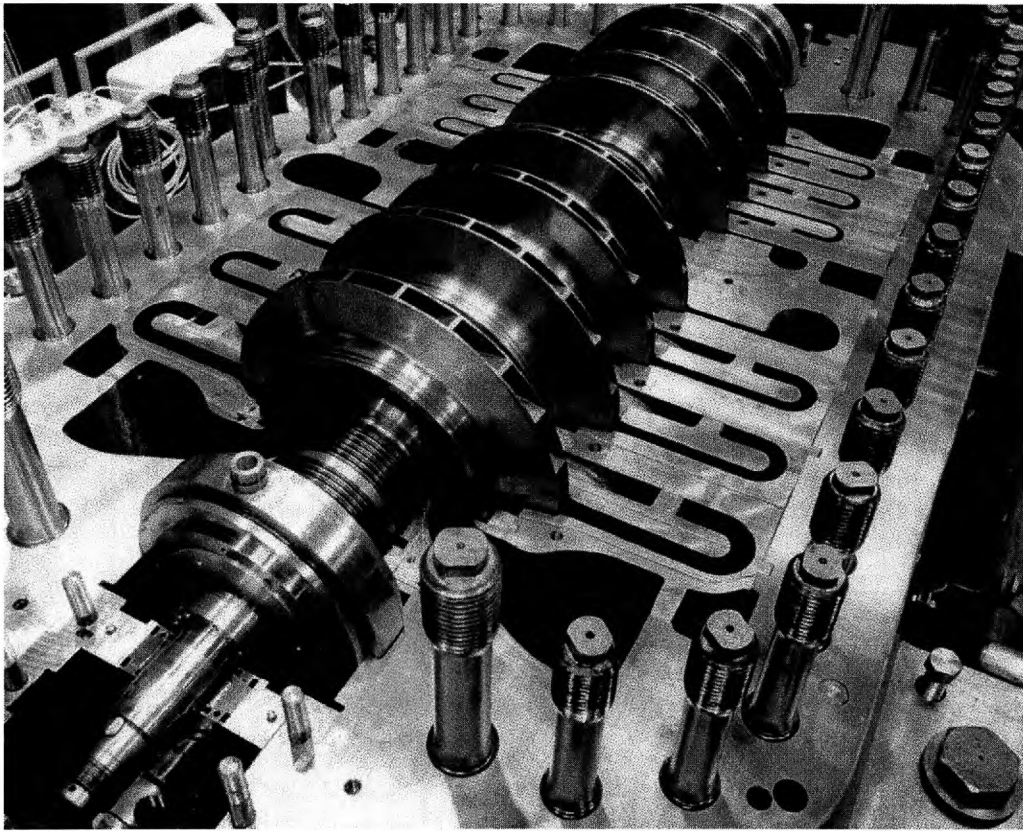


Figure 35: A Single-Shaft Industrial Centrifugal Compressor [11]

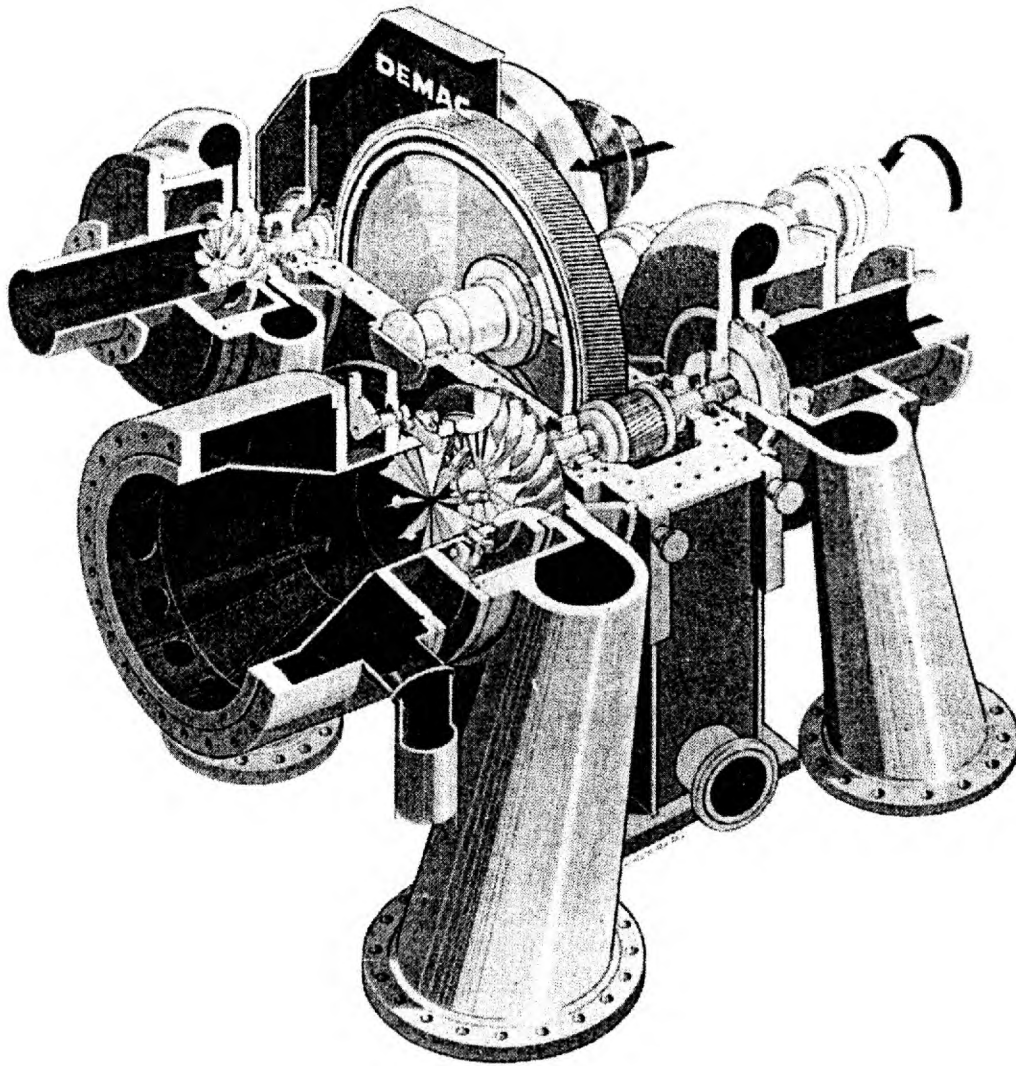


Figure 36: An Integrally-Geared Centrifugal Compressor [38]

3.2 Stage Design

Effort was taken to design each stage in a modular fashion, maintaining the same components and features between stages. The only differences between stages are the blade height of the rotors and diffusers, the length and diameter of the rotor bores, and the outer diameter of the inlet ducting. This simplifies design, analysis and manufacturing. The multistage arrangement is shown in **Figure 37**.

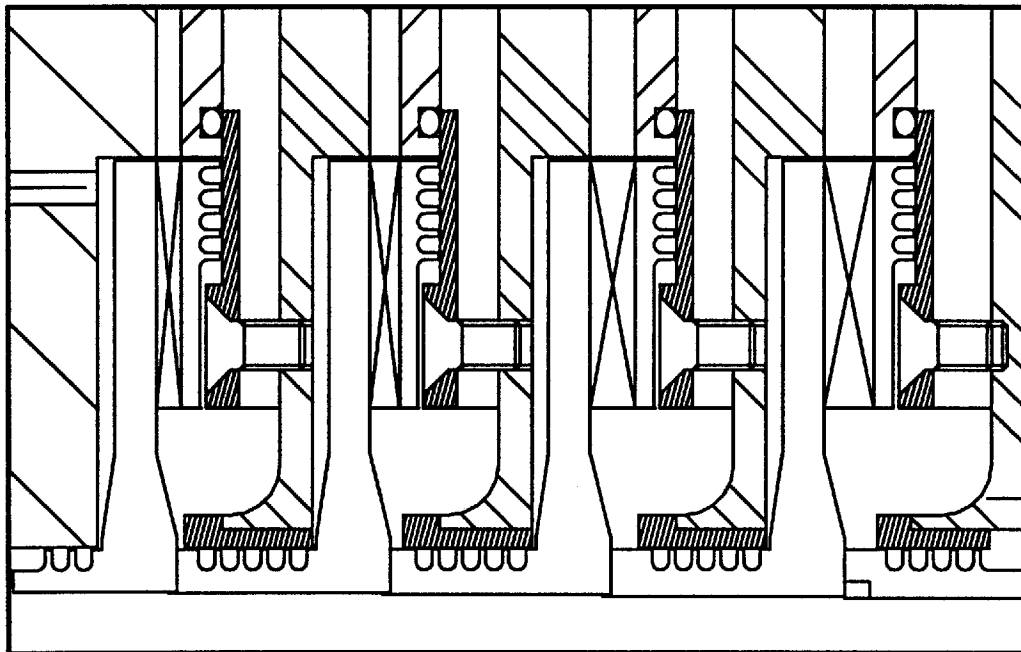


Figure 37: Multistage Arrangement

A multistage test rig was ultimately designed around the CFD predictions of the final high speed design. As the design pressure ratio predicted was 1.57, it was determined that 4 stages would be necessary to meet the requirements established in Section 1.3. All of the stages were mounted on the same shaft. Face seals were located on the shroud and conventional, straight-through labyrinth seals were used to seal between stages. The inlet flow was directed inwards by four radial ducts. The ducting was designed for low axial length, which was necessary due to rotordynamic constraints. The low mach number flow in the inlet ducting indicated that low losses could be expected.

3.3 Stage Matching

In a multistage machine, the later stages must run at lower specific speed due to the lower volume flow rate at the inlet. While it would be beneficial to design a new blade profile for each stage, this would be significantly more effort and would complicate manufacturing. Alternatively, a common rotor and diffuser profile was used for each stage, with the blade

height scaled proportionally to inlet specific volume.

Experimental results of the chosen stage design were not available prior to the design of the multistage compressor. The initial single stage CFD predictions, as shown in **Figure 32**, were used for the stage matching. Based on a PR of 1.58, an efficiency of 58% and an intercooler effectiveness $\eta_{IC} = 0.8$ after the 2nd stage, the blades were cropped proportionally to the volumetric flow at stage inlet. The blade heights are shown in **Table 15**.

Property	Stage 1	Stage 2	Stage 3	Stage 4
Blade Height	3.0mm	2.3mm	1.4mm	1.1mm

Table 15: Stage Blade Heights

Cumpsty [28] gives a good explanation of the phenomenon of off-design matching. **Figure 38** shows the operating points of the first and last stages of a compressor, with each stage having identical flow characteristics. At point (a), the compressors are matched. At (b), the first stage compressor is producing a lower pressure ratio, which in turn increases the volumetric flow into the second compressor, further lowering the output pressure. At (c), the increased pressure ratio of the first stage, creates a low volumetric flow rate into the final stage. Point (d) indicates that a choking final stage can prevent the onset of stall. Otherwise, a compressor with a plenum upstream will surge when operating at points with a positive pressure slope. This is due to the plenum expanding back downstream and causing a reversal of flow. The low specific speed machine is even more sensitive to matching issues, because leakage has a significant effect on stage throughput.

3.4 Stage Sealing

In order to achieve adequate sealing between stages, close running clearances are required. However, running tight clearances increases the likelihood of a seal touching down, potentially causing damage to the rig. In order to minimize the damage, either an abradable

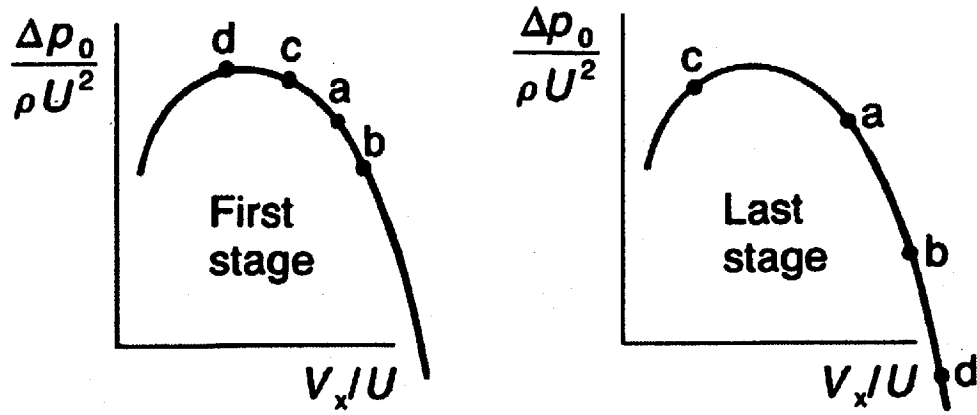


Figure 38: Multistage Matching [28]

seal or a rub-tolerant seal is required. PEEK (Polyetheretherkeytone) was used, as it is rub-tolerant and able to withstand working temperatures of up to 250°C [39]. Running clearances were in the range of 35 to 50 microns for the shaft seals. The shroud seals were touching down at zero speed, but as the rotors deflected up to 100 microns away from the stationary surface, as predicted by FEA analysis described in section 5.1.2, the running clearance was dependant on the speed of the rig. Future rotors could be redesigned to deflect less, or not at all.

3.5 Intercooling

Basic calculations determined that, at the efficiency and pressure ratio predicted, intercooling would be required to keep temperatures down. **Figures 39 & 40** show the isentropic efficiency and maximum temperature for a range of intercooler configurations, assuming adiabatic compression and ducting. Air at the inlet is at standard conditions and each stage produces a PR of 1.58 at a stage isentropic efficiency of 60%.

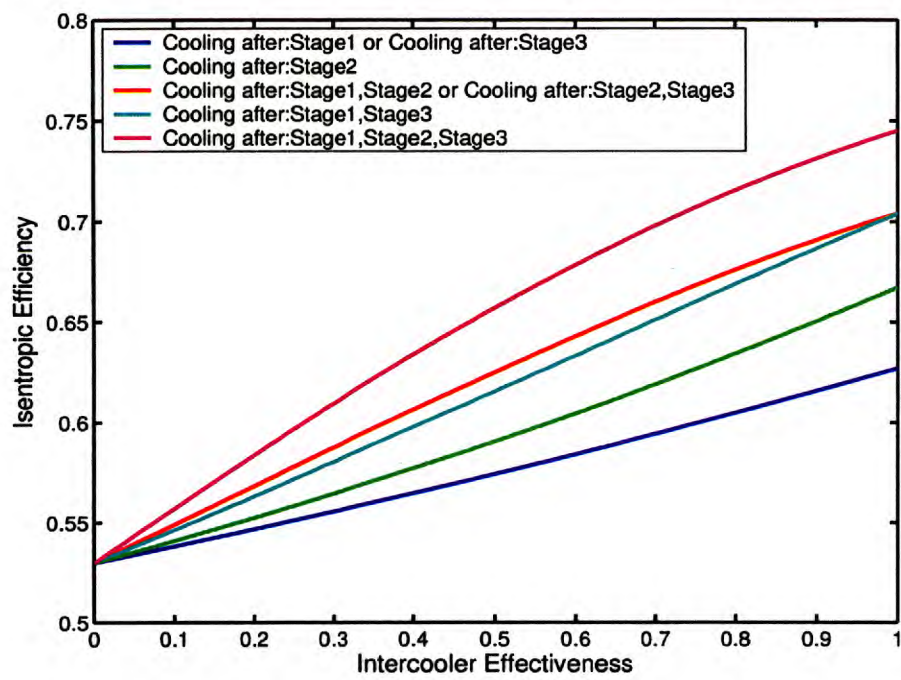


Figure 39: Efficiency vs. Intercooler Arrangement and Effectiveness

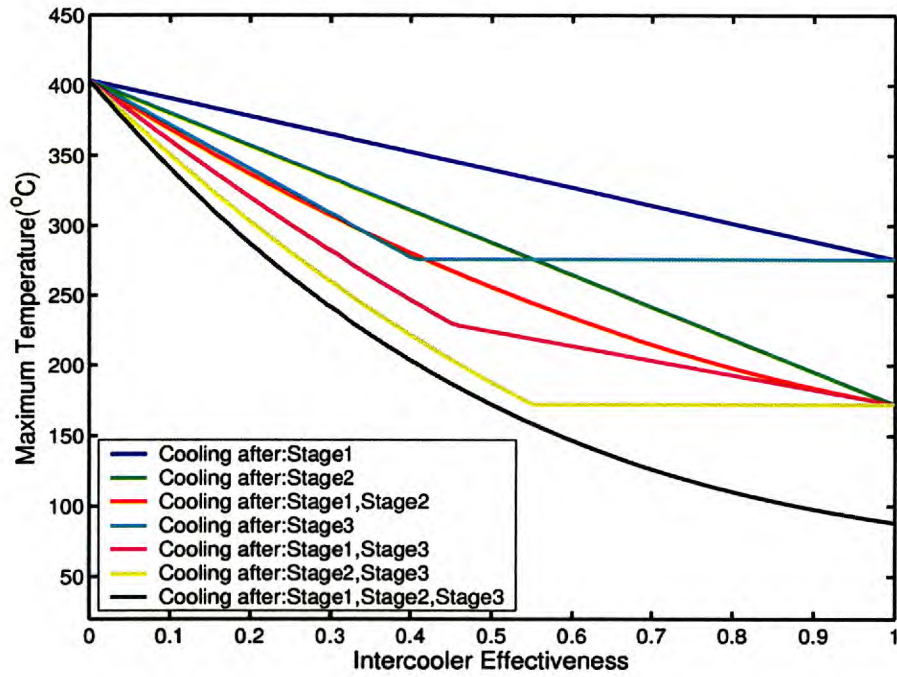


Figure 40: Maximum Temperature vs. Intercooler Arrangement and Effectiveness

It can be seen that if only one stage is intercooled, intercooling after the second stage has the greatest effect on maximum temperature for $\eta_{IC} > 0.55$ and the greatest effect on overall efficiency at all η_{IC} . If more intercoolers are used, the maximum temperature will drop further and efficiency will increase.

3.6 Design Summary

The design described in this chapter was developed simultaneously with the multistage simulation routine described in the next chapter. The design constraints and considerations described in chapter 5 also influenced the design. What has been described is the final product of iteration of design, modelling, analysis and simulation.

4 Multistage Simulations

A multistage arrangement of the forward-swept compressor was introduced in Chapter 3. In the present research, a system model was programmed in MATLAB [40] to calculate the effects of various phenomena on the performance, range and efficiency. The phenomena of primary interest are:

- Compressor Matching

In a multistage compressor, the outlet of one stage will determine the inlet of the upstream stage. Effects of off-design are compounded and multistage compressors will have lower range than the individual stages.

- Leakage

With a low mass throughput, the leakage between stages becomes more significant to the range, efficiency and flow dynamics.

- Windage

With a low power of compression, the loss due to frictional force acting on exposed faces of the rotor is proportionally higher than in a compressor with a high power.

- Intercooling Effectiveness

Intercooling will have a large effect on the inlet flow density of upstream compressors, especially if done externally and at only one stage.

In a multistage turbo compressor, the matching of stages is crucial to performance and operating range. Mismatching effects are cumulative and compounded through the stages. Additionally, a proportionally large amount of leakage occurs in a low specific speed compressor, which complicates the analysis and matching. Windage is of concern due to the low work done per stage. Cooling can also have an effect on the matching, whether deliberate (by means of an intercooler), or through inter-stage or environmental heat transfer. **Figure 41** shows a series of stages, with one individual stage highlighted for clarity. The flow path of both compression and leakage is superimposed.

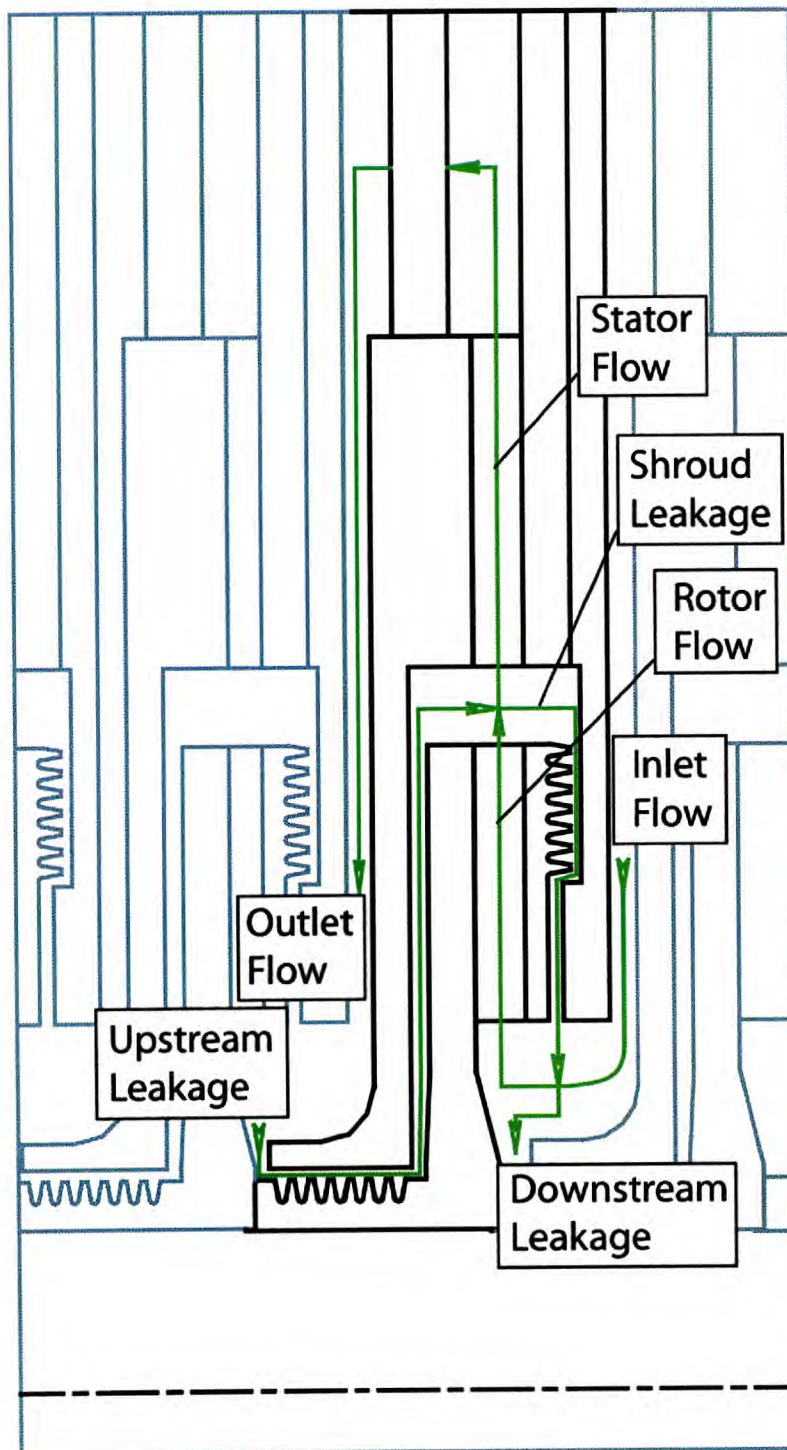


Figure 41: Single Stage

This chapter describes the modelling methods used for predicting the performance of a machine, given the compressor performance maps of the individual stages as well as other critical parameters. By treating each stage as a module, a common code base can be used to evaluate machine designs of differing size, speed, matching methods, number of stages, individual compressor maps, and seal clearances. Simulations were performed for designs based on CFD and experimental analysis, as well as optimized designs.

4.1 Object-Oriented System Model Overview

Object oriented programming allows sets of variables and functions to be grouped together within virtual components, known as 'objects'. A class is essentially a template which forms a structure for any number of objects. An object is an incidence of a class. The consolidation of a group of variables and functions into an object allows modular programming, convenient structure and reuse of code. Each class will normally have many functions associated with it, which are common to every object of that class. The variable values can differ between individual objects, but the number of variables between objects of the same class is constant.

Object oriented programming differs from a simple subroutine because each object, by belonging to a specific class, has all of the relevant variables and subroutines contained within it. This allows the object to be a 'black-box', with the inner workings not of concern to the user. A far more comprehensive description of object oriented programming is described in [41]. The 'Gas' class is presented as an example of a basic class.

Because of the significant leakage between stages, it is not possible to analyze the multistage compressor in a direct, stage-by-stage fashion. It is also not possible to determine the losses in a linear way because of the iterative methods of calculating the windage and leakage. Therefore, an iterative solution is required for system modelling. Furthermore, it was required to simulate many configurations and variations. This is the motivation for using the object oriented approach. The following classes were used:

- Machine
- Stage
- Compressor
- Rotor
- Stator
- Labyrinth Seal
- Drum
- Disc
- Map
- Gas

Each 'Machine' object contains an array of sequential 'Stage' objects, manages the gas flow between them, and determines the total machine performance. **Figure 42** shows the objects used to model the components of each stage, both superimposed on the drawing and as separate entities. The structure and analysis of each class are described in the following sections.

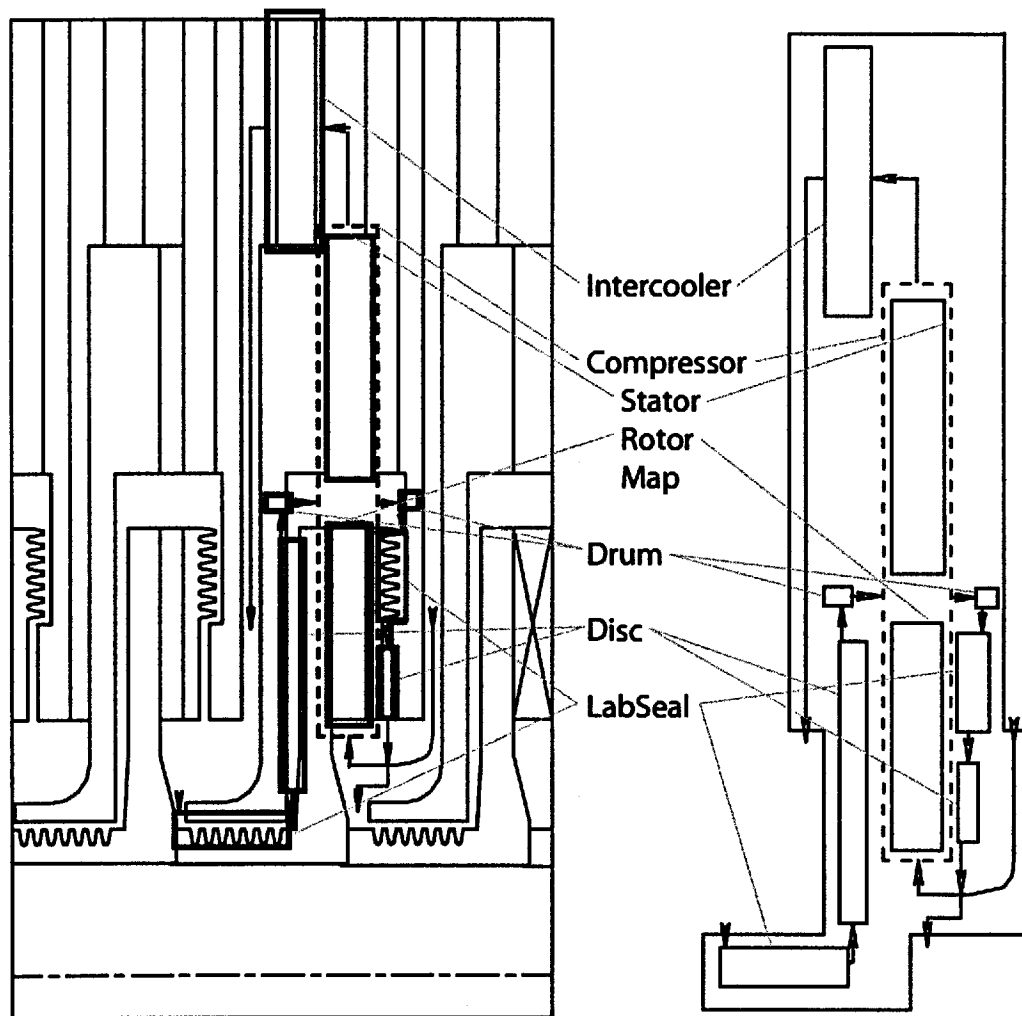


Figure 42: Single Stage Object Layout

This is shown as a block diagram in the following figures. **Figure 43** shows the notation used to describe a single object of type 'Class Of Object', named 'Object Name'.



Figure 43: Object Figure Notation

Figure 44 shows how a single 'Machine' object interacts with any number of stage objects.

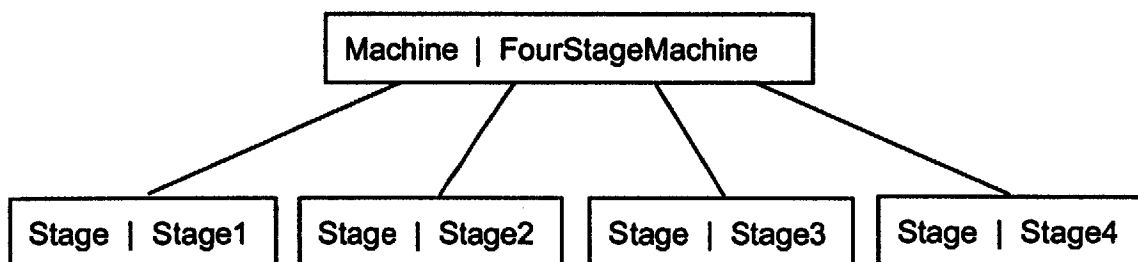


Figure 44: Machine-Stage Object Interaction

Figure 45 shows the structure of a single 'Stage' object, and how it interacts with all of the objects necessary to simulate an actual compressor stage.

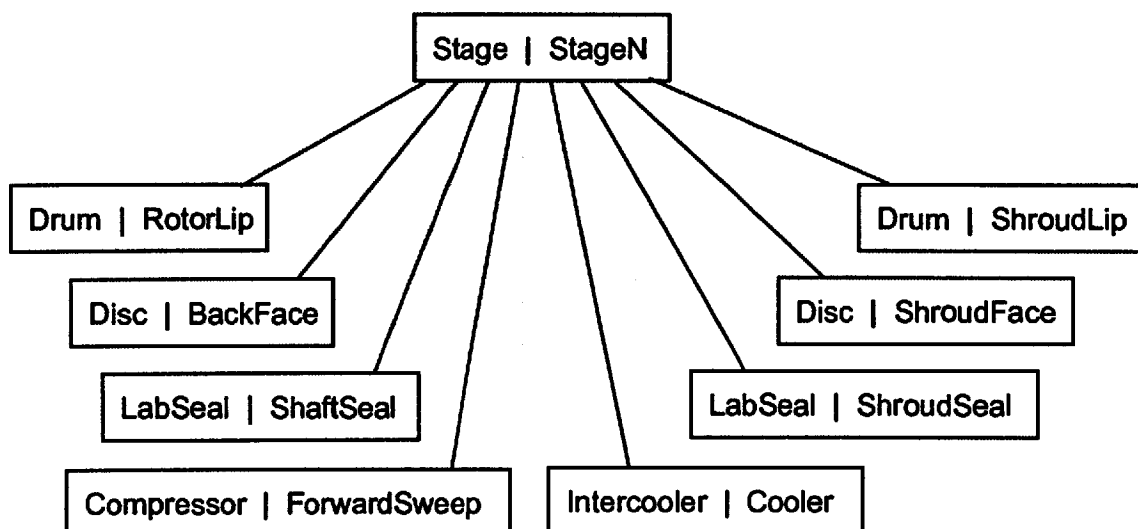


Figure 45: Stage Object Structure

Figure 46 shows the connection between the 'Compressor' object and its two constituent objects, as well as how any object interacts with a 'Gas' object, which is passed through to all the other objects.

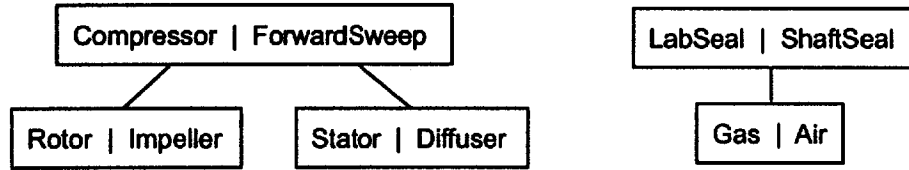


Figure 46: Compressor Object Structure and Gas Object Usage

4.2 Class Descriptions

The individual classes are described here using a 'bottom-up' approach. The nomenclature used is specific to each class.

4.2.1 Gas Class

The Gas class is the simplest class used in this simulation. It will be described both in the context of its place in the overall simulation and as an example of a simple class. It contains the physical constants and state dependant properties of the gas mixture it represents. It is used to calculate the properties based on a given temperature and pressure. The class structure is useful because all that is required to make a new Gas object are the physical properties. All of the functions, internal variables and structure are common.

Creation: When an object is created, all of its internal variables are created as well. For any object of the Gas class, the following variables are initialized at creation, based on input data:

- *R*: The gas constant for the gas being modelled.

- $[T_{ref}]$: $1 \times n$ matrix containing temperatures of known properties. Must be monotonic and increasing.
- $[\mu_{ref}]$: $1 \times n$ matrix containing dynamic viscosity values at corresponding temperatures in T_{ref}
- $[k_{ref}]$: $1 \times n$ matrix containing conductivity values at corresponding temperatures in T_{ref}
- $[c_{p_{ref}}]$: $1 \times n$ matrix containing specific heat values at corresponding temperatures in T_{ref}
- T_{amb} : Ambient temperature
- P_{atm} : Atmospheric pressure

The following variables are created with no initial value:

- ρ : The gas density at operating point (OP).
- c_p : Specific heat at OP.
- k : Conductivity at OP
- ν : Kinematic viscosity at OP
- μ : Dynamic viscosity at OP
- γ : Ratio of specific heats at OP

Data Flow: As a relatively simple class, the operation of the gas class contains two steps.

1. Calculate the new variables for the given OP
2. Extract these new variables

In order to update the parameters, a function *Calculate* is used. This is a function of the temperature and pressure at the OP.

$$GasObject_{updated} = Calculate(GasObject_{initial}, T_{OP}, P_{OP}) \quad (40)$$

The properties at the OP are then extracted through specific functions. For example, *GetGamma* is used to extract the γ variable:

$$\gamma = GetGamma(GasObject_{updated}) \quad (41)$$

This provides a clean and concise method for calculating the variables of interest. For further class descriptions, these two steps will be consolidated.

Calculations: The density, ρ , is calculated from the ideal gas law using inlet conditions and R . μ , k and c_p are calculated through linear interpolation, based on the vectors created at initialization. For example, if i_1 is the index of the largest entry in $[T_{ref}]$ less than T_{OP} and i_2 is the index of the smallest entry in $[T_{ref}]$ greater than T_{OP} , then **Equation 42** shows how k_{OP} is calculated:

$$k_{OP} = [k_{ref}]_{i_1} + ([k_{ref}]_{i_2} - [k_{ref}]_{i_1}) \frac{T_{OP} - [T_{ref}]_{i_1}}{[T_{ref}]_{i_2} - [T_{ref}]_{i_1}} \quad (42)$$

This can be abbreviated as a function *Interp1* using **Equation 43**, which is the expression of the function as used in MATLAB. μ and c_p can be similarly interpolated.

$$k_{OP} = Interp1([T_{ref}], [k_{ref}], T_{OP}) \quad (43)$$

γ is calculated from R and c_p . The kinematic viscosity is calculated using **Equation 44**

$$\nu = \frac{\mu}{\rho} \quad (44)$$

4.2.2 Map Class

Each object of the Map class contains one performance map. This can be associated with either a single stage or a whole machine and is based on either experimental or computational results. The Map class is used for calculating the pressure ratio and efficiency at a given mass flow rate and shaft speed, through interpolation between points. Map objects also produce graphs of their associated data, corrected for a given inlet temperature and pressure.

Creation: Each Map object requires 4 $m \times n$ matrices as well as reference temperature and pressure:

- $[\dot{m}_{ref}]$: Mass flow matrix at referred conditions
- $[RPM_{ref}]$: Shaft speed matrix at referred conditions
- $[PR]$: Pressure ratio matrix
- $[\eta]$: Efficiency matrix
- T_{ref} : Temperature corresponding to data
- P_{ref} : Pressure corresponding to data

Each row corresponds to a line of constant speed, and each column corresponds to a constant mass-flow. The reference temperature and pressure, P_{ref} and T_{ref} , are required in order to perform calculations corresponding to varying inlet conditions. Finally, a figure number is given which is used for managing output plots. $[\dot{m}_{ref}]$ must contain m identical columns containing positive entries, with $[\dot{m}_{ref}]_{i,j+1} > [\dot{m}_{ref}]_{i,j}$. $[RPM_{ref}]$ must contain n identical rows containing positive entries, with $[RPM_{ref}]_{i+1,j} > [RPM_{ref}]_{i,j}$. At points of surge and choke, $[PR]$ contains the indeterminate MATLAB entity NaN , while $[\eta]$ contains zeros.

Data Flow: Once created, a Map object is static and cannot be changed by any objects or functions. The function it serves is to produce a PR and efficiency for a given mass

flow, shaft speed, inlet temperature and inlet pressure, along with the range. A plot of the corresponding point on a redimensionalized map can also be produced. This is also used to produce the compressor map figures shown.

Calculations: For a given operating point of $\dot{m}_{op}$, RPM_{op} , T_{inlet} and P_{inlet} , the first computation is to use $[\dot{m}_{ref}]$ and $[RPM_{ref}]$ to create corrected matrices, $[\dot{m}]$ and $[RPM]$ corresponding to T_{inlet} and P_{inlet} . This is done through the following scalar multiplications:

$$[\dot{m}] = [\dot{m}_{ref}] \cdot \frac{\sqrt{T_{inlet}/T_{ref}}}{P_{inlet}/P_{ref}} \quad (45)$$

$$[RPM] = [RPM_{ref}] \cdot \sqrt{T_{inlet}/T_{ref}} \quad (46)$$

To aide in interpolation, an additional matrix $[S]$ is created:

$$[s]_{i,j} = \frac{[\dot{m}]_{i,j} - [\dot{m}]_{i,1}}{[\dot{m}]_{i,m} - [\dot{m}]_{i,1}} \quad (47)$$

$[\dot{m}_{ref}]$ can be used in place of $[\dot{m}]$ in **Equation 47**, giving the same result. Each entry in $[s]$ corresponds to the fraction along the range of $[\dot{m}]$. Two vectors containing the data at speed lines above and below RPM_{op} can be extracted from the corrected matrices and designated by *hi* and *lo* subscripts. Interpolation using $[RPM^2]$ can be used to create vectors for corrected mass flow, PR and efficiency at the given speed:

$$PR_{RPM}(s) = PR_{lo}(s) + (PR_{hi}(s) - PR_{lo}(s)) \left(\frac{RPM_{op}^2 - RPM_{lo}^2}{RPM_{hi}^2 - RPM_{lo}^2} \right) \quad (48)$$

$$\eta_{RPM}(s) = \eta_{lo}(s) + (\eta_{hi}(s) - \eta_{lo}(s)) \left(\frac{RPM_{op}^2 - RPM_{lo}^2}{RPM_{hi}^2 - RPM_{lo}^2} \right) \quad (49)$$

$$\dot{m}_{RPM}(s) = \dot{m}_{lo}(s) + (\dot{m}_{hi}(s) - \dot{m}_{lo}(s)) \left(\frac{RPM_{op}^2 - RPM_{lo}^2}{RPM_{hi}^2 - RPM_{lo}^2} \right) \quad (50)$$

This is shown graphically in **Equation 51** and **Figure 47**

$$\frac{l}{L} = \frac{RPM_{op}^2 - RPM_{lo}^2}{RPM_{hi}^2 - RPM_{lo}^2} \quad (51)$$

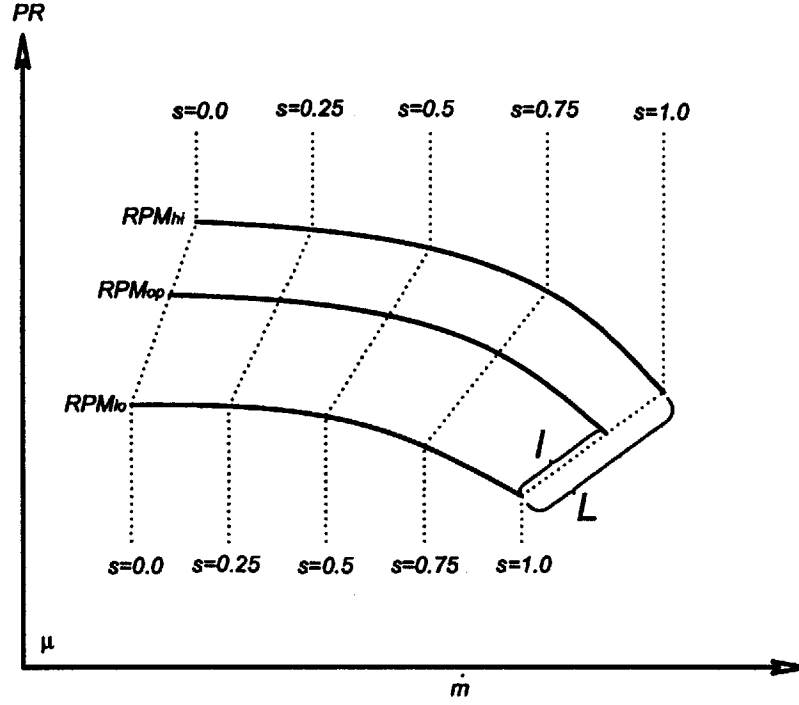


Figure 47: Map Interpolation Method

$\dot{m}_{RPM}$, PR_{RPM} , and η_{RPM} can then be used to interpolate PR_{OP} and η_{OP} from the mass flow, $\dot{m}_{OP}$, along with range and indications of stall and choke. This method is not exact, as the only way to accurately predict the performance at a given rotational speed is through experimentation or CFD simulation. However, this method is required to account for changes in referred speed due to changing temperatures. For experimentally determined maps, all points at top mass flow are taken at fully open valve, and interpolation between choked points will be reasonably accurate. The accuracy at points away from choke will be compromised, especially if data is not collected all the way to the point of surge at each speed line.

4.2.3 Rotor and Stator Class

Each compressor object currently contains a Rotor and Stator object. These hold all of the geometric variables needed to define the geometry of the impeller and diffuser, including the blade profile, diffuser profile and blade height. Currently, all performance prediction is done by maps obtained either by CFD computations or by experiment. However, it is intended that eventually, empirical and design data will exist, allowing multistage machines to be modelled without requiring maps. However, that is currently outside of the scope of the software, and the Rotor and Stator objects are left idle during simulations. The Compressor class uses only the given map or design point data for performance prediction. To clarify, the Rotor and Stator classes do not contain variables relevant to windage and sealing, only variables relevant to defining the compression region.

4.2.4 Compressor Class

The compressor class is used for calculating the outputs of the compressor being modelled, based on the thermodynamic equations discussed in Section 2.1.

Creation: Each compressor object contains one object of each of the following classes: Map, Stator, Rotor and Gas (however, as mentioned earlier, the Rotor and Stator objects are not utilized and are included only for future convenience). All of these objects are input at creation. The Map object determines the behavior of the compressor, while the Gas object contains the relevant gas properties. The only other required input parameter is the impulse exponent, I , which is described shortly.

- I : The impulse exponent: $\log_{PR}(PR_{stator})$.

Data Flow: The following input variables are required in order to perform each iteration:

- ω : Current shaft speed
- $P_{rotor,in}$: Inlet pressure of rotor

- $T_{rotor_{in}}$: Inlet temperature of rotor
- $\dot{m}_{rotor_{in}}$: Mass flow into rotor
- $P_{stator_{in}}$: Inlet pressure of stator
- $T_{stator_{in}}$: Inlet temperature of stator
- $\dot{m}_{stator_{in}}$: Mass flow into stator

The shaft speed is merely relayed by the Stage object. The rotor and stator inlet parameters are provided by the Stage object as well. The inlet pressure of the stator is simply the stage inlet pressure, however, the rotor inlet mass flow differs from the stage inlet mass flow due to associated leakage from the shroud seal and to the downstream shaft seal. The rotor inlet temperature is the temperature of the mixed gas. The following output variables are computed:

- $\dot{m}_{lim}$: A vector containing the lower and upper mass flow limits at the current shaft speed and inlet conditions
- $T_{rotor_{out}}$: Total Temperature at rotor outlet
- $P_{rotor_{out}}$: Static pressure at rotor outlet
- $T_{stator_{out}}$: Total Temperature at stator outlet
- $P_{stator_{out}}$: Total pressure at stator outlet
- H : Power consumed by compression.

The stator inlet pressure is the rotor outlet pressure from the previous iteration. Again, the temperature and mass flow differ from rotor outlet to stator inlet due to leakage from the upstream shaft seal and leakage to the shroud seal.

Calculations: First, the gas object is used to determine the relevant gas properties for the inlet conditions, namely γ and c_p . Then, the map object is used to determine $\dot{m}_{surge}$, $\dot{m}_{choke}$, $PR_{compressor}$ and $\eta_{compressor}$.

- $PR_{compressor}$: The current total-to-total PR of the whole compressor
- PR_{rotor} : The current total-to-static PR of the rotor
- PR_{stator} : The current static-to-total PR of the stator

The impulse exponent, I , has been devised as an alternative to the more commonly used quantity of 'reaction', as it is calculated from only the pressure ratio, and not from a ratio of enthalpies. It is a measured result, rather than a design parameter. It is needed to calculate the pressure at the rotor outlet, which determines the pressure difference across the seals. Although it is unlikely that the impulse remains constant at all operating points, it is kept constant for the simulation, as little work has been done to characterize the static pressure at the rotor outlet. Early CFD results indicated an impulse exponent value of 0.6. $PR_{compressor}$ and $\eta_{compressor}$ are calculated as follows.

$$PR_{stator} = PR_{compressor}^{(I)} \quad (52)$$

$$PR_{rotor} = PR_{compressor}^{(1-I)} \quad (53)$$

An efficiency is then assumed to be equal for both the rotor and stator, and is computed using **Equation 54**, using the following substitutions: $k = (\gamma - 1)/\gamma$, $K_r = PR_{rotor}^k - 1$, $K_s = PR_{stator}^k - 1$, $K_c = PR_{compressor}^k - 1$

$$\eta_{rotor} = \eta_{stator} = \frac{-K_r - K_s + \sqrt{K_c + 2K_r K_s (1 + 2\eta_{compressor} K_c)}}{2K_r K_s} \quad (54)$$

This formula uses the same principals as the concept of polytropic efficiency. This allows the rotor and stator to be treated as separate entities while maintaining the same

overall efficiency. It was found that this was insignificant for the pressure ratios of concern. With the rotor and stator pressure ratio and efficiency defined, the rotor and stator outlet temperatures can be calculated, along with the total power consumption, using the equations from Section 2.1

4.2.5 LabSeal Class

The LabSeal class is used for representing both the conventional labyrinth seal used for shaft sealing, as well as the less common face seals. It is used to calculate the leakage flow as well as the windage losses within the seals.

Creation: Upon creation of a LabSeal object, all of the relevant geometric parameters of each labyrinth seal groove are saved as constant values. These include:

- N_T : The number of teeth contained in the seal.
- $[R_T]$: A $1 \times N_T$ matrix containing the radius at each tooth
- $[D_C]$: A $1 \times N_T$ matrix containing the radial clearance at each tooth
- $[D_P]$: A $1 \times N_T$ matrix containing the pitch distance of each tooth
- $[D_T]$: A $1 \times N_T$ matrix containing the depth of each tooth
- $[D_W]$: A $1 \times N_T$ matrix containing the width of each groove

Figure 48 and **Figure 49** show these parameters for both the shaft and face seal. Each LabSeal object also contains one object of class Gas, which is used to determine the properties of the working fluid.

Data Flow: Each iteration, T_{in}, P_{in}, P_{out} and ω are input and the following variables are calculated:

- $\dot{m}_{leakage}$: The mass flow present in the seal.
- $[P_{seal}]$: A $1 \times N_T$ matrix containing the pressure after each edge.

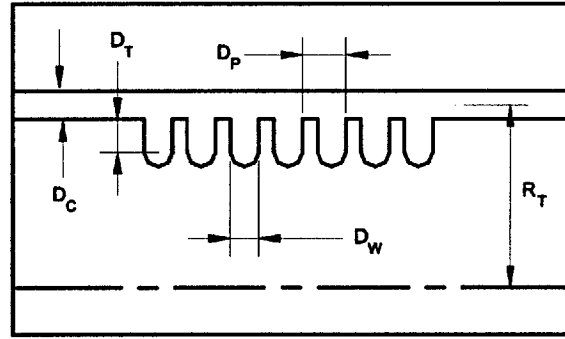


Figure 48: Shaft Seal Parameters

- T_{out} : The gas temperature at the outlet.
- H : The power consumed due to windage.

Calculations: Labyrinth seals restrict gas flow from high pressure to low pressure, by allowing gas to flow through a very narrow annular gap, between the seal lip and the stationary surface, forcing the flow to accelerate. A pressure drop then occurs when the gas is expanded irreversibly into the cavity in the next groove. The leakage through labyrinth seals was originally investigated by Elgi [42]. Most calculations for labyrinth seal leakage are based on a combination of theory and experimental results. Most of the previous work done on labyrinth seals has been on the shaft type. With face type seals, the clearance area varies between knife tips, due to changes in diameter. Correlations for cylindrical seals were found that allowed variable cross-sectional areas to be analyzed, as no face seal models have been found.

Yucel and Kazakia [43] have a method of cylindrical type labyrinth seal analysis that is particularly suited to extrapolation to face type analysis, because it calculates the pressure loss through each seal lip, rather than treating the whole seal as one entity. This way each seal can be given a different area, which is the case with face-type seals. The downside is that an iterative solution must occur as the calculations are based on the mass flow. However, the calculations are simple enough that this can be incorporated without requiring excessive computational resources. Many relationships are given, but the paper

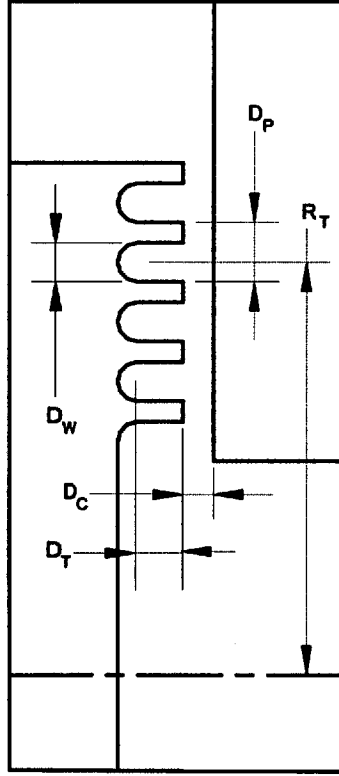


Figure 49: Face Seal Parameters

as a whole does not discuss the numerical implementation or the iterations involved. The following describes the inner workings of the labyrinth function that are based on the method of Yucel and Kazakia.

N_T is the number of teeth in the seal, and C_0 is the coefficient of pressure recovery in the seal based on experimental observations, and taken as constant at 0.611. The kinetic energy carryover coefficient, μ_0 is equal to 1 for the first stage and determined by the following expression for all the other stages:

$$\mu_0 = 1 + .0791(N_T - 1) \quad (55)$$

Although it is desired to calculate the leakage based on the pressure difference across the seal, the method calculates the pressure loss based on the mass flow. This is done for a range of mass flows stored in a $m \times 1$ matrix $[m_{lab}]$. The seal pressures are calculated

and stored in the $m \times N_T$ matrix $[P_{lab}]$.

$$[P_{lab}]_{i,1} = P_{in} \quad (56)$$

$$[P_{lab}]_{i,j+1} = \sqrt{([P_{lab}]_{i,j})^2 - \left(\frac{[\dot{m}_{lab}]_i}{C_0 \mu_0 * [A]_j} \right)^2 RT_{in}} \quad (57)$$

The matrix of gap areas at each seal $[A]$, is calculated from $[R_T]$ and $[D_C]$. The pressure after the last seal tooth is stored in matrix P_{N_T} .

$$[P_{N_T}]_i = [P_{lab}]_{i,N_T+1} \quad (58)$$

The mass flow is then interpolated, as abbreviated in **Equation 43**:

$$\dot{m} = \text{interp1}([P_{N_T}], [\dot{m}_{lab}], P_{out}) \quad (59)$$

The pressure across the seal $[P_{seal}]$ can then be interpolated using

$$[P_{seal}]_j = \text{interp1}([\dot{m}_{lab}], [P_{lab}]_{i=1:m,j}, \dot{m}) \quad (60)$$

As gas passes through the labyrinth seal, it is heated through windage. Millward and Edwards developed correlations that predict the amount of power loss and accompanying temperature rise through a labyrinth seal given gas properties, mass flow and labyrinth seal geometry [44].

$$[C_w]_j = \frac{\dot{m}}{\frac{1}{2}\nu[R_T]_j} \quad (61)$$

$$[\rho]_j = \frac{[P_{seal}]_j R}{T_{in}} \quad (62)$$

$$[T]_j = T_{in} \quad (63)$$

$$[Re]_j = \frac{[\rho]_j \omega [R_T]_j^2}{\nu} \quad (64)$$

$$[C_{ms}]_j = 6 \cdot 10^{-2} \left(\frac{[C_w]_j}{[Re]_j} \right)^{.55} N_T^{-.65} \quad (65)$$

$$H_{windage} = \sum_{j=1}^{N_T+1} \pi \omega^3 \cdot [\rho]_j \cdot [R_T]_j^4 \cdot [C_{ms}]_j ([L_{pitch}]_j + 2[D_T]_j) \quad (66)$$

$$T_{out} = T_{in} + \frac{H_{windage}}{\dot{m} c_p}; \quad (67)$$

4.2.6 Disc Class

Windage is the phenomenon of friction acting on a disc spinning in a viscous medium. Many correlations based on theory and experiments are available. The Disc class calculates the windage generated by radial flow past a rotating disc. It is used to represent the back face of the compressor and the shroud face inside the face seals.

Creation: Similarly to the LabSeal class, creation of a Disc object creates a contained object of class 'Gas', along with the relevant geometric parameters of the windage gap:

- R_{inner} : Inner diameter of disc.
- R_{outer} : Outer diameter of disc.
- s : Width of gap.

These parameters are shown visually in **Figure 50**

Data Flow: The windage class uses the geometric variables specified at creation, along with inlet conditions $\dot{m}$, T_{in} , P_{in} and ω to determine the following variables:

- T_{out} : Inner diameter of disc.

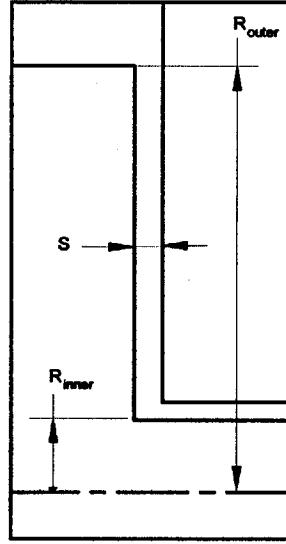


Figure 50: Disc Windage Gap Parameters

- P_{out} : Outer diameter of disc.
- H : Power consumed by windage.

Calculations: The most simple windage case is a disc spinning in an enclosed medium with no mass flow. This is also known as virtual zero mass flow. The design of the gas compressor only has leakage flows acting on windage surfaces, as the frictional effects acting within the rotor passages are an intrinsic part of the rotor analysis.

A modified form of the Reynolds number is used in windage calculations.

$$Re_w = \frac{\omega R_{outer}^2}{\nu} \quad (68)$$

For the mass flows present, the tangential velocity of the disc is much greater than the radial velocity component due to leakage, so the no-flow conditions of [45] can be used. The windage loss of each face is finally calculated using **Equation 69**. A correction factor F uses superposition to account for the effect of the inner diameter. The flow regimes considered and the best empirical equations are described in **Table 16**.

$$H_{windage} = \frac{1}{2} C_m F \rho \omega^3 R_{outer}^5 \quad (69)$$

$$F = 1 - \left(\frac{R_{inner}}{R_{outer}} \right)^5 \quad (70)$$

$$T_{out} = T_{in} + \frac{H_{windage}}{c_p} \quad (71)$$

Regime	Flow Condition	Equation
I	Laminar, Fully-Developed Flow	$Cm_I = 2\pi(s/R_{outer})^{-1} Re_w^{-1}$
II	Laminar, Undeveloped Flow	$Cm_{II} = 3.7(s/R_{outer})^{-0.1} Re_w^{-0.5}$
III	Turbulent, Fully-Developed Flow	$Cm_{III} = 0.08(s/R_{outer})^{-0.133} Re_w^{-0.25}$
IV	Turbulent, Undeveloped Flow	$Cm_{IV} = 0.102(s/R_{outer})^{0.1} Re_w^{-0.2}$

Table 16: Disc Windage Regimes and Empirical Equations

The windage regimes are plotted graphically in **Figure 51**.

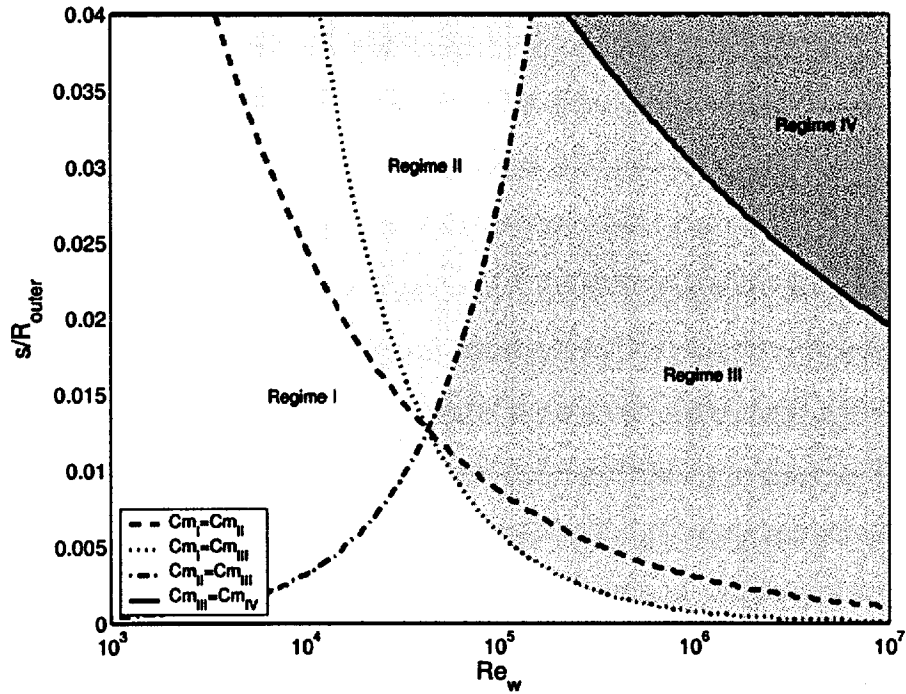


Figure 51: Disc Windage Regimes

4.2.7 Drum Class

The Drum class was not implemented, due to the low contribution to total power consumption. Modelling difficulty is caused by the low *gap/radius* ratio, and low power consumption is expected as a low power consumption is predicted for the adjacent disc windage and the lips are relatively short. However, this will change if tip speeds are raised. What follows is a description of the techniques used for predicting drum windage losses, and a rough analysis of the losses in each stage. The correlations used can be found in [46] and [47]

Similarly to the disc windage analysis described in the previous section, drum windage is split into four categories, described in Table 17. Laminar and turbulent conditions are both considered along with the presence of Taylor vortices, which are illustrated in Figure 52.

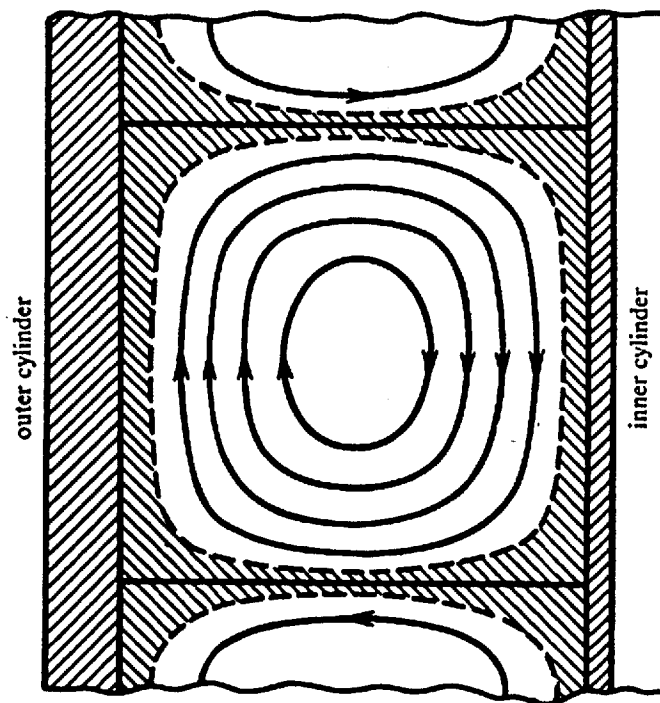


Figure 52: Taylor Vortices [46]

Regime	Flow Condition
I	Laminar Flow
II	Laminar Flow With Taylor Vortices
III	Turbulent Flow
IV	Turbulent Flow With Taylor Vortices

Table 17: Drum Windage Regime Descriptions

The drum windage operating regime for a specific condition is dependant on the Reynolds number for windage, Re_w and Taylor number, Ta , which is given by **Equation 72**.

$$Ta = \frac{2V_a s}{\nu} \quad (72)$$

Where V_a is the mean axial velocity. The regimes are demarcated in **Figure 53**. **Figure 54**, from [46], shows how the shear stress can be calculated for conditions with Taylor vortices. The solid, curved lines correspond to a rotating cylinder under turbulent conditions, while the straight, solid lines correspond to the theoretical equations for a rotating cylinder under laminar conditions. The value of s/r is only 0.0057 for the compressor rotors used, which is less than the minimum value plotted, 0.0168. Nevertheless, this can give us an indication of the drum windage losses present. **Table 18** shows the calculations made for each stage, based on the highest pressure ratio predicted and flow regime IV.

Stage	1	2	3	4
Re_w	145	217	326	489
Ta	303	303	303	303
$\log(v_2 s / \nu)$	3.3	3.5	3.6	3.8
$\log(\tau_2 / \rho v_2^2)$	-2.7	-2.8	-2.9	-2.9
W	14.8	18.9	24.1	30.7

Table 18: Stage Drum Windage Calculations

The Reynolds number and Taylor number confirm that the first stage is in Regime II, the last two stages are in Regime IV, and the second stage is in between. However,

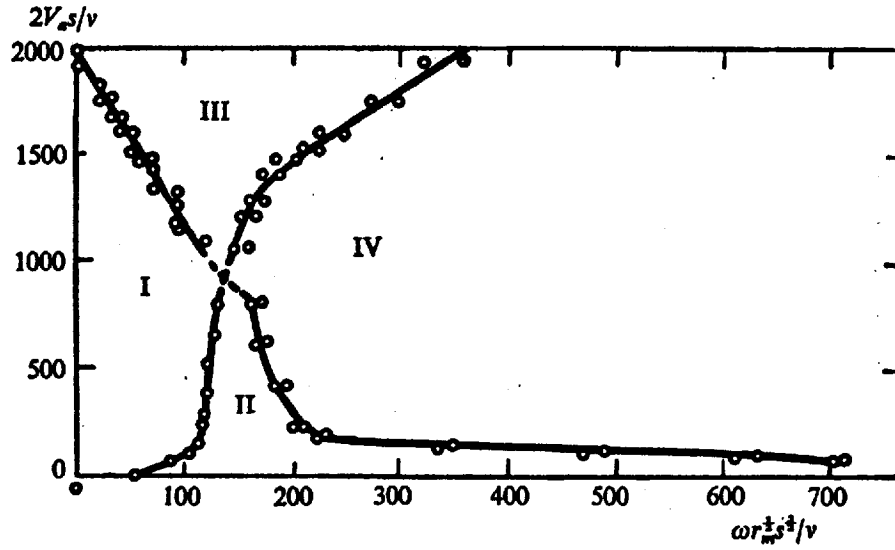


Figure 53: Drum Windage Regimes [46]

the 'worst-case' scenario is Regime IV, so this analysis is over conservative. The value of $\log(v_2 s/\nu)$ falls in the range of 3.3 to 3.8, which Figure 54 shows is relatively insensitive to changes in s/r . The shear stress predicted can be used to calculate a total drum windage power loss of 88.4W. Given the uncertainties of this model with respect to the gap to radius ratio and the relative insignificance of the predicted power loss, it was decided to not use it within the analysis framework.

Nevertheless, the structure for a Drum class has been created, and Drum objects are incorporated into the simulation. The outlet conditions are simply set equal to the inlet conditions and the power consumption is zero. The implementation is identical to the disc class.

4.2.8 Cooler Class

Intercooling is modelled by using a simple coefficient of effectiveness η_{IC} and a simple pressure loss, ΔP , defined by Equation 73 and Equation 74. These can be related to a $1 \times n$ matrix, $[Q]$, the volumetric flow rate. If $[Q]$ is used, then η_{IC} and ΔP should be

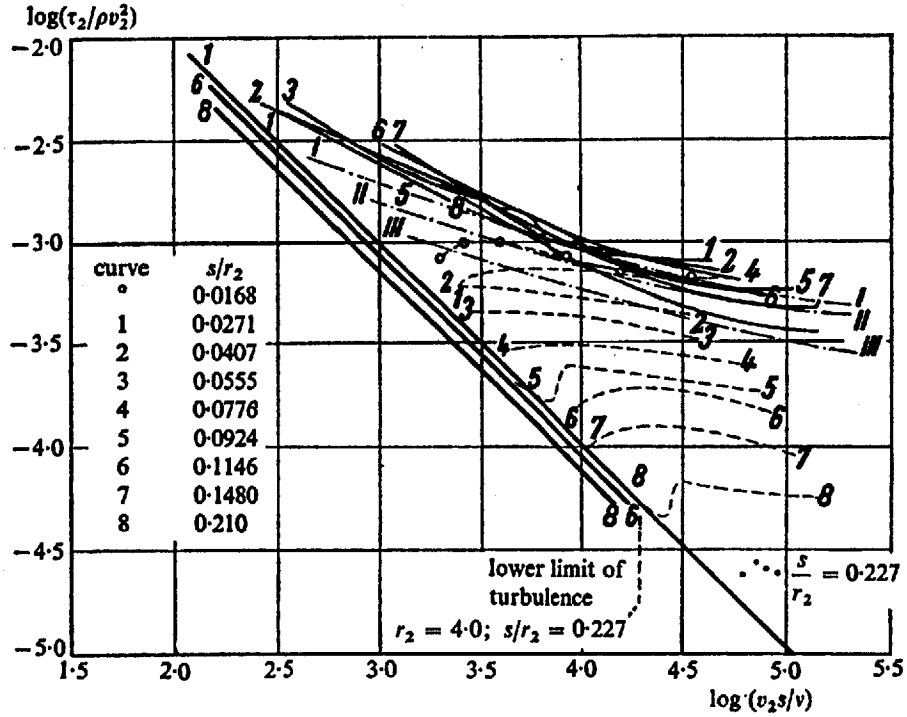


Figure 54: Drum Windage Data [46]

equally sized matrices for the purposes of interpolation.

$$\eta_{IC} = \frac{T_{in} - T_{out}}{T_{in} - T_{atm}} \quad (73)$$

$$P_{out} = P_{in} - \Delta P \quad (74)$$

ΔP and η_{IC} are established at object creation. Calculation requires T_{in} , T_{atm} and P_{in} , calculating P_{out} and T_{out} . Additionally, the volumetric flow rate is required if the intercooler performance is interpolated.

This method is adequate for modelling simple heat exchangers of known effectiveness and pressure loss. For modelling custom, integrated intercoolers, a finite difference model was programmed. The model and results (which were not integrated within the object-oriented framework) are discussed in Appendix B.

4.2.9 Stage Class

The Stage class manages the interactions of its contained objects by iterating the temperatures, pressures and mass flows while the solution converges. Total stage power consumption is also calculated.

Creation: A categorization of the objects contained within each Stage object is shown in Table 19. These are all input at object creation.

Object	Class	Contained by	Description
stage	Stage	Machine	Manages interaction of contained objects
forwardsweep	Compressor	Stage	Contains rotor, stator and map objects.
rotor	Rotor	Compressor	parameters of rotor
stator	Stator	Compressor	parameters of stator
map	Map	Compressor	Performance map of compressor
shaftseal	LabSeal	Stage	Shaft Seal Model
shroudseal	LabSeal	Stage	Shroud Seal Model
backface	Disc	Stage	Windage model of rotor
shroudface	Disc	Stage	windage model of shroud
rotorback	Drum	Stage	windage model of rotor tip
shroudlip	Drum	Stage	windage model of shroud tip
intercooler	Cooler	Stage	Intercooler model
air	Gas	All	Contains properties of working fluid

Table 19: Object Description

Data Flow: The flow of data in and out of each stage object is shown in Figure 55.

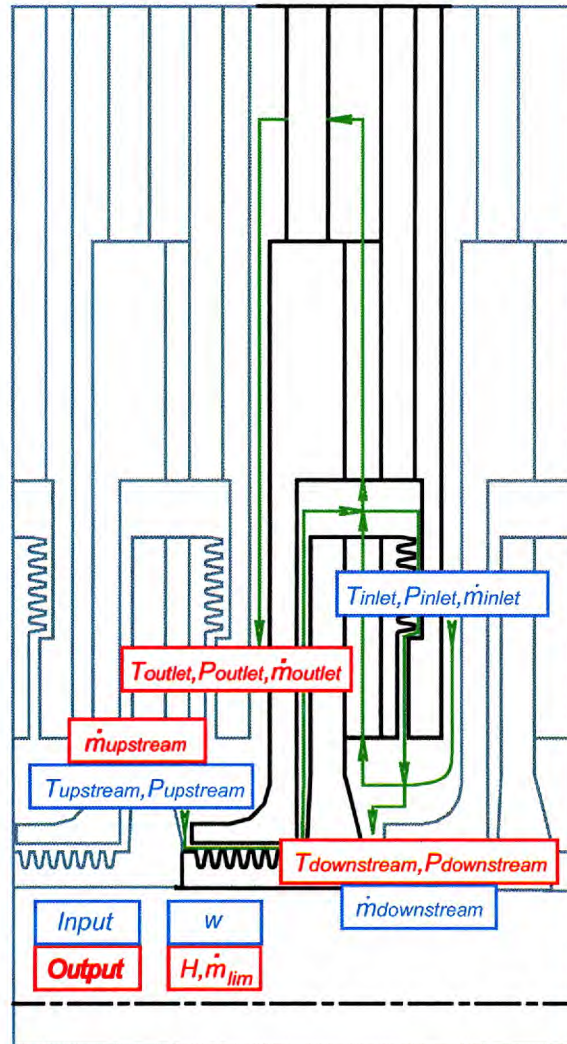


Figure 55: Stage Input and Output

The input variables of each stage calculation consist of:

- RPM : Shaft speed.
- P_{inlet} : Pressure at stage inlet.
- T_{inlet} : Temperature at stage inlet.
- $\dot{m}_{inlet}$: Mass flow at stage inlet.
- $P_{upstream}$: Pressure upstream of shaft seal.
- $T_{upstream}$: Temperature upstream of shaft seal.
- $\dot{m}_{downstream}$: Mass flow of leakage from inlet.

The output variables consist of:

- H : Power consumed.
- P_{outlet} : Pressure at stage outlet.
- T_{outlet} : Temperature at stage outlet.
- $\dot{m}_{outlet}$: Mass flow at stage outlet.
- $P_{downstream}$: Pressure at mixed inlet of compressor.
- $T_{downstream}$: Temperature at mixed inlet of compressor.
- $\dot{m}_{upstream}$: Mass flow of leakage from upstream of shaft seal.
- $\dot{m}_{surge}$: The lower mass flow limit for the given speed and inlet conditions.
- $\dot{m}_{choke}$: The upper mass flow limit for the given speed and inlet conditions.

Calculations: The station abbreviations are shown in **Figure 56**. Before iteration can begin the stage needs to be initialized. For this purpose, temperatures and pressures at rotor outlet and stator outlet are determined only by the rotor and stator and not by leakage flows. Mass flow through rotor and stator are set equal to the stage inlet mass flow and leakage flows are set equal to some initial guess. Iteration can then begin.

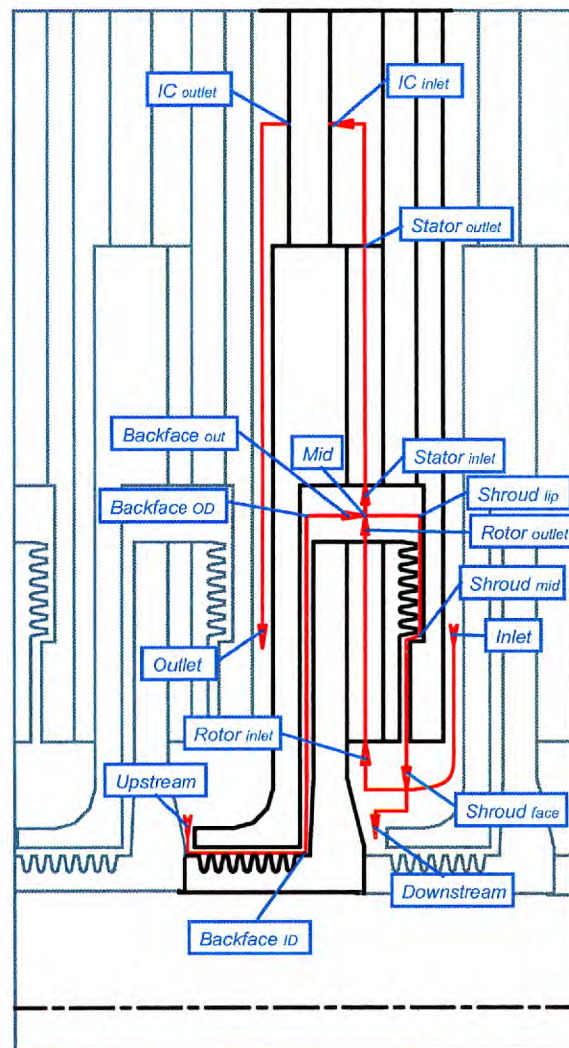


Figure 56: Stage Station Abbreviations

Inlet Calculations: Each iteration involves a number of steps. The pressure at the compressor inlet is equal to the stage inlet. This is also the pressure seen by the downstream stage.

$$P_{rotor\,in} = P_{downstream} = P_{inlet} \quad (75)$$

The mass flow at the rotor inlet is dependant on the stage inlet, the leakage from the shroud and the leakage to the downstream stage. In order to keep the iterations stable, a relaxation factor, K , is used. The best combination of stability and convergence time was found to be for $K = 0.5$. This gives equal weighting to the previous value of $\dot{m}_{rotor\,in}$ and the newly calculated value.

$$\dot{m}_{rotor\,in} = (1 - K)\dot{m}_{rotor\,in} + K(\dot{m}_{inlet} + \dot{m}_{shroud} - \dot{m}_{downstream}) \quad (76)$$

The temperature of the rotor inlet and of the downstream leakage is a flow-weighted average of the temperature:

$$T_{rotor\,in} = T_{downstream} = \frac{T_{inlet}\dot{m}_{inlet} + T_{shroud\,out}\dot{m}_{shroud}}{\dot{m}_{inlet} + \dot{m}_{shroud}} \quad (77)$$

Shroud Calculations: The 'shroudlip' object is used to determine the conditions at the outer diameter of the shroud seal, based on the conditions between the rotor and diffuser and the previous leakage through the shroud.

$$T_{mid}, P_{mid}, \dot{m}_{shroud}, \omega \rightarrow \boxed{\text{Drum} \mid \text{shroudlip}} \rightarrow P_{shroudlip}, T_{shroudlip}, H_{shroudlip} \quad (78)$$

The 'shroudseal' object is then used to determine the new level of shroud leakage and the conditions inside of the labyrinth seal on the shroud. The previous value of $P_{shroudmid}$, the pressure at the inside of the labyrinth seal, is used to determine the pressure across

the seal.

$$P_{shroudlip}, T_{shroudlip}, P_{shroudmid}, \omega \rightarrow \boxed{\text{LabSeal} \mid \text{shroudseal}} \rightarrow T_{shroudmid}, \dot{m}_{shroud}, H_{shroudseal} \quad (79)$$

The windage acting on the area inside of the labyrinth seal is calculated by the 'shroud-face' object.

$$T_{shroudmid}, P_{shroudmid}, \dot{m}_{shroud}, \omega \rightarrow \boxed{\text{Disc} \mid \text{shroudface}} \rightarrow T_{shroudout}, P_{shroudout}, H_{shroudface} \quad (80)$$

The difference in pressure across the inner face of the shroud is then calculated using **Equation 81**

$$\Delta P_{shroudface} = P_{shroudout} - P_{shroudmid} \quad (81)$$

The value of $P_{shroudmid}$ is then updated. Again, the convergence is stabilized by the relaxation factor, K .

$$P_{shroudmid} = (1 - K)P_{shroudmid} + K(P_{rotorin} - \Delta P_{shroudface}) \quad (82)$$

The total windage for the shroud is then calculated based on the individual contributions of the three objects.

$$H_{shroud} = H_{shroudface} + H_{shroudseal} + H_{shroudlip} \quad (83)$$

Backface Calculations: The calculations for the backface are similar to the shroud calculations. The values passed from the upstream stage and the previous pressure at the inner diameter of the backface, P_{ID} are used to calculate the leakage using the 'shaftseal' object.

$$P_{upstream}, T_{upstream}, P_{ID}, \omega \rightarrow \boxed{\text{LabSeal} \mid \text{shaftseal}} \rightarrow T_{ID}, \dot{m}_{upstream}, H_{shaftseal} \quad (84)$$

The leakage is then used to determine the windage across the back face and outer diameter of the rotor using the 'rotorback' and 'rotorlip' objects.

$$T_{ID}, P_{ID}, \dot{m}_{upstream}, \omega \rightarrow \boxed{\text{Disc} \mid \text{rotorback}} \rightarrow T_{OD}, P_{OD}, H_{rotorback} \quad (85)$$

$$T_{OD}, P_{OD}, \dot{m}_{upstream}, \omega \rightarrow \boxed{\text{Drum} \mid \text{rotorlip}} \rightarrow P_{backfaceout}, T_{backfaceout}, H_{rotorlip} \quad (86)$$

The pressure across the back face and lip is then calculated.

$$\Delta P_{backface} = P_{backfaceout} - P_{ID} \quad (87)$$

This pressure difference is used to determine the new pressure at the inner diameter of the back face. The relaxation factor, K , is used to stabilize the convergence.

$$P_{ID} = (1 - K)P_{ID} + K(P_{mid} - \Delta P_{backface}) \quad (88)$$

The total windage of the back face and shaft seal is calculated.

$$H_{backface} = H_{shaftseal} + H_{rotorback} + H_{rotorlip} \quad (89)$$

Compressor Calculations: The compressor object is represented by the 'forwardsweep' object and is used to update the conditions at rotor exit and stator exit based on the conditions at rotor inlet and the conditions at the midpoint between the rotor and stator. The range of the compressor at the given speed and inlet conditions is also returned.

$$\begin{array}{c}
P_{rotor_{inlet}}, T_{rotor_{inlet}}, \dot{m}_{rotor_{inlet}}, P_{mid}, T_{mid}, \dot{m}_{stator_{inlet}}, \omega \\
\downarrow \\
\boxed{\text{Compressor | forwardsweep}} \\
\downarrow \\
P_{rotor_{outlet}}, T_{rotor_{outlet}}, P_{stator_{outlet}}, T_{stator_{outlet}}, \dot{m}_{lim}
\end{array} \tag{90}$$

The pressure at the midpoint is determined solely by the rotor outlet pressure. This is likely to not be accurate, as the leakage will most certainly affect the compressor operation. However, the specifics of this aspect of operation are not yet known.

$$P_{mid} = P_{rotor_{outlet}} \tag{91}$$

The midpoint temperature is the temperature weighted by mass flow from the rotor outlet and shaft seal leakage. If there is upstream leakage across the shaft seal or a pressure loss in the rotor, this will not be true, but these conditions will only be present when the machine is not operating properly.

$$T_{mid} = \frac{T_{rotor_{outlet}} \dot{m}_{rotor_{outlet}} + T_{backface_{out}} \dot{m}_{upstream}}{\dot{m}_{rotor_{outlet}} + \dot{m}_{upstream}} \tag{92}$$

The mass flow at the stator inlet is determined by the rotor outlet flow and the leakages across the shaft seal and the upstream leakage. The constant K is once again used to stabilize the convergence.

$$\dot{m}_{stator_{inlet}} = (1 - K)(\dot{m}_{stator_{inlet}}) + K(\dot{m}_{rotor_{outlet}} + \dot{m}_{upstream} - \dot{m}_{shroud}) \tag{93}$$

Intercooler Calculations: The mass flow outlet is determined by the stator outlet as no leakage is present in the intercooler.

$$\dot{m}_{outlet} = \dot{m}_{stator_{outlet}} \quad (94)$$

The 'intercooler' object is used to calculate the outlet temperature and pressure.

$$T_{stator_{out}}, P_{stator_{out}}, \dot{m}_{outlet}, T_{atm} \rightarrow \boxed{\text{Cooler | intercooler}} \rightarrow T_{outlet}, P_{outlet} \quad (95)$$

Total Stage Power Consumption Calculation: The total power consumption of the stage is then calculated.

$$H_{stage} = H_{compression} + H_{shroud} + H_{backface} \quad (96)$$

4.2.10 Machine Class

The Machine class contains a $1 \times N$ array of 'Stage' objects, corresponding to the number of stages in the machine being simulated. A machine object manages the interaction of these stages along with other necessary parameters, such as the sealing pressure at the final stage.

Creation: Upon creation, a machine object requires the following inputs:

- N : The number of stages.
- $[Stages]$: A $1 \times N$ array containing Stage objects.
- $P_{sealing}$: The pressure of the sealing air at the final stage
- $T_{sealing}$: The pressure of the sealing air at the final stage
- $\dot{m}_{guessleak}$: An initial guess of the leakage flows within the machine at design point.
- $fluid$: An object of type Gas, containing the properties of the working fluid

Data Flow: Before any calculations can be done, the machine object must be initialized. This is done at $\dot{m}_{design}$ and ω_{design} with inlet conditions of T_{amb} and P_{atm} stored in the *fluid* object. $1 \times N$ arrays are created containing all of the data output for each stage:

- $[H]$: Power Consumed at each stage.
- $[P_{outlet}]$: Pressure at stage outlet.
- $[T_{outlet}]$: Temperature at stage outlet.
- $[\dot{m}_{outlet}]$: Mass flow at stage outlet.
- $[P_{downstream}]$: Pressure at mixed inlet of compressor.
- $[T_{downstream}]$: Temperature at mixed inlet of compressor.
- $[\dot{m}_{upstream}]$: Mass flow of leakage from upstream of shaft seal.
- $[F_{surge}]$: An array of boolean variables indicating if a stage, i , is surging ($[F_{surge}]_i = 1$) or not ($[F_{surge}]_i = 0$).
- $[F_{choke}]$: An array of boolean variables indicating if a stage, i , is choked ($[F_{choke}]_i = 1$) or not ($[F_{choke}]_i = 0$).
- $[\eta]$: Isentropic Efficiency.
- $[\dot{m}_{recirc}]$: Recirculation at each stage required to prevent stall.

When iteration is to begin, the following parameters are required:

- P_0 : Machine inlet pressure.
- T_0 : Machine inlet temperature.
- $\dot{m}_0$: Machine inlet mass flow.
- RPM : Machine shaft speed.

Calculations: Each stage object in $[Stages]$ is then calculated in order. The input used for the i^{th} stage is shown in Table 20

Input	$i = 1$	$1 < i < N$	$i = N$
RPM	RPM	RPM	RPM
$\dot{m}_{inlet}$	$\dot{m}_0 + [\dot{m}_x]_i$	$[\dot{m}_{outlet}]_{i-1} + [\dot{m}_x]_i - [\dot{m}_x]_{i-1}$	$[\dot{m}_{outlet}]_{i-1} + [\dot{m}_x]_i - [\dot{m}_x]_{i-1}$
P_{inlet}	P_0	$[P_{outlet}]_{i-1}$	$[P_{outlet}]_{i-1}$
T_{inlet}	$\frac{T_0 \dot{m}_0 + [T_{outlet}]_i [\dot{m}_x]_i}{[\dot{m}_{inlet}]_i}$	$\frac{[T_{outlet}]_{i-1} [\dot{m}_{outlet}]_{i-1} + [T_{outlet}]_i [\dot{m}_x]_i}{[\dot{m}_{inlet}]_i}$	$\frac{[T_{outlet}]_{i-1} [\dot{m}_{outlet}]_{i-1} + [T_{outlet}]_i [\dot{m}_x]_i}{[\dot{m}_{inlet}]_i}$
$\dot{m}_{downstream}$	0	$[\dot{m}_{upstream}]_{i-1}$	$[\dot{m}_{upstream}]_{i-1}$
$P_{upstream}$	$[P_{downstream}]_{i+1}$	$[P_{downstream}]_{i+1}$	$P_{sealing}$
$T_{upstream}$	$[T_{downstream}]_{i+1}$	$[T_{downstream}]_{i+1}$	$T_{sealing}$

Table 20: Input Parameters for the i^{th} Stage

$[\dot{m}_x]$ contains the recirculation at each stage. After the calculation of stage i , $[F_{surge}]_i$ and $[F_{choke}]_i$ are determined. The following conditional analysis is performed:

$$\text{if } [\dot{m}]_i < [\dot{m}_{surge}]_i, \text{ then } [F_{surge}]_i = 1, [F_{choke}]_i = 0, [\dot{m}_x]_i = [\dot{m}_{surge}]_i - [\dot{m}_x]_{i-1} \quad (97)$$

$$\text{if } [\dot{m}_{surge}]_i \leq [\dot{m}]_i \leq [\dot{m}_{choke}]_i, \text{ then } [F_{surge}]_i = 0, [F_{choke}]_i = 0, [\dot{m}_x]_i = 0 \quad (98)$$

$$\text{if } [\dot{m}_{choke}]_i < [\dot{m}]_i \text{ then } [F_{surge}]_i = 0, [F_{choke}]_i = 1, [\dot{m}_x]_i = 0, PR_{compressor} = 1.001 \quad (99)$$

Recirculation from the stage outlet back to the stage inlet is added to stop the compressor from surging. Any choked compressors are given a low pressure ratio of 1.01. These points are flagged as choked or stalled until conditions change back to a normal operating condition. This aides the stability of the convergence. Stalled and choked points are not included in the results and associated maps.

The efficiency is calculated from the isentropic power consumption for the total pressure achieved. If there is leakage into the compressor from the sealing air, the efficiency should be determined from the inlet mass flow using H_{s1} . If there is leakage out of the compressor due to insufficient sealing from the sealing air, the efficiency should be determined from the inlet mass flow, using H_{s2} . If there is no mass flow at the HP seal, then a nominal isentropic power consumption can be calculated from $H_{s0} = H_{s1} = H_{s2}$:

$$H_{s1} = [\dot{m}_{outlet}]_N \left(\frac{\gamma RT_0}{\gamma - 1} \right) \left[\left(\frac{[P_{outlet}]_N}{[P_{inlet}]_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]; \quad (100)$$

$$H_{s2} = [\dot{m}_{inlet}]_1 \left(\frac{\gamma RT_0}{\gamma - 1} \right) \left[\left(\frac{[P_{outlet}]_N}{[P_{inlet}]_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]; \quad (101)$$

The appropriate H_s is then used along with the sum of the stage power consumptions:

$$\eta_s = \frac{H_s}{\sum_{i=1}^N [H]_i} \quad (102)$$

The solution is then iterated until there is less than 0.5% change in the solution per iteration for all compressor mass flows, temperatures and pressures and less than 2.0% change for leakage mass flows. All solutions that were found to converge do so within 30 iterations. Each OP is run for 60 iterations before being considered to be non-converging.

4.3 Simulation Limitations

There are known simulation limitations due to both difficulties in implementing certain phenomena as well as unknowns in the component models.

Seal-Windage Interactions: It is unknown how the windage and seals will interact. Interactions are thought to be insignificant, as seal performance is generally considered to be independent of shaft speed. However, it would be beneficial to confirm this for this particular geometry using CFD.

Leakage-Compressor Interactions: It is unknown how the leakage from the shaft seal and to the shroud will effect the flow between the rotor and stator. This is not currently possible due to difficulties in CFD modelling of the compressor, even without the influence of leakage.

Lack of Drum windage modelling: Due to lack of a suitable drum windage model, the effects of drum windage at the rotor and shroud tips have been ignored. It would be beneficial to use CFD to assess both disc and drum windage specific to this geometry.

Face Seal Pumping: The face seal leakage does not take into account the pumping action from the rotor. This will likely reduce the losses, but again, this is not thought to be significant due to high tangential velocity of the rotor relative to the radial gas velocity.

4.4 Simulation Operation

The classes described above were utilized for the simulation of the multistage forward-swept compressor. It must be mentioned that the design and simulation were performed concurrently, based on the single stage CFD results of Vine. The simulation described is that of the arrangement implemented into the test rig, described later in chapter 5.

4.4.1 Compressor Arrangement

The arrangement described in section 3 was simulated using this analysis. The first simulations that were performed used the CFD results of Vine described in section 2.6.3 for the individual stage maps, with the volumetric flow scaled proportional to blade height, as described in section 3.3. The nominal shaft and shroud seal clearances were chosen at the expected clearances $50\mu m$ and $100\mu m$ respectively. The effect of seal clearance on efficiency was also investigated. An intercooler was simulated at an effectiveness of 100%

after the second stage, as the water-air intercooler used in the experimental work was oversized for the required heat transfer.

4.4.2 Convergence of Solution

Figure 57, **Figure 58** and **Figure 59** show the convergence of temperature, pressure and mass flow for the machine described above, at $\dot{m} = 12g/s$ and $60kRPM$. It can be seen that the temperature and pressure converge very quickly, but it can take many more iterations for the mass flow to stabilize.

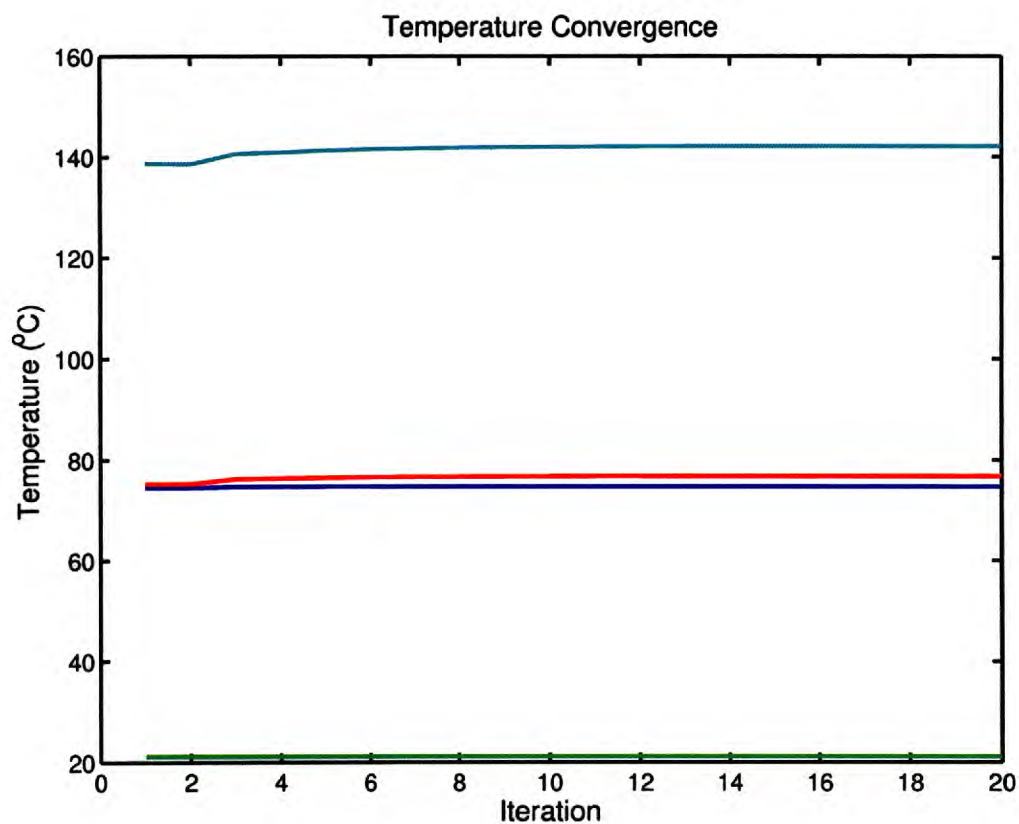


Figure 57: Temperature Conversion

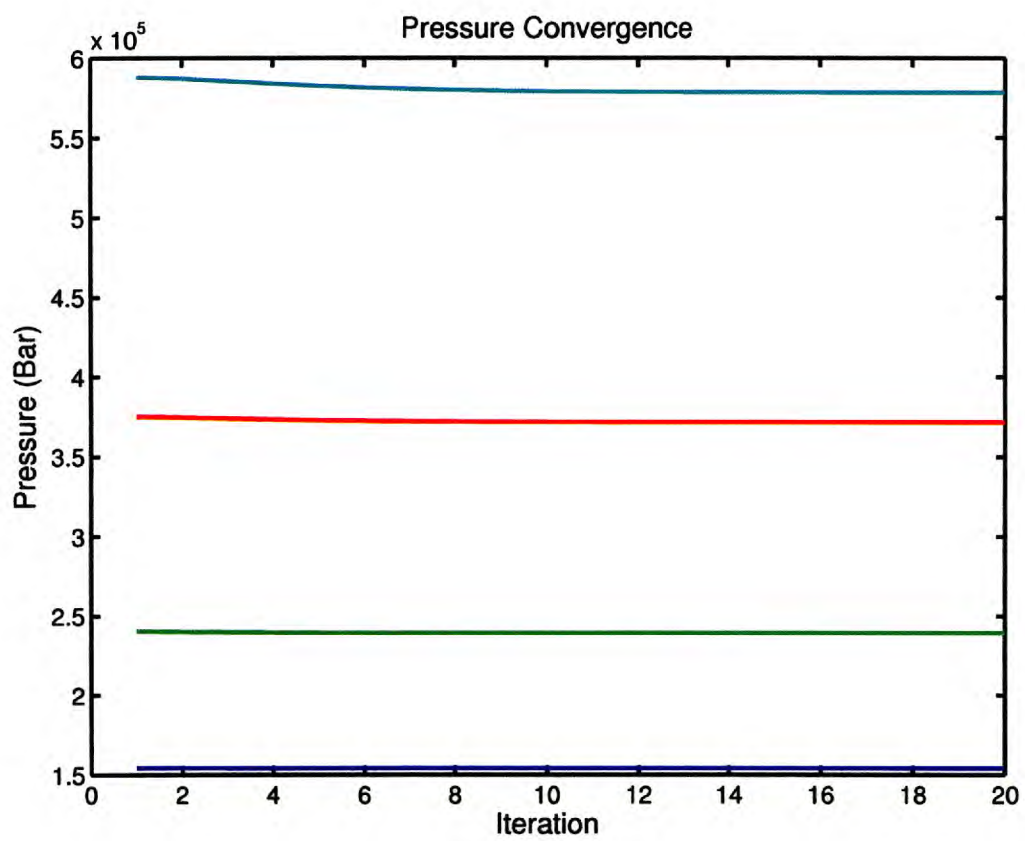


Figure 58: Pressure Conversion

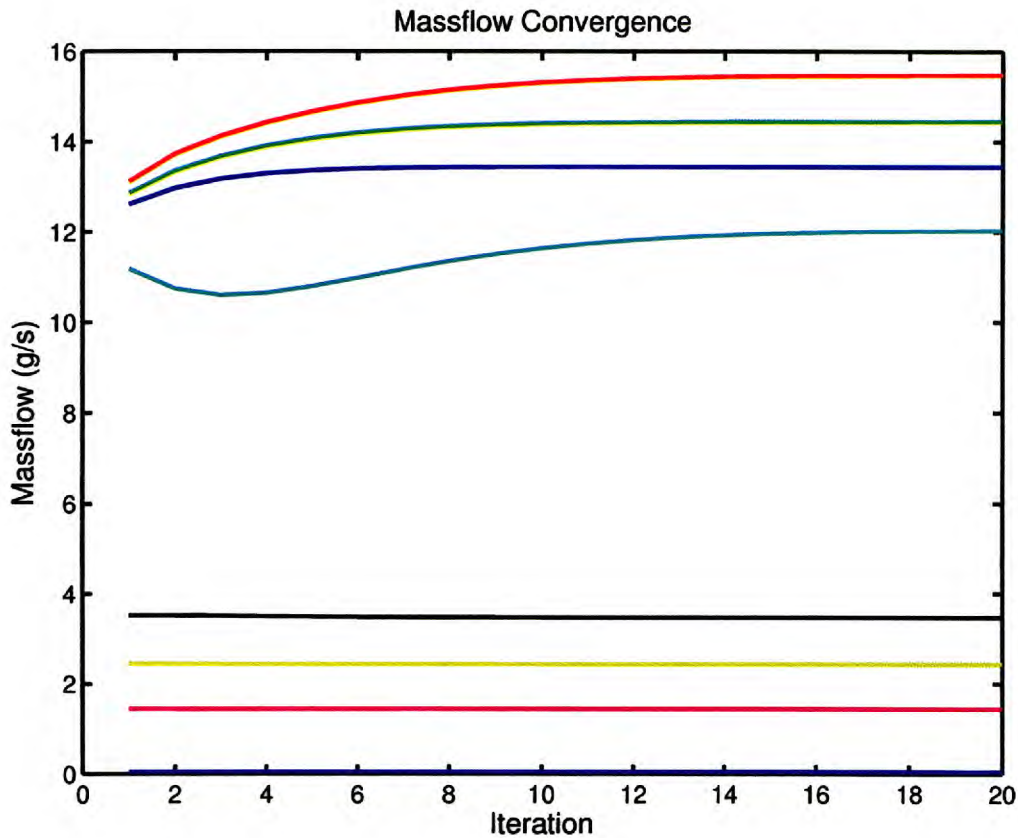


Figure 59: Mass-flow Conversion

4.5 Simulation Results

4.5.1 Compressor Map

The simulated compressor map is shown in **Figure 60**. This is compiled by running the described code at different speeds and mass flows. The performance of each individual stage is entered via an input file, and is based on the CFD predictions of Vine [35]. The shaft and shroud seal clearances are 50mm and 100mm, respectively. The range is less than that of the single stage map as the off-design operation changes the inlet conditions of the upstream stages.

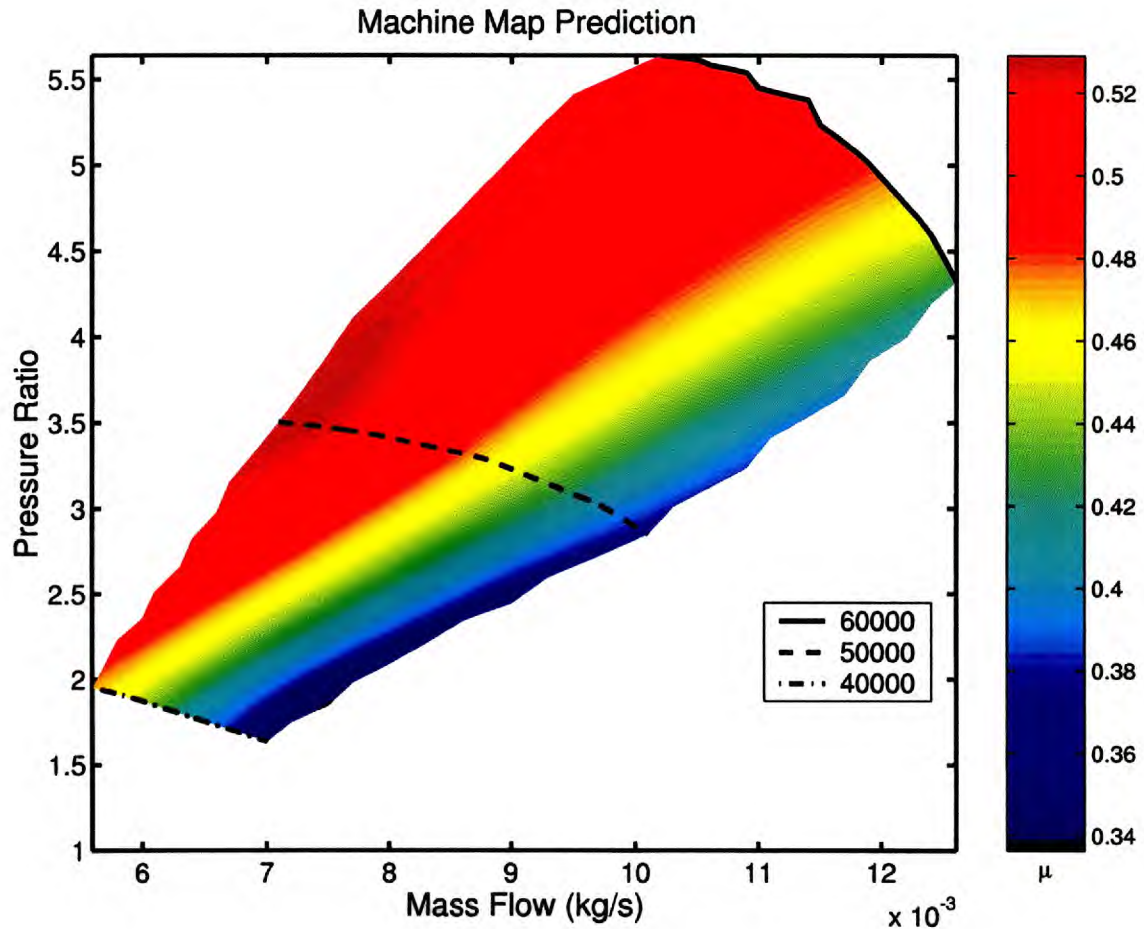


Figure 60: Simulation Results for Initial Design

4.5.2 Effect of Seal Clearance on Performance

The effects of seal clearance were investigated by simulating the compressor by varying both shaft and shroud seal clearance $\pm 50\%$. The effect that this has on both range and efficiency can be seen by comparing the simulation in **Figure 60** to the map of the reduced clearance case shown in **Figure 61** and the increased clearance case shown in **Figure 62**. The case of increased clearance did not converge at high speeds due to excessive leakage, indicating the importance of tight seal clearances in any experimental work.

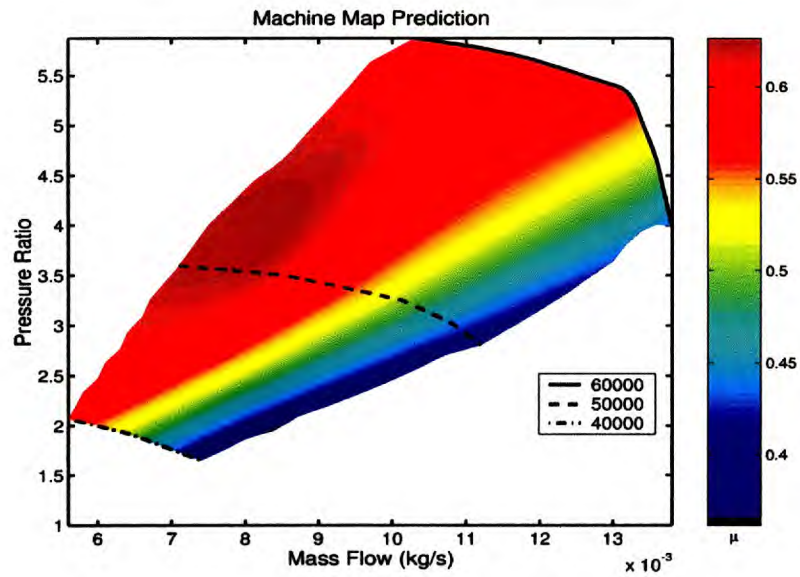


Figure 61: Simulation Results for Initial Design With Seal Clearance of $25\mu m$ and Shroud Seal Clearance of $50\mu m$.

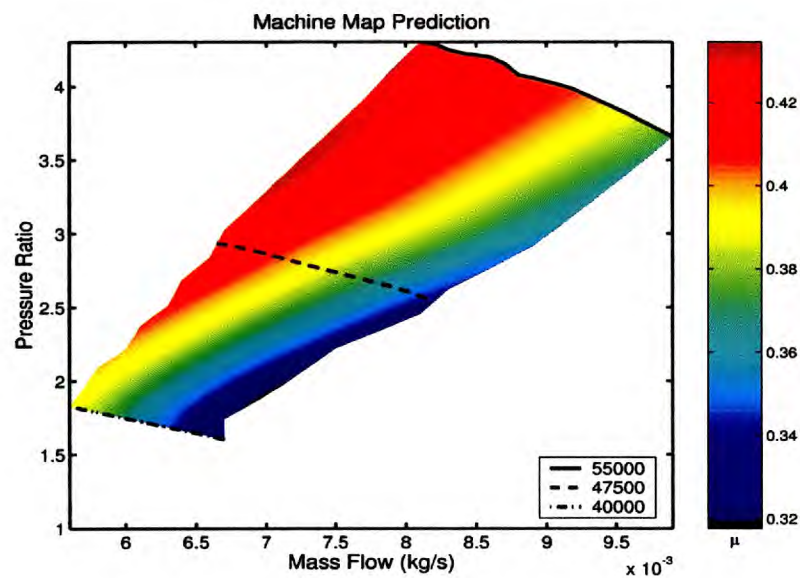


Figure 62: Simulation Results for Initial Design With Seal Clearance of $75\mu m$ and Shroud Seal Clearance of $150\mu m$.

4.5.3 Multistage Simulation Based On Single Stage Experiments.

Simulations were attempted for the compressor based on the single stage experiments at 60kRPM, which were performed after the multistage compressor had been designed and manufacture had begun. Simulations based on single stage test results indicated that the multistage compressor should not function properly at any points. However, this proved to be false, as the upstream stages have a stabilizing effect on downstream stages. This is explained further in chapter 7.

4.5.4 Further Simulations Based on Multistage Experiments

Further simulations are described in section 7.3. These simulations are based on individual stage data gathered from the test rig described in the following chapter. These results generally agree with the experimental observations at operating points where none of the stages surge or choke. These results are also used to reevaluate the stage matching. A design with updated stage matching is simulated in section 7.4.

5 Test Rig Design and Manufacturing

A test rig was designed and built to evaluate the performance of the multistage compressor arrangement developed in chapter 3 and simulated in chapter 4. It was intended to perform the following functions:

1. To provide a prototype compressor for natural gas fuelling of a 100kW microturbine, meeting the requirements established in section 1.4 and of the design described in chapter 3.
2. To experimentally investigate the matching of low specific speed compressors, in order to assess the computational matching methods described in section 4.

Air is chosen as the working fluid for reasons of safety and convenience. A delivery of 12 g/s at 4.5 bar was chosen as a design target. A schematic of the test rig is shown in **Figure 63**, a 3D exploded view is shown in **Figure 64** and a 3D cutaway is shown in **Figure 65**. A photo is shown in **Figure 66**. The four stages are identical in blade and diffuser profile but differ in blade height, as shown in **Table 15**. The shaft is driven at the high pressure end by a variable speed motor and gearbox. It is designed to run with either oil-lubricated or grease-packed bearings. Air blown seals are implemented in such a way that no oil is introduced into the compression region. A counterflow intercooler has been salvaged from a compressor of approximately 15kW of power consumption, and is installed after the 2nd stage. The three dimensional models of all components were created using Pro-Engineer [48] parametric modelling software. The detailed drawings were produced from the solid models for the purposes of manufacturing. These are shown in appendix C .

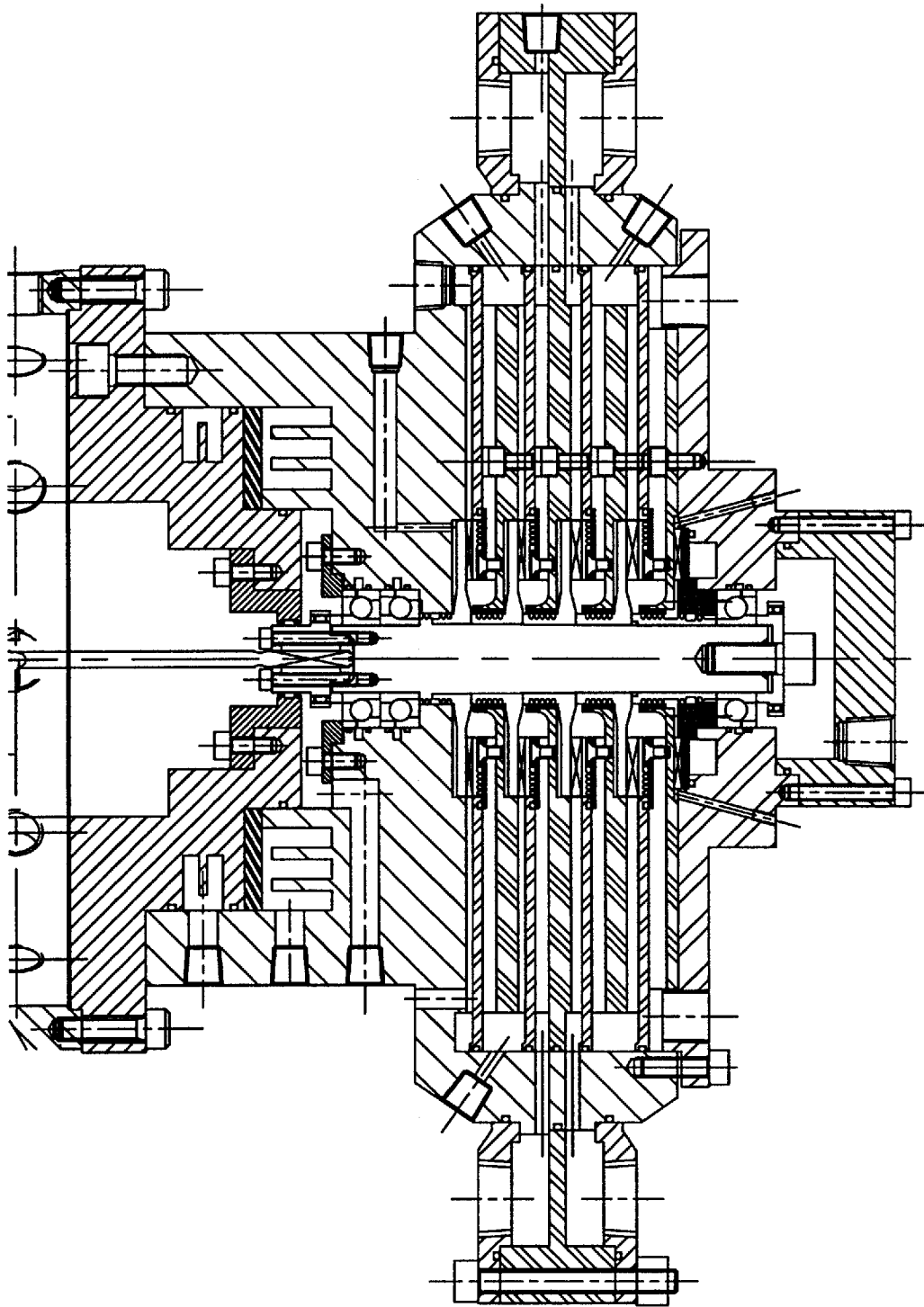


Figure 63: Detailed Drawing of Test Rig

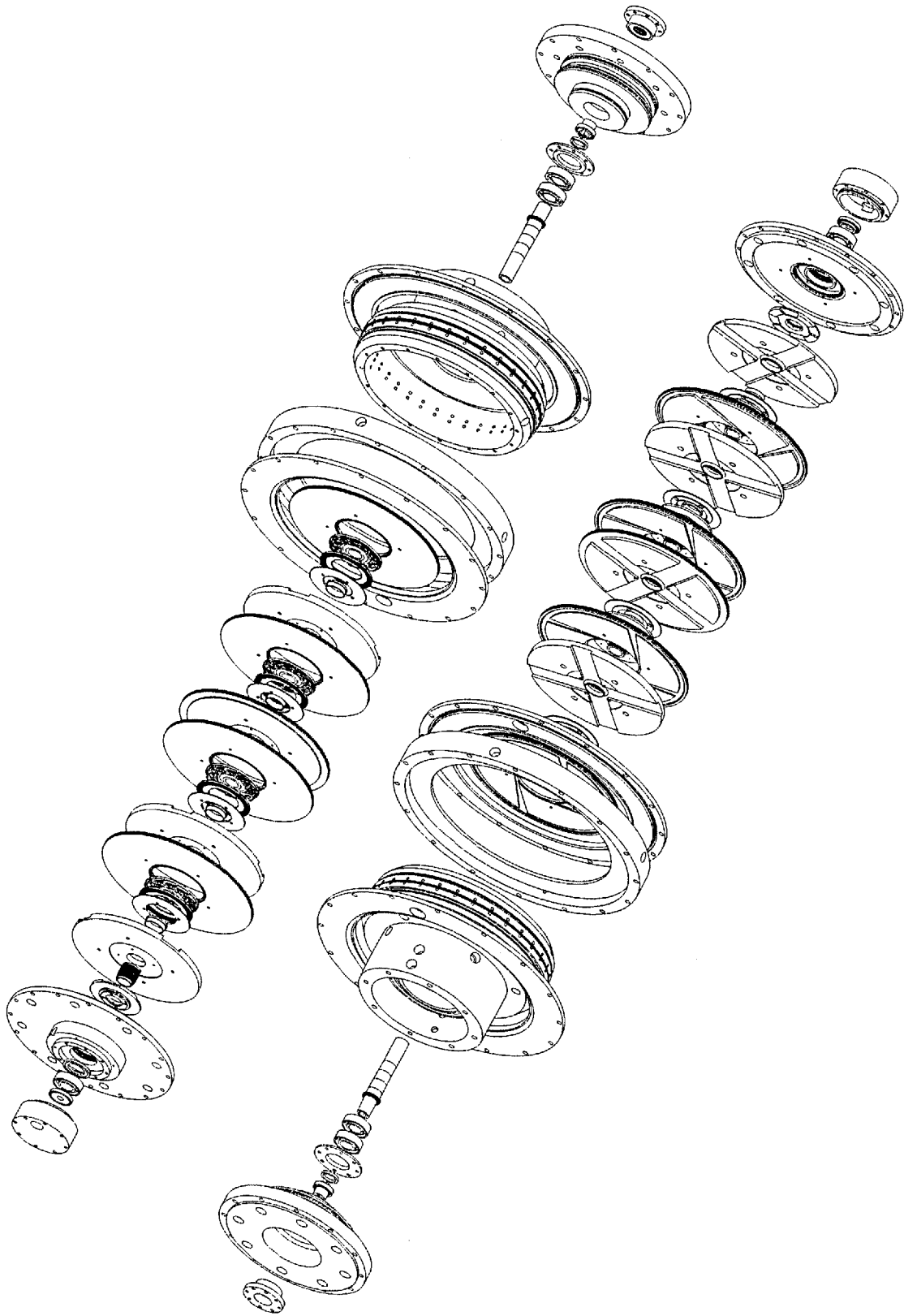


Figure 64: Test Rig Exploded Views

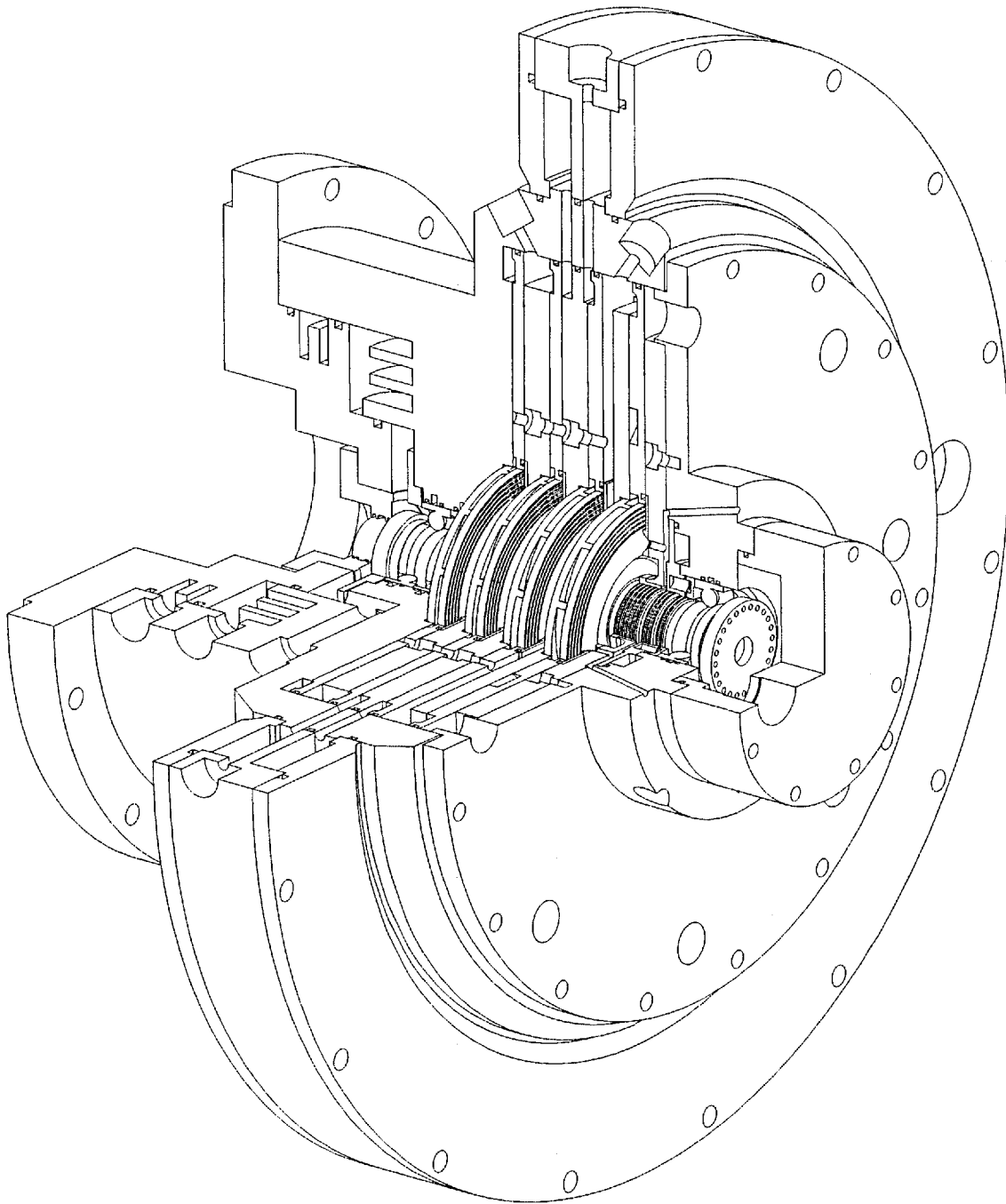


Figure 65: Test Rig Cut-Away View

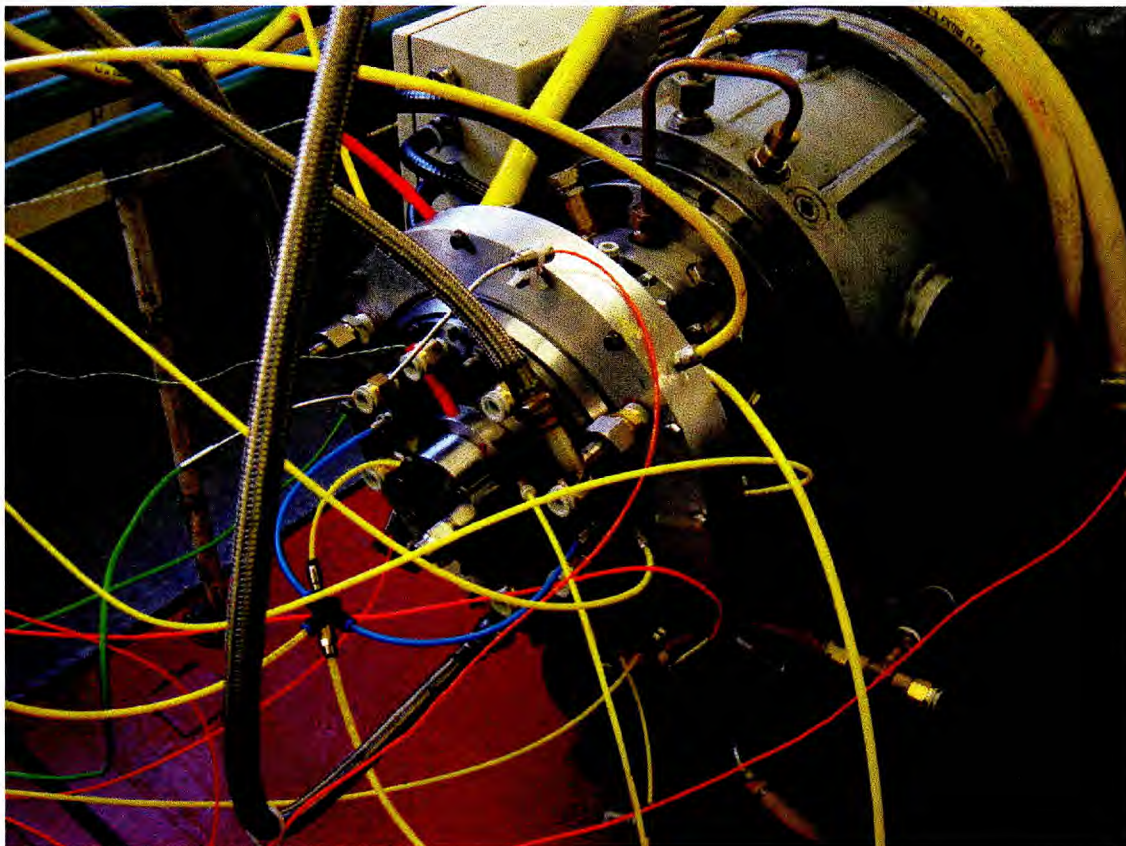


Figure 66: Rig Photograph

5.1 Rotors

The blade profile of the rotors are all based on the same blade cross section, shown in **Figure 30**, to aide in manufacturing and analysis. Stress analysis and CFD have been performed for the first stage, and past analysis has shown that cropped blades of similar cross section have lower stresses and slightly lower aerodynamic performance.

5.1.1 Fabrication Method

For mass production of a similar gas compressor, it would be desirable to cast the rotors from aluminium. However, this is not feasible for low production numbers. Instead, the rotors and shrouds were CNC machined as separate parts and vacuum-brazed together. **Figure 67** shows the steps in the brazing process. Two separate parts made from 416

S21 stainless steel are shown in (A). The blades of the rotor fit into a lip in the shroud and the parts are brazed together and heat treated to a yield stress of 500MPa (B). After brazing, the lip is removed with a cylindrical grinder (C). The rotors are assembled on the shaft, and the lip is removed on each rotor. The face seals are also trimmed to the correct axial length. This ensures a minimal amount of face seal run-out in the assembled compressor. A photograph of the assembled compressor rotor is shown in **Figure 68**.

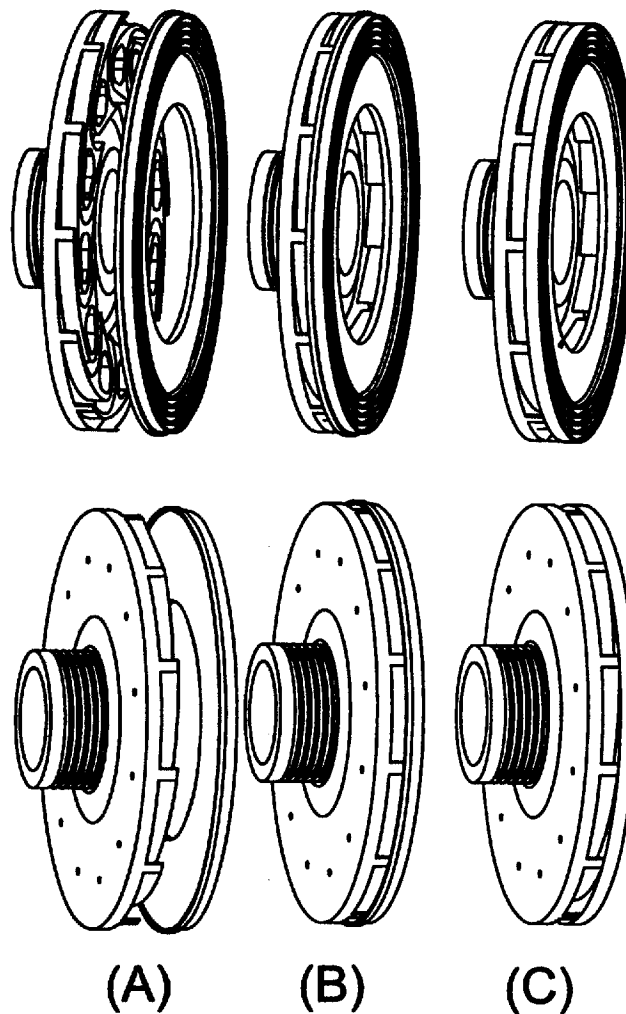


Figure 67: Rotor Assembly Process
(A) Components, (B) Brazed, (C) Machined

The blades are hollow in order to allow excess brazing material to escape the gap and the holes in the rotor allow air to escape the hollow blades. The hollow blades also lower

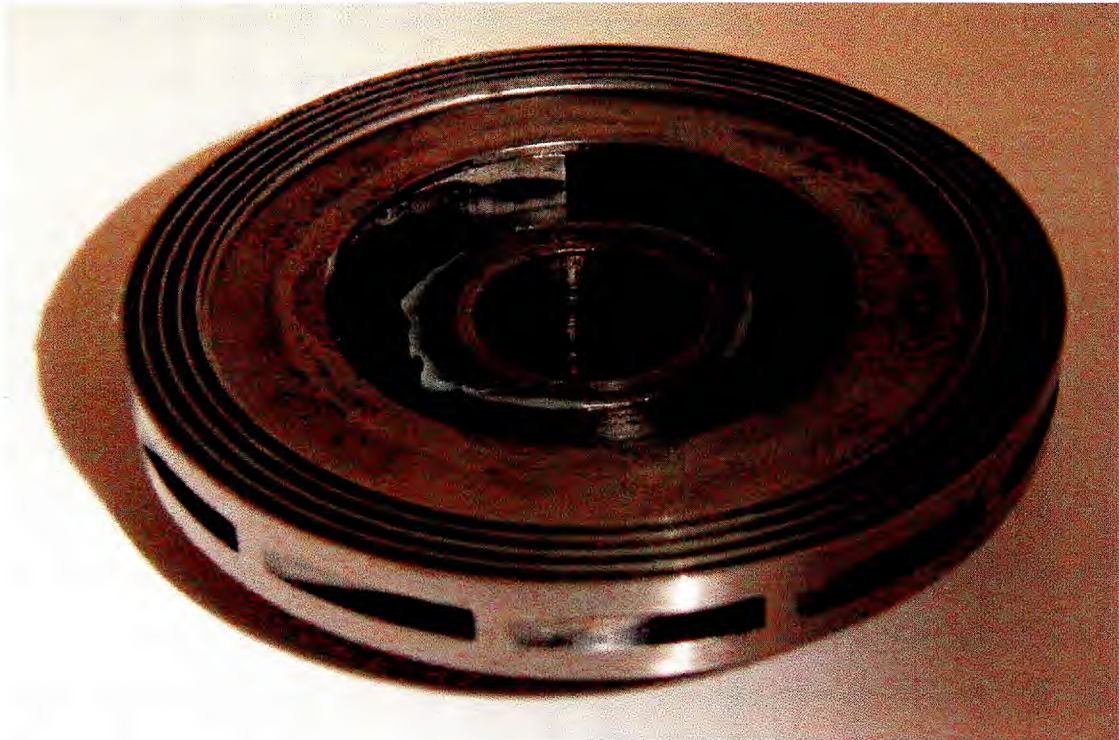


Figure 68: Photo of Assembled Rotor

the blade mass, reducing stress and raising natural frequencies. Steel has advantages as a rotor material, such as hardness, durability and temperature resistance, which are important for a test rig without established operating conditions. The labyrinth seals were manufactured by using a 1mm wide cutting tool with a semicircular tip. The face seal is at a high diameter due to difficulties in accurately manufacturing low diameter circular grooves in a face.

5.1.2 Structural Analysis

Stress analysis has been performed using ANSYS [49], a commercially available, finite-element analysis (FEA) package. The rotor is imported as an IGES file, and meshed using tetrahedral elements. The grooves in the labyrinth seals are square bottomed, due to complications of meshing toroidal surfaces. The rotor is constrained in the axial and tangential directions at one point at the bore on the seal end of the rotor.

Figure 69 shows the Von-Mises equivalent stress, expressed in kPa. The maximum stress of 400 MPa at the bore gives a safety factor of 1.25 on yielding. Higher stresses are detected at discrete nodes, however this is due to perfect stress concentrations which will not be present in reality. Even if yielding at the bore does occur, the toughness of the material should only cause local plastic deformation and not failure. The fatigue life will need to be evaluated for subsequent designs, but was not of a concern in the test rig, due to the low number of hours spent testing.

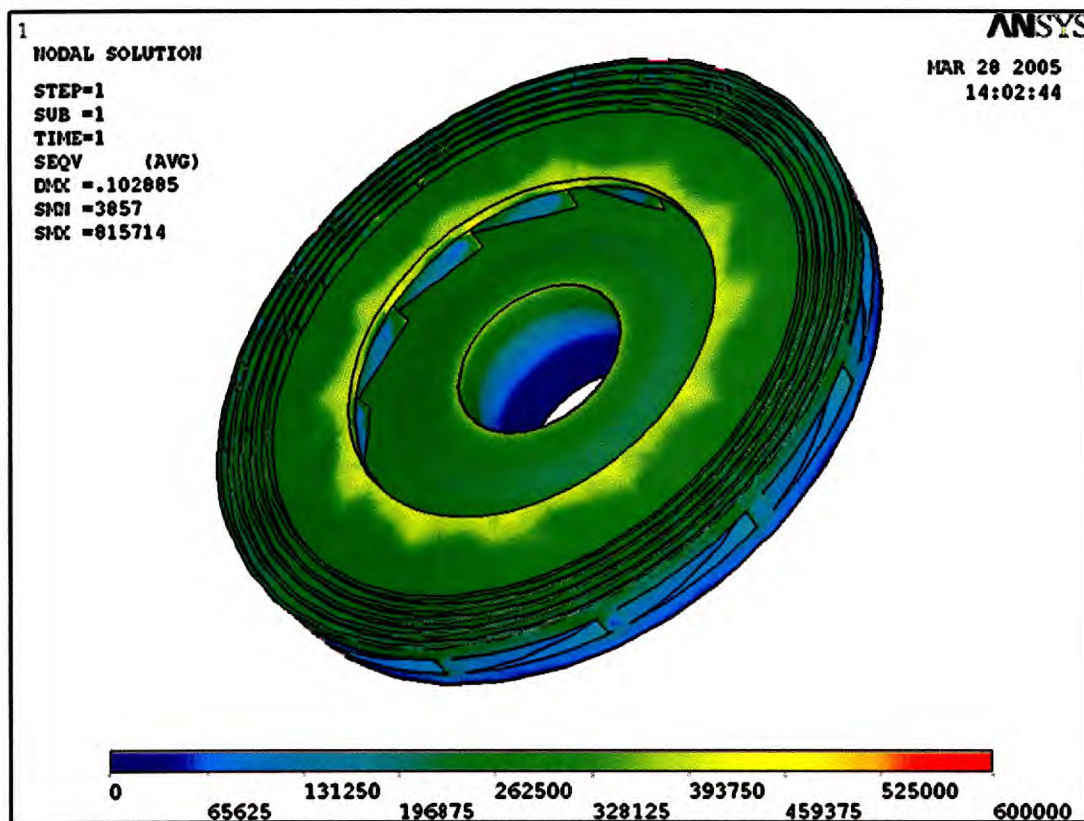


Figure 69: Von Mises Stress in KPa

Figure 70 and **Figure 71** show the axial and total displacements of the rotor, expressed in mm. The radial displacement predicted in the shaft seals is less than a micron, however the maximum axial displacement of the rotor seals is $94\mu m$, increasing the clear-

ance at higher speeds. Tip clearance variation was simulated in section 4.5.2. More work can potentially be done to attempt to minimize this additional clearance, possibly at the expense of increased stress or axial length.

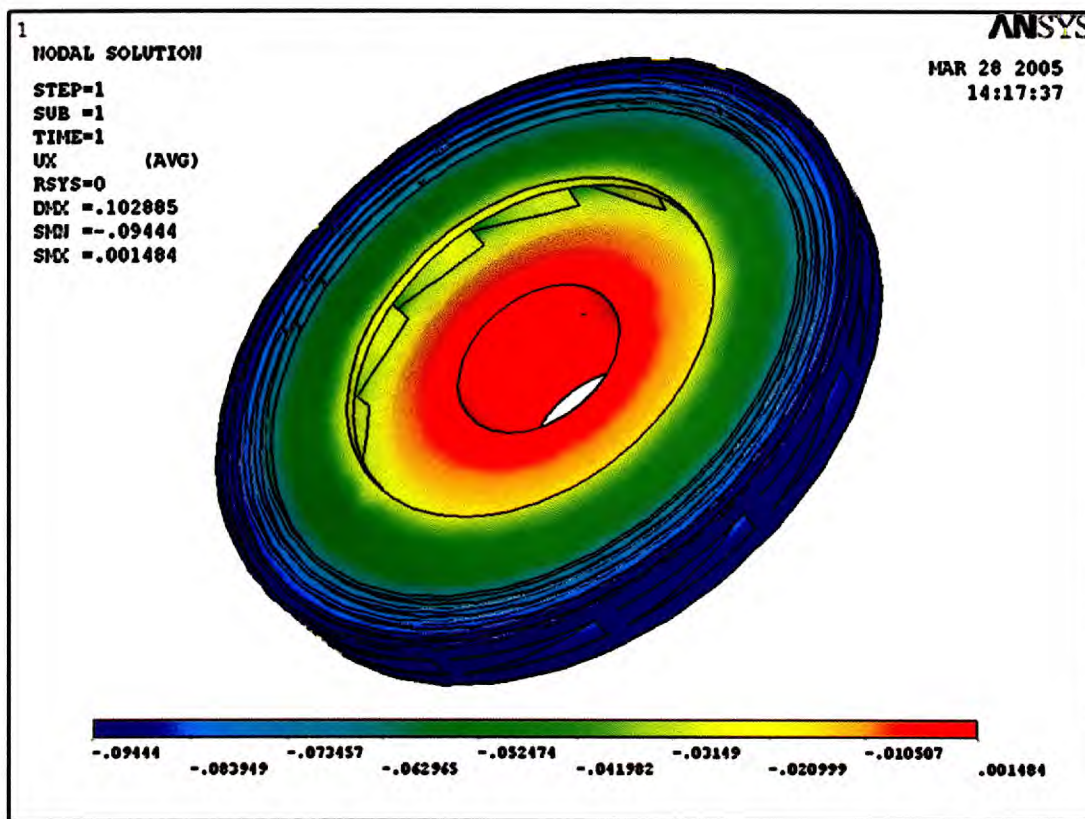


Figure 70: Axial Displacement of Rotor in mm

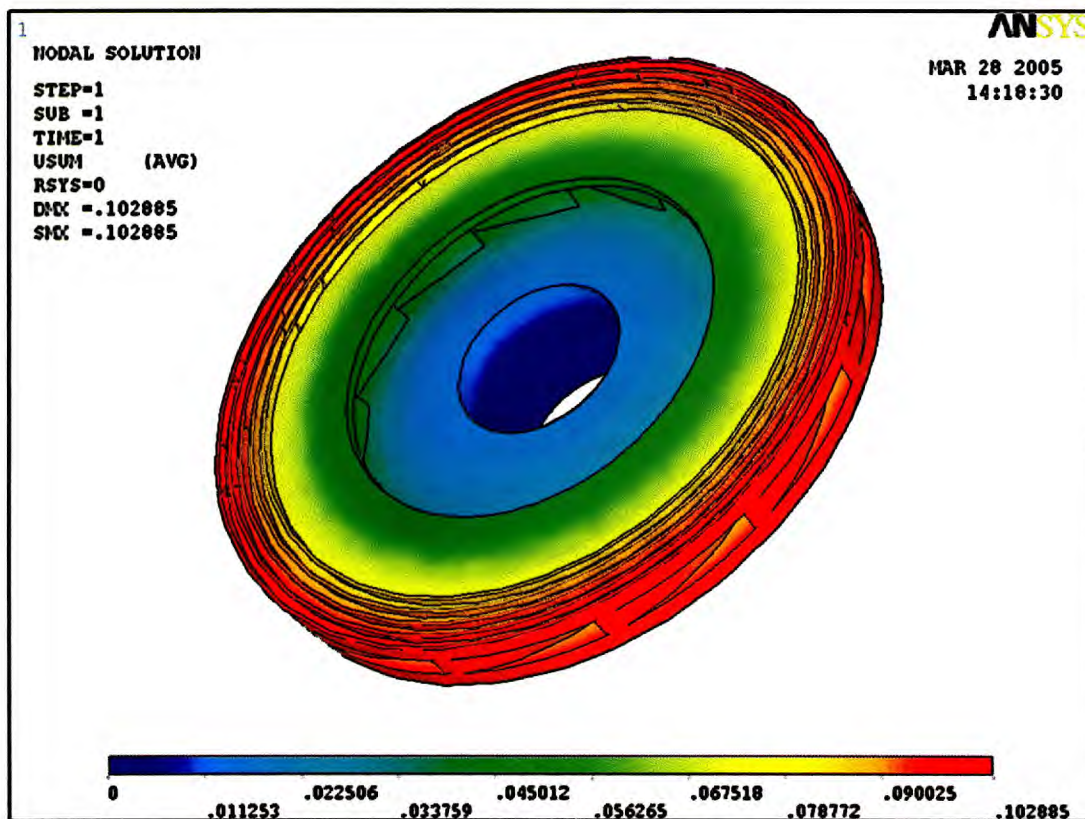


Figure 71: Total Displacement of Rotor in mm

5.2 Bearings

A pair of 17mm, angular-contact, 25°, ABEC9, ceramic-balled, bearings is used on the drive side to take the axial load, and a similar single bearing on inlet side is fitted with a set of springs to provide the pre-load. This bearing size and arrangement was chosen so that the bearings were large enough to resist the expected end load, but small enough to enable grease lubrication. The spring pre-load provides nearly constant preload, even at varying temperatures. The bearings are lubricated with Tellus 37 gearbox oil, supplied from radial holes through the outer race. This oil is of higher viscosity than the specification of the bearings, however it is common to the gearbox drive, simplifying the required ancillaries and eliminating any problems of oil-mixing.

The pressure on the back side of the compressor is measured, and the pressure of the sealing air supply can be adjusted in order to determine the effect of sealing pressure on pressure ratio and mass flow. This ensures that the aerodynamics of the rotors are unaffected by the sealing air and that the shaft seal is functioning properly. The sealing air escapes from the other side of the seal through the bearing, which ensures that the bearing oil does not build up. This is shown in **Figure 72**.

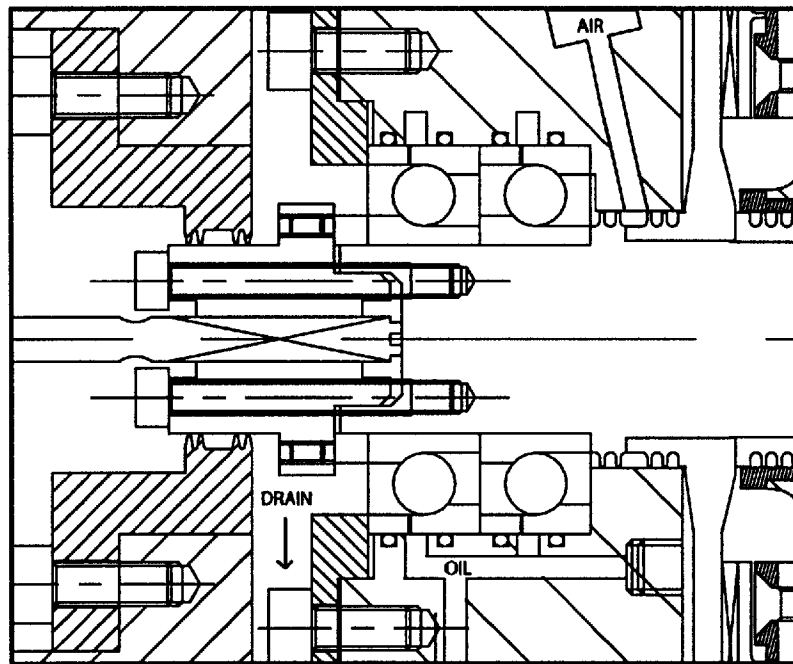


Figure 72: Thrust Bearing Arrangement

The inlet side bearing is air blown with a seal that leaks out to atmosphere on both sides. An additional set of seals protects the compressor inlet from any residual kinetic energy from the seals, so as to not affect the flow. This is shown in **Figure 73**

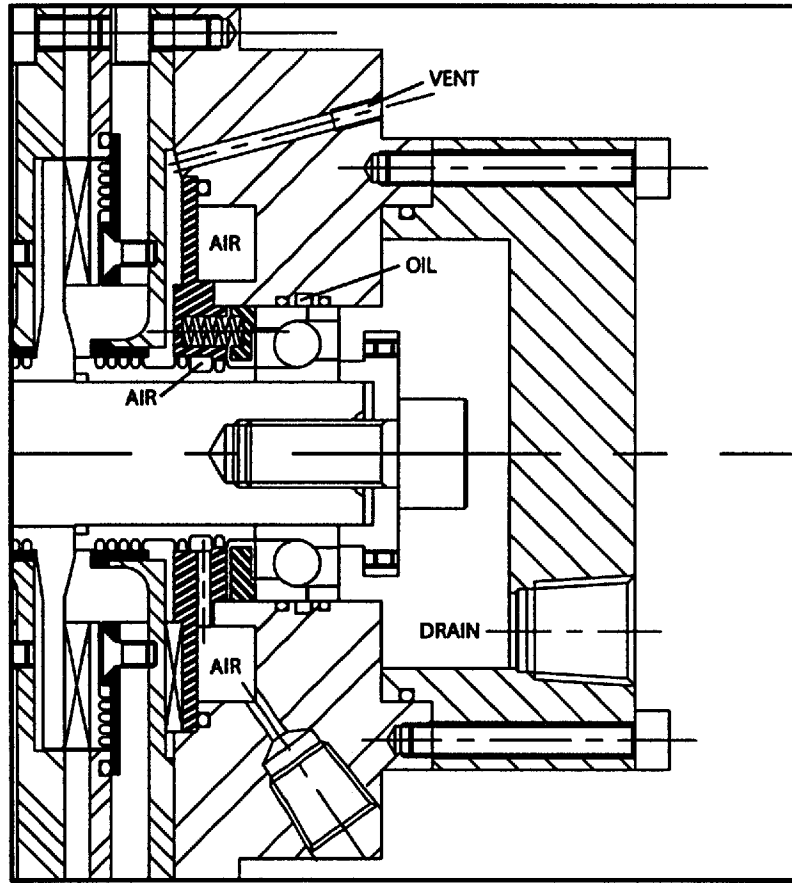


Figure 73: Preload Bearing Arrangement

The calculations of bearing life are performed using the methods described in [50]. The maximum thrust load is calculated at 220N. The spring pre-load is set at 20% of this value. The radial force of imbalance is calculated at 78N with 2g*mm of imbalance, almost 10 times that measured by the balancing machine. The dynamic load, P_e , is calculated using **Equation 103**

$$P_e = (F_r)X + (F_{PL} + F_T)Y \quad (103)$$

F_r , F_{PL} and F_T are the radial, preload and thrust loads during operation. For bearings with 25° contact angles, X and Y are determined by the condition in **Equation 104**.

If:

$$\frac{F_{PL} + F_T}{F_r} > 0.68 \quad (104)$$

then,

$$X = 0.41 \quad (105)$$

and

$$Y = 0.87 \quad (106)$$

The bearings being used have a dynamic load capacity, $C_{33} = 6000N$, and a multiplier $K = 1.8$ for the tandem arrangement. These are used to predict the life in cycles, L_{10} and the life in hours, L_{10h} .

$$L_{10} = \left(\frac{K \cdot C_{33}}{P_e} \right)^3 \cdot 10^6 \quad (107)$$

$$L_{10h} = \frac{L_{10}}{60 \cdot (RPM)} \quad (108)$$

These calculations show an expected life of 3 years. The effect of any end load deviation is shown in **Figure 74**.

5.3 Rotor Dynamics

An analysis of the rotor dynamics is required in order to ensure that the shaft and bearings do not vibrate excessively. An FEA analysis is used to determine the natural frequencies of the shaft and bearing arrangement with respect to shaft speed. These modes are potentially excited by the imbalance in the shaft, which has a driving frequency equal to the frequency of rotation. As there are no provisions for increasing the damping present, rotational frequencies equal to the natural frequency of operation must be avoided.

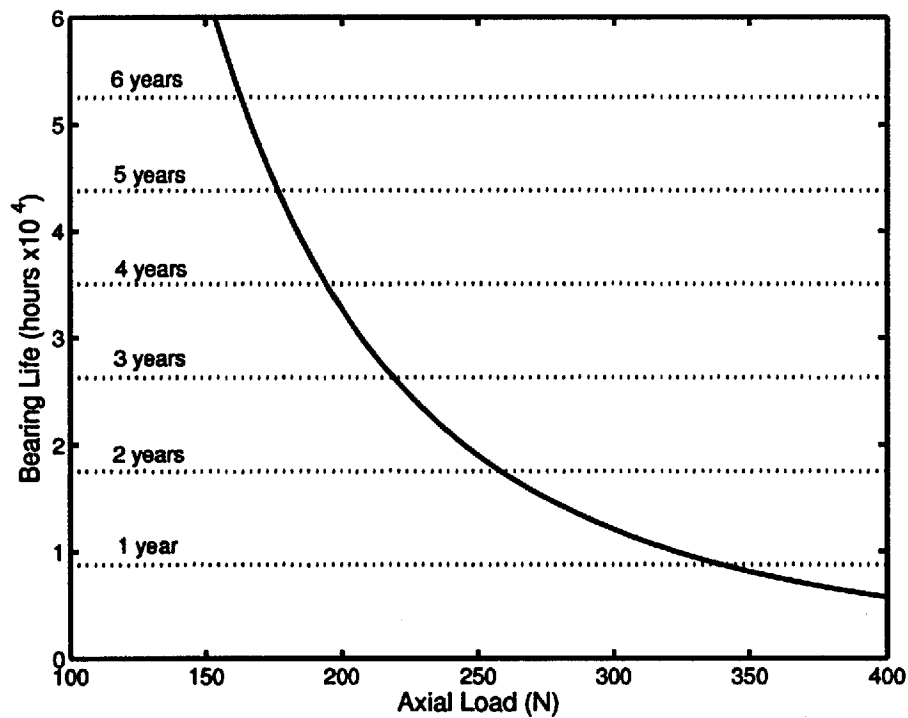


Figure 74: Bearing Life vs. End Load

The shaft was modelled using axisymmetric elements with equivalent inertias in the form of pipe elements as shown in **Figure 75**. Pipe elements are chosen for their ability to model gyroscopic effects. The rotors are modelled using a reduced stiffness, 10% of that of the material used, as they will have little effect on the stiffness of the arrangement. The model, which uses an ANSYS script file written by Leontopolous [51], produces the Campbell diagram shown in **Figure 76**. This shows the natural frequencies plotted vs. shaft speed, along with the '1X' line of excitation due to imbalance. The first natural frequency crosses the 1X line at 69,000 RPM. This rigid body mode occurs at such a high frequency as the shaft length and overhung masses are kept to a minimum. No natural frequencies associated with the test rig were detected during normal operation.



Figure 75: Rotor Dynamics Model Constructed From Pipe Elements

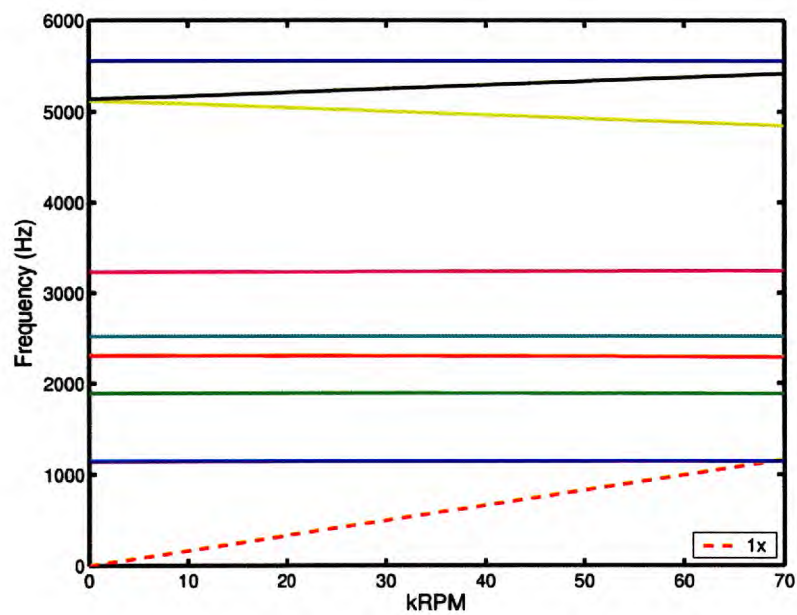


Figure 76: Campbell Diagram showing modes and 1x line

5.4 Diffusers and Ducting

For the first stage, the inlet ducting is bolted to the preload bearing housing. For the final stage the diffuser is added to the assembly on its own. All intermediate stationary components are added as 'stator' sub-assemblies, as shown in **Figure 77**. Each stator consists of an inlet ducting, a rotor seal, a shaft seal and a diffuser (A). They are assembled in the manner shown in (B). The first and third stators direct the flow from the diffuser directly into the inlet duct, and therefore are machined as an assembly, as shown in (C). The second stator directs the flow through a set of radial holes to a plenum, which feeds the air to the inter-cooler. The flow returns in a similar manner. Therefore, the second stator remains unmachined as shown in (B). The main housing is used to align the rotors for assembly.

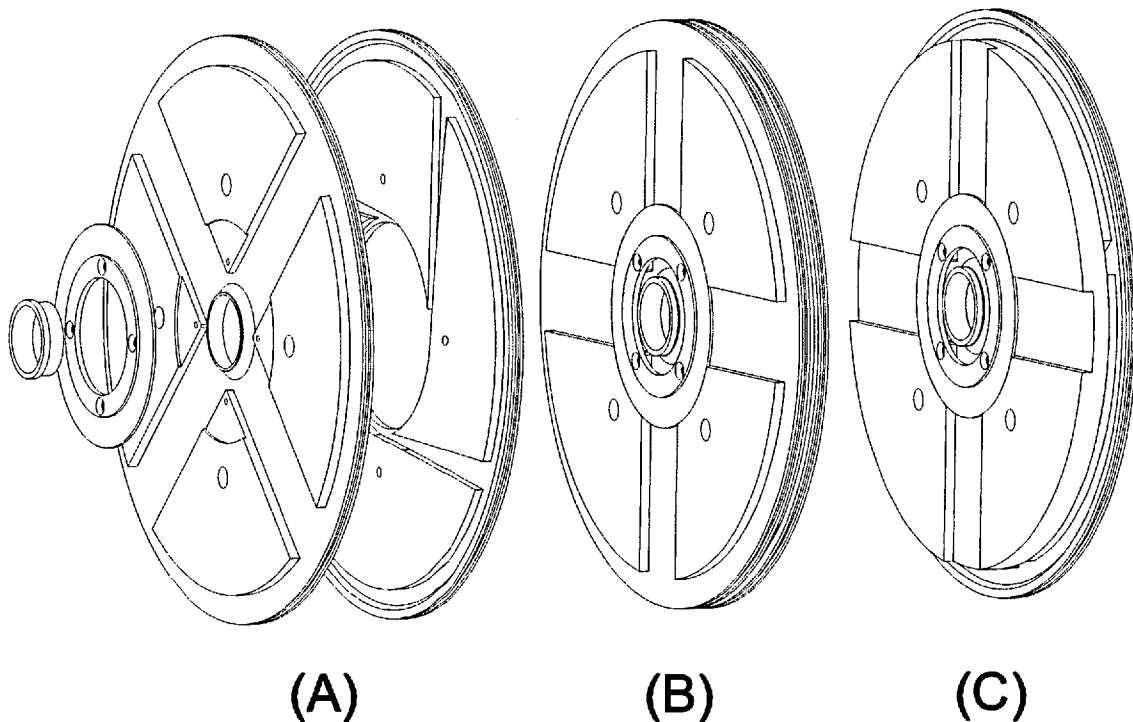


Figure 77: Stator Assembly Process
(A) Components (B) Assembly (C) Machined Assembly

The stators are held in place by friction caused by clamping loads and are removed

by jacking screws. O-rings seal between stages at the outer diameter. The diffuser passages are polished by hand to reduce the frictional losses. The shaft seal insert is bonded in place. Both inlet ducting and diffuser are surface ground to an appropriate thickness in order to achieve good alignment between rotor and stator. The surface grinding also ensures that the stators are flat, which is important as the tolerances will stack up.

5.5 Intercooling

While a design with an incorporated intercooler would be very compact, it does not allow the room necessary for instrumentation, so the intercooling was implemented externally, through a large counterflow intercooler originally used for a reciprocating compressor, shown in **Figure 78**.

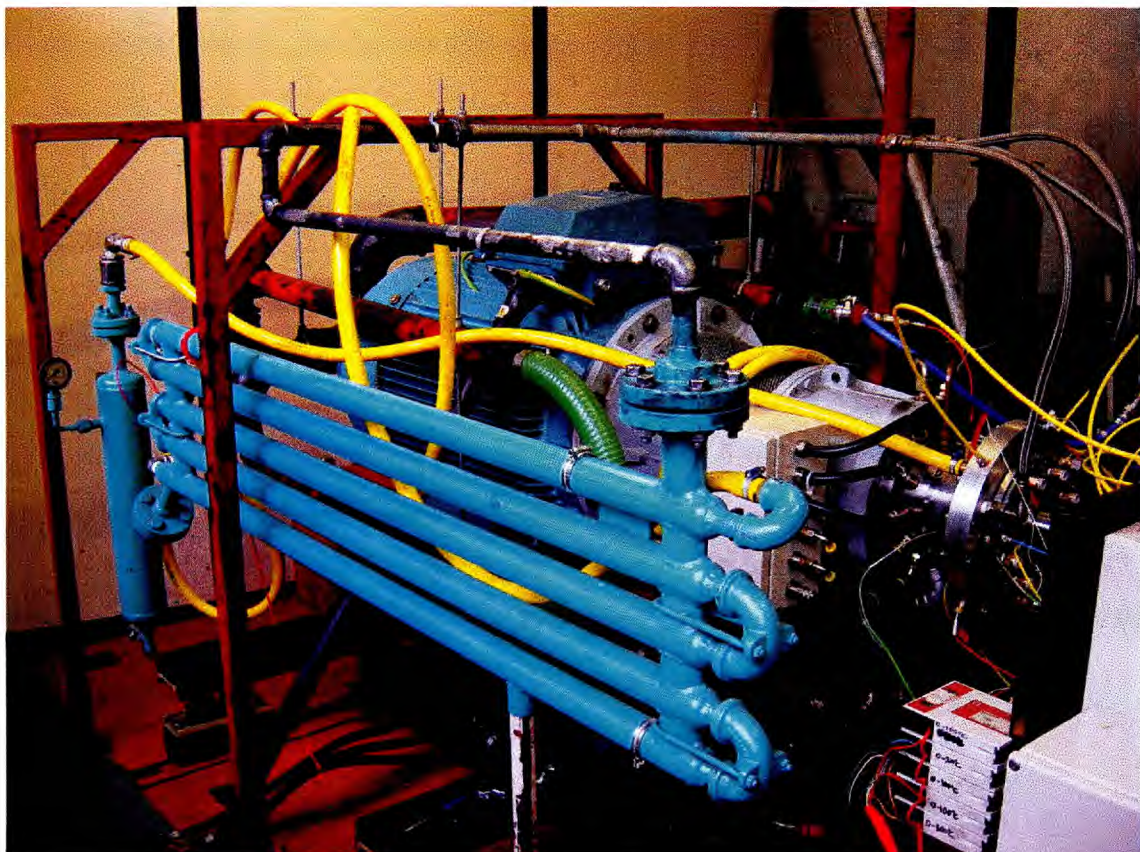


Figure 78: Intercooler

At the outer diameter of the test rig are two plenums. One collects the air after the second stage where it is fed to the intercooler through 3 braided hoses. The cooled air returns from the intercooler to the test rig into the other side. This is shown in **Figure 79**

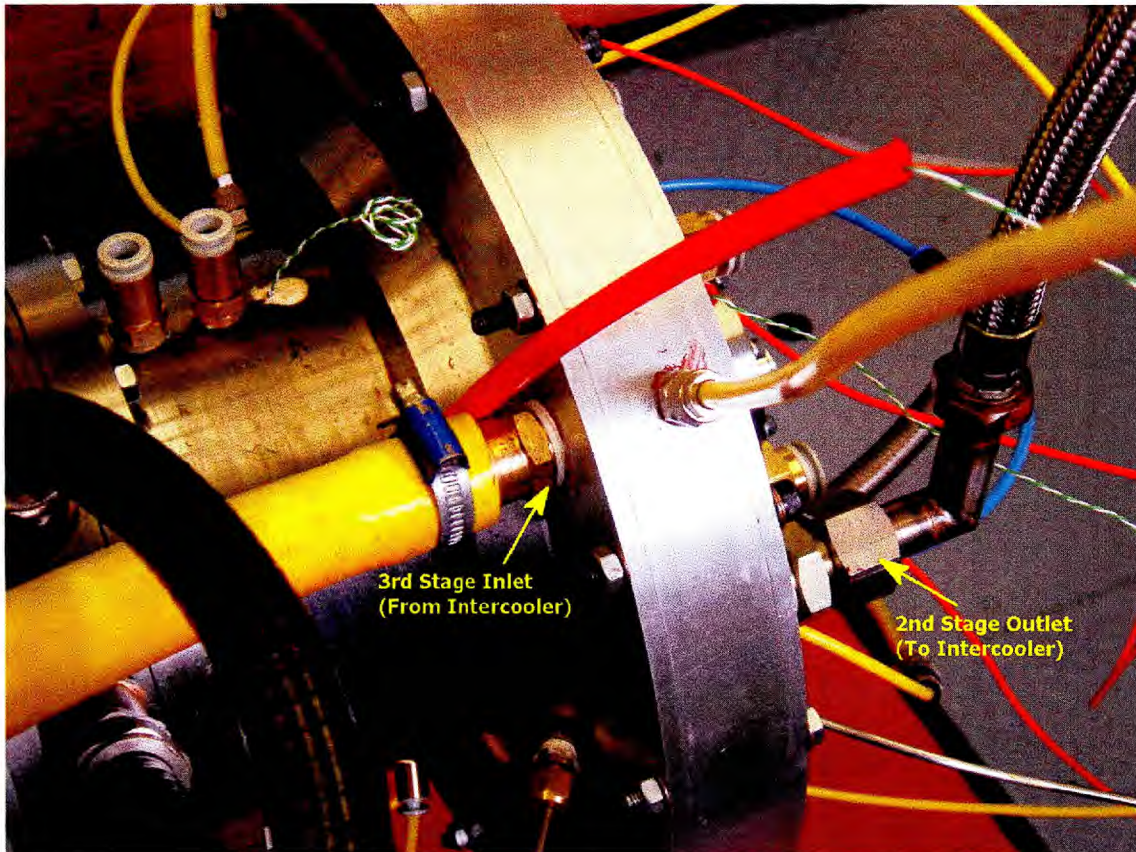


Figure 79: 2nd Stage Exhaust Collector and 3rd Stage Inlet

This arrangement is useful, as it allows the first two stages to be tested independently of the last two stages and vice-versa. This allows for greater characterization of the individual compressor stages.

5.6 Compressor Assembly

In order to assemble the compressor, a small jig is built into the housing. The component containing the female labyrinth seal on the square drive is also used to clamp the

outer race of a bearing used during the assembly. This arrangement is shown in **Figure 80**.

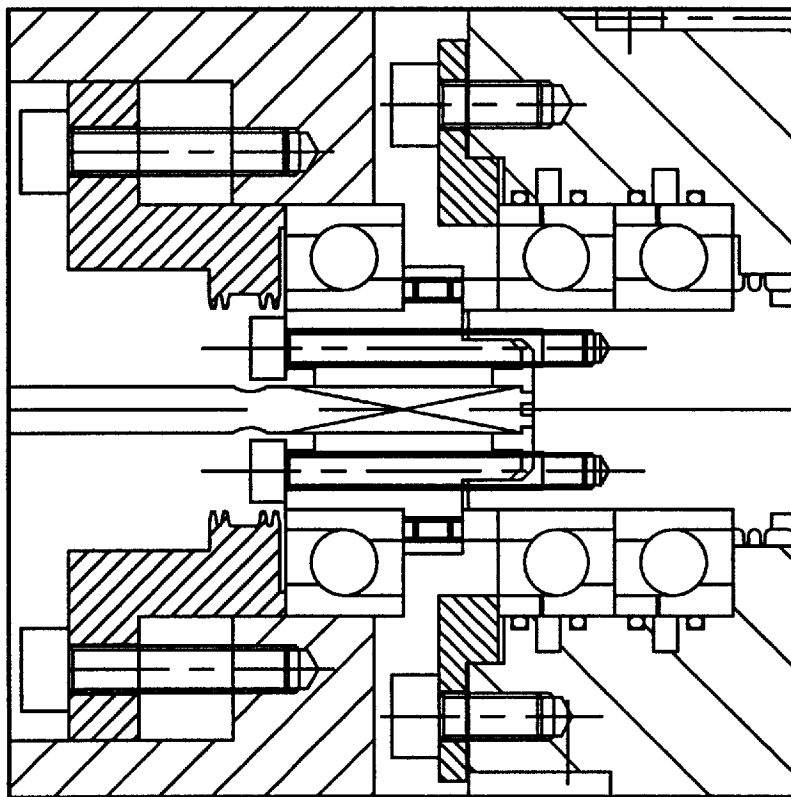


Figure 80: Assembly Bearing Arrangement

This holds the shaft straight during the assembly process and supports it as the rotors are fitted. Once the preload bearing has been added, the assembly bearing is removed. This allows seal clearances and rotor-diffuser alignment to be inspected as the rig is assembled. Many partial assemblies were performed as face seals and diffusers were ground to fit. The angular position of the preload bearing housing is essentially arbitrary, as all of the instrumentation and fluid delivery contained is independent of orientation. After the assembly bearing is removed, bearing resistance is assessed by hand with the preload housing at various angular positions. The preload housing is bolted in the angular position that introduces as little shaft resistance as possible, optimizing shaft alignment.

5.7 Dynamic Balancing

In order to reduce the effect that shaft imbalance will have on the vibrations present, the shaft must be balanced. Balancing was performed using a CEMB ZE50 TCI dynamic balancing machine. No balancing of individual rotors was required. The test rig was balanced as an assembly, as shown in **Figure 81**. The compressor was driven as a turbine using compressed air. Balancing planes exist at each end of the shaft, which will accommodate *M2* grub screws and short cap head screws with masses of .04g and .16g, respectively. The imbalance was $0.1g \cdot mm$ for the drive side and $0.2g \cdot mm$ for the non-drive side, which will cause radial loads of 4N and 8N at full speed. The intercooler plenum, mounting flange, and other non-necessary parts are removed for balancing to reduce the stationary inertia and improve access to the shaft.

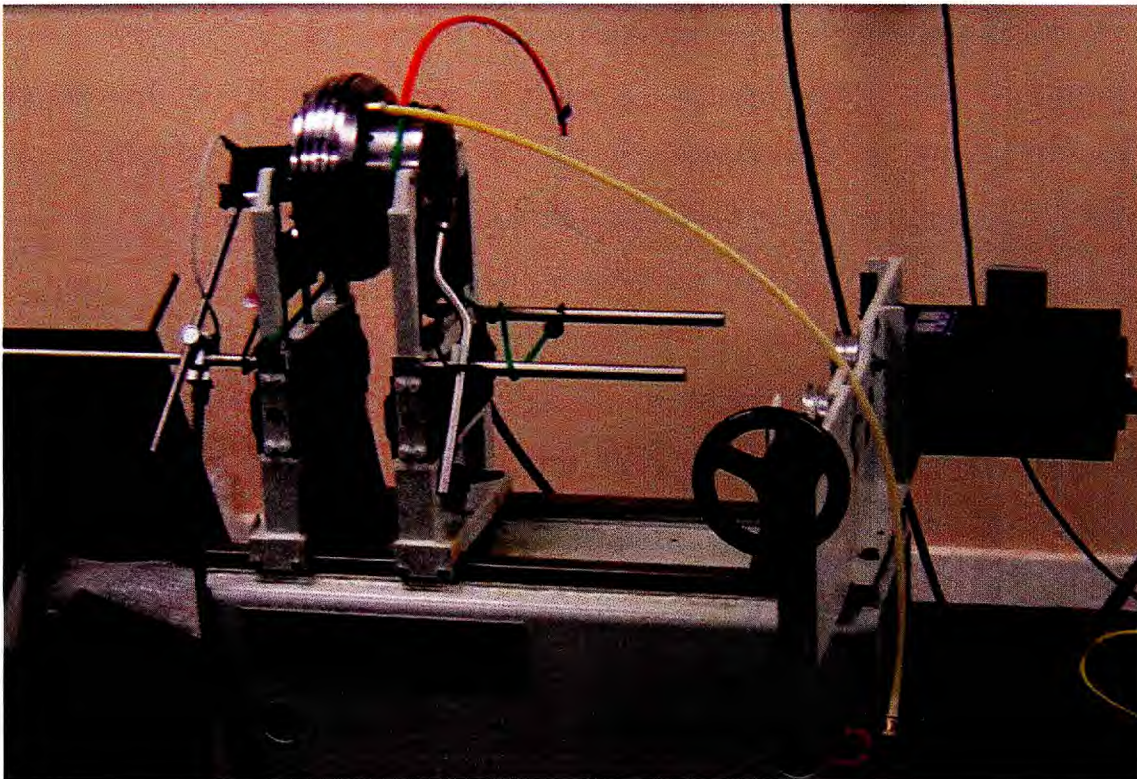


Figure 81: Rig Balancing

5.8 Motor, Gearbox and Drive

The test rig is driven by a double epicyclic gearbox fitted to an inverter-driven, variable speed motor. A schematic of the low speed coupling, gearbox and high speed coupling is shown in **Figure 82**. The motor is a conventional synchronous motor produced by ABB and capable of 110kW at 3000RPM. The gearbox has a ratio of 27:1 and a speed rating of 80kRPM (model F4183 from Compact Orbital Gears [52]).

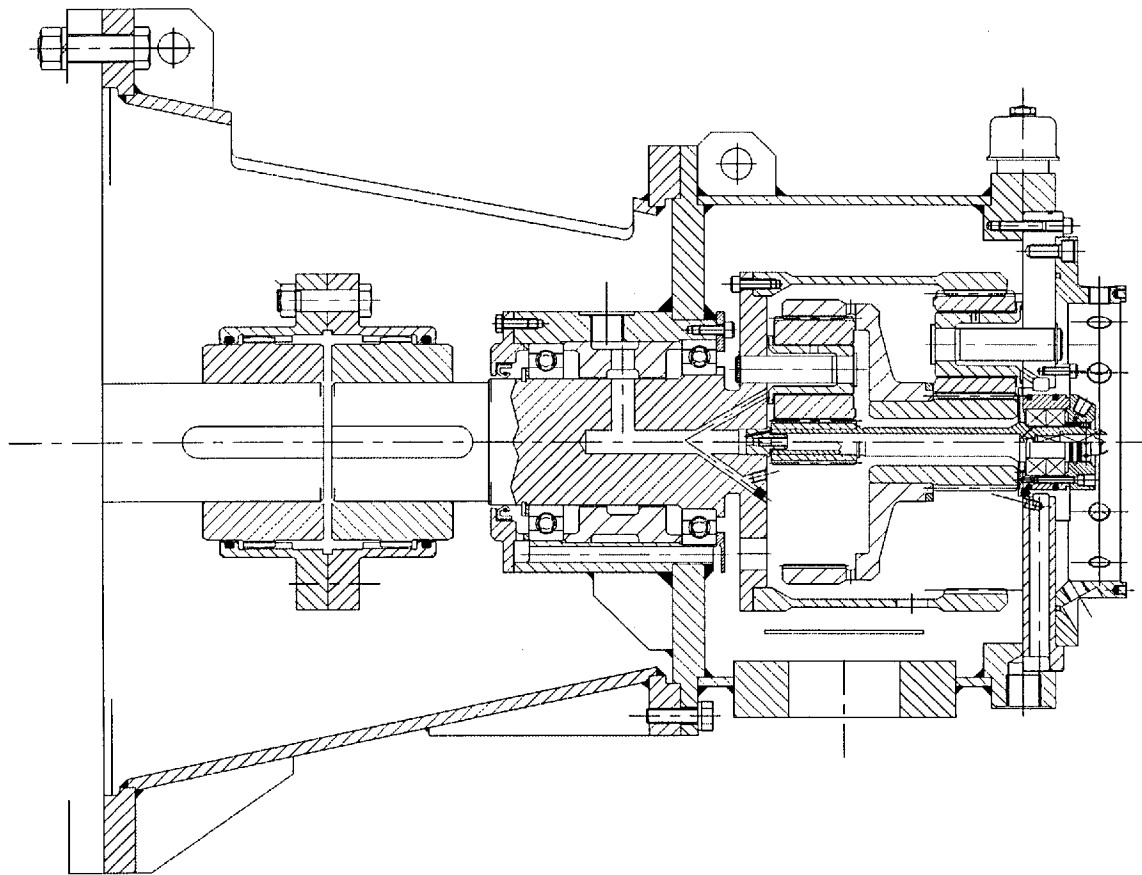


Figure 82: Gearbox by Compact Orbital Gears [52]

The gearbox has been modified to use a square drive coupling to provide the torque to the test rig, as shown in **Figure 83**. The spool piece is 4mm and 6mm across flats on the test rig side and gearbox side, respectively. It has been designed to shear close to the test rig in the event of high torque caused by test rig failure. The spool piece has been nitrided to a depth of 50 microns to increase surface hardness. This will minimize wear on the

more expensive mating components in the gearbox. Both the spool piece and the female coupling are made from EN3B steel and have been heat treated to a yield stress of 900MPa.

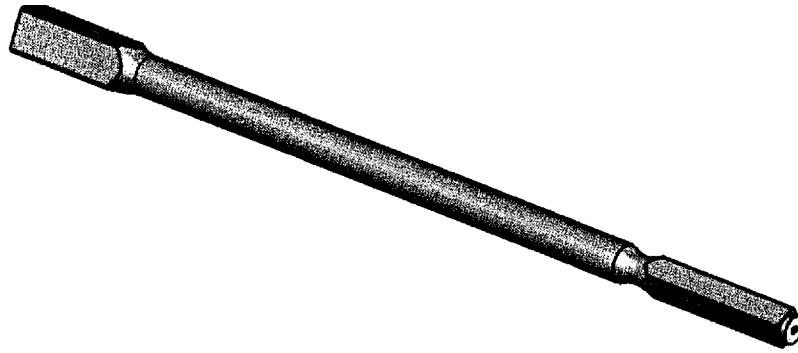


Figure 83: High Speed Coupling

6 Instrumentation, Experimentation and Test Schedule

This section describes the experimental setup, the various testing configurations, the instrumentation required and the operation of the test rig.

6.1 Measurements

The location of all pressure and temperature measurement points are shown in **Figure 84**. Pressure is maintained by a valve which is operated manually from outside the test cell, connected by approximately 3 meters of 25mm tubing. The air then passes through a large duct to a mass flow meter located in a different test cell. A velometer is fitted to the test rig in order to measure vibrations.

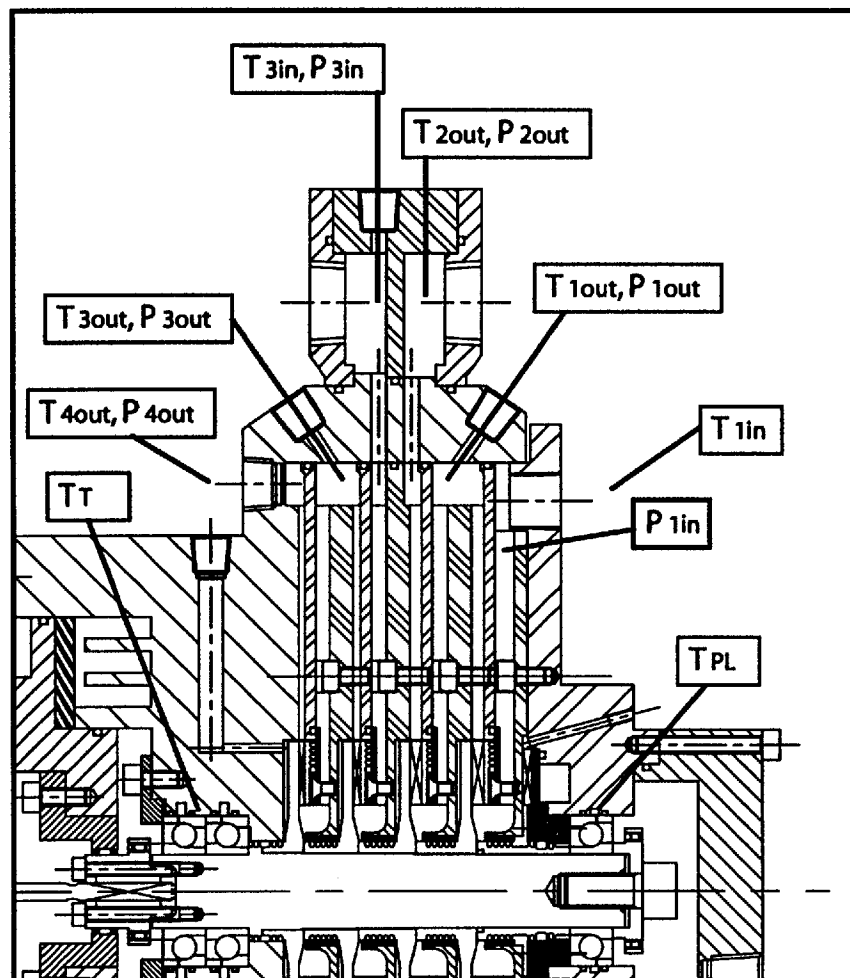


Figure 84: Measurement Locations

6.2 Test Configurations

The externally controlled valve and external intercooler allow the test rig to be utilized in different configurations.

6.2.1 4 Stage Compressor

The normal operational configuration for the test rig is shown in **Figure 85**. The inlet of the first stage is open to the atmosphere, the 2nd stage outlet is ducted through the intercooler and back to the 3rd stage inlet. The 4th stage outlet is connected to the throttling valve and then through the main mass flow meter.

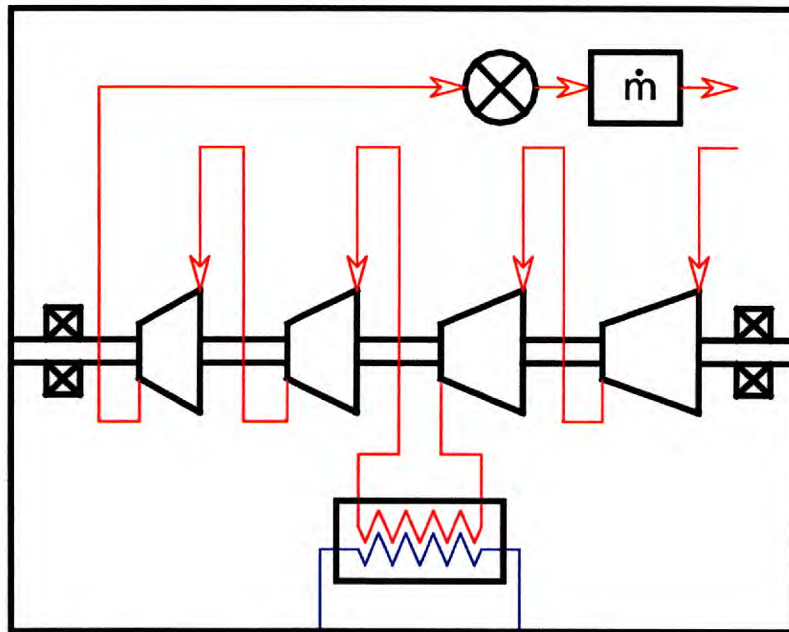


Figure 85: 4 Stage Configuration

6.2.2 2 Stage Compressor

In order to test the first two stages independently, the intercooler outlet is connected directly to the throttling valve, bypassing the last two stages, as shown in **Figure 86**. The second stage is affected by the leakage to the 3rd stage inlet, which is at atmospheric pressure.

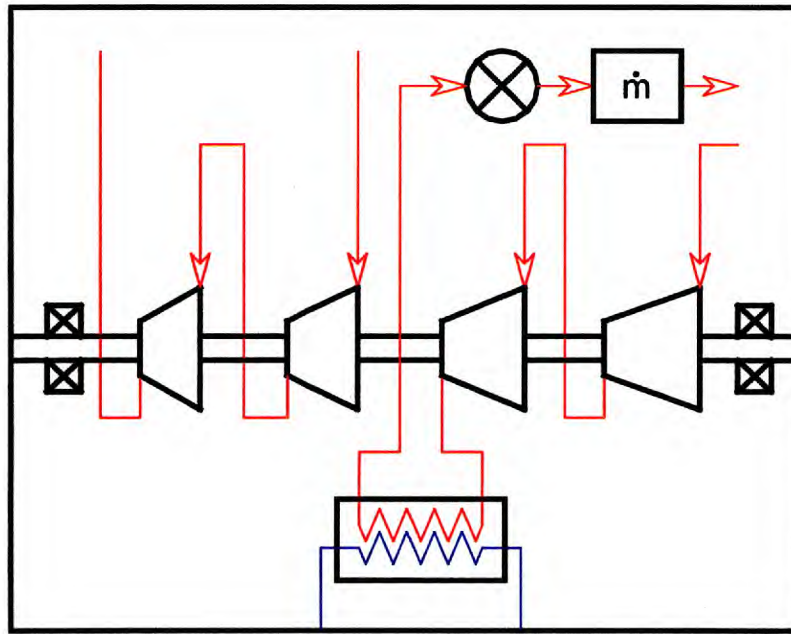


Figure 86: 2 Stage Configuration

6.2.3 4 Stage Vacuum Pump With Inlet Throttling

Inlet throttling was initially used to determine the suitability of the 4 stage compressor for vacuum pump applications. However it provided useful insight into the surge dynamics of the 4 stage machine. In this configuration, the 8 inlet ports are connected to the throttling valve through a manifold. The 4th stage exhaust is open to the atmosphere. The auxiliary mass flow meter is connected upstream of the throttling valve and protected by a filter. This is done because the lowered mass flow of the inlet valved configuration is below the minimum mass flow of the main mass flow meter. This also allows the 4th stage to exhaust directly to the atmosphere.

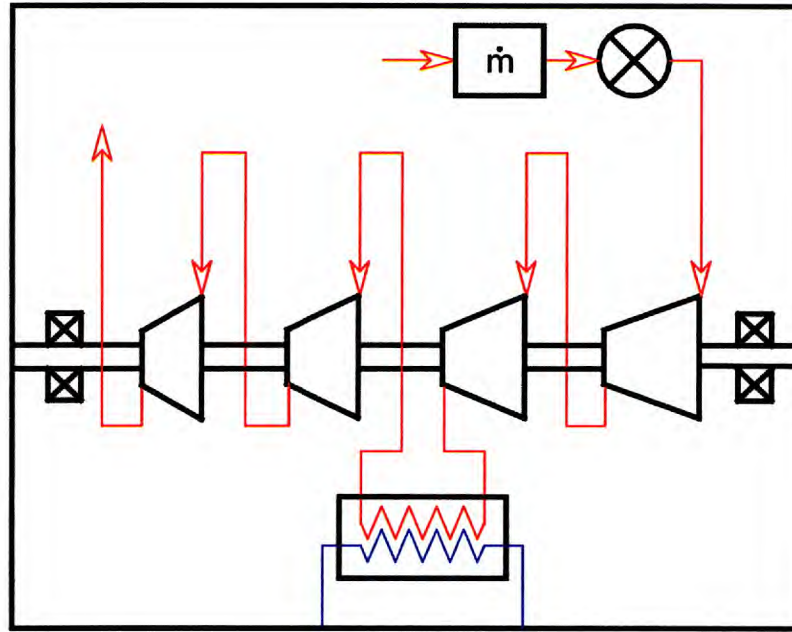


Figure 87: 4 Stage Configuration With Inlet Throttling

6.3 Instrumentation

Temperature and pressure are measured at appropriate points throughout the rig, and the mass flow rate is measured at the outlet, after being throttled. **Table 21** lists all of the required measurements and the instruments used. **Table 22** shows all of the required measurements for rig diagnostics.

6.3.1 Pressure Sensors

The pressure sensors are all connected to static pressure tapings in the test rig via nylon tubing and fittings, with the exception of the final stage pressure measurement which is connected by copper tubing. All measurements are at regions where the flow is expected to be less than Mach 0.2, minimizing the discrepancy between static and total pressure. The sensors measuring P_{1i} and P_{1o} read absolute pressure and all other sensors measure gauge pressure. The ambient absolute pressure is measured at the beginning of the test, allowing conversion between gauge and absolute. Many of these pressure sensors were already available, and the combination used was chosen to minimize additional purchases.

Property	Symbol	Instrument	Range	Accuracy
Stage 1 Inlet Temp	T_{1i}	PRT	0 – 100°C	±1°C
Stage 1 Outlet Temp	T_{1o}	K Type	0 – 1000°C	±2°C
Stage 2 Outlet Temp	T_{2o}	PRT	0 – 300°C	±1°C
Stage 3 Inlet Temp	T_{3i}	PRT	0 – 100°C	±1°C
Stage 3 Outlet Temp	T_{3o}	K Type	0 – 1000°C	±2°C
Stage 4 Outlet Temp	T_{4o}	PRT	0 – 300°C	±1°C
Stage 1 Inlet Pressure	P_{1i}	Druck PTX1400	0.8 – 1.2barabs	±0.20kPa
Stage 1 Outlet Pressure	P_{1o}	Druck PTX1400	0 – 1.6barabs	±0.75kPa
Stage 2 Outlet Pressure	P_{2o}	Druck PTX1400	0 – 2.5barg	±1.3kPa
Stage 3 Inlet Pressure	P_{3i}	Druck PTX1400	0 – 1.6barg	±0.8kPa
Stage 3 Outlet Pressure	P_{3o}	Druck PTX1400	0 – 4.0barg	±2.0kPa
Stage 4 Outlet Pressure	P_{4o}	Druck PTX1400	0 – 7.5barg	±3.8kPa
Stage 4 Rotor Pressure	P_{4m}	Druck PTX1400	0 – 10barg	±5.0kPa
Volumetric Flow	$\dot{V}_{out}$	Swirl Flow Meter	0 – .0361m ³ /s	1.8 × 10 ⁻⁴ m ³ /s
Flowmeter Temperature	T_{fm}	PRT	0 – 200°C	±1°C
Flowmeter Pressure	P_{fm}	DRUCK PTX600	0 – 10barabs	±0.80kPa
Auxiliary Mass Flow	$\dot{m}_{aux}$	Cole-Parmer 16 Series	0 – 256SLPM	±1%

Table 21: Measurement Instrumentation

Property	Symbol	Instrument	Range	Rig Threshold
Preload Bearing Temp	T_{PL}	K Type	0 – 200°C	120°C
Thrust Bearing Temp	T_T	K Type	0 – 200°C	120°C
Rig Vibrations	V_{vib}	Bently Nevada (PN 330500)	0 – 6mm/s	2mm/s

Table 22: Diagnostic Instrumentation

6.3.2 Platinum Resistance Thermometers and Temperature Transmitters

Platinum Resistance Thermometers (PRT's) were chosen due to their accuracy when compared to K-type thermocouples. Temperature transmitters measure the resistance of the PRT and convert the measurement into a 4 – 20mA signal scaled to a preset range, as shown in Table 21.

6.3.3 K type Thermocouples

K type thermocouples were used in the first and third stage outlet, as the reduced space did not allow the 50mm wetted length specified by the PRT supplier. Additionally, these temperatures were not to be used in calculating the machine overall power consumption

as individual stage mass flow measurements could not be attained, so the lower accuracy is unimportant.

K type thermocouples were also used to measure bearing temperatures, as the bearing temperature measurement was used only for condition monitoring. Welded-tip thermocouples allowed temperature measurements to be made in the bearing housing, only 0.5mm from the bearing surface.

6.3.4 Velometer

The velometer used is a Bently-Nevada model 'PN 330500' affixed to a removable stud. The stud is affixed to the test rig on the preload bearing housing with cyanoacrylate adhesive. Frequencies attributed to the gearbox can be measured in this arrangement, so it is expected that excitations anywhere in the test rig will be apparent. The output is monitored by a frequency analyzer, which gives both the measured value in *mm/s* as well as the frequency components.

6.3.5 Flowmeters

The main mass flow meter is an ABB Trio-Whirl ST 40 volumetric flow meter. It is fitted with a pressure sensor and PRT, as shown in **Table 21**, in order to calculate the mass flow rate. A large ducting system is located between the valve and the flowmeter, which causes high frequency fluctuations in volumetric flow to be unresolved. The temperature loss in the ducting causes the temperature at the mass flow meter to be close to ambient, and the pressure is approximately atmospheric. This establishes a minimum mass flow measurement of 3 g/s, which is less than the minimum measurement taken.

The auxiliary mass flow meter, a Cole-Parmer 16 Series Mass Flow Meter, is designed for mass flow measurements of less than 5.12 g/s. It is used for the inlet throttling test. This flow meter works on the principle of measuring the pressure drop across a restriction

designed to induce laminar flow. The device is calibrated over the range of volumetric flows. Internal pressure and temperature sensors are used to convert the volumetric flow rate measured to a mass flow rate.

6.4 Data Acquisition

Two data acquisition systems were used for the conversion of electrical signals to numerical values and the display and logging of those values. All temperature and pressure measurements taken at the test rig were logged by the main data acquisition system, located at the rig control panel. The signals for the PRT's and pressure sensors were 4-20 mA signals, corresponding to the lower and upper limits of measurement. The thermocouple signals were measured directly by the A/D system. The second data acquisition system logged $\dot{V}$, T_{fm} , and P_{fm} and was located in a separate test cell. The data was compiled by using time stamps to match up the data points. The velometer signal was processed by a DI-2200 real time FFT analyzer and a rolling spectral analysis showed frequencies of excitation. Mass flow measurements from the auxiliary flowmeter were recorded by hand from the LCD display.

6.5 Calculations

6.5.1 Mass Flow Calculation

The mass flow calculation of the main mass flow meter is done using Van der Waals equation of state. This is probably unnecessary for these purposes, as the mass flow is not used for the calculation of efficiency or power input, but it is implemented in the mass flow measurement system, which is otherwise used for a larger range of pressures and temperatures.

$$\left(P_{fm} + \frac{a}{v^2}\right)(v - b) = R_{air}T_{fm} \quad (109)$$

$$a = \frac{27R^2T_{cr}^2}{64P_{cr}} \quad (110)$$

$$b = \frac{RT_{cr}}{8P_{cr}} \quad (111)$$

$$\dot{m} = \frac{\dot{V}}{v} \quad (112)$$

6.5.2 Pressure Ratio Calculations

Pressure Ratio Calculations use static to static measurements taken at points of low velocity. They include the pressure losses in the inlet and outlet ducting of each stage. The calculations are performed using the following equations. The inlet pressure measurement is used to determine the atmospheric pressure before the rig is started.

$$P_{atm} = (P_{1i})_{RPM=0} \quad (113)$$

$$PR_1 = \frac{P_{1o}}{P_{1i}} \quad (114)$$

$$PR_2 = \frac{P_{2o} + P_{atm}}{P_{1o}} \quad (115)$$

$$PR_3 = \frac{P_{3o} + P_{atm}}{P_{3i} + P_{atm}} \quad (116)$$

$$PR_4 = \frac{P_{4o} + P_{atm}}{P_{3o} + P_{atm}} \quad (117)$$

$$PR = \frac{P_{4o} + P_{atm}}{P_{1i}} \quad (118)$$

$$PR_{12} = \frac{P_{2o} + P_{atm}}{P_{1i}} \quad (119)$$

$$PR_{34} = \frac{P_{4o} + P_{atm}}{P_{3i} + P_{atm}} \quad (120)$$

6.5.3 Power Consumption

The machine power consumption is difficult to determine. The power consumption of the motor is available, but the losses in the motor drive, gearbox, and bearings are all included and are greater than the expected compressor power consumption. Torque measurement was considered, but the combination of low torque and high speed made it unfeasible. An energy balance was initially planned, however interstage heat transfer, described later, made this impossible. There is currently no reliable method for calculating the power consumption.

6.6 Control and Operation

For each test, the speed was brought up slowly to 30kRPM and tested throughout the range of the machine by adjusting the valve from wide open throttle (WOT) up to the point of surge. Points of steady state pressure and temperature were flagged by time stamps. Temperature never achieved steady state at any point due to the increasing temperature of the gearbox, the high thermal mass of the test rig, and the low mass flow of the working fluid. At the onset of surge, the valve was quickly opened, except during the inlet throttling tests. The speed was increased by increments of 5kRPM up to the design speed of 60kRPM.

6.6.1 Vibration Monitoring

Test rig vibration measurements were presented as a dynamic response showing both the amplitude vs. frequency graph, the frequency and amplitude readings of the most prominent frequencies, and the overall vibration level. A vibration at the frequency of shaft operation is a normal product of imbalance, but multiples of the running frequency indicate either a misalignment, a shaft or rotor touchdown, or some other test rig anomaly.

When oil-lubricated bearings were used, the vibrations always remained below 1.5mm/s along with low vibration at the running frequency and no discernable vibrations at frequency multiples. In successive tests, the vibration at the running frequency increased

noticeably, but not significantly. This could be attributed to an increase in imbalance, which would be caused by relative radial movement between rotating components.

The test rig was also run with grease packed bearings. It was also attempted to run with a grease packed preload bearing during the inlet throttling test. During both of these tests, the vibration measurements at 50-55kRPM increased dramatically. Aside from dry friction between mated parts, the only damping present in the rig comes from the bearing lubrication, and the o-rings used to seal the oil inlet passage on the drive side. It is thought that the damping in the greased bearing configuration is insufficient.

6.6.2 Bearing Monitoring

Even with grease packed bearings, the temperature measurements for the preload and thrust bearings never exceeded 77°C and 107°C respectively. This is well below the limit of 120°C specified by the manufacturer. The heat of compression, the bearing losses, and windage losses all contribute to the increased temperature. The higher temperature at the thrust side is likely caused by the high temperature of the gearbox and the thrust loading of the bearings. The temperature continued to rise throughout each test, due to the buildup of heat in the oil system, caused by inadequate cooling. Additional cooling could reduce the oil system temperature and allow the test rig to be run for longer periods of time.

7 Results and Analysis

Contrary to the multistage simulations of section 4.5.3, which were based on the single stage experimental results of Vine, the multistage compressor operated effectively over a substantial range. The behavior of the forward-swept compressor within a multistage arrangement was characterized and many aspects of the multistage simulation routine were validated. Complications of interstage heat transfer and mismatching negatively affected performance, but simulations predict better performance for a redesigned compressor.

It is not prudent to present the compressor maps using non-dimensional or referred conditions, as the temperature at each stage inlet is not known accurately. Furthermore, the calculations used in formulating a corrected map assume adiabatic conditions, which were not present. Additionally, efficiency cannot be calculated accurately. Therefore all maps show pressure ratio plotted with respect to total machine throughput, and efficiency measurements are not included.

7.1 Experimental Results

The upstream stages appear to have a stabilizing effect on the first stage. The first stage is able to produce much lower flowrates than the nearly identical single stage experimental test rig mentioned in section 2.6.4. This can be seen by comparing the first stage range of operation in the different configurations.

7.1.1 4 Stage Compressor

The compressor in the 4 stage configuration produces a maximum pressure ratio of 2.58 at a mass flow of 4.74 g/s and a speed of 60kRPM. The compressor map of the 4 stage configuration is shown in **Figure 88**.

The individual stage pressure ratios are shown in **Figure 89**. The first two stages are

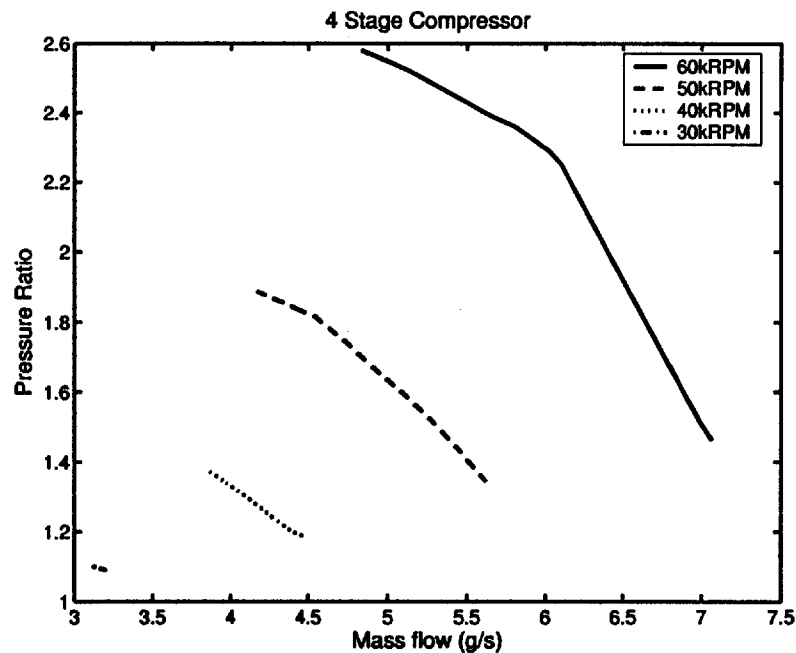


Figure 88: Performance Map of 4 Stage Compressor

operating within a mass flow region characterized by a consistent pressure ratio. The last two stages are operating much closer to choke and the effect that this has on their pressure ratios is apparent. The combined pressure ratio of the first two stages is shown in Figure 90, which shows a negative pressure ratio slope.

7.1.2 2 Stage Compressor

During the operation of the 4 stage compressor, the first stage is throttled all the way down to surge, which is the lower limit of the mass flow range. However, neither of the first two stages exhibits the substantial drop in pressure ratio at high mass flow normally attributed to high incidence angles and choke. The two stage compressor configuration was tested to allow the first two stages to operate over a larger part of their flow range. The tests were conducted similarly to the 4 stage configuration, however the valve provided a significant pressure loss even in the fully open position. An additional data point was taken at each speed with the valve bypassed and the intercooler outlet connected to

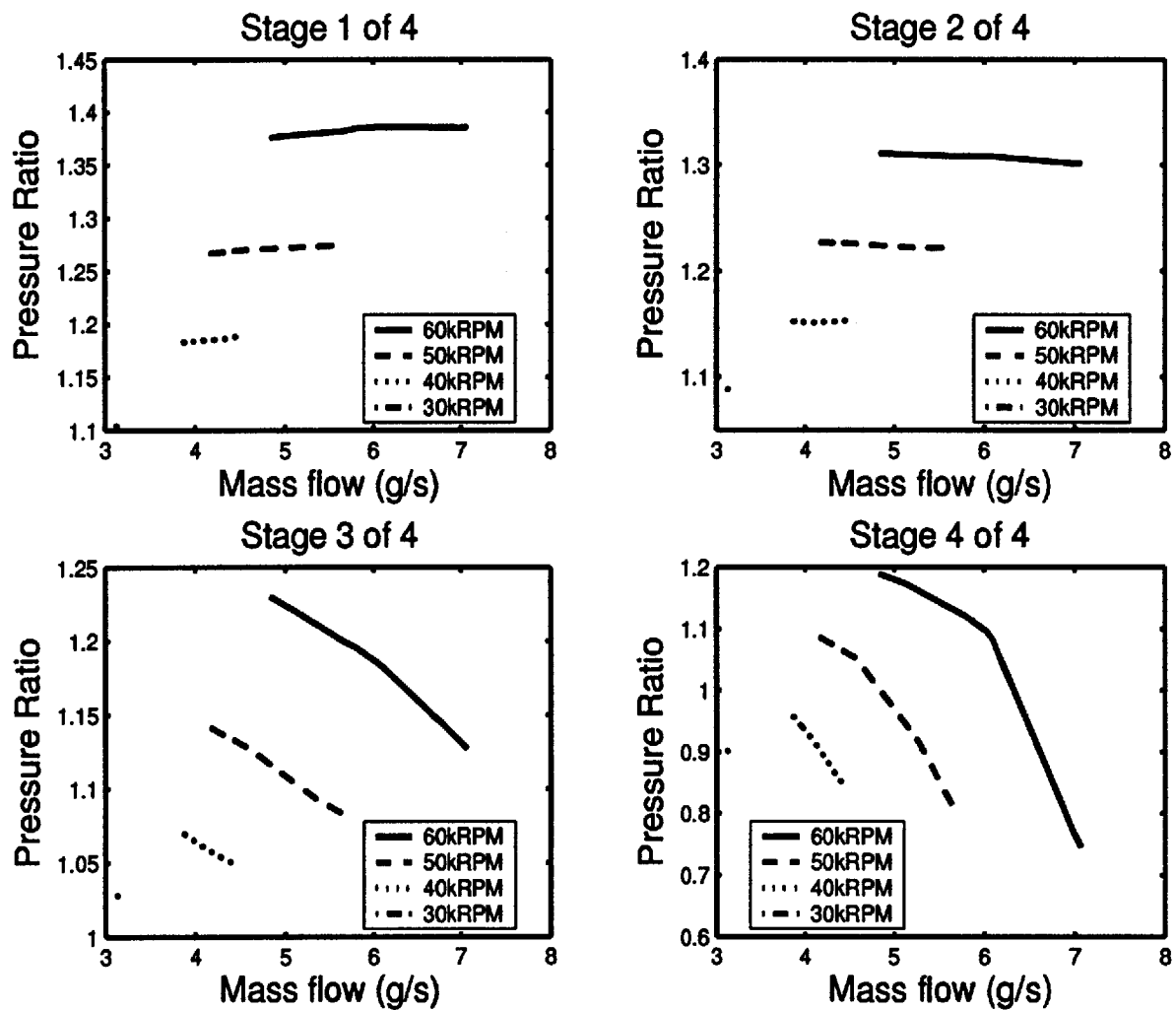


Figure 89: Individual Stage Performance of 4 stage Compressor

the ducting to the mass flow meter. The results from this test are shown in **Figure 91**.

The measurements acquired at increased flow allow a greater characterization of the first stage range, as shown in **Figure 92**. Both stages run very close to choke, but the most interesting effect is the increased propensity of the first stage to surge. At 60kRPM the first stage surges at 7.16 g/s with one upstream stage and at 4.74 g/s with 3 upstream stages. The single stage experimental results indicated the compressor surges at 10 g/s with only a throttling valve upstream.

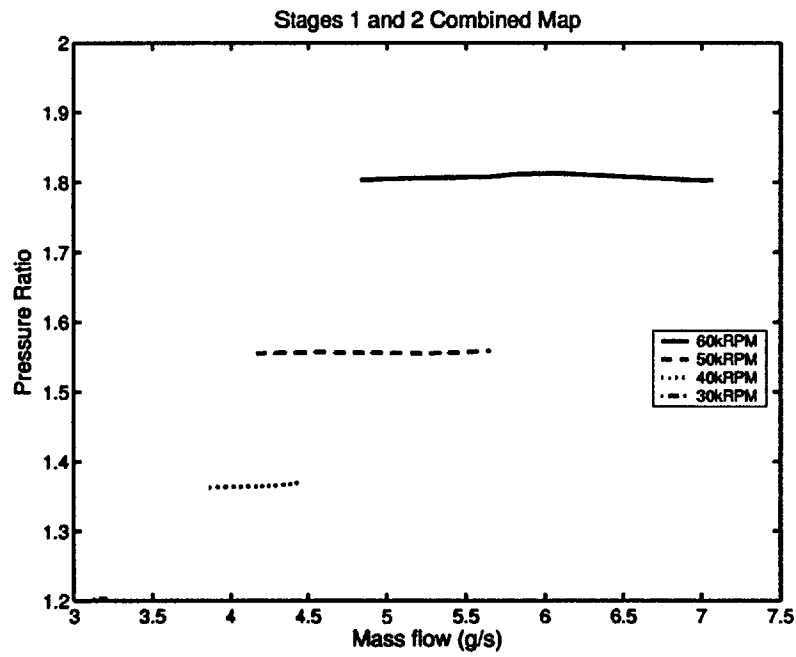


Figure 90: Performance of First 2 Stages of 4 Stage Configuration

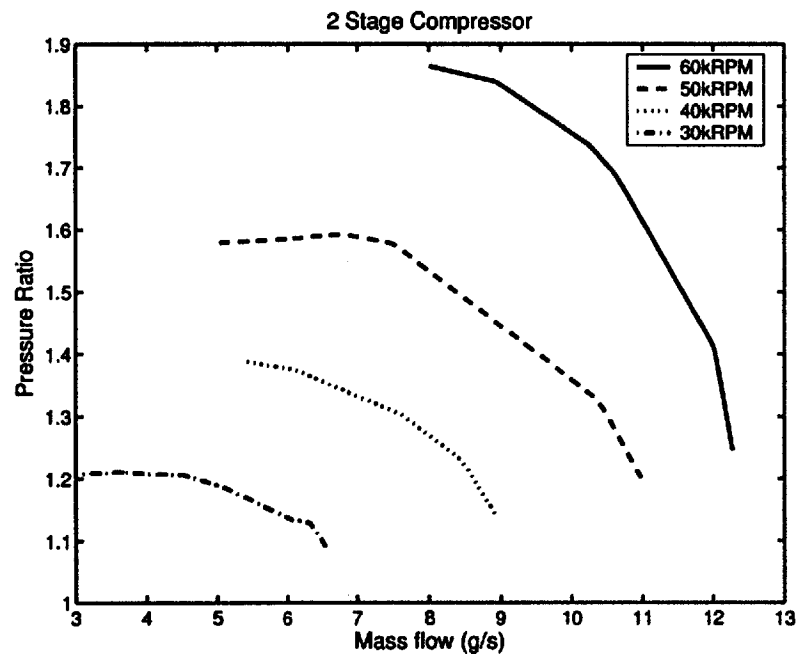


Figure 91: 2 Stage Compressor Performance

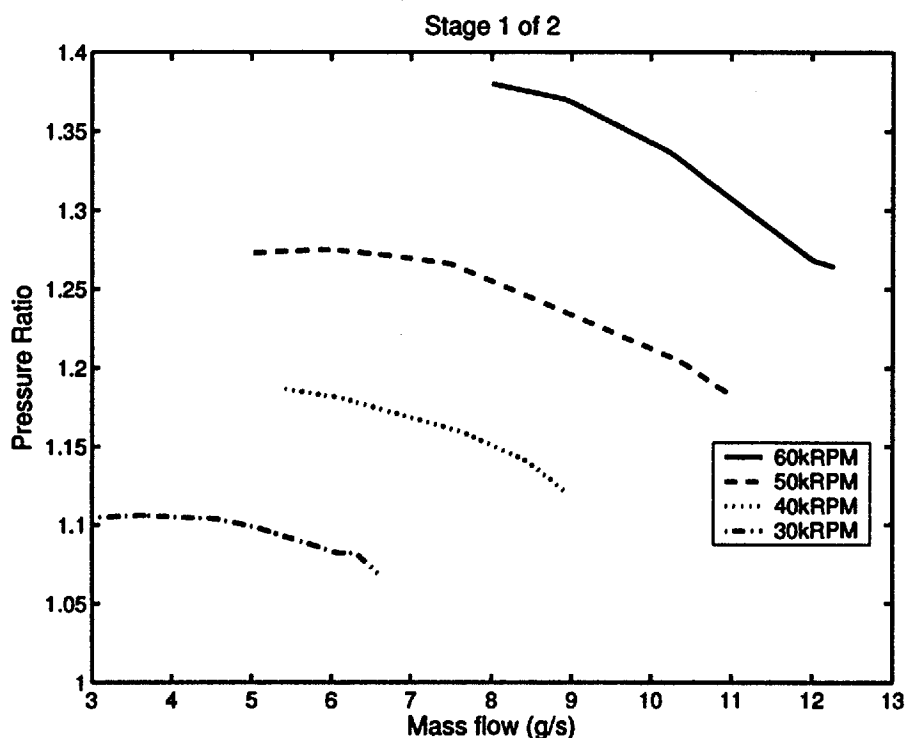


Figure 92: 1st Stage Compressor Performance within 2 Stage Test Rig

7.1.3 Temperatures

Although the temperatures increase continuously as the rig operates, due to heat accumulation in the rig and oil tank, a correlation can be seen between operational speed and temperature. The temperature plots shown in **Figure 93** show the minimum and maximum temperature measured at each speed, split into pre-intercooler measurements and post-intercooler measurements.

The most important measurement here is the inlet temperature of the rig. This is measured by a PRT suspended close to the test rig inlet. As the test rig heats up, the ambient air is heated as it is drawn past the test rig. It is also likely that the inlet air is further heated as it passes through the inlet ducting before the first stage rotor. This has a negative effect on the performance of the first stage, and this effect will carry through the compressor.

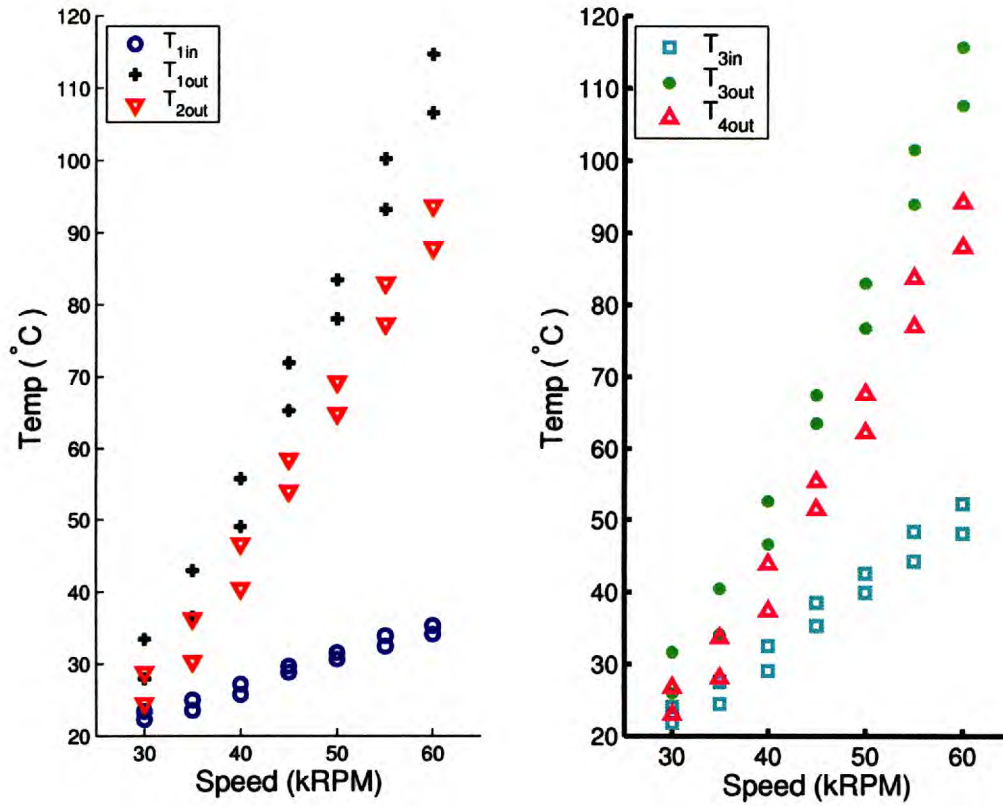


Figure 93: Temperature Minimum and Maximum at Each Measurement Speed

Also interesting is the increase in temperature at T_{1out} and T_{3out} , compared to the drop in temperature at T_{2out} and T_{4out} . A PRT in the outlet ducting is used to measure the temperature and much heat has been lost by the time the air reaches it. This could be through the casings to the gearbox or through the flexible silicone tubing.

7.1.4 Inlet Throttling

The steady state results of the inlet throttled configuration are not particularly interesting, as a slightly reduced pressure ratio in the first two stages leads to choking in the final stage. However, the surging is much higher frequency and lower amplitude than observed in the outlet throttled tests. At both 40kRPM and 48kRPM, the throttling valve can be fully closed, reducing the net mass flow to zero. A pressure ratio is still maintained, although it

oscillates in a periodic manner. **Figures 94 & 95** show the pressure at each measurement point as a function of time while the valve is gradually closed. **Figure 95** also shows the pressure recover as the valve is rapidly opened.

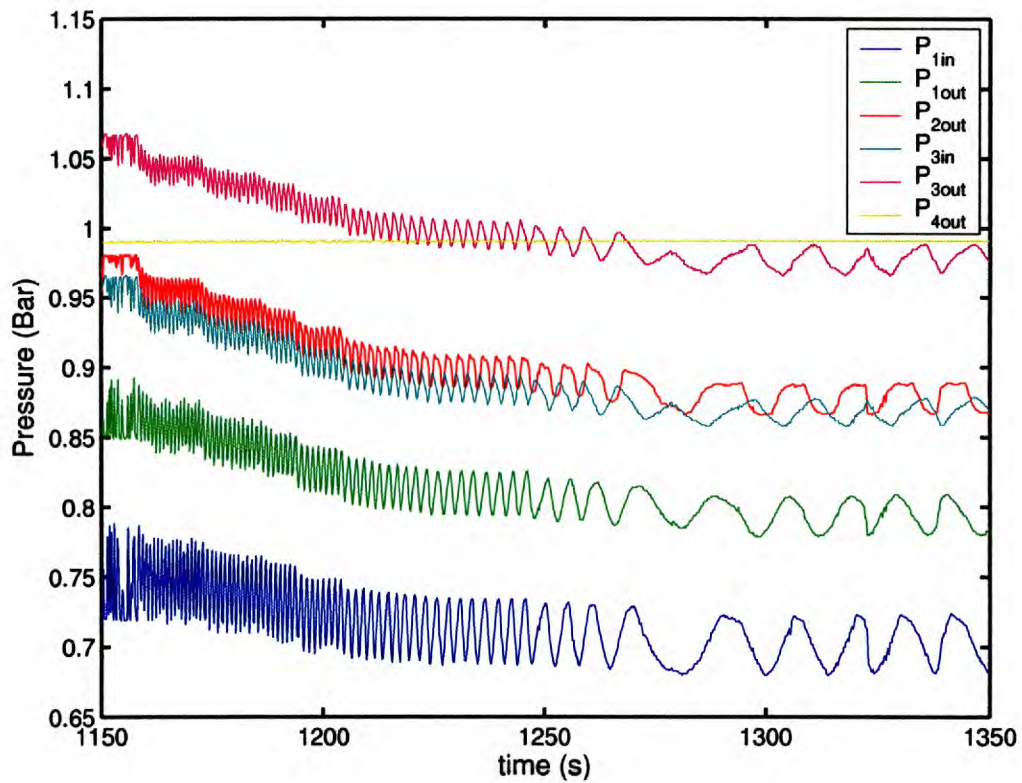


Figure 94: Surging at 40kRPM with Increasing Inlet Throttling

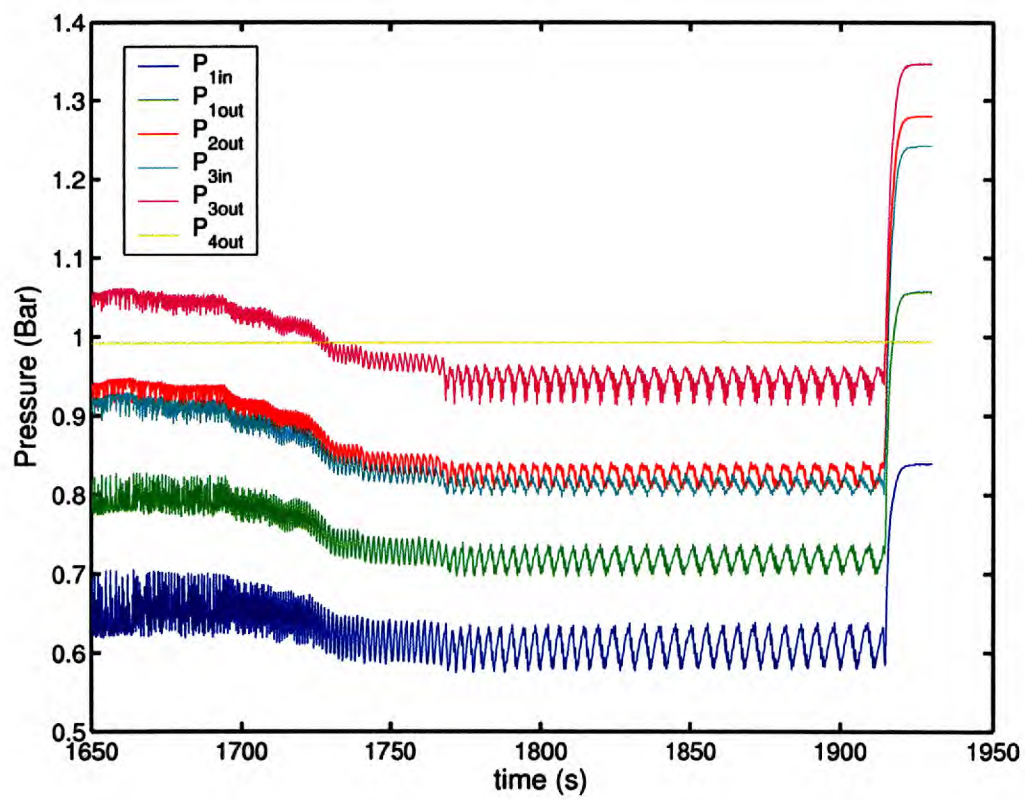


Figure 95: Surging at 48kRPM with Increasing Inlet Throttling

7.2 Analysis of Machine Performance

The main trends observed in the multistage forward-swept test rig data included :

- The resistance to surge of the first stage is increased with an increasing number of upstream stages.
- The pressure ratio of the first stage is below that measured in the single stage rig.
- Temperature measurements indicate that adiabatic conditions are not present within the rig.
- Throttling the compressor at the inlet does not cause an audible surging noise and the flow can be reduced to zero.

7.2.1 Non-Adiabatic Conditions

Highly non-adiabatic conditions eliminate the energy balance method as a basis for calculating the power consumption. However, the high levels of heat transfer within the test rig indicate sufficiently high heat transfer coefficients for the casings to affect the cooling of the working fluid.

If the isentropic efficiency is 50%, the PR is 1.38 and the inlet temperature is 35°C , the temperature after the 1st and 2nd stages would be 94°C and 165°C , respectively, if the compressor operated adiabatically. The difference between these temperatures and the temperatures measured indicate that heat is being lost from the 2nd and 4th stages. The air in the 1st and 3rd stages is likely to be heated prior to entering the rotor. The lack of temperature measurements close to the rotor inlets and outlets make any quantifications impossible.

A simple heat transfer analysis, based on the equations in [54], shows the contribution made by the radial holes in counteracting the heat transfer in the intercooler. Based on the temperatures measured in the plenum, and a constant casing temperature, 16°C of

temperature change occurs in these holes alone. The heat transfer rates in the inlet passages and diffusers are likely to be of this order, but are harder to quantify due to the geometric complexities.

Although the intercooler is known to work effectively, the gas is likely re-heated by the time it reaches the 3rd stage rotor inlet. Although temperature measurements indicate a low intercooler effectiveness, the air at the intercooler outlet was cool to the touch during early testing of the two stage configuration, giving anecdotal evidence of a high effectiveness. The intercooler effectiveness during a 4 stage test is shown in **Figure 96**. The reduction in calculated effectiveness must be caused by the reheating of the air in the plenum prior to measurement. The pressure loss has a linear relationship to the volumetric flow at the inlet, as shown in **Figure 97**.

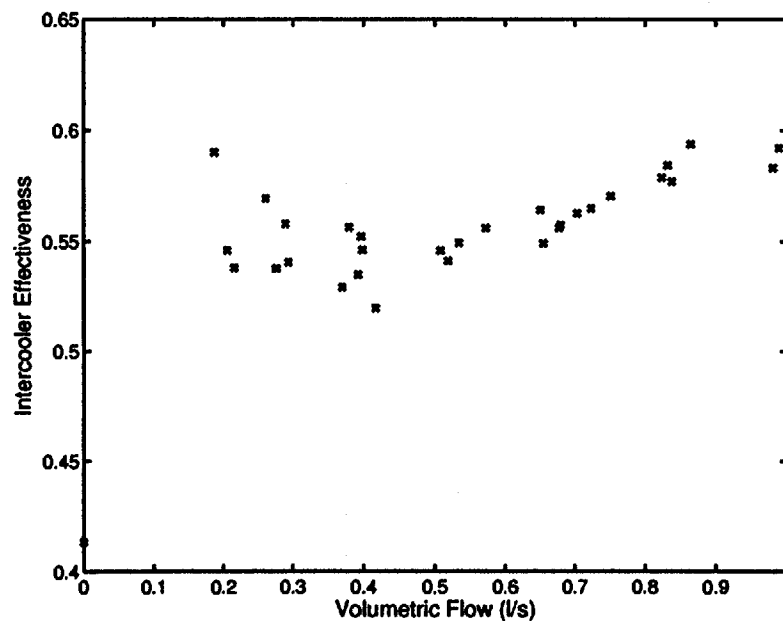


Figure 96: Intercooler Effectiveness

In order to reduce power consumption and increase PR, future testing could focus on achieving cooling between each stage, through liquid cooling of the compressor casings. However, this will still not allow the energy balance method to be used for calculating

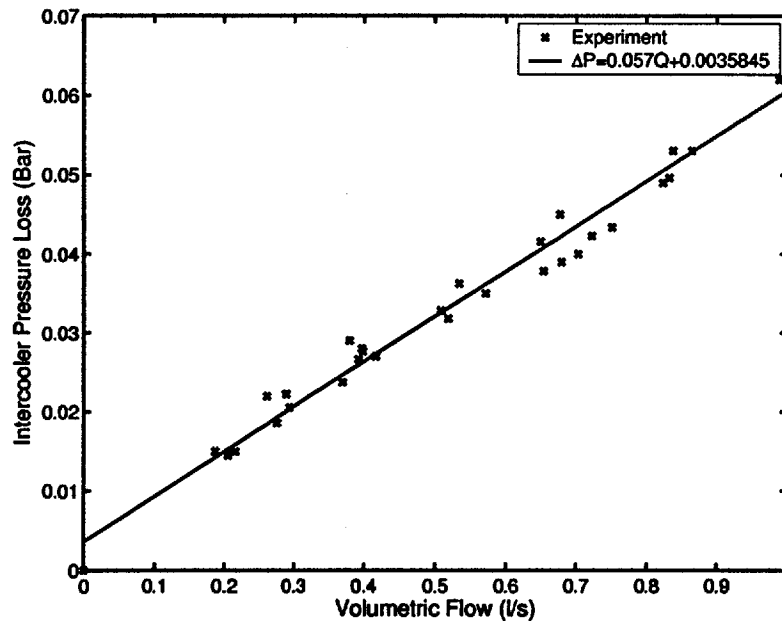


Figure 97: Intercooler Pressure Loss

individual stage power consumptions. The total machine power consumption could be measured if the thermal capacity of the cooling liquid is included and the test rig is insulated and thermally isolated from the gearbox.

7.2.2 First Stage Range

Figure 98 shows a composite map of the first stage which uses results from both the 2 stage and 4 stage configurations. Examination of the separate test results reveal that the lower mass flow results come from the 4 stage configuration. The two stage rig surges at these operation points. This indicates that the first stage point of surge is heavily dependant on upstream stages. Upstream stages allow the first stage to operate at points where it would normally surge. Additionally, the two stage test rig surges at lower mass flow than the high speed single stage test rig described in section 2.6.4.

The 1st stage is not able to approach choke during the 4-stage test, as the last 2 stages are heavily choked and the flowrate cannot increase any further. Therefore, the 2 stage

configuration is required to determine the points at high mass flow. The first stage cannot be brought up to the point of full choke, because the second stage chokes first.

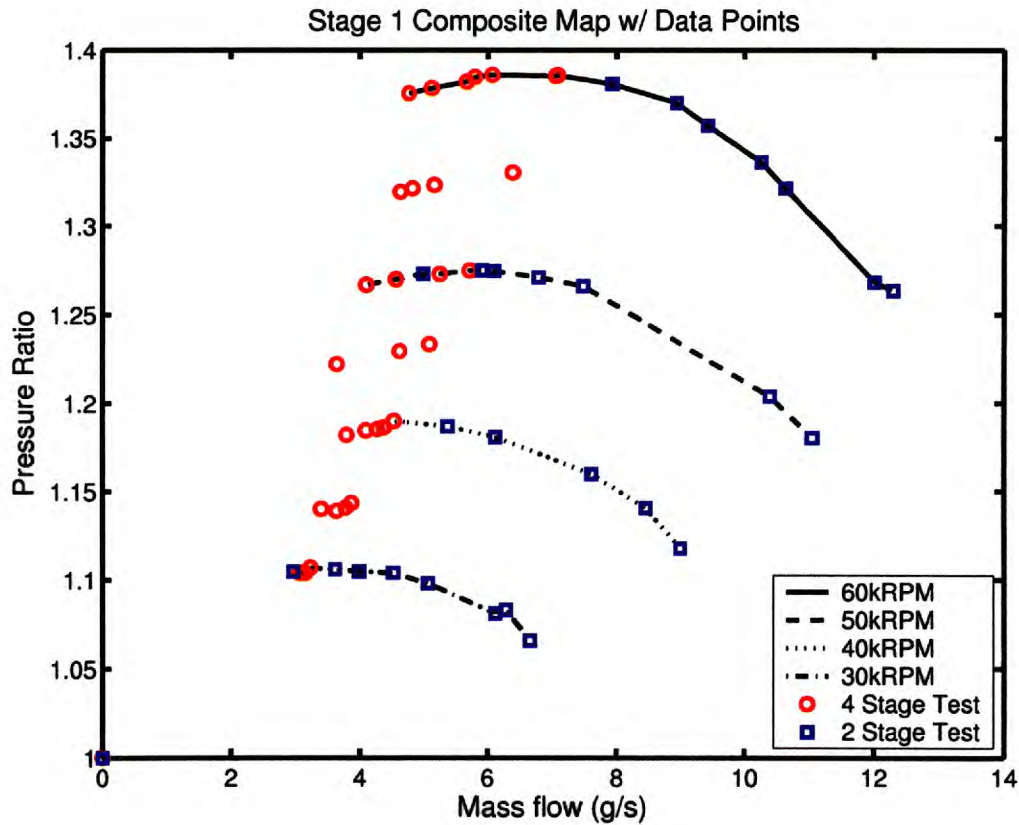


Figure 98: Stage 1 Performance: Both Tests

7.2.3 First Stage Pressure Ratio

The pressure ratio of the first stage is lower than in the single stage experiment. The most significant cause is likely to be the increased inlet temperature. Unfortunately, the exact inlet temperature cannot be determined, due to high heat transfer. The single stage experiment had a totally axial inlet up to the rotor, as it was an overhung design, allowing air to reach the rotor more directly. Additionally, the single stage test rig was driven by a cold air turbine, which kept casing temperatures low. Conversely, the multistage test rig situates the rotors between bearings, which necessitates the use of additional casings. These casings heat both the ambient air close to the inlet and the air flowing radially to

the first stage rotor inlet. Assuming the single stage experiment allowed the air to reach the rotor adiabatically at the ambient temperature of 20°C , the maximum value of T_{1in} of 35°C causes a drop in referred speed from 60kRPM to 58.5kRPM. Interpolation of the single stage experiments gives a pressure ratio of 1.405 at 58.5kRPM, down from 1.42 at 60kRPM. Additional heating is likely to occur in the inlet ducting, which will reduce stage pressure ratio further. Interpolation of the single stage test results gives a PR of 1.38 at 55.7kRPM. If adiabatic conditions were present, this would be dimensionally similar to the first stage operating at a inlet temperature of 67°C at 60kRPM. The exact contribution of increased inlet temperature to the discrepancy in pressure ratio cannot be determined.

7.2.4 Surging With Inlet Valve.

The compressor does not overheat, vibrate excessively or enter into deep surge when the inlet valve is fully closed. This gives it advantages as a fuel gas compressor, as it may be isolated from the fuel supply while running without serious damage occurring. This may be beneficial to the microturbine start-up procedure, whether or not the fuel gas compressor is directly coupled or driven electrically.

Furthermore, if **Figures 94 and 95** are examined, the first two stages appear to surge in phase with each other, while the 3rd and 4th stage pressures oscillate in phase with the 3rd stage inlet. This phase shift indicates that the intercooler provides significant volume between the second and third stage. This will facilitate surging of the first two stages.

7.2.5 Total Machine Range

It can be seen from **Figure 91** that the 2 stage configuration surges when the total compressor pressure slope gets close to flat, which causes flow instability. The pressure slope of the 4 stage configuration is still negative at the point of surge, however, **Figure 90** shows the pressure map of the first two stages. It is likely that the surging is initiated by air in the intercooler expanding back against the first two stages. It may be that if the

intercooler is eliminated, the range of the four stage configuration could be extended.

7.3 Re-simulation of Compressor

From the observations made with regard to the range of the first stage, it can be seen that the simulations based on the single stage test results were overly restrictive, due to the lack of steady state data at low mass flow. During the entire 4 stage test the mass flow rate was always lower than the mass flow at which the single stage test rig surged. If the compressor map shown in **Figure 98** is used as the basis for a new compressor simulation, the simulation technique can be readapted. However, a few assumptions must be made:

- **The mass flow of each stage will be scaled with blade height as before:** This must be done because the full compressor range is only determined for the first stage. This allows the first stage to be used as a basis for the other stages.
- **Because the efficiency can not be accurately determined with the available test results, an efficiency of 50% is used at all points:** Although tested at 3X scale, the experimental single stage map in **Figure 34** indicates that this is may not be an unreasonable achievement in future testing, especially far from the point of choke.
- **The inlet temperature is set at 35°C:** The simulation has the same limitation of increased inlet temperature as the test rig.
- **The pressure ratio of each stage must be adjusted in order to account for the lower pressure ratios in the later stages:** The pressure ratios of each stage is lower than the previous stage, likely due to increased surface to area ratio within the passages. In order for the later stages to be scaled, the pressure ratio at each point is raised to the power of an exponent, z , between 0 and 1. There is merely a correction factor which scales first stage pressure about 1, allowing it to be used to represent the other stages in a simulation.

Suitable exponents were determined by iteration and are presented in **Table 23**.

Stage	Blade Height (in mm)	exponent (z)	$PR = 1.38^z$
1	3.0	1.00	1.38
2	2.3	.88	1.328
3	1.4	.73	1.265
4	1.1	.63	1.224

Table 23: Stage Exponents

The overall machine performance and the individual stages' performance are presented in **Figure 99** and **Figure 100**.

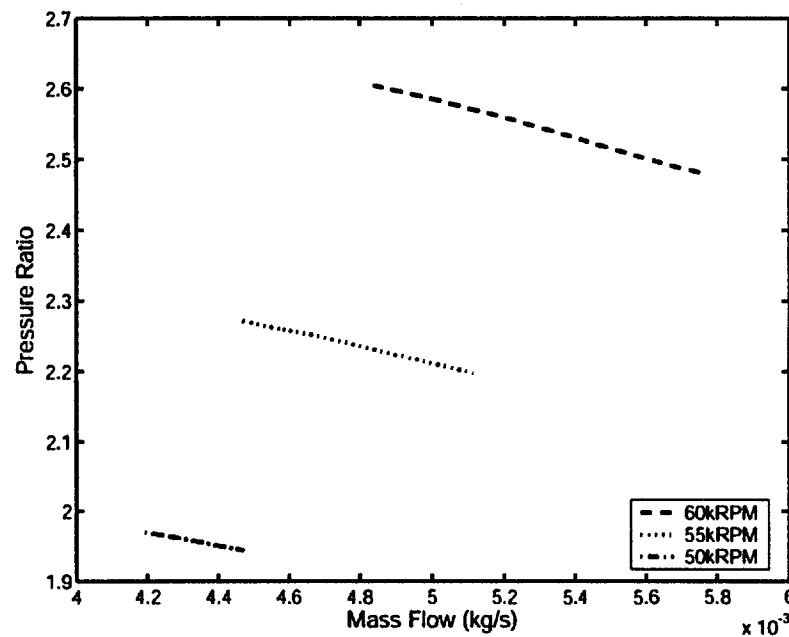


Figure 99: Simulated Compressor Performance, Based on Composite First Stage Map

If compared to the experimental results shown in **Figure 88** and **Figure 89**, it is observed that the lower end of the range is accurately represented, but the simulation does not model the compressor at higher mass flow rates and lower speeds. Upon closer examination of the 4th stage, many points of the experimental plot have a pressure loss at the last stage. Points close to choke could not be measured either for the first stage or during the single stage tests described in section 2.6.4. If the compressor could be tested at higher mass flows, these points could likely be represented.

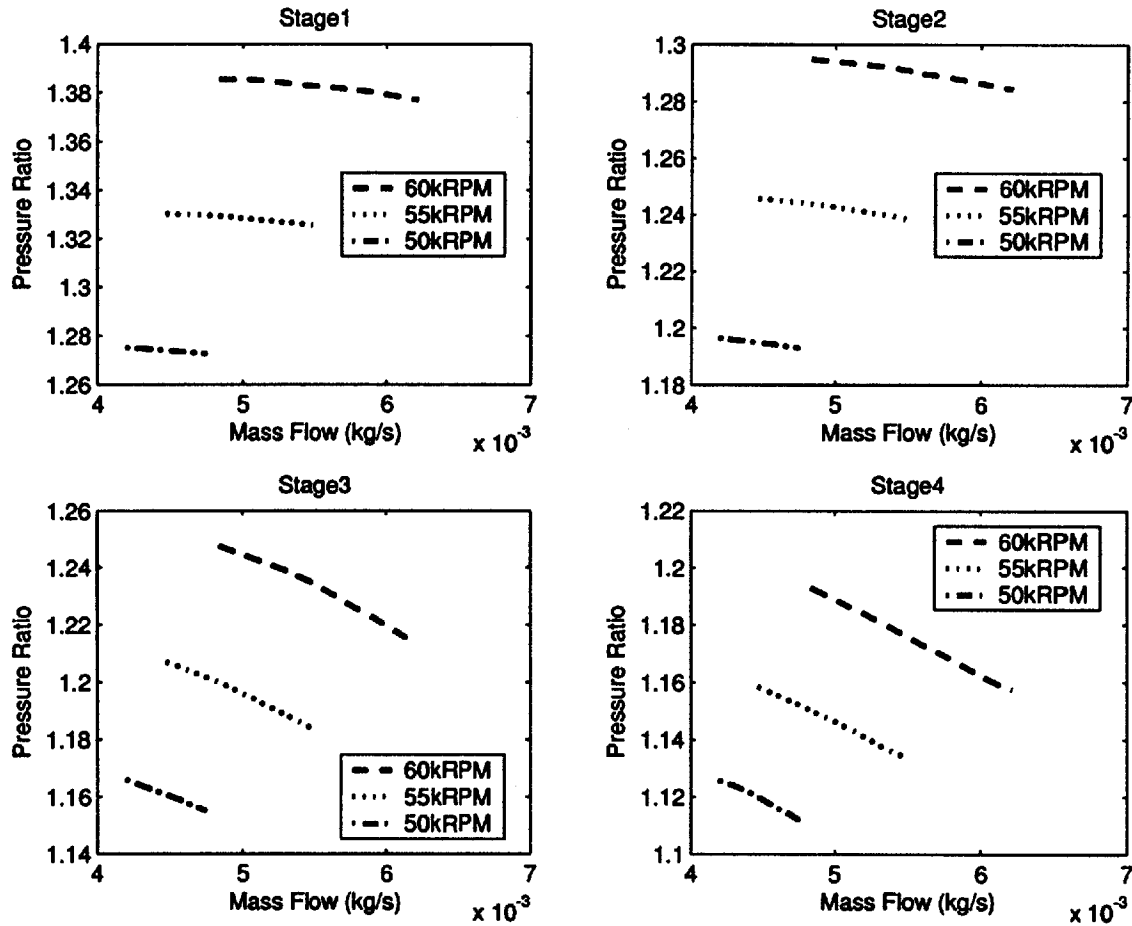


Figure 100: Simulated Stage Performance, Based on Composite First Stage Map

7.4 Compressor Redesign

If the compressor is redesigned with new stage matching, both the peak pressure ratio and range can be increased. This is due to both taller blades and better matching. New blade heights presented in Table 24 were chosen based on the peak pressure ratios expected. The compressor is then re-simulated with the approximations made above. The new exponents are linearly interpolated by blade height based on Table 23. The intercooling is increased to 0.8 for each stage. The inlet temperature remains at 35°C. The resulting machine simulation is shown in Figure 102. The peak pressure ratio of 3.0 occurs at a mass flow of 5g/s and an efficiency of 35%.

Property	Stage 1	Stage 2	Stage 3	Stage 4
Blade Height	3.0mm	2.4mm	1.7mm	1.5mm

Table 24: Stage Blade Heights

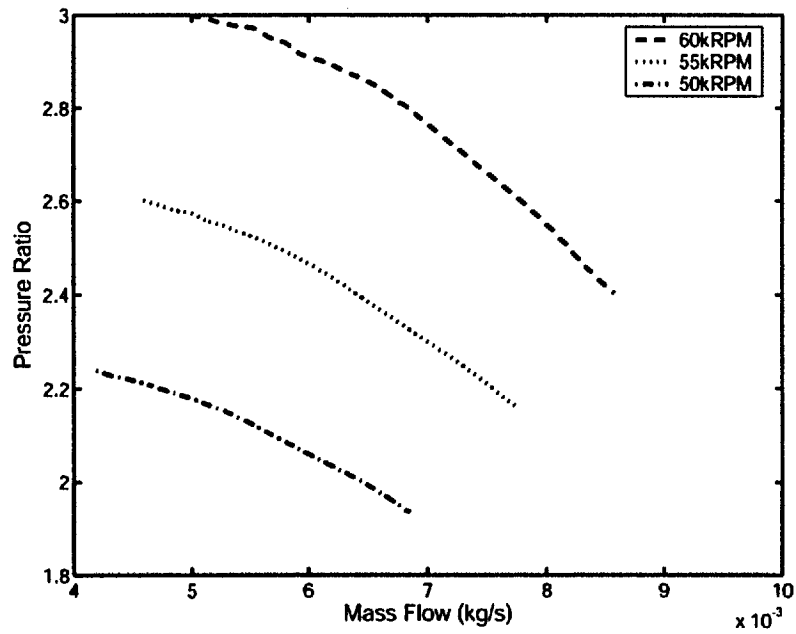


Figure 101: Simulated Performance of Redesigned Compressor

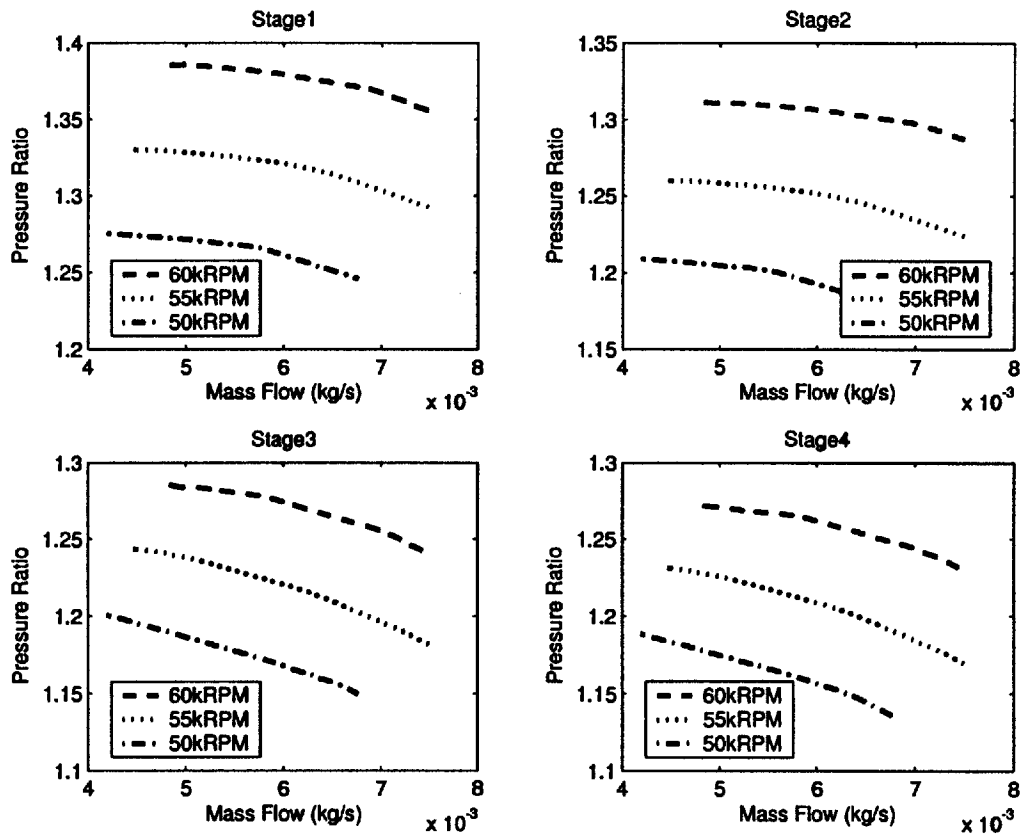


Figure 102: Simulated Individual Stage Performance of Redesigned Compressor

It is not known if the upstream stages will provide the same level of surge resistance to the first stage as was present in the experimental test rig. As all stages maintain a negative pressure slope, the flow is likely to be stable. Nevertheless, at more appropriate stage matching, a performance increase is predicted.

7.5 Uncertainty Analysis

A study of the propagation of uncertainty showed that the measurements of pressure ratio and total mass throughput are sufficiently accurate. The effect of temperature on the energy-balance efficiency calculation is shown.

7.5.1 Pressure Ratio Uncertainty

The pressure ratio for a stage, given inlet and outlet gauge pressures, P_{IN} and P_{OUT} , and atmospheric pressure measurement, P_{REF} , is calculated by **Equation 121**. The uncertainty of the calculated values are given by **Equation 122** through **Equation 125**, based on uncertainties of pressure measurements $U_{P_{IN}}$, $U_{P_{OUT}}$, and $U_{P_{REF}}$. The calculated uncertainty is shown in **Table 25**

$$PR = \frac{P_{OUT} + P_{REF}}{P_{IN} + P_{REF}} \quad (121)$$

$$U_{PR} = \sqrt{\left(U_{P_{OUT}} \frac{\delta PR}{\delta P_{OUT}}\right)^2 + \left(U_{P_{IN}} \frac{\delta PR}{\delta P_{IN}}\right)^2 + \left(U_{P_{REF}} \frac{\delta PR}{\delta P_{REF}}\right)^2} \quad (122)$$

$$\frac{\delta PR}{\delta P_{OUT}} = \frac{1}{P_{IN} + P_{REF}} \quad (123)$$

$$\frac{\delta PR}{\delta P_{IN}} = -\left(\frac{P_{OUT} + P_{REF}}{(P_{IN} + P_{REF})^2}\right) \quad (124)$$

$$\frac{\delta PR}{\delta P_{REF}} = \frac{1}{P_{IN} + P_{REF}} - \left(\frac{P_{OUT} + P_{REF}}{(P_{IN} + P_{REF})^2}\right) \quad (125)$$

Calculation	PR_1	PR_2	PR_3	PR_4	PR_{12}	PR_{34}	PR
Nominal Value	1.39	1.31	1.24	1.19	1.82	1.48	2.59
Uncertainty	0.008	0.012	0.013	0.021	0.013	0.023	0.038

Table 25: Peak Pressure Ratio Uncertainty Results

7.5.2 Mass Flow Uncertainty

Although the specific volume at the mass flow meter was calculated using Van Der Waal's correction, the uncertainty analysis shows that the use of the ideal gas law is sufficient, even if the error of R is one percent. At the nominal operating point, the uncertainty of total compressor mass flow is 0.24g/s. The individual stage measurements are not known,

due to the inability to measure the leakage flow rates.

$$v = \frac{RT_{fm}}{P_{fm}} \quad (126)$$

$$U_v = \sqrt{\left(U_{T_{fm}} \frac{R}{P_{fm}}\right)^2 + \left(-U_{P_{fm}} \frac{RT_{fm}}{P_{fm}^2}\right)^2 + \left(U_R \frac{T}{P}\right)^2} \quad (127)$$

$$\dot{m} = \frac{\dot{V}}{v} \quad (128)$$

$$U_{\dot{m}} = \sqrt{\left(-U_v \frac{\dot{V}}{v^2}\right)^2 + \left(U_{\dot{V}} \frac{1}{v}\right)^2} \quad (129)$$

Calculation	$\dot{V}$	T_{fm}	P_{fm}	R	v	$\dot{m}$
Nominal Value	$6 \times 10^{-3} m^3/s$	$283K$	$101000Pa$	$287J/kg \cdot K$	$0.81m^3/kg$	$7.4g/s$
Uncertainty	$0.18 \times 10^{-3} m^3/s$	$1K$	$800Pa$	$2.87J/kg \cdot K$	$0.011m^3/kg$	$0.24g/s$

Table 26: Mass Flow Uncertainty Results

7.5.3 Efficiency Error

Due to the interstage heat transfer and the input of heat from the gearbox, efficiency calculations using the energy balance method are determined to be inaccurate. **Figure 103** shows the dependance of the calculated efficiency on the temperature of the outlet gas for the first stage at the operating point. The uncertainty can not be predicted, due to the complex heat transfer phenomena within the test rig. It is therefore not prudent to make efficiency calculations based on the energy balance method, either for the individual stages, or for the machine as a whole.

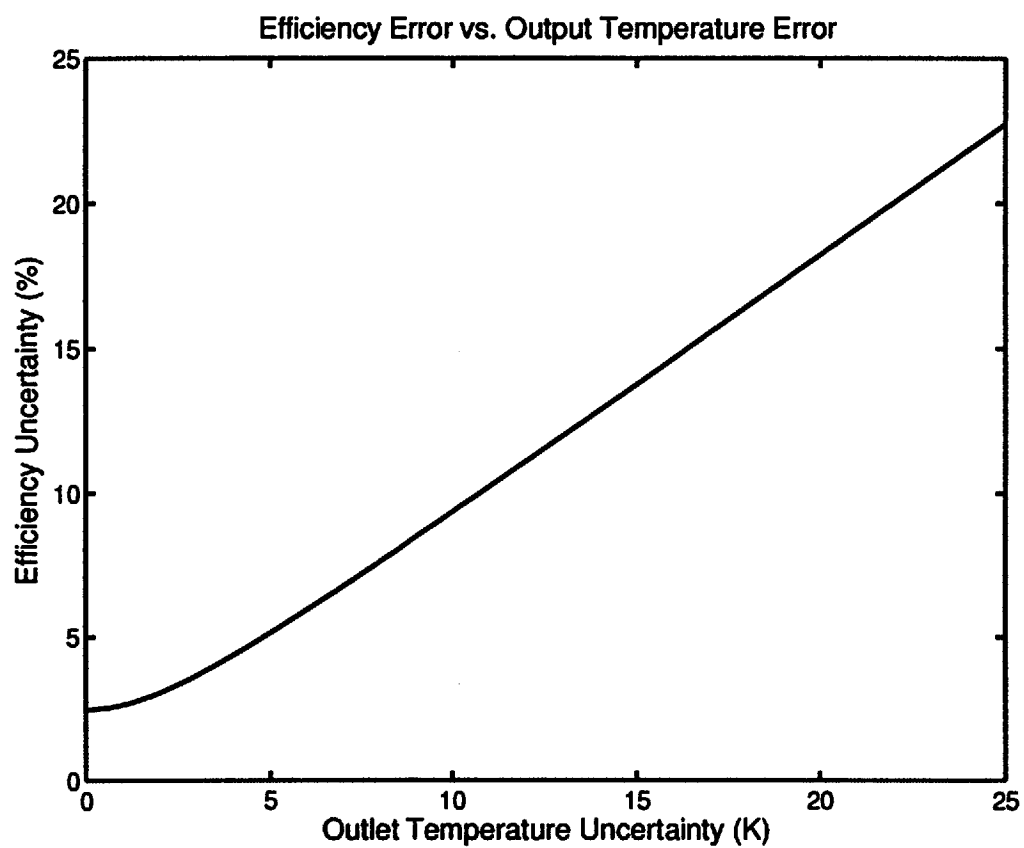


Figure 103: Dependence of Efficiency Calculation on Temperature Measurement

8 Conclusions

In this section the research is reviewed, the achievement of the stated objectives is evaluated, and recommendations are made for future work.

8.1 Review of Work

8.1.1 Microturbine Fuelling Requirements

In chapters 1 and 2, the requirements for a fuel gas compressor were investigated. The suitability of current compression technology was evaluated, and it was found that positive displacement machines are required for the low flow rates required. A new type of turbo-compressor was introduced, and the associated research was summarized.

8.1.2 The Multistage Forward-Swept Compressor

The multistage forward-swept compressor was introduced in section 3. The compressor was designed to deliver a flowrate and pressure ratio required for fuelling a microturbine with natural gas, but at a similar shaft speed. The stages are arranged similarly to a single shaft industrial compressor.

8.1.3 Simulation and Predictions

The multistage simulation code presented in chapter 4 was developed to allow single stage performance predictions to be used in the evaluation of a multistage compressor design. The test rig was designed using CFD predictions for the single stage performance, which were subsequently found to be overly optimistic in terms of pressure ratio and surge margin. The revised results, based on single stage experiments, indicated that the multistage compressor design should be surging at the first stage while choked at the last and therefore have no useable range. Further simulations were performed using revised first stage results from the multistage experiments.

8.1.4 Experimental Testing

Chapter 5 details the test rig design and manufacture and chapter 6 describes the experimental setup and procedure. A multistage compressor was successfully tested up to the design speed of 60,000RPM in both a 4 stage and 2 stage configuration.

8.1.5 Experimental Results

Chapter 7 presents the results of the testing. It is found that the first stage performance is lower than previous experiments and the later stage performance is lower still, due to reduced blade height and flow mismatching. However, the individual stage mass flow range is found to increase as upstream stages are added. A pressure ratio of 2.58 at 4.74 g/s is achieved for the 4 stage compressor. A compressor redesign is predicted to have higher pressure ratio and improved range.

8.2 Evaluation of Objectives

The goals of this work, presented in section 1.4, were:

- To accurately model a multistage implementation of the forward-swept compressor, generating design tools applicable to gas compressors for the range of microturbine operating powers.
- To develop a gas compressor which serves as both a prototype gas compressor for fuelling a microturbine of 100kW output power and as a test rig for simulation model validation.

8.2.1 Design of a Prototype Fuelling compressor

Unfortunately, the goal of producing a compressor suitable for supplying the pressure and flow requirements of a 100kW microturbine was not realized. The pressure ratio of 2.6 falls far short of the requirement of 4.5. The mass flow delivered was 4.8 g/s at full pressure ratio while the volumetric flow rate equivalent to natural gas is 12g/s. The reduced mass flow is mostly caused by the mismatching of the later stages. The low stage pressure ratios

cause both the low overall pressure ratio and the mismatching in the later stages.

However, this work has generated a much better understanding of the problems involved with designing and testing the multistage forward-swept compressor. With single stage improvements, improved cooling and a greater understanding of the effect of the multistage arrangement on compressor range, it may be possible to build a properly matched multistage compressor that will meet the required mass flow rate and pressure ratio.

8.2.2 Characterization of the Multistage Arrangement

The multistage arrangement has been adequately simulated, although simulation predicts less range than is observed in practice. A more accurate assessment of the single stage range is needed to improve the simulations. A major shortcoming is in accounting for the internal heat transfer between stages, which is detrimental to both efficiency and pressure ratio. However, it has been shown that significant heat transfer between working fluid and casing is possible, and this may be useful for introducing provisions for intercooling within the compressor in later designs.

Additionally, it was observed that upstream stages can aid in suppressing surge in the first stage. This implies that a larger range may be expected of individual stages than single stage testing would indicate. This will aid in the difficult task of compressor matching, allowing a design point closer to the peak stage pressure ratio than would otherwise be attempted. If the external intercooler is eliminated, the range may increase further.

8.3 Future Recommendations

There is still much to be learned about both the forward-swept centrifugal compressor and its implementation in a multistage arrangement. Advancements in individual stage performance must be implemented in future test rigs.

8.3.1 Test Rig Modifications

The multistage experimental test rig can still be used to produce useful data of interest in this area of research. During the course of the described research, advancements have been made in the single stage design, which have been entirely limited to changes in diffuser geometry. Therefore, all stages can benefit from an increased pressure ratio and efficiency by implementing new diffusers with the current rotors.

The rig has been designed to accommodate changes in stage blade height, and the 3rd and 4th stage rotors could be replaced with rotors of increased blade height, provided diffusers were also replaced. This will reduce the effects of choking seen in the current design.

Additionally, the introduction of additional cooling facilities for the gearbox and test rig oil supply will reduce overall temperatures and increase the cooling. This should increase the compressor pressure ratio and reduce the degree of mismatching. A heat shield between the compressor and gearbox will produce similar benefits. This will also allow liquid cooling passages in the compressor to be effectively utilized.

8.3.2 Investigation of Minimization of Seal Deflections Through Rotor Design

Currently, the face seals on the rotor deflect away from the stationary face due to centrifugal loading. It may be possible to design the back face of the rotor such that this deflection is reduced. This may increase the stresses in the rotor, but current stresses are well below the yield stress of available materials.

8.3.3 Numerical Prediction of Internal Heat Transfer

Because of the relatively low flowrates and high solidity casings of the forward-swept compressor design, the heat transfer between stages is significantly higher than in conventional turbomachines. In order to accurately assess the effects of internal heat transfer between stages, work should be done on the application of the finite difference heat transfer model,

as described in Appendix B, to the compressor and internal geometry. This will allow both a greater characterization of the test rig heat transfer and an assessment of intercooling possibilities.

8.3.4 Investigation of Alternate Applications

Some other applications that have been identified as possible uses for the multistage forward-swept compressor are:

Hybrid Turbocharging and Electric Supercharging: The reduced speed of the forward-swept compressor allows much lower speeds for a given mass flow rate. The low speed is attractive because motors and generators are expensive at the shaft speeds present in conventional automotive turbochargers.

Fuel Cell Compressor: Fuel cells can be made more compact for a given power if the air is pressurized. However, they can not be turbocharged in the same manner as an internal combustion engine because the exhaust gas is not hot enough to drive the turbine. The concepts of hybrid turbocharging and electric supercharging can be carried over to fuel cell pressurization.

Industrial Gas Compression: Currently, the final stages in industrial centrifugal compressors have a low blade height and low pressure ratio. The forward-swept compressor has advantages in efficiency and pressure ratio at low volume flows.

The suitability of the multistage forward-swept compressor for these applications should be investigated.

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Appendices

A Stacked Rotor Design

A previous design, patented by Pullen [53], is shown in **Figure 104**. This was analyzed using the techniques described in Chapter 4 prior to the implementation of off-design analysis. The simulations showed that the seal leakage was too large, due to each seal experiencing the entire stage pressure ratio at high diameter. This was analyzed prior to the development of the high forward sweep concept, when windage caused considerable losses, due to higher rotor diameter. The high forward sweep development made this approach less useful than the separate rotor design.

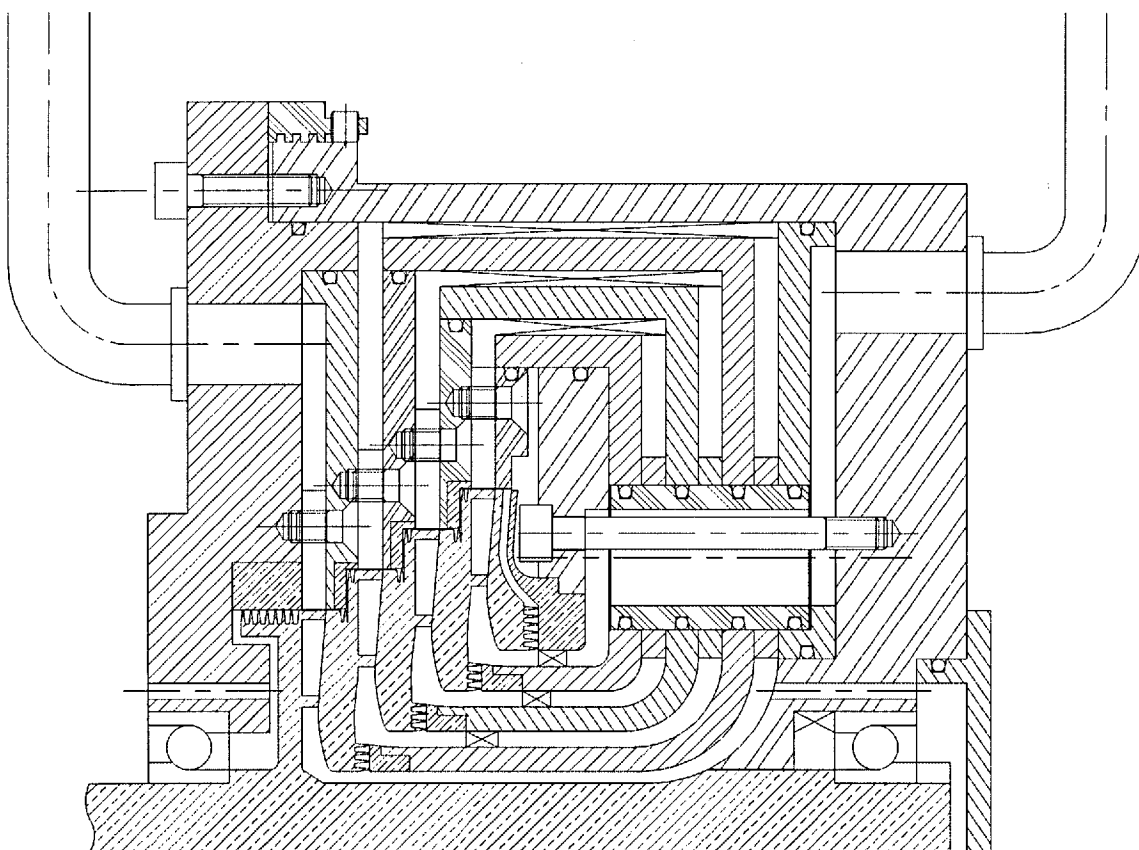


Figure 104: Stacked-Rotor Design

B Intercooling Analysis

Initially, the feasibility of using a custom intercooler integrated into the test rig was investigated. Due to the added complexity of construction it was decided to use an external intercooler instead. This reduced the amount of manufacture, improved the modularity of the test rig, and made instrumentation access easier. However, as interstage heat transfer was found to be high, issues of heat transfer should be readdressed. This section presents the initial findings of intercooler investigations and describes the design tools that were created.

B.1 Basic Design

The intercooler is an important part of the rig design. It is necessary to keep the gas temperatures within acceptable limits, and to reduce interstage heat transfer. By specifying an intercooling effectiveness, there is no thought to how the performance is achieved, only what results are expected. It is important to realize that the size of the intercooler increases dramatically with increased effectiveness, due to the increased area required to move heat from low temperature gas. It is therefore only possible to optimize the amount of effectiveness for a given intercooler size. This chapter describes the basic design and simulation tools that were developed in the process of optimization and analysis. Intercooler effectiveness is defined as:

$$\eta_{IC} = \frac{T_{inlet} - T_{outlet}}{T_{inlet} - T_{atm}}$$

There are two parts of the intercooler, the ducts that transfer heat from the compressed gas to the casing, and the fins that transfer the heat from the casing to the air. The design of the intercooler is motivated by the following phenomena.

1. Air-cooling is desirable, as the elimination of any ancillaries necessary for liquid cooling brings benefits to the machine design in terms of cost, simplicity, reliability and size.

2. It is desirable to have the intercooler integrated into the casing in order to minimize the potential for leakage and to minimize the overall size of the machine.
3. Given a reasonable cross-section of flow, the flow in the intercooler is predominantly laminar due to the low mass flow. This implies that the heat transfer coefficient is unaffected by surface roughness and gas velocity.
4. The air cooling on the outer fins does not have any mass flow restrictions and it can therefore be sent at high velocity across a large surface area.

The fins can be of similar shape to those found on many low speed conventional motors, as the heat transfer is easily modulated by increasing the air flow. However, the gas velocity in the internal channels is inversely proportional to the flow through them, at a given mass flow rate. The heat transfer is therefore going to be maximized by increasing the wetted area.

These conditions allow the temperature of the conducting barrier to be kept well below the temperature of any gases that require intercooling and very close to ambient. This means that the gas flowing through the concentric channel arrays can be kept in close physical proximity, and that the maximum heat transfer occurs for a given pressure loss and surface area.

The gas flows through internal passages created by fins located on concentric annuli. The heat flows out through these annuli and to external fins which are air cooled through the use of an external fan. The effectiveness of the intercooler is determined by the cooling fan power and its complexity, which in turn is limited by manufacturing aspects, as described in the next section.

B.2 Manufacturing Restrictions

There are two ways that the intercooler can be manufactured in production. The first method would be to create complex extrusions that fit within each other. The heat sinks present in computer applications use this technology. The second option would be to use

corrugated foil that has been soldered together in the manner similar to compact heat exchangers, such as radiators and intercoolers used in the automotive industry.

B.2.1 Extruded Aluminium

Fully finned aluminium annuli produced by extrusion would be an option for production, but are unfeasible for a one off. Still, there are a few options that would simulate an extruded part. However, expert knowledge is required for the extrusion and die design, and the extrusion process. There are three solutions for creating a prototype to simulate an extruded component.

1. Milled Finned Annuli

The passages could be machined with a slitting saw. Technician experience and advice is the best data as to what is achievable. A slitting saw of 1/16" is considered the limit of what is practical. The smallest available slitting saw is 1/32", but it is too flexible to accurately cut a large number of slits. This process would produce an accurate simulation, but would be very time consuming and expensive.

2. Spark Eroded Finned Annuli

As the components would all be of a constant cross-sectional profile, they could be spark-eroded by using a CNC wire. This would again be time consuming and expensive and would create a very rough surface finish.

3. Extruded Cyclic Sections

Axially symmetric extrusions could be created, which would more accurately simulate the production process than a part machined from solid. It would be much easier than a whole section because the die would be much smaller and have no hollow parts. They would be held in place by a centrally located annulus. It would most likely be less expensive than the other options, but an extrusion expert would still be required to design the die.

B.2.2 Corrugated Foil

Intercoolers made from corrugated foil are common in automotive applications. Thin, corrugated foil is created by rolling sheet through shaped rollers. This foil is sometimes ribbed to encourage boundary layer separation and increase heat transfer. The sheets are then brazed together. This is the cheapest option for mass production heat exchangers. A custom designed intercooler of this type could be integrated into a production machine. A custom design of this type would be difficult for a one-off, but is an option. It could be very simple to manufacture, as it would be assembled by hand and brazed in an oven, but it would be imprecise for the same reason.

B.3 Model Fundamentals

Many choices of modelling methods present themselves concerning the analysis of the intercooler. The simplest model of any heat exchanger is an analytical one that assumes perfect thermal conductivity across any walls, and infinite resistivity in the direction along the walls. This assumes no conduction in the direction of flow, and no thermal resistance due to the pipe, which are not unjust assumptions for a simple heat exchanger. The primary problems with this type of model is the inclusion of only two flows and the inability to model complex shapes. The cross-section has at least two gas flows and one cooling air flow. At the other end of the modelling spectrum is CFD, which can produce reliable solutions, especially for laminar flow, but is computationally intensive, and therefore not suited as a design tool. The solution that was chosen is a compromise between the two. As the cross-section of the conducting metal is complicated, but the cross section of the individual gas flow paths is not, it was chosen to use a finite difference conduction model of the intercooler and empirically derived convection coefficients for the individual gas flows.

B.4 The Finite Difference Model

The finite difference equations for steady state conduction are essentially an energy balance. Each node has a temperature that is applicable to the 1-4 areas surrounding it. The number of areas corresponding to a node is dependant on the nodes position on the grid.

Each area is bounded by other areas and boundary lines. This is shown in **Figure 105**.

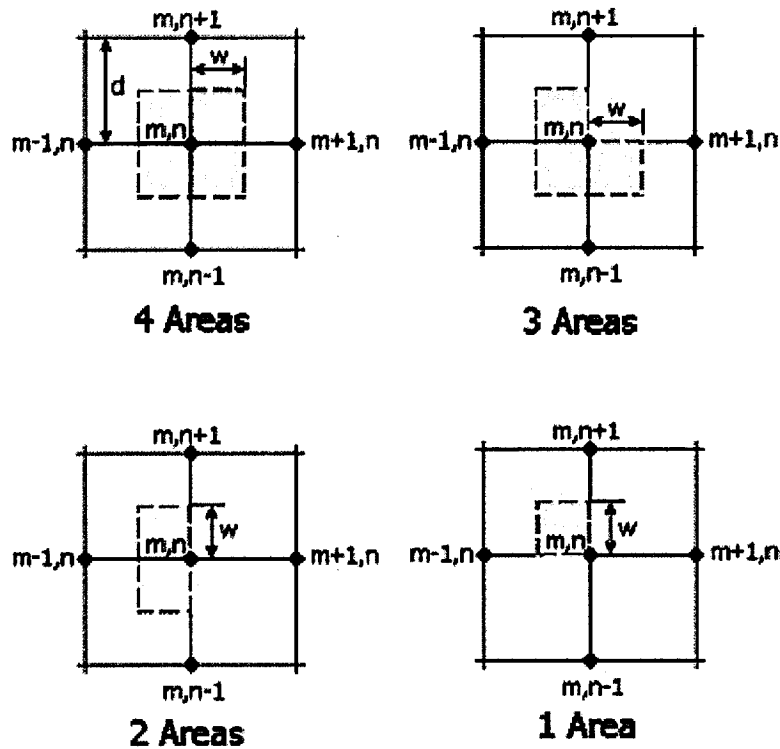


Figure 105: Different Element Types

Boundary lines come in four types.

1. Conduction from continuous solid
2. Conduction from contact surface
3. Convection from fluid
4. Symmetrical boundary

The convention that is used here is that heat flux into the area is positive. Because no internal heating or transient effects are present, the total heat flux into the total area governed by a node must be equal to zero. As the temperatures at the various nodes are not known, a system of equations must be established. These take the form:

$$[X] \cdot [T] + [S] = 0 \quad (130)$$

The following sections describe the heat transfer phenomena present in each of the boundary types. The formulation of the matrix is then discussed.

B.4.1 Conduction From A Continuous Solid

Conduction from a continuous solid requires the use of Newton's law of conduction,

$$q'' = k \frac{\delta T}{\delta x} \quad (131)$$

The finite difference equivalent for heat transfer across the green boundary line shown in the '4 Areas' section of figure 105 is:

$$q = k \cdot w \cdot t \frac{T_{(m,n+1)} - T_{(m,n)}}{d} \quad (132)$$

where k is the coefficient of conduction of the metal and t is the thickness of the cross-section. Analogous boundary equations exist for heat transfer across any boundary surface between two continuous solid elements.

B.4.2 Conduction From Contact Surface Resistance

When two surfaces are in contact, it is appropriate to use a contact resistance, which accounts for surface finish, material, contact pressure, and interspatial fluid. The contact resistance can be used to calculate the heat flow with **Equation 133**. **Table 27** shows the surface resistances of various contacts.

$$q'' = R_{contact} \Delta T, \quad (133)$$

Or for the green border shown in **Figure 105** with 2 elements:

$$q = w \cdot t \cdot R_{contact} (T_{(m,n)} - T_{contact}) \quad (134)$$

where $T_{contact}$ is the temperature of the material in contact.

Contact Material	Interfacial Fluid/Material	Interfacial Contact Pressure	Surface Roughness	Thermal Resistance $R(10^{-4}m^2 * K/W)$
Steel	Vacuum	$100kN/m^2$	Various	6 – 25
Steel	Vacuum	$10000kN/m^2$	Various	.7 – 4
Copper	Vacuum	$100kN/m^2$	Various	1 – 10
Aluminium	Vacuum	$100kN/m^2$	Various	1.5 – 5
Aluminium	Air	$100kN/m^2$	$10\mu m$	2.75
Aluminium	Hydrogen	$100kN/m^2$	$10\mu m$	.7
Aluminium	Glycerine	$100kN/m^2$	$10\mu m$	.265
Aluminium	indium foil filler	$100kN/m^2$	unknown	.07
Aluminium	lead coating	0	unknown	.01 – .1

Table 27: Contact Resistance of Various Surfaces, Incropera and Dewitt [54]

B.4.3 Forced Convection From a Fluid

If the boundary occurs at the interface between the metal and a convecting fluid, the application of Newton's law of cooling is appropriate

$$q'' = h \cdot (T_s - T_{inf}) \quad (135)$$

where T_{inf} is a nominal temperature of the fluid. In the case of a pipe, it is the mean temperate in the cross section. In an immersed body, it is the temperature away from the thermal boundary layer. For the finite difference analysis, it is calculated as

$$q = w \cdot t \cdot h(T_{(m,n)} - T_{inf}) \quad (136)$$

The coefficient of conduction, h , is calculated through empirical equations, which are non-dimensionalized through Reynolds, Nusselt and Prandtl numbers: Re , Nu and Pr .

$$Re = \frac{\rho \cdot u \cdot D}{\mu} \quad (137)$$

$$Nu = \frac{h \cdot D}{k_{gas}} \quad (138)$$

D is the hydraulic diameter, which, for a non circular section, is defined as four times

the cross-sectional area, divided by the wetted perimeter:

$$D = \frac{4A}{P_{wetted}} \quad (139)$$

The Prandtl number, Pr , is a non-dimensional, temperature-dependant material property.

Laminar Flow

If the condition of **Equation 140** is true, the flow is considered laminar. The coefficient of convection then depends on the profile of the flow.

$$Re_D = \frac{\rho \cdot u \cdot D}{\mu} < 2300 \quad (140)$$

If the condition of **Equation 141** is satisfied, then the flow is fully developed and the Nusselt number can be extracted from interpolating the results from **Table 28** [54].

$$\frac{Re \cdot Pr \cdot D^{.33333 \cdot \frac{D_{vs}}{D_s} \cdot 14}}{l} < 2 \quad (141)$$

Cross Section Type	Side Ratio length/width	Nusselt Number Uniform q_s''	Nusselt Number Uniform T_s
Circular	n/a	4.36	3.66
Square	1:1	3.61	2.98
Rectangular	1.43	3.73	3.08
Rectangular	2.0	4.12	3.39
Rectangular	3.0	4.79	3.96
Rectangular	4.0	5.33	4.44
Rectangular	8.0	6.49	5.60
Rectangular	∞	8.23	7.54

Table 28: Nusselt Numbers for Laminar Flows, Incropera and Dewitt [54]

If the condition of **Equation 141** is not satisfied, the flow is still in the entry region. The Nusselt number is then determined by **Equation 142**.

$$Nu = .86X \frac{Re_D * Pr^{.33333}}{L/D} \left(\frac{\mu_{mean}}{\mu_{surface}} \right)^{.14} \quad (142)$$

The factor, X , is used to extrapolate the Nusselt numbers for fully developed square profiles to flow in the entry region. This is shown in **Equation 143**

$$X = \frac{Nu_{fullydeveloped_{circularprofile}}}{Nu_{fullydeveloped_{actualprofile}}} \quad (143)$$

Pipe losses are calculated by obtaining the friction factor from the moody chart and the Reynolds number. The friction factor for laminar, fully developed flow can be expressed as:

$$f = \frac{Re_D}{64} \quad (144)$$

Equation 145 is then used to determine the pressure gradient.

$$\frac{dP}{dx} = f \frac{\rho * V^2}{2D} \quad (145)$$

However, this is only applicable to fully developed flow. No empirical relations were found for heat transfer in the entry region. It was decided to make the assumption that pressure gradient is proportional to heat transfer, as both are dependant upon the amount of disturbance in the flow. To calculate pipe loss in a fully developed region, the pipe loss was multiplied by the ratio:

$$\frac{Nu_{EntryRegion}}{Nu_{FullyDeveloped}} \quad (146)$$

This assumption is crude, but as the pressure loss for laminar flow in this case was not significant, it is considered to be acceptable.

Turbulent Flow

If the condition of **Equation 140** is not satisfied, the flow is considered turbulent or

transitional. The entry length of turbulent flow is typically short and lacks the empirical analysis tools that fully developed transitional flow has. It is conservative in terms of heat transfer to use fully developed flow equations for the entry region. The moody chart is required in order to obtain the friction factor, which is necessary for both heat transfer and pipe loss calculations. The moody chart can also be given as an analytical relationship for turbulent flow, shown in **Equation 147**, however it cannot be solved explicitly [55].

$$\frac{2.51}{Re\sqrt{f}} = \frac{-1}{\sqrt{f} + 2 * \log \frac{Roughness}{3.7 * D}}; \quad (147)$$

And the Nusslet number can be determined from the empirical relation:

$$Nu = \frac{\frac{f}{8} (Re_D - 1000) Pr}{1 + 12.7 \frac{f}{8} 5 \left(Pr^{\frac{2}{3}} - 1 \right)} \quad (148)$$

The convection coefficient can be determined from the Nusselt Number using **Equation 138**. The gradient of the pressure drop can be determined by **Equation 145** and the friction factor.

B.4.4 Symmetrical Boundary

In order to reduce the amount of computational resources required for solving the model, it is desirable to use axisymmetric boundary conditions. This is the easiest boundary condition to implement, because it assumes that the temperature gradient across the boundary condition is zero, which implies that no heat flows. This requires no changes to the model and the accuracy can be verified by examining the temperature profile at the boundary conditions and making sure that no gradient is found.

B.5 Matrix Formulation

Each node is given a distinct number $N_{(m,n)}$. There are N_{total} nodes. The system matrix, $[X]$, has dimensions $N_{total} \times N_{total}$. $[T]$, the temperature matrix, is a column vector of height N_{total} where the value of the N^{th} element corresponds to the temperature of node N. $[S]$ is another column matrix, which represents sources and sinks from the compressed gas and cooling air. The N^{th} row of X, along with the N^{th} value of S, constitutes the energy

balance equation for node N. The following example includes all 4 types of boundaries and illustrates the formulation of the matrix.

B.5.1 Example of Matrix Formulation

Figure 106 is an example that illustrates all of the types of boundary conditions. The equations for two nodes are formulated and implemented in the matrices. In order to solve any problem, the equations for all nodes must be determined.

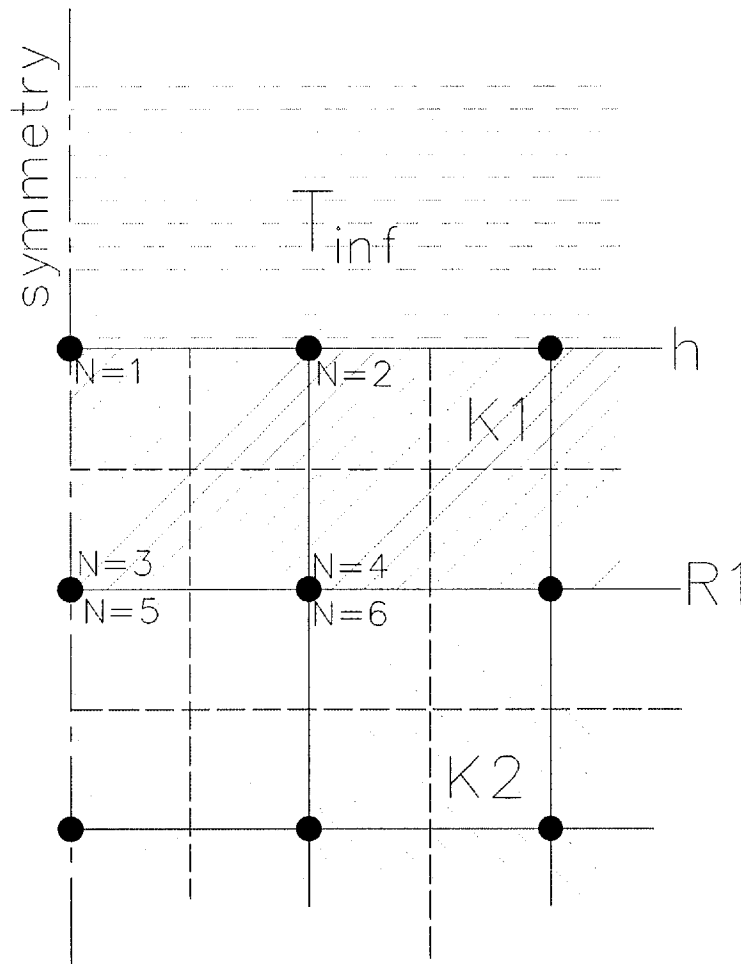


Figure 106: Intercooler Sample Analysis

Node 1 governs one area, which has four boundaries: 2 conduction, 1 convection, 1 symmetry. Two different materials are present with thermal conductivities k_1 and k_2 and a contact resistance R . The nodes form square areas for the sake of simplicity. One fluid

is present with convection coefficient, h . Nodes 3 and 4 belong to material 1, Nodes 5 and 6 belong to material 2. Nodes 3 and 5 are coincident, as are nodes 4 and 6. The section has thickness, t . The equation of thermal equilibrium for Node 1 is:

$$(T_2 - T_1) \frac{k_1 * w * t}{L} + (T_3 - T_1) \frac{k_1 * w * t}{L} + (T_{inf} - T_1) * h * w * t = 0 \quad (149)$$

which simplifies to:

$$\left(-2 \frac{k_1 * w * t}{L} - h * w * t \right) T_1 + \left(\frac{k_1 * w * t}{L} \right) T_2 + \left(\frac{k_1 * w * t}{L} \right) T_3 + (h * w * t) T_{inf} = 0 \quad (150)$$

This corresponds to setting row 1 of X equal to:

$$\left[\left(-2 \frac{k_1 * w * t}{L} - h * w * t \right) \quad \left(\frac{k_1 * w * t}{L} \right) \quad \left(\frac{k_1 * w * t}{L} \right) \quad 0 \quad 0 \quad \dots \quad 0 \quad 0 \right] \quad (151)$$

and row 1 of S equal to $h * w * t * T_{inf}$

For later evaluation of the energy transfer in and out of the metal it is useful to have a matrix [Z], which contains any coefficients related to convection, as convection is the only boundary of the conductive matrix that allows heat to pass through. Row 1 of Z is equal to $-h * w * t$.

The equation of thermal equilibrium for Node 3 is:

$$(T_1 - T_3) \frac{k_1 * w * t}{L} + (T_4 - T_3) \frac{k_1 * w * t}{L} + (T_5 - T_3) * R * w * t = 0 \quad (152)$$

which simplifies to:

$$\left(\frac{k1 * w * t}{L}\right) T1 + \left(-2\frac{k1 * w * t}{L} - R * w * t\right) T3 + \left(\frac{k1 * w * t}{L}\right) T4 + (R * w * t) T5 \quad (153)$$

This corresponds to setting row 3 of X equal to:

$$\left[\left(\frac{k1 * w * t}{L}\right) \quad 0 \quad \left(-2\frac{k1 * w * t}{L} - R * w * t\right) \quad \left(\frac{k1 * w * t}{L}\right) \quad (R * w * t) \quad \dots \quad 0 \quad 0 \right] \quad (154)$$

and row 3 of S equal to zero.

The matrices X and S are formed in this way. This is done automatically in the Matlab program based on user-defined geometric parameters. The temperature matrix T can then be calculated though matrix inversion and multiplication.

$$[T] = [X]^{-1} \cdot -[S] \quad (155)$$

The heat flow into each element can then be calculated:

$$Q = Z * A * T + S \quad (156)$$

A is a row vector with N_{total} columns, each equal to 1.

Each element is bordered by at most 1 fluid, so the heat transfer to the appropriate fluid can be determined. This will determine the temperature of the fluid for the next cross-section. The fluid temperatures are iterated to convergence.

B.6 Intercooler Results

A problem-specific meshing program was written in Matlab, which can be adapted to any number of concentric finned annuli. Iterations of intercooler design led to the schematic

shown in **Figure 107**. An integrated intercooler is utilized after the 2nd stage. The other stages are not cooled. It was believed that this gives the best combination of compactness, minimization of peak temperature and efficiency. A sample cross-section is shown in **Figure 108**. The intercooler cools the gas from the second stage outlet temperature of 459 K down to 317 K, giving an intercooler effectiveness of 85%.

These techniques and programs could be used to design an intercooler for total machine cooling, or to evaluate the heat transfer within the experimental test rig casings.

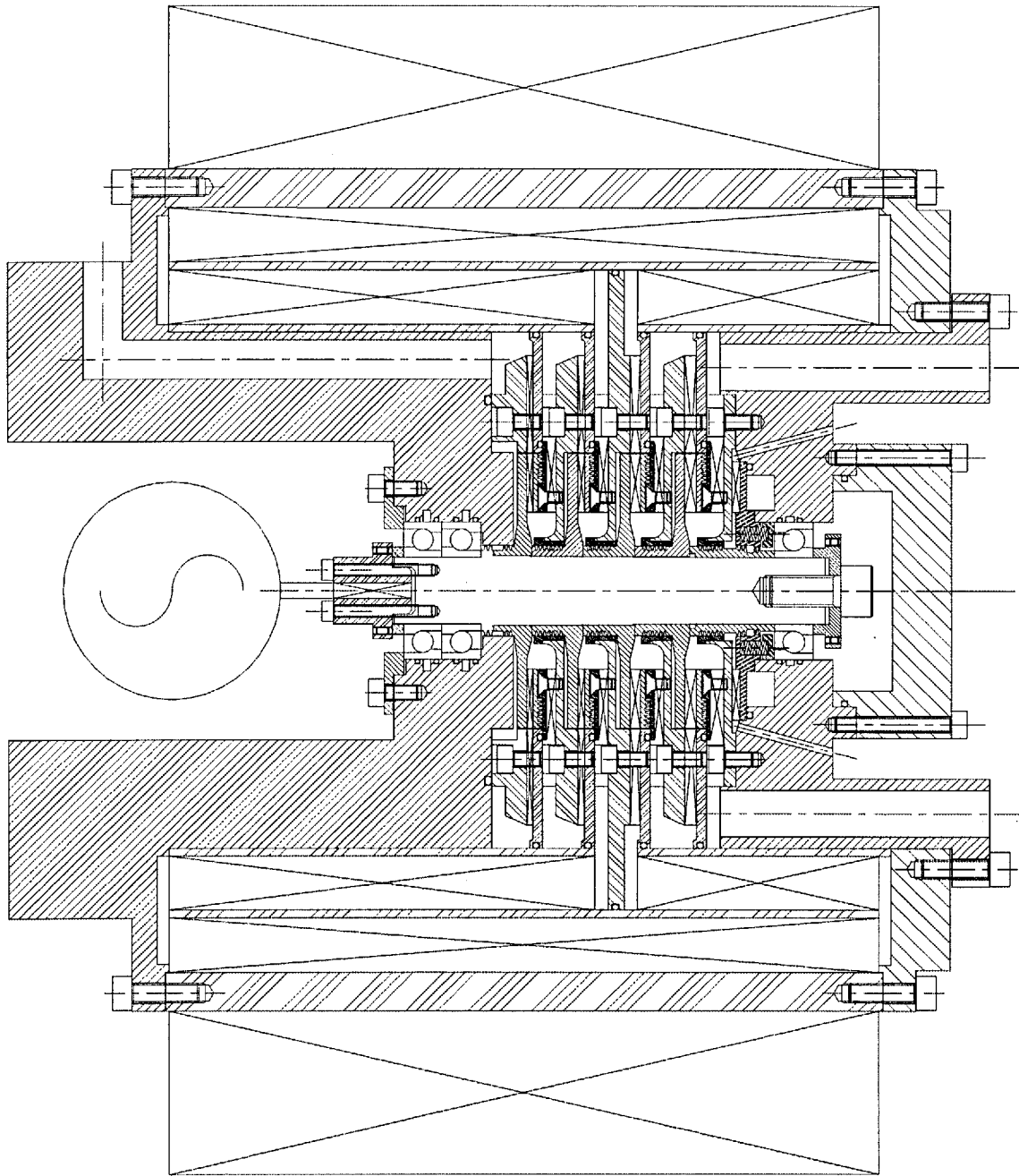


Figure 107: Rig Schematic with Integrated Intercooler

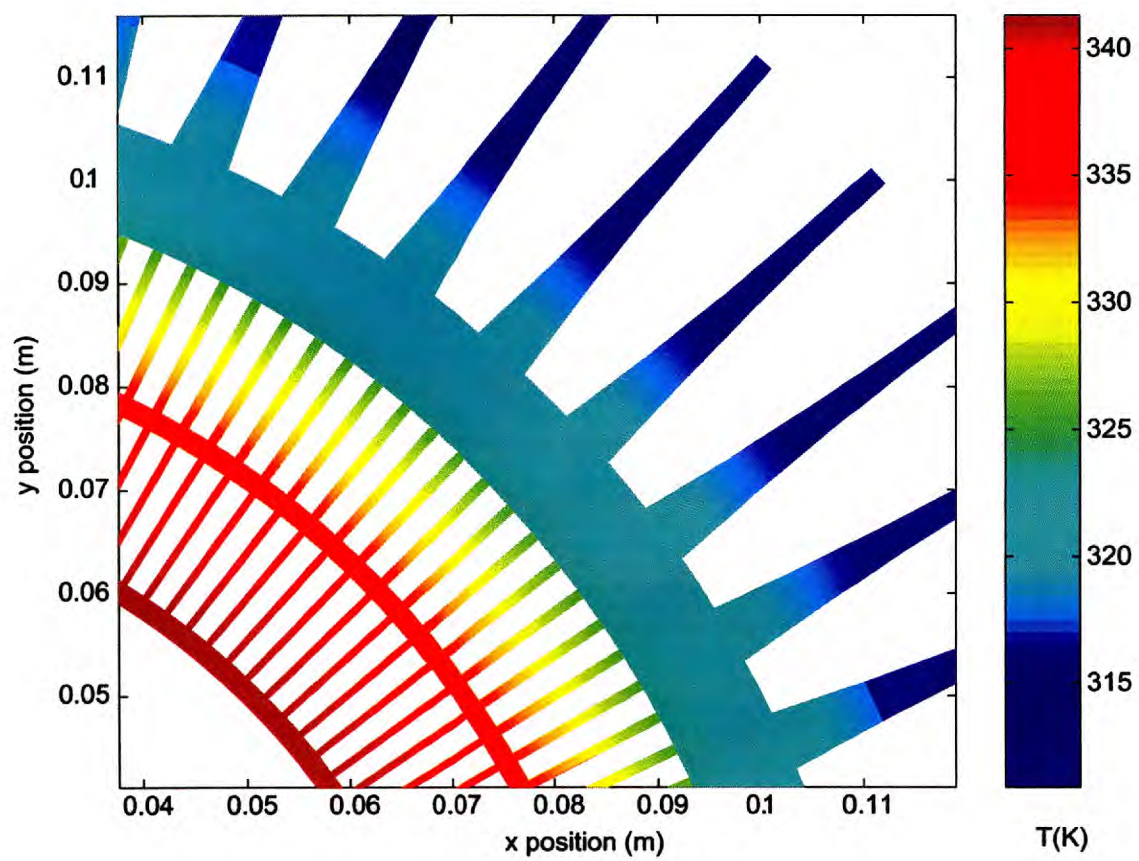


Figure 108: Sample Cross-Section of Intercooler

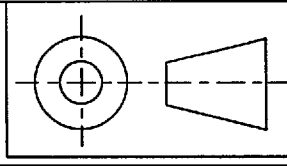
C Detailed Drawings

Detailed drawings used for component manufacture are shown on the following pages.

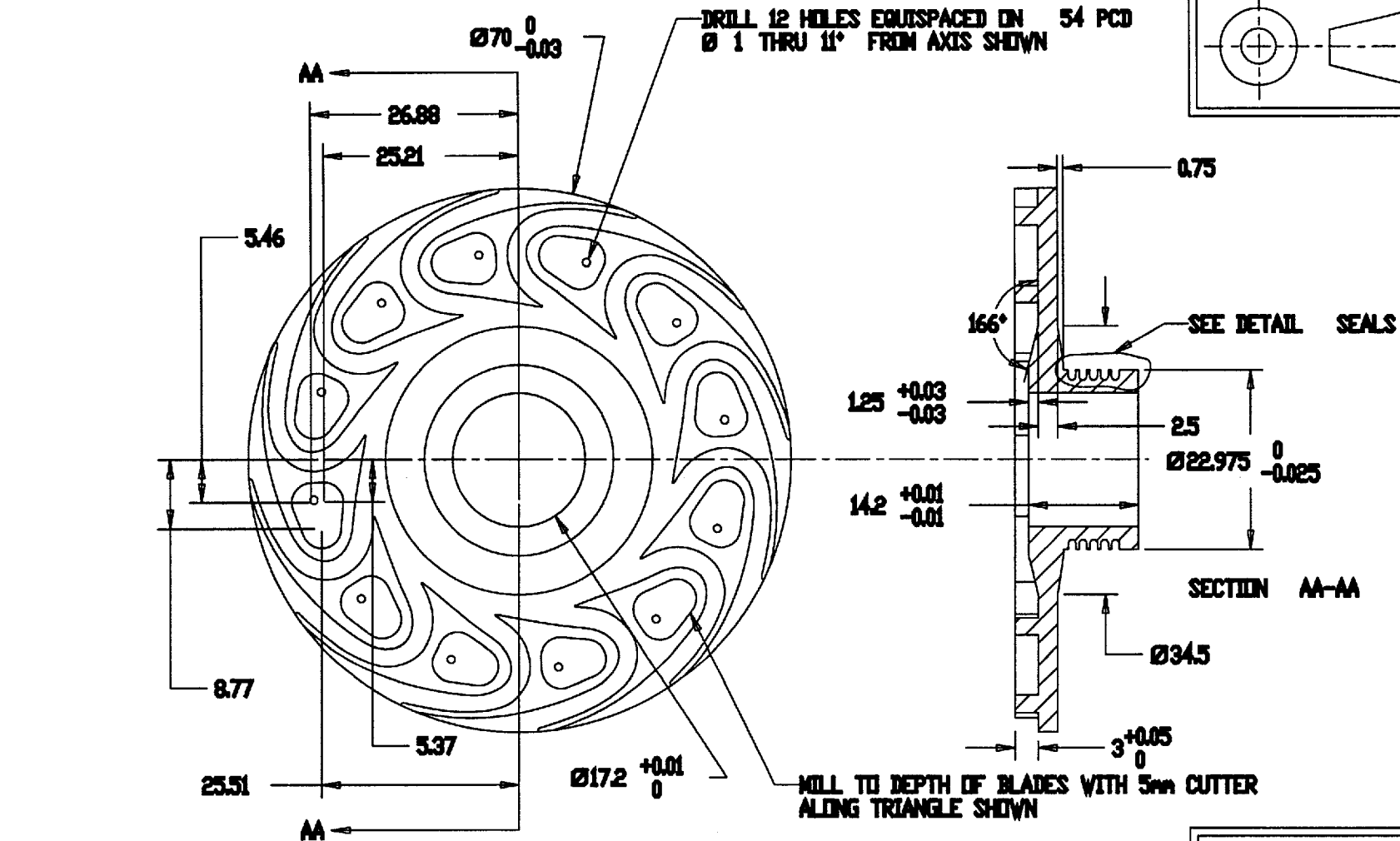
Table 29 lists the components and their drawing numbers.

Component	Drw. No.	Component	Drw. No.
Shaft	1.1	IGV 1	3.1.1
Rotor 1	1.2.1	IGV 2	3.1.2
Rotor 2	1.2.2	IGV 3	3.1.3
Rotor 3	1.2.3	IGV 4	3.1.4
Rotor 4	1.2.4	Stator 1	3.2.1
Shroud 1	1.3.1	Stator 2	3.2.2
Shroud 2	1.3.2	Stator 3	3.2.3
Shroud 3	1.3.3	Stator 4	3.2.4
Shroud 4	1.3.4	Stator Block	3.3
Seal Array	1.4	Abradable Rotor Seal	3.4
Square Drive	1.5	Abradable Shaft Seal	3.5
Rotor Clamp	1.6	Preload Housing	4.1
Race Washer	1.7	Spring Race	4.2
Main Housing	2.1	Spring Stop	4.3
Outer Race clamp	2.2	Oil Cap	4.4
Oil Insulator	2.3	Inner Volute	5.1
Mounting Flange	2.4	Outer Volute	5.2
Square Seal	2.5		

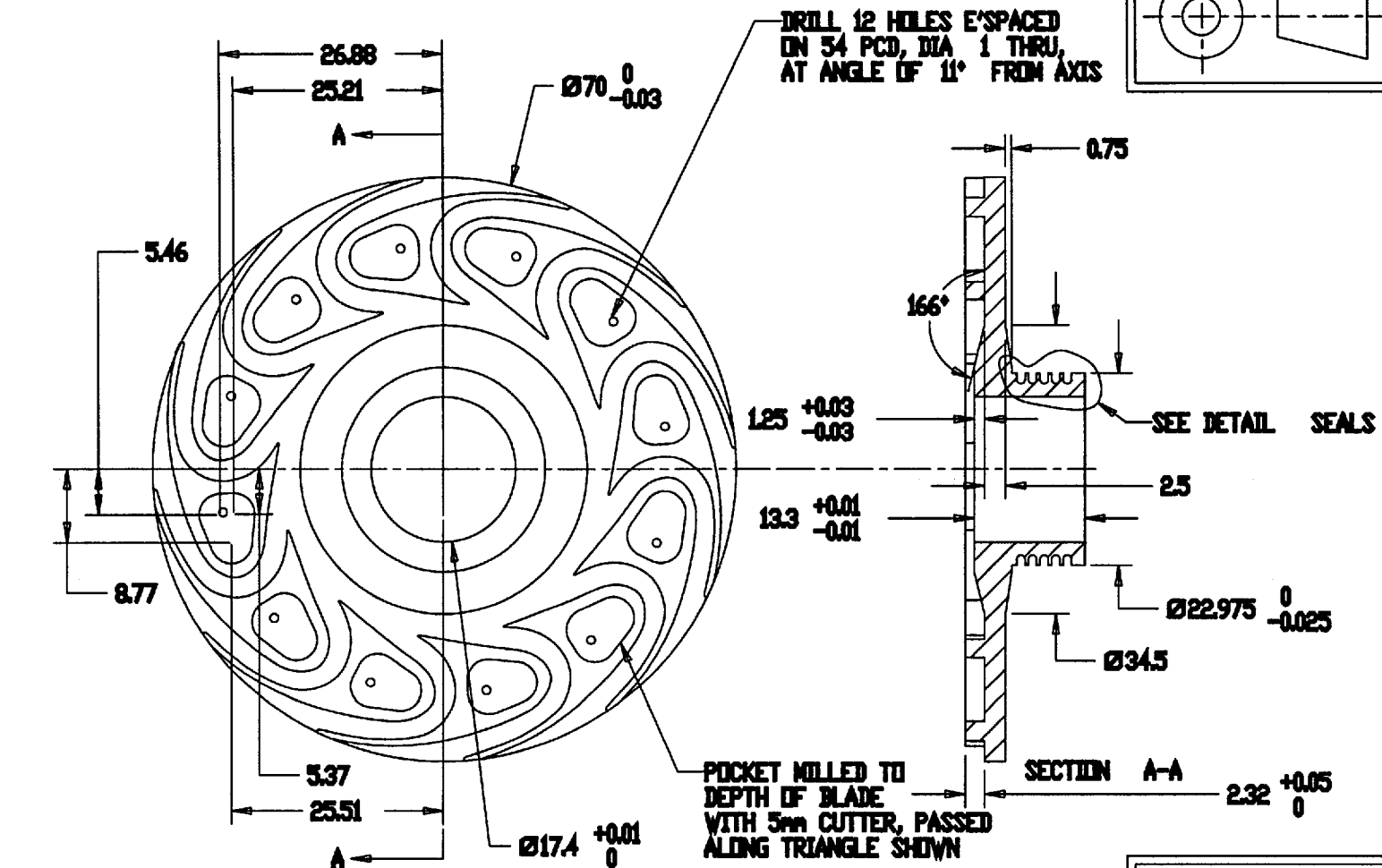
Table 29: List of Components



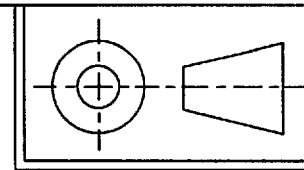
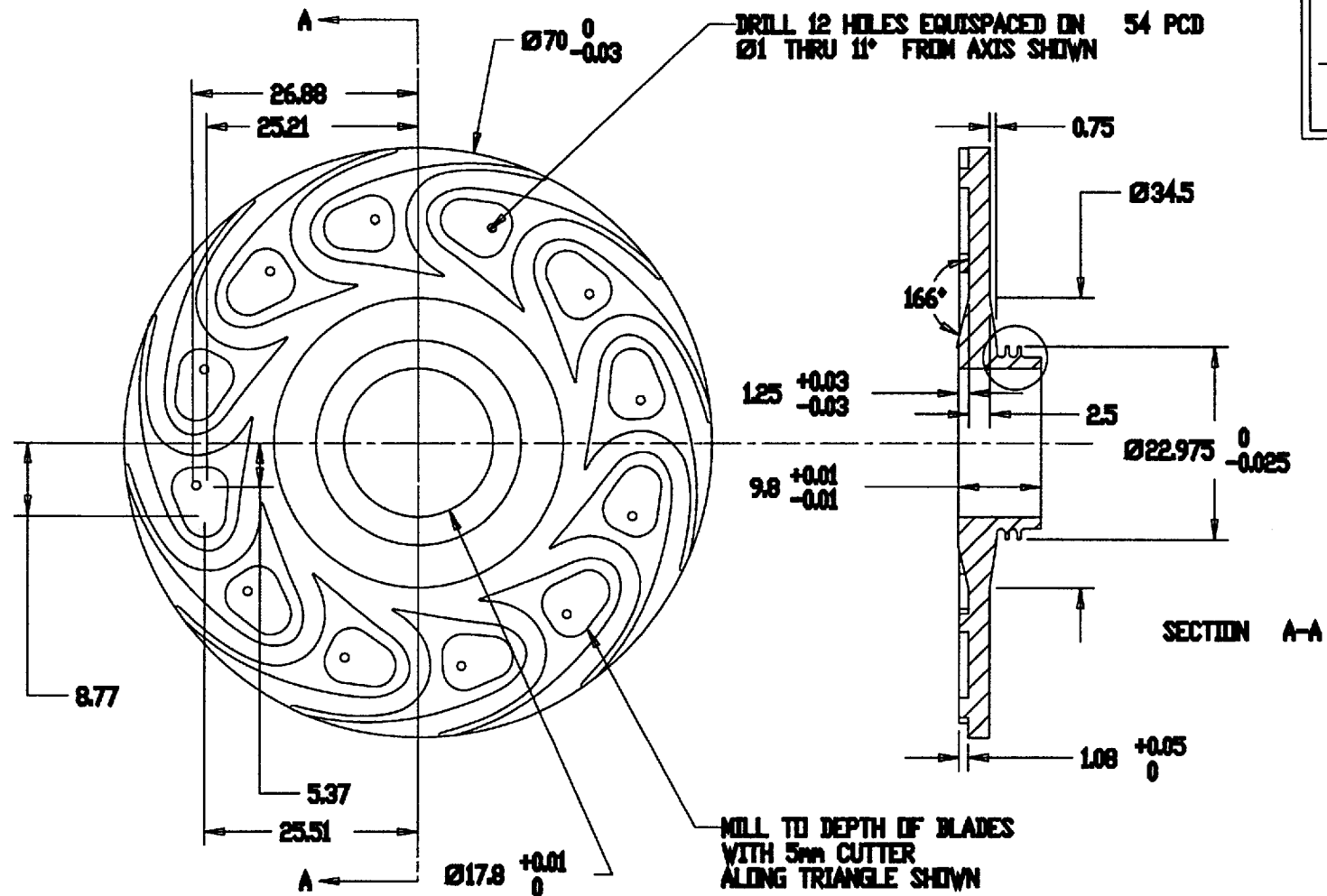
017.8 ±0.01 0.2mm FOR GRINDING		No. Off: 2 (1 spare)
Part Name: Shaft		Sheet: 1/1
Part No: IC-GCMS-1.1		Version: 1
Material: EN 16T		Finish: Ground
General Tolerance: +/- 0.1mm		Edge Chamfers: 0.2 mm
All dimensions in mm		Corner Chamfers: 0.1mm
DRN by W E Thornton (w.thornton @cacuk)		Date:
CHK by:		Date:



Part Name: ROTOR 1	No. OF 1
Part No: IC-GCMS-121	Sheet 1/1
Material: Stainless Steel (410 S21)	Version 1
General Tolerance: +/- 0.1mm	Finish
All dimensions in mm	Edge Chamfers: 0.2 mm
DRN by W E Thornton (w Thornton @ cacuk)	Corner Chamfers: 0.1 mm
CHK by:	Date:
	Date:



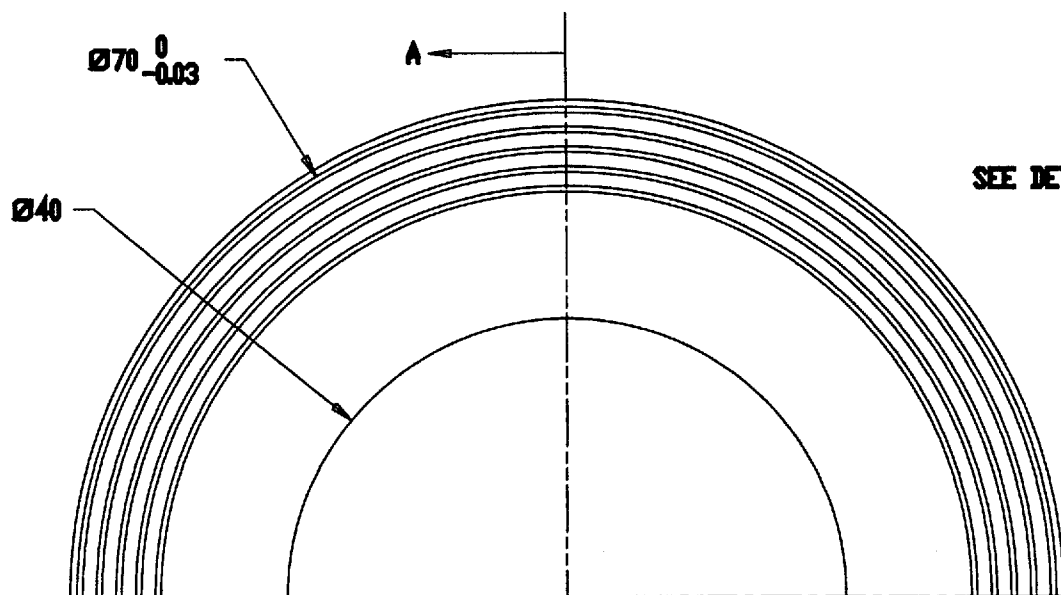
Part Name: ROTOR 2	No. DPA: 1
Part No: IC-GCMS-12.2	Sheet: 1/1
Material: STAINLESS STEEL (410 S21)	Version: 1
General Tolerance: +/- 0.1mm	Finish:
All dimensions in mm	Edge Chamfers: 0.2 mm
DRN by V E Thornton (v.thornton@icacul)	Corner Chamfers: 0.1mm
CHK by:	Date:
	Date:



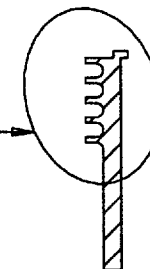
SECTION A-A

DETAIL SCALE 8.000

No. DFP 1	
Part Name: ROTOR 4	Sheet: 1/1
Part No: IC-GCMS-124	Version: 1
Material: STAINLESS STEEL (410 S21)	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@scacuk)	Date:
CHK by:	Date:

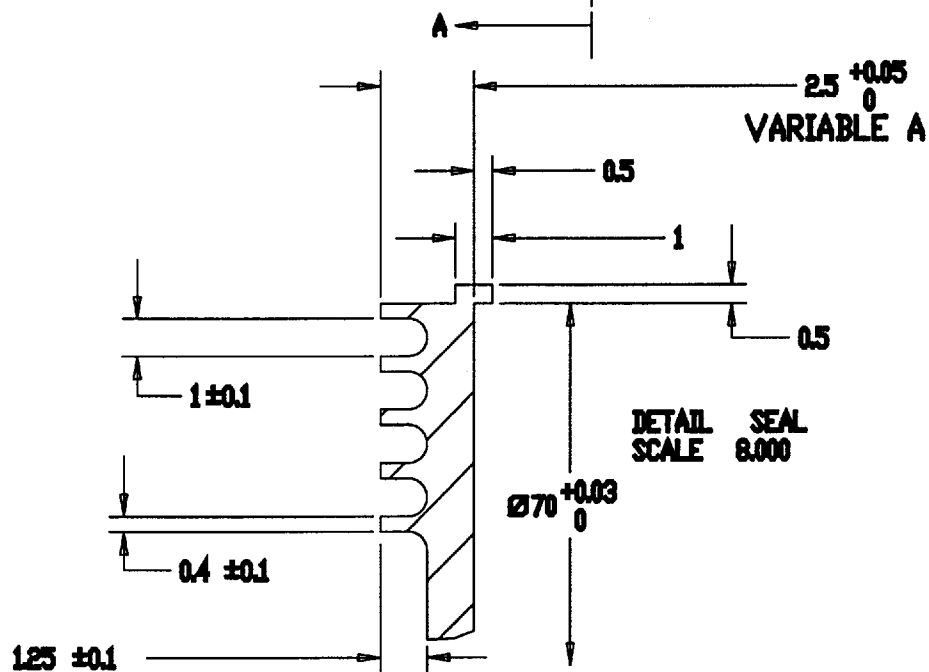


SEE DETAIL SEAL



SECTION A-A

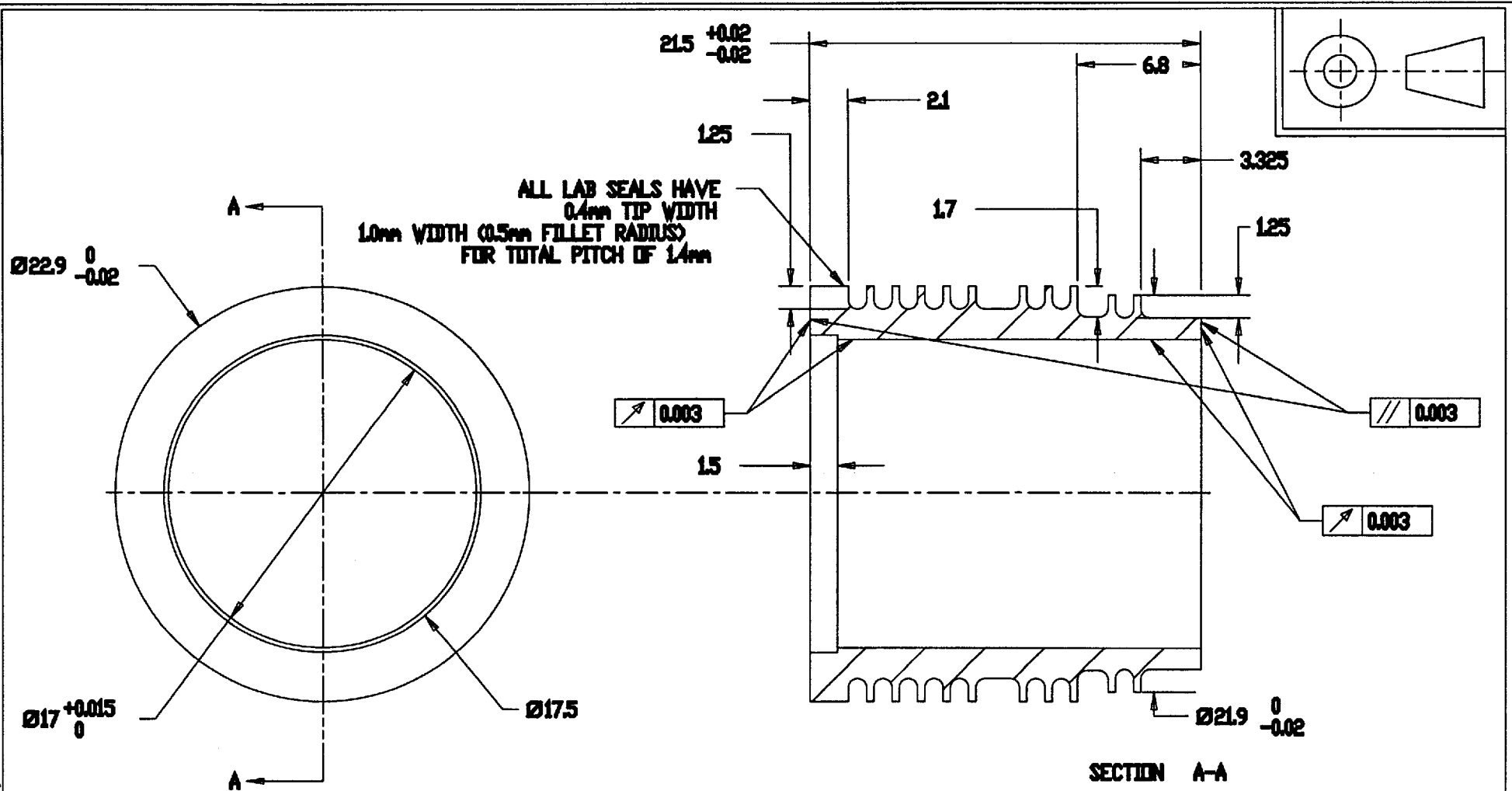
VARIABLE A	
IC-GCMS-1.31	: 2.50
IC-GCMS-1.32	: 2.88
IC-GCMS-1.33	: 2.92
IC-GCMS-1.34	: 2.92



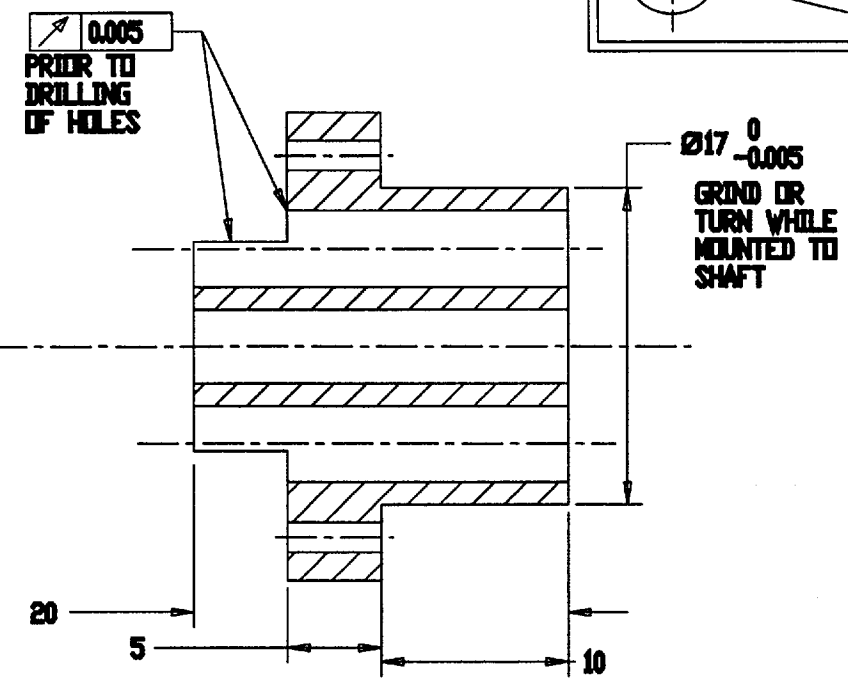
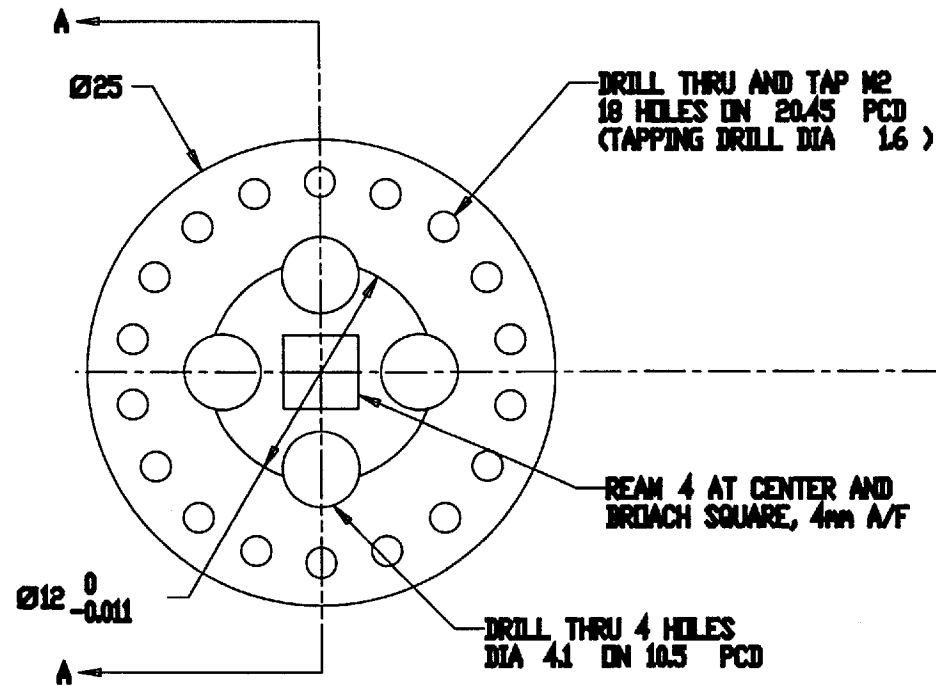
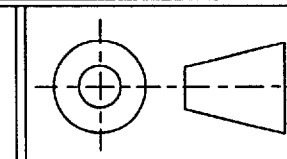
TOTAL THICKNESS MAY VARY BETWEEN PARTS

HALF SECTION SHOWN
PART IS SYMMETRIC

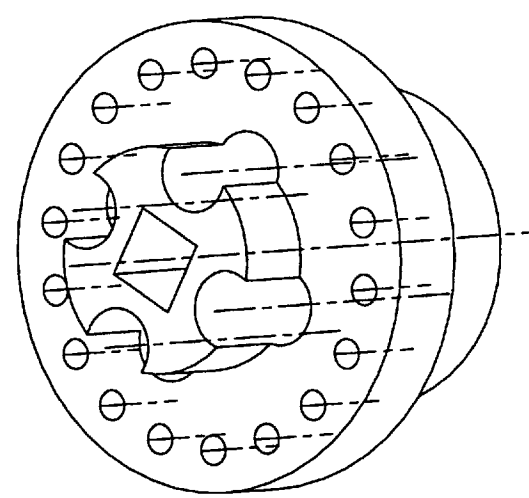
Part Name: Shroud		No. DfP: 1+1+1+1
Part No: IC-GCMS-1.3X		Sheet: 1/1
Material: Stainless Steel (410 S21)		Version: 0
General Tolerance: +/- 0.1mm		Finish:
All dimensions in mm		Edge Chamfers: 0.2 mm
DRN by V E Thornton (v.thornton@cacul)		Corner Chamfers: 0.1mm
CHK by:		Date:
		Date:



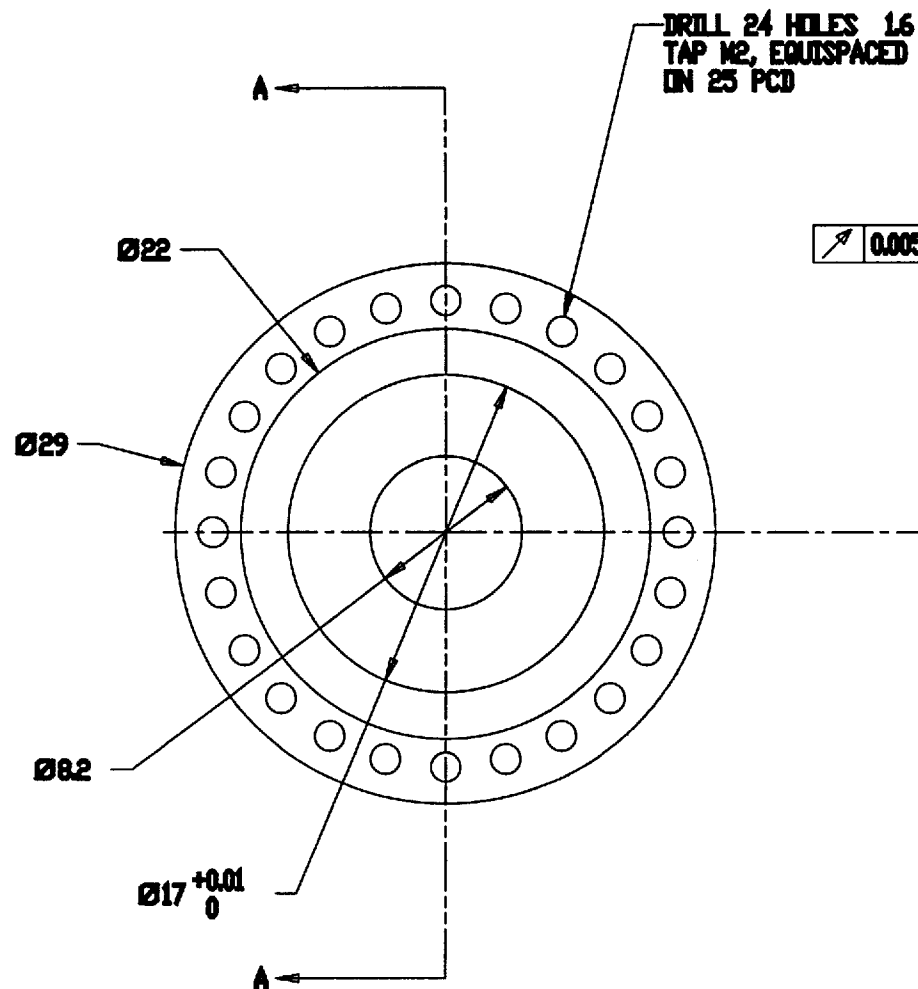
No. OFF: 2 (1 Spare)	
Part Name: Seal Array	Sheet: 1/1
Part No: IC-GCMS-14	Version: 1
Material: EN 16T	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@icacuk)	Date:
CHK by:	Date:



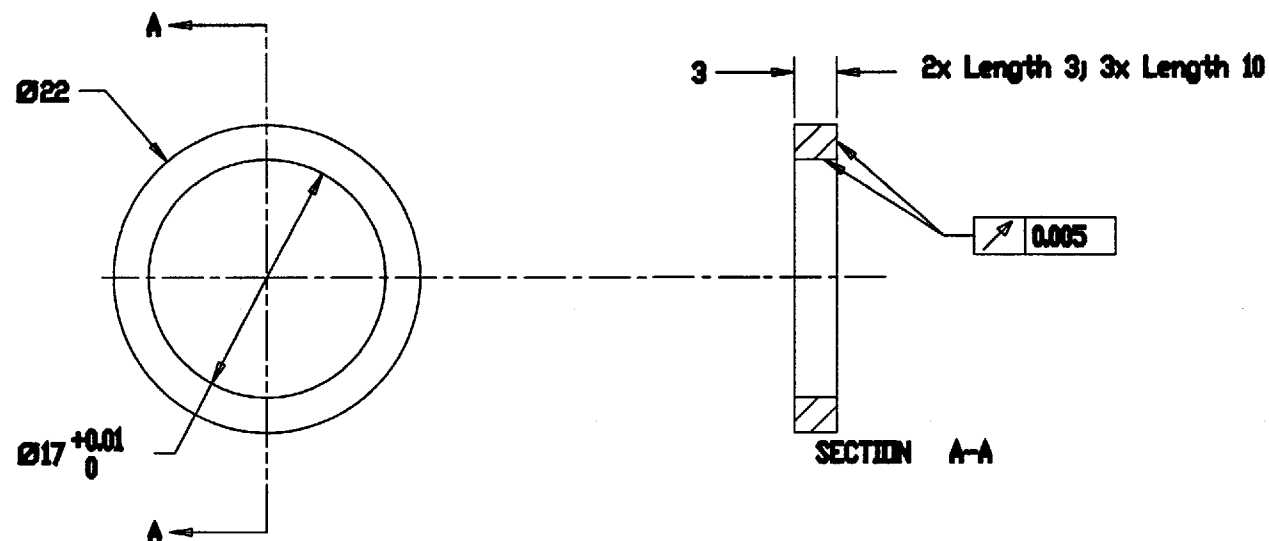
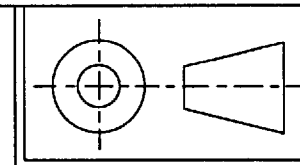
SECTION A-A



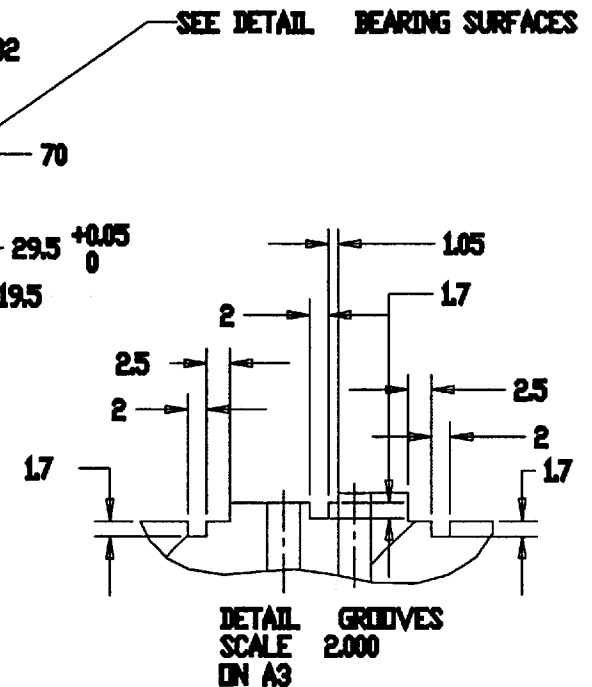
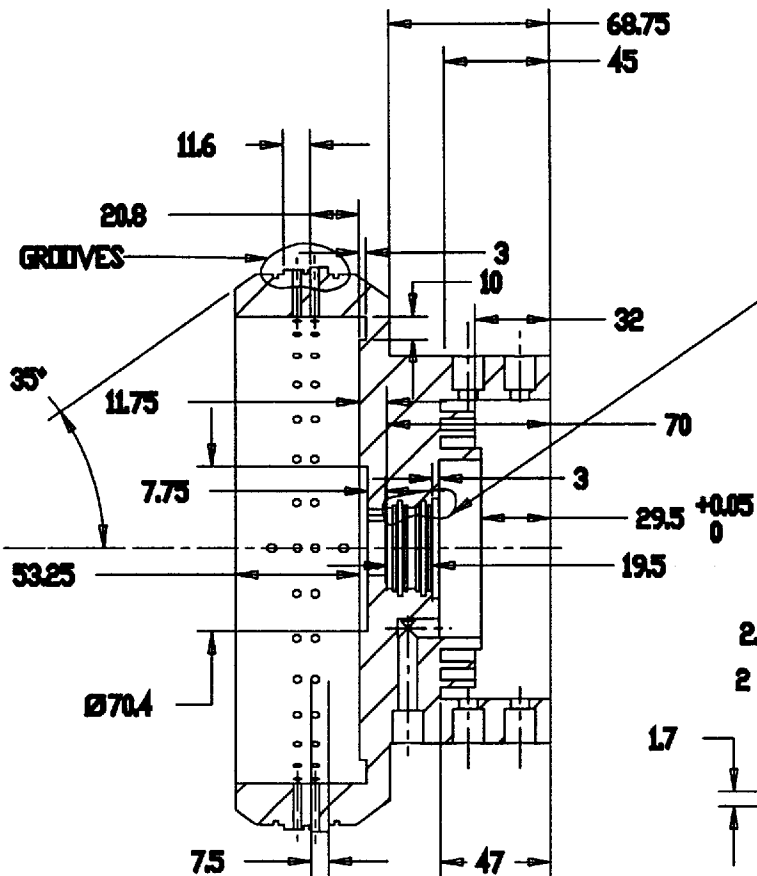
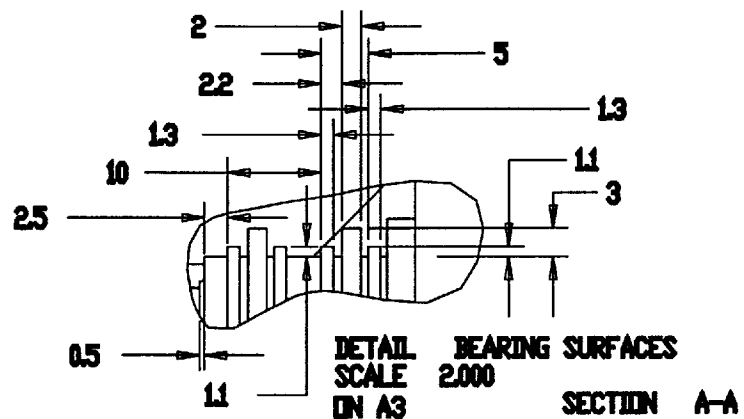
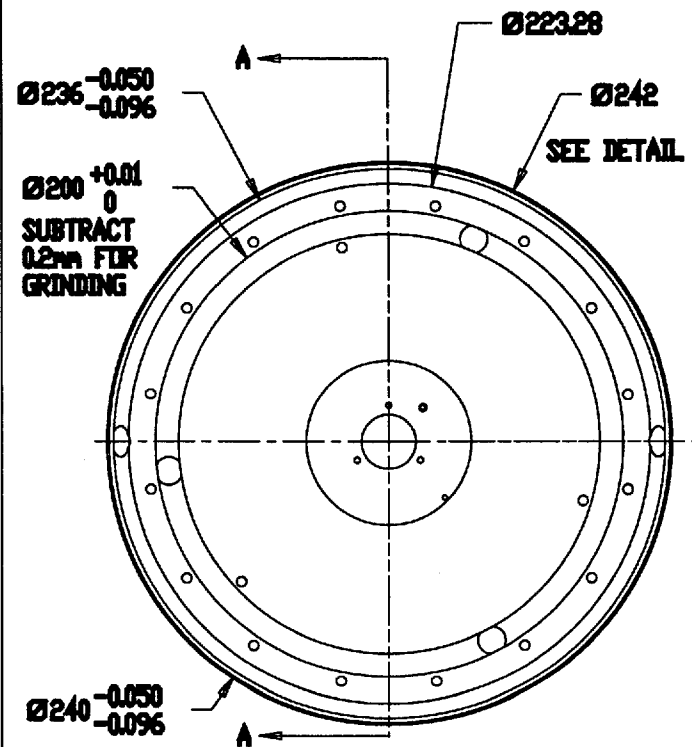
No. Diff 2 (1 spare)	
Part Name: Square Drive	Sheet: 1/1
Part No: IC-GCMS-15	Version: 1
Material: EN 40B Nitrided	Finish: NITRIDED
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@icacuk)	Date:
CHK by:	Date:



No. DFP 1	
Part Name: Rotor Clamp	Sheet: 1/1
Part No: IC-GCMS-16	Version: 1
Material: EN 16T	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@cacuk.co.uk)	Date:
CHK by:	Date:

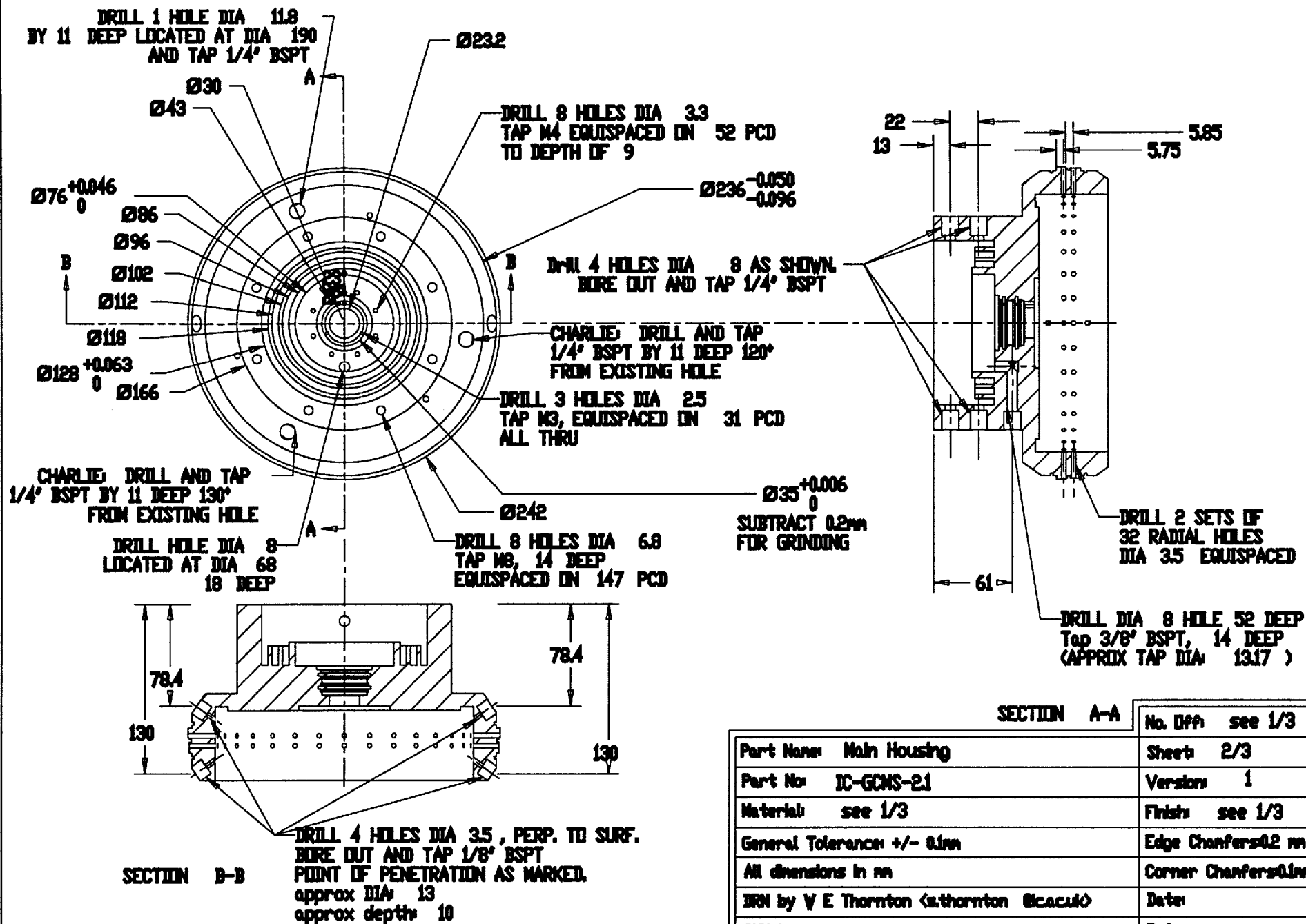


No. Dfth 3+2	
Part Name: Race Washer	Sheet: 1/1
Part No: IC-GCMS-1.7	Version: 1
Material: EN 16T	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@icacul.co)	Date:
CHK by:	Date:



No. OFA	1
Sheet	1/3
Version	1
Finish	Bearing Surfaces Ground Last
Edge Chamfers	0.2 mm
Corner Chamfers	0.1 mm
Date:	
Date:	

Part Name:	Main Housing
Part No:	IC-GCMS-2.1
Material:	EN 24T
General Tolerance:	± 0.1 mm
All dimensions in mm	
DRN by	V E Thornton (v.thornton@icacuk)
CHK by:	



SECTION A-A		No. DFN	see 1/3
Part Name:	Main Housing	Sheet:	2/3
Part No:	IC-GCMS-2.1	Version:	1
Material:	see 1/3	Finish:	see 1/3
General Tolerance:	+/- 0.1mm	Edge Chamfers:	0.2 mm
All dimensions in mm		Corner Chamfers:	0.1mm
DRN by	V E Thornton (s.thornton @cacul)	Date:	
CHK by:		Date:	

DRILL HOLE DIA 4, 62.5 DEEP
COUNTER-BORE DIA 8, 47 DEEP
TAP 1/8" BSPT

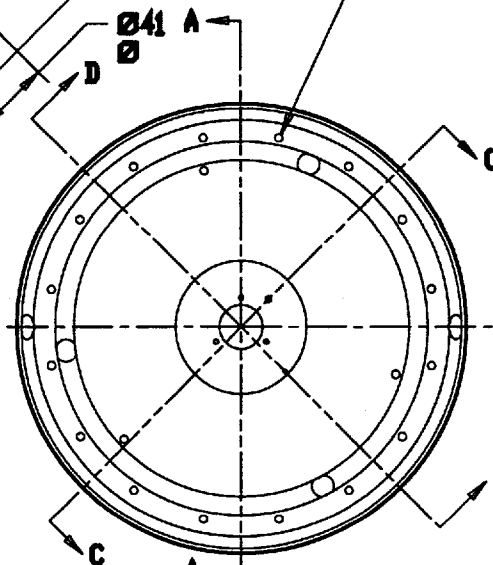
DRILL HOLE DIA 2
25.4 DEEP
COUNTER-BORE DIA 3.3
6 DEEP, TAP M4

SECTION C-C

DRILL 16 HOLES DIA .42
EQUISPACED ON 208 PCD
TAP M4, 13 DEEP

GEOMETRIC TOLERANCES

DRILL HOLE DIA 2, 75.78" to surface
COUNTER-BORE DIA 5, 51 DEEP
COUNTER-BORE DIA 8.75 21 DEEP
TAP 1/8" BSPT, 19 DEEP
DEPTHS MEASURED FROM SURFACE
AND ALONG CENTERLINE
(Tap approx DIA 9.72)



0.004

0.0006
0.0006

0.0012

0.005

0.003
0.003

0.008

SECTION A-A

DRILL HOLE DIA 2 TO .3mm FROM SURFACE
COUNTER-BORE DIA 8.75, 50 DEEP
TAP 1/8" BSPT

DO NOT BREAK THRU
LEAVE AT LEAST .3 mm OF MATERIAL
DRILL HOLE PRIOR TO GRINDING

DRILL HOLE DIA 2
37" FROM CENTERLINE,
LOCATED AS SHOWN

DRILL HOLE,
DIA 8.75
52 DEEP
TAP 1/8" BSPT

DRILL HOLE, DIA 2
THRU TO RADIAL HOLE

SECTION D-D

No. Off. see 1/3

Part Name: Main Housing

Sheet: 3/3

Part No. see 1/3

Version: 1

Material: see 1/3

Finish: see 1/3

General Tolerance: +/- 0.1mm

Edge Chamfers: 0.2 mm

All dimensions in mm

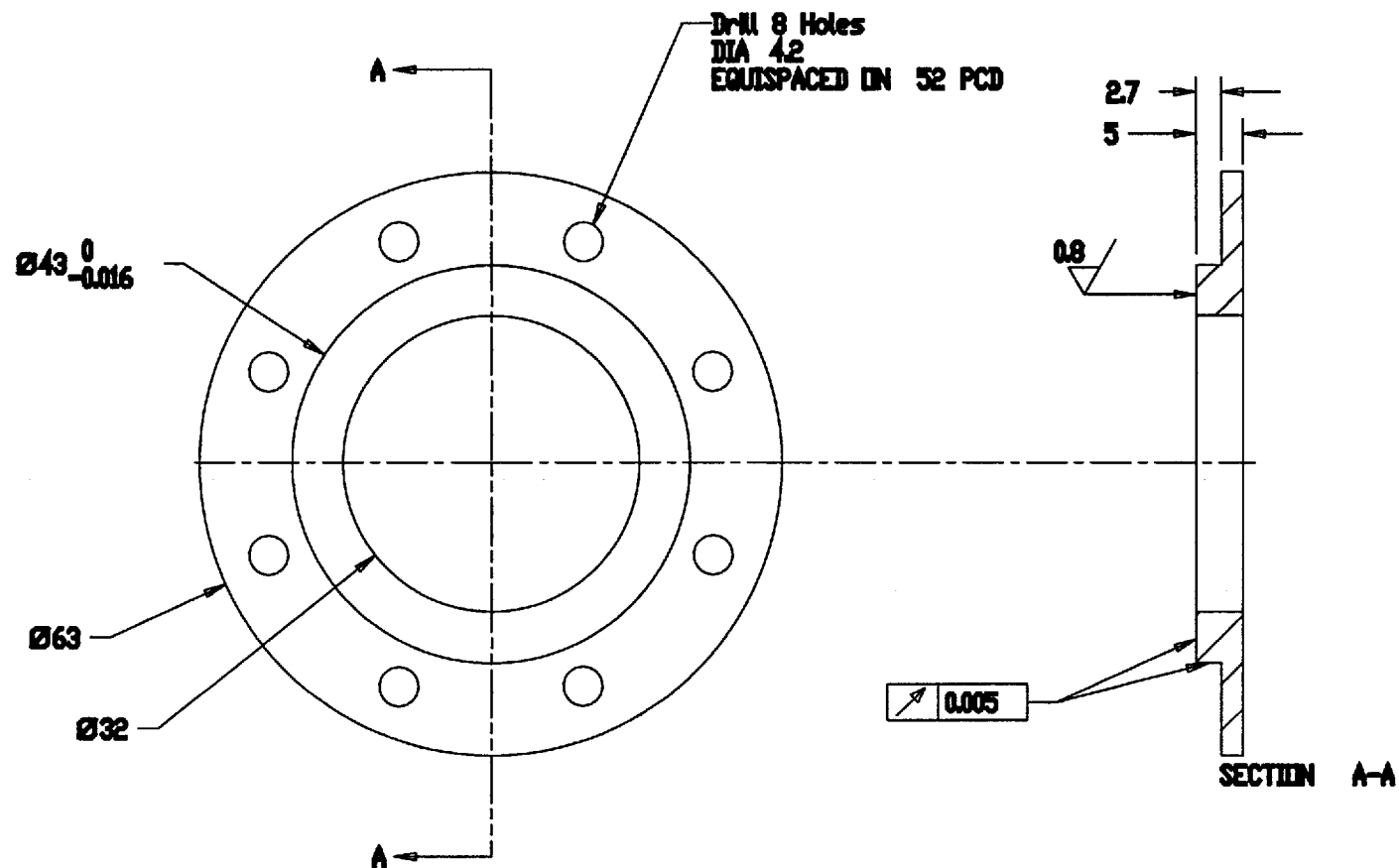
Corner Chamfers: 0.1mm

DRN by V E Thornton (vthornton@cacul)

Date:

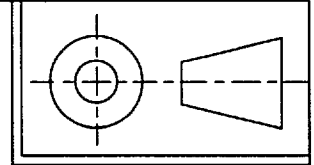
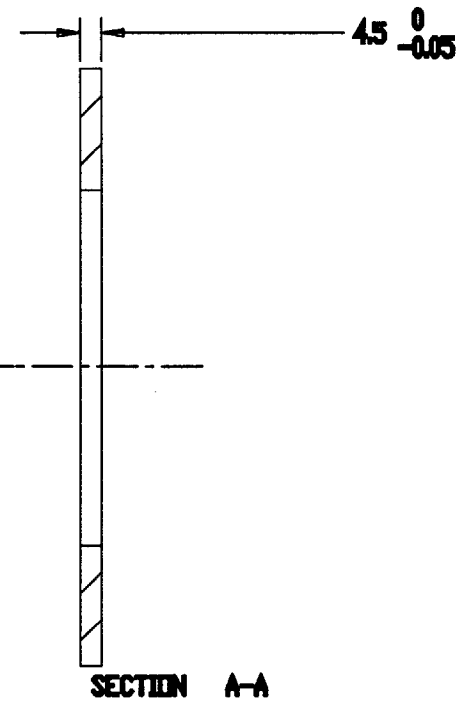
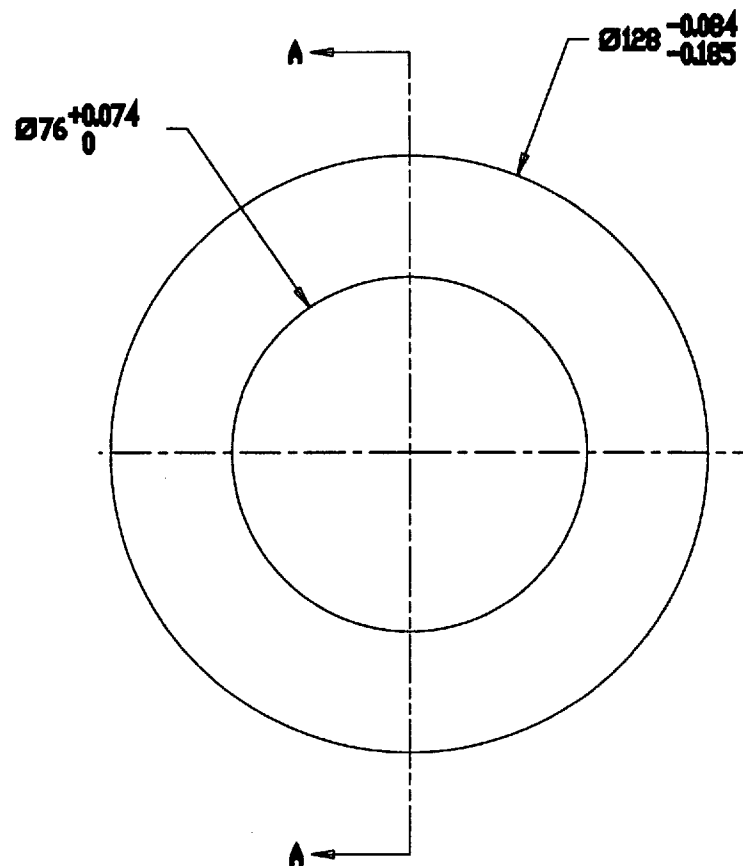
CHK by:

Date:

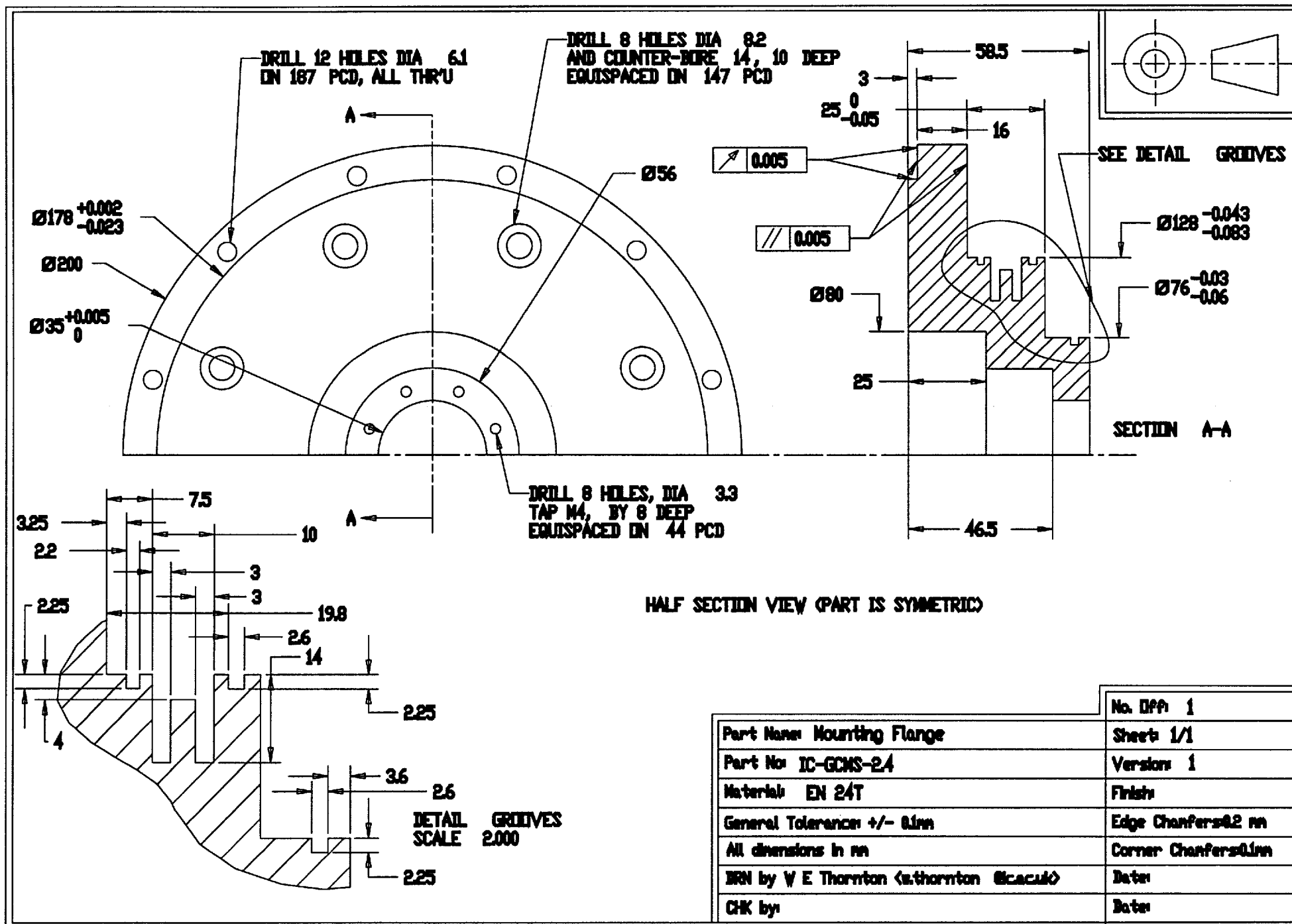


No. Off: 1
Sheet: 1/1
Version: 1
Finish:
Edge Chamfers: 0.2 mm
Corner Chamfers: 0.1 mm
Date:
Date:

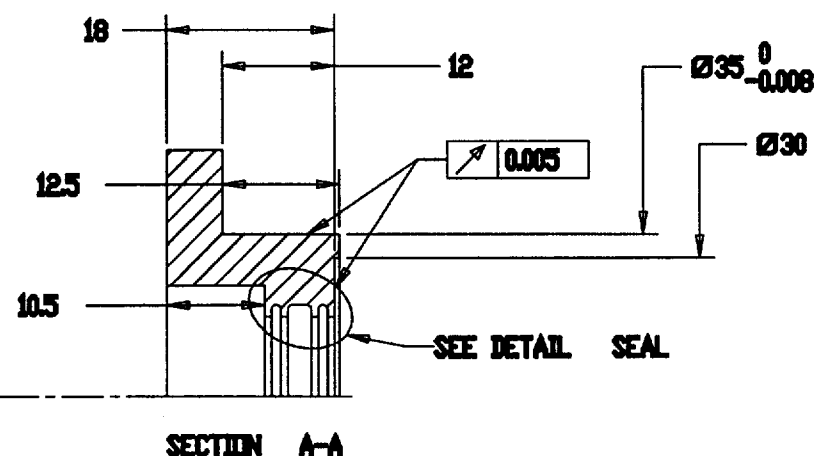
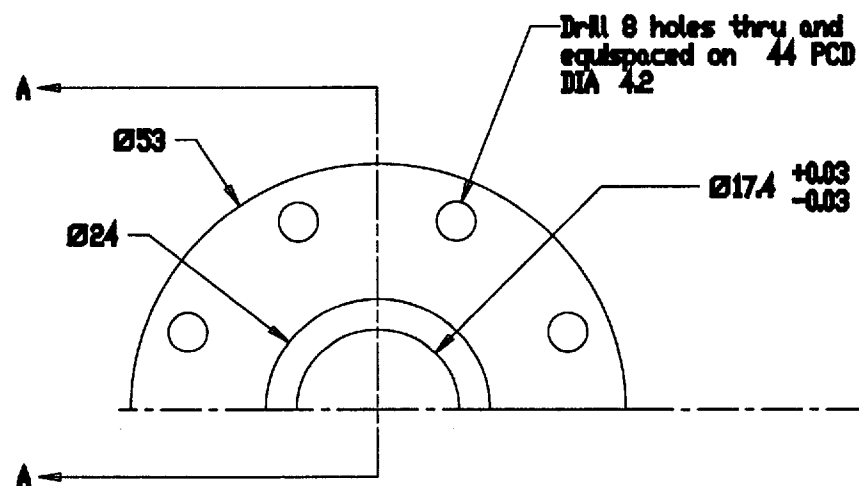
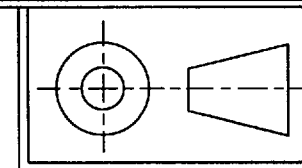
Part Name: Outer Race Clamp
Part No: IC-GCMS-2.2
Material: EN 16T
General Tolerance: +/- 0.1mm
All dimensions in mm
DRN by V E Thornton (v.thornton@cacuk)
CHK by:



No. Dpp: 1	
Part Name: Oil Insulator	Sheet: 1/1
Part No: IC-GCMS-23	Version: 1
Material: Tufnol	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (s.thornton@icacul.co)	Date:
CHK by:	Date:



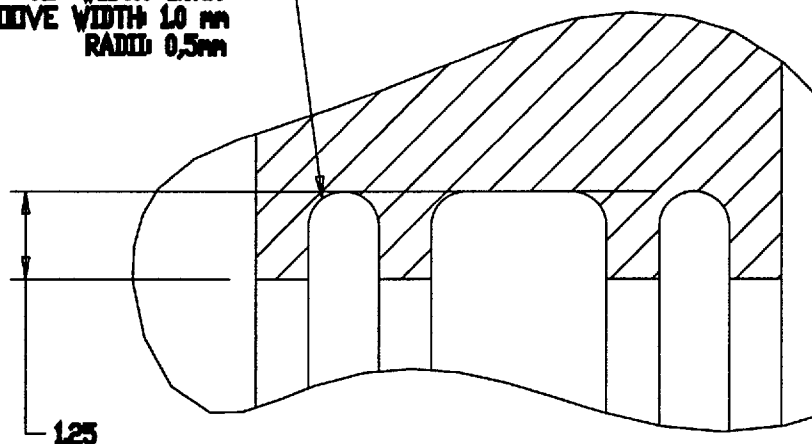
No. DFP 1
Part Name: Mounting Flange
Part No: IC-GCMS-24
Material: EN 24T
General Tolerance: $\pm 0.1mm$
All dimensions in mm
DRN by W E Thornton (w.thornton@caculo)
CHK by:
Sheet 1/1
Version 1
Finish:
Edge Chamfers: 0.2 mm
Corner Chamfers: 0.1mm
Date:
Date:



SECTION A-A

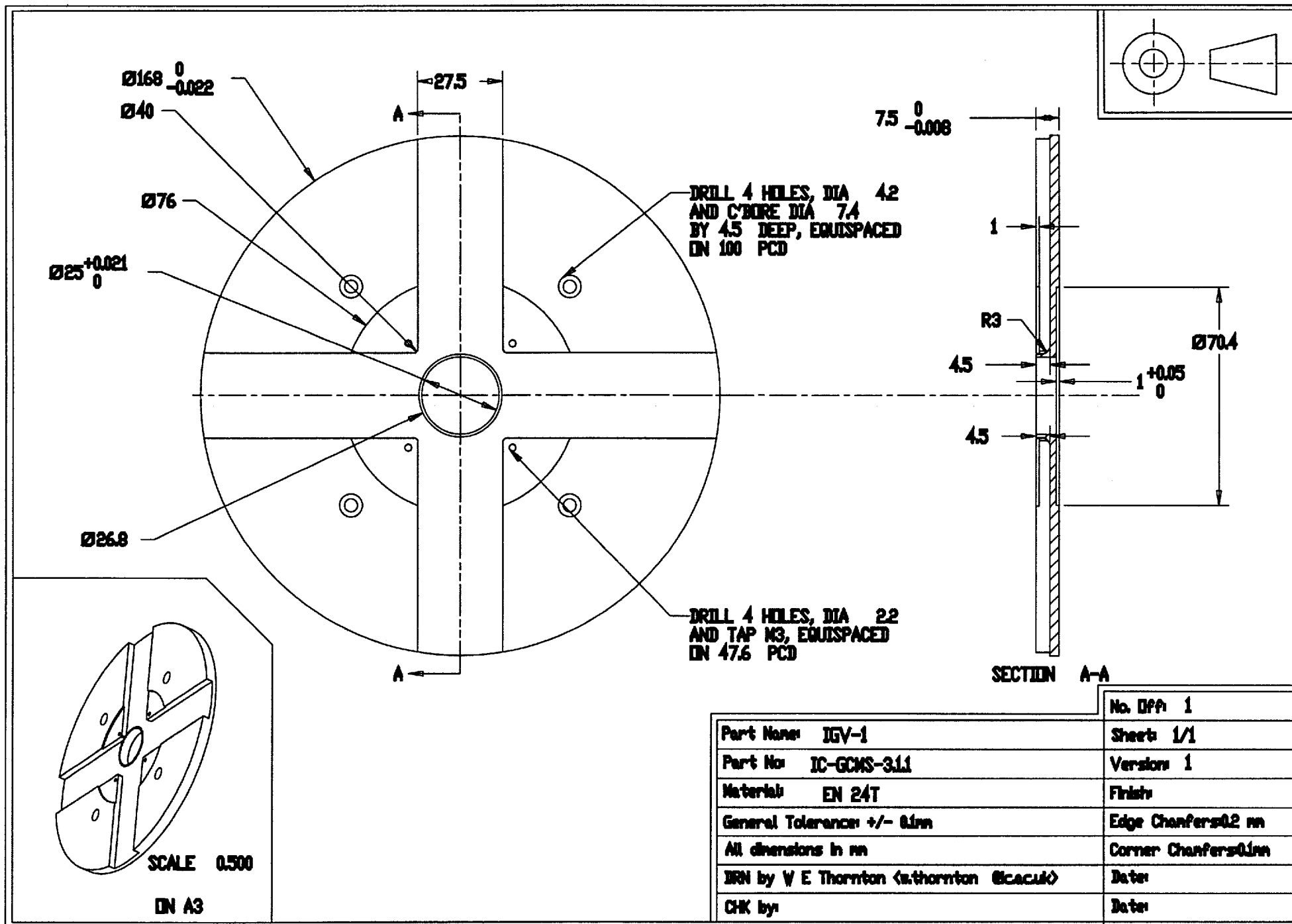
HALF SECTION VIEW (PART IS SYMMETRIC)

ALL SEALS:
TIP WIDTH: 1.0mm
GROOVE WIDTH: 1.0 mm
RADIUS: 0.5mm



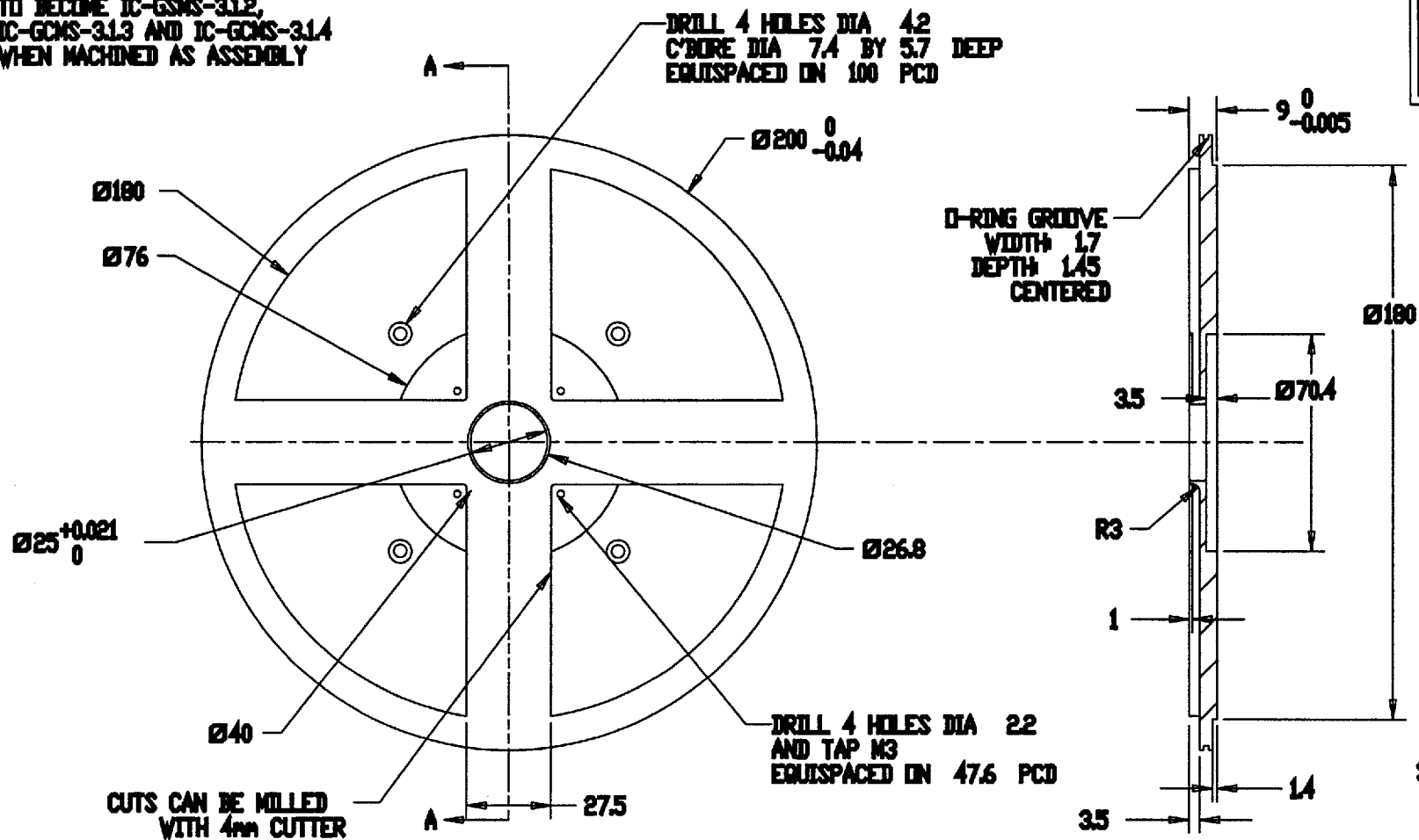
DETAIL SEAL
SCALE 15.000

		No. Diff: 1
Part Name: Square Seal		Sheet: 1/1
Part No: IC-GCMS-2.5		Version: 1
Material: EN 16T		Finish:
General Tolerance: +/- 0.1mm		Edge Chamfers: 0.2 mm
All dimensions in mm		Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@cacul)		Date:
CHK by:		Date:



Part Name: IGV-1	No. DfP: 1
Part No: IC-GCMS-3.1.1	Sheet: 1/1
Material: EN 24T	Version: 1
General Tolerance: +/- 0.1mm	Finish:
All dimensions in mm	Edge Chamfers: 0.2 mm
DRN by V E Thornton (v.thornton@scacul)	Corner Chamfers: 0.1mm
CHK by:	Date:
	Date:

TO BECOME IC-GCMS-3.12,
IC-GCMS-3.13 AND IC-GCMS-3.14
WHEN MACHINED AS ASSEMBLY



No. Off: 3

Part Name: IGV BLANK

Sheet: 1/1

Part No: IC-GCMS-3.1X

Version: 1

Material: EN 24T

Finish:

General Tolerance: +/- 0.1mm

Edge Chamfers: 0.2 mm

All dimensions in mm

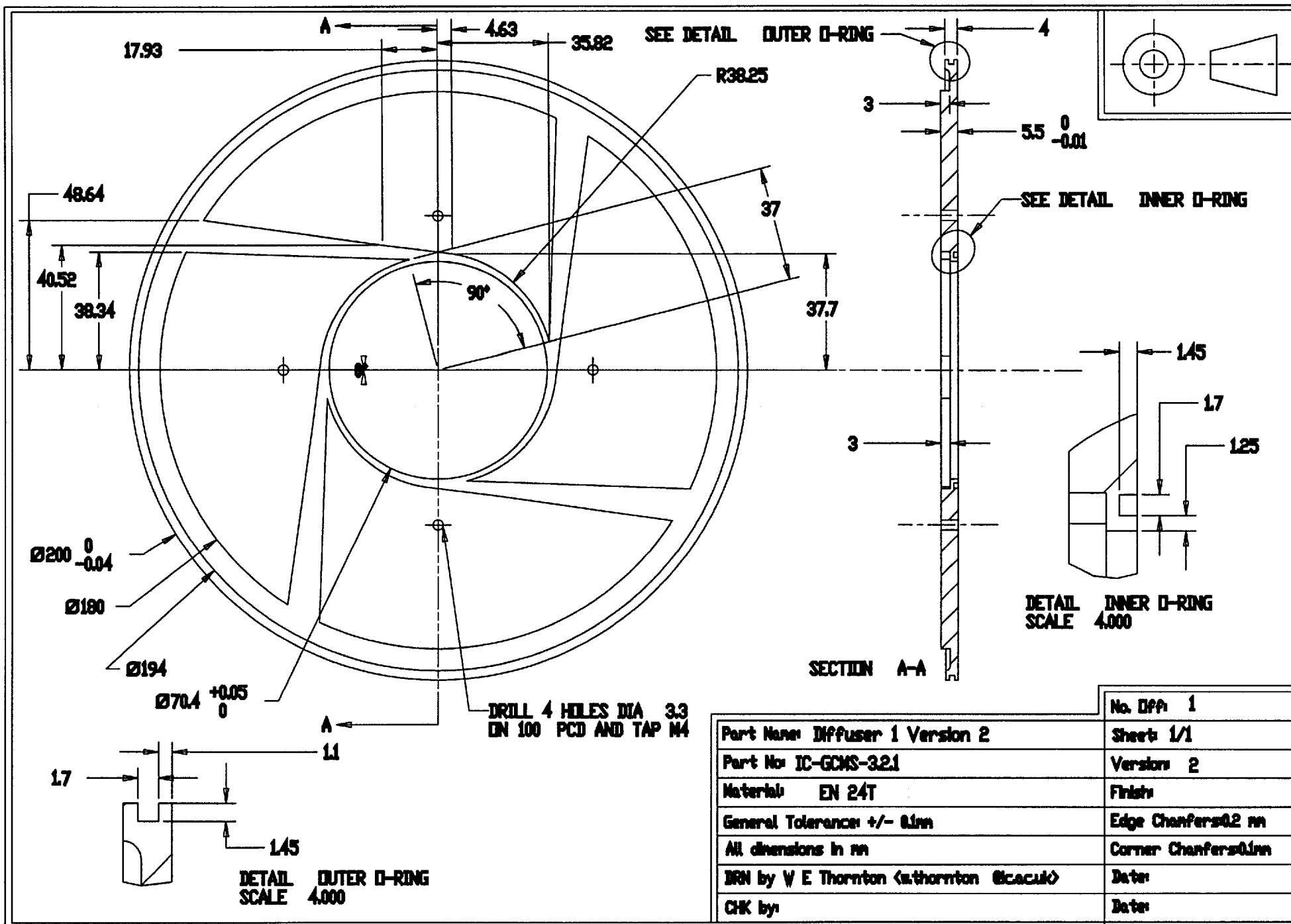
Corner Chamfers: 0.1mm

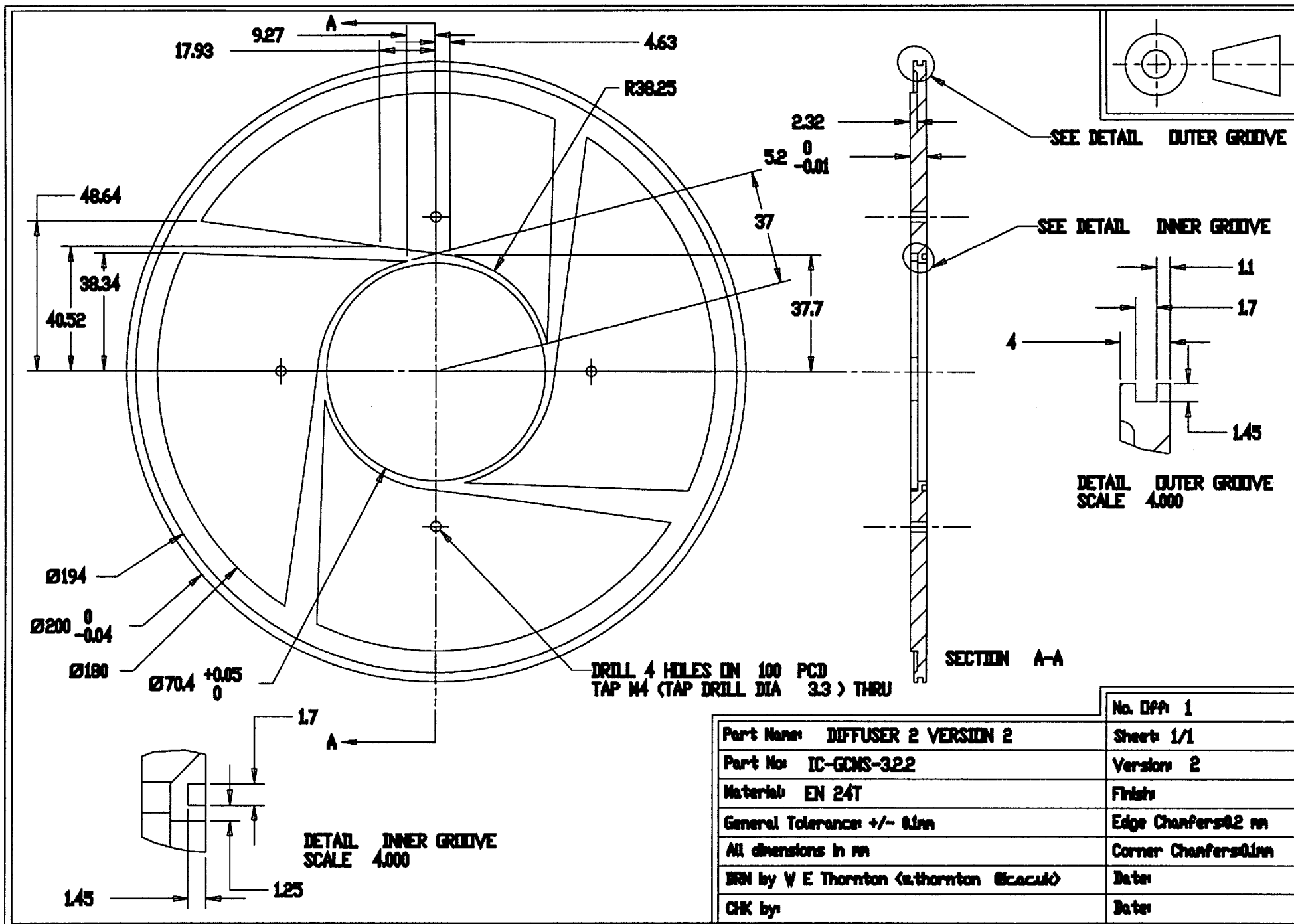
DRN by V E Thornton (authornton @cacul)

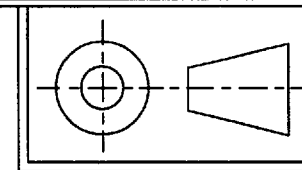
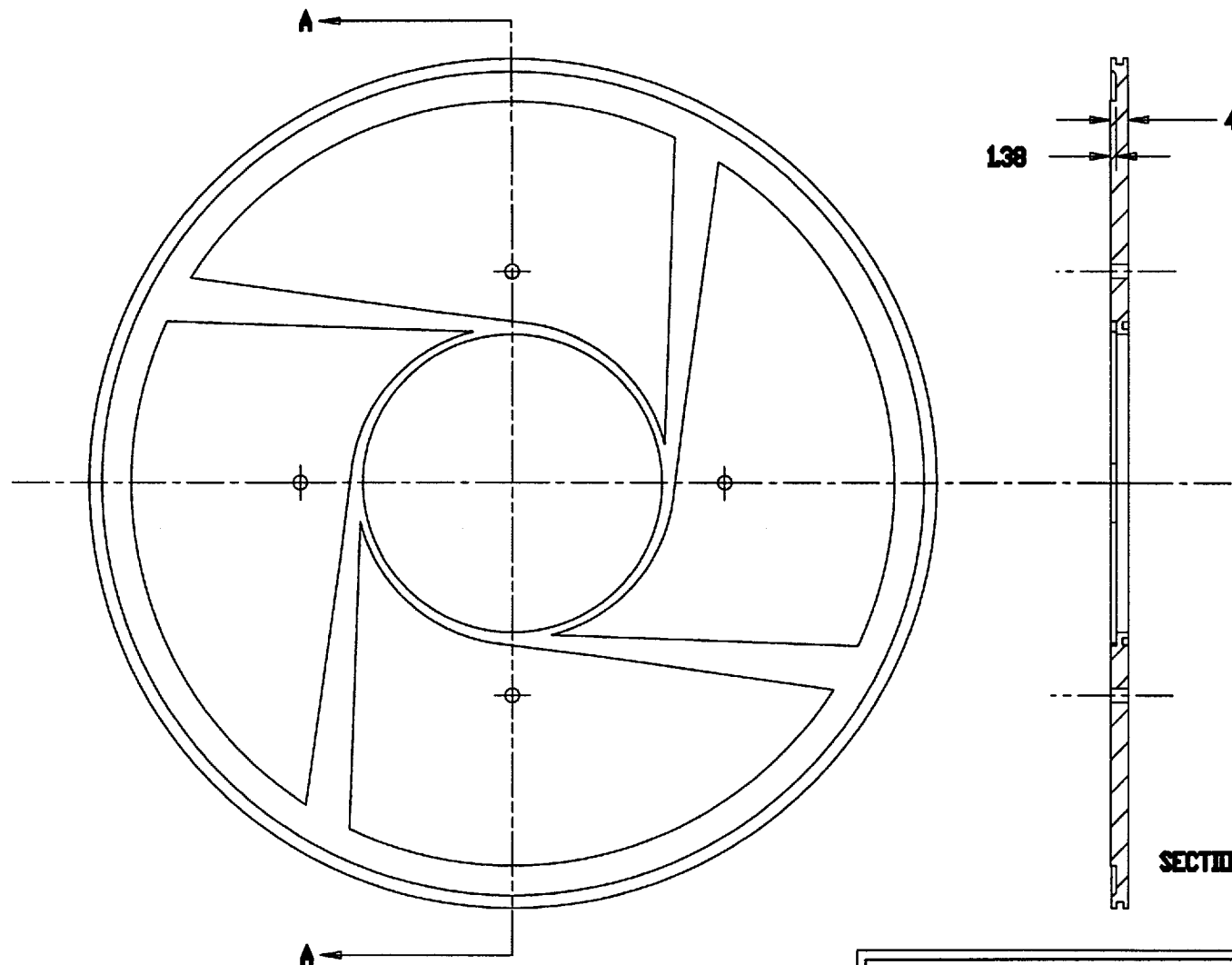
Date:

CHK by:

Date:

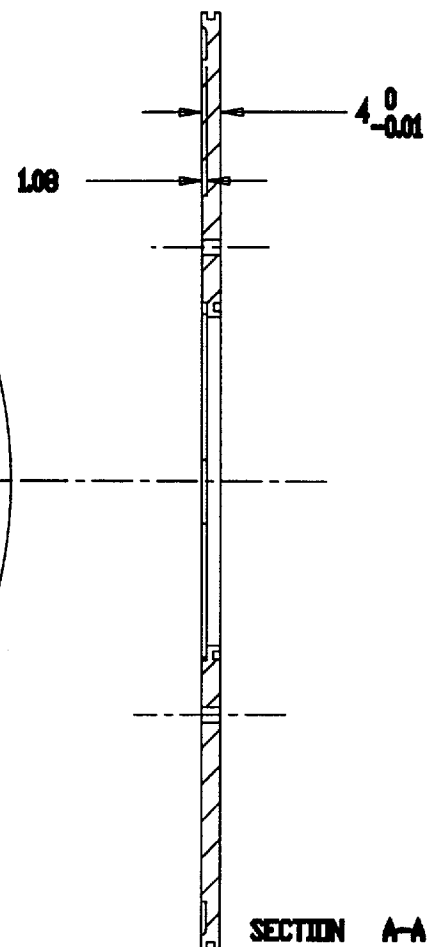
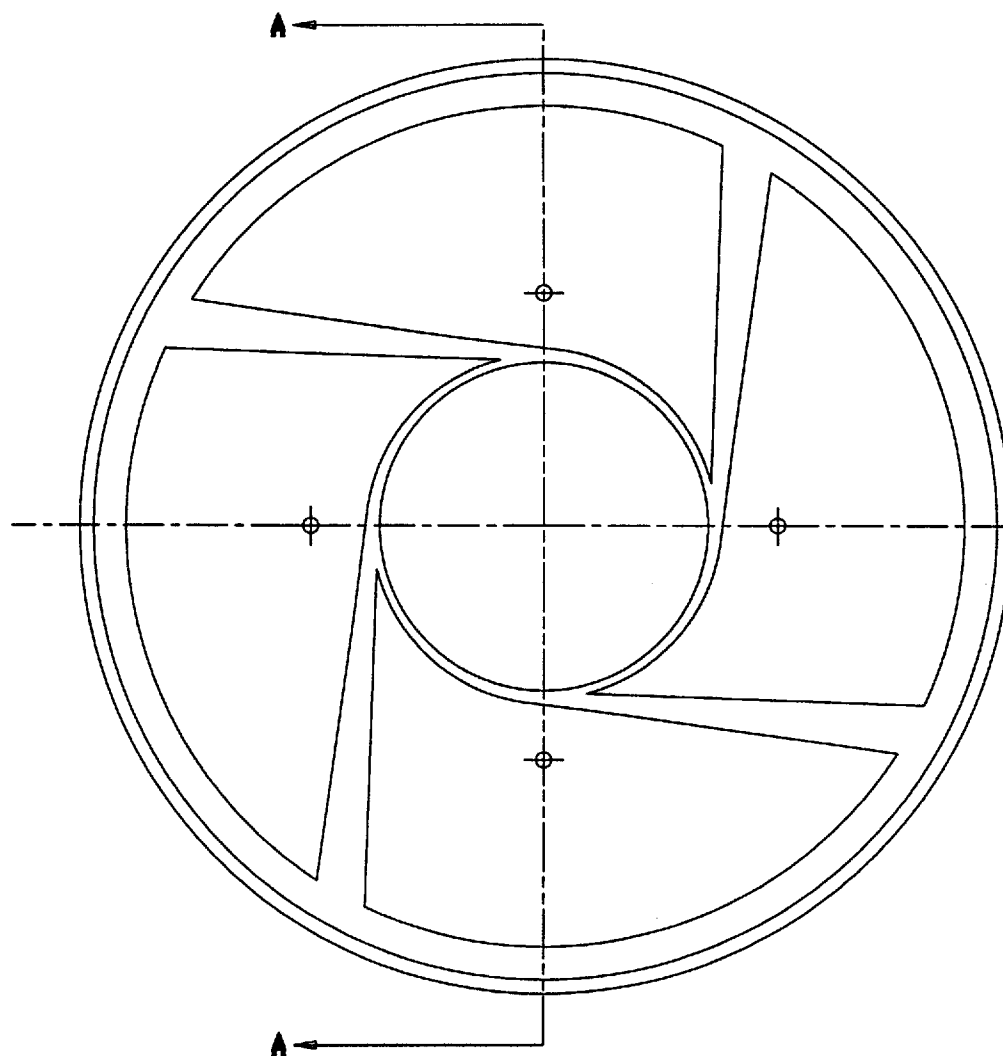
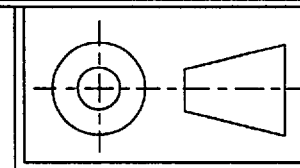




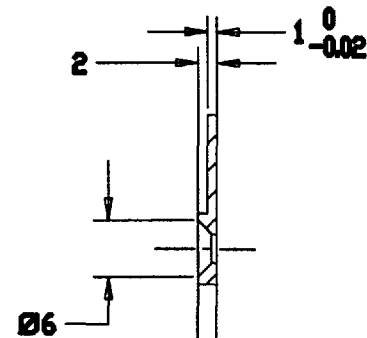
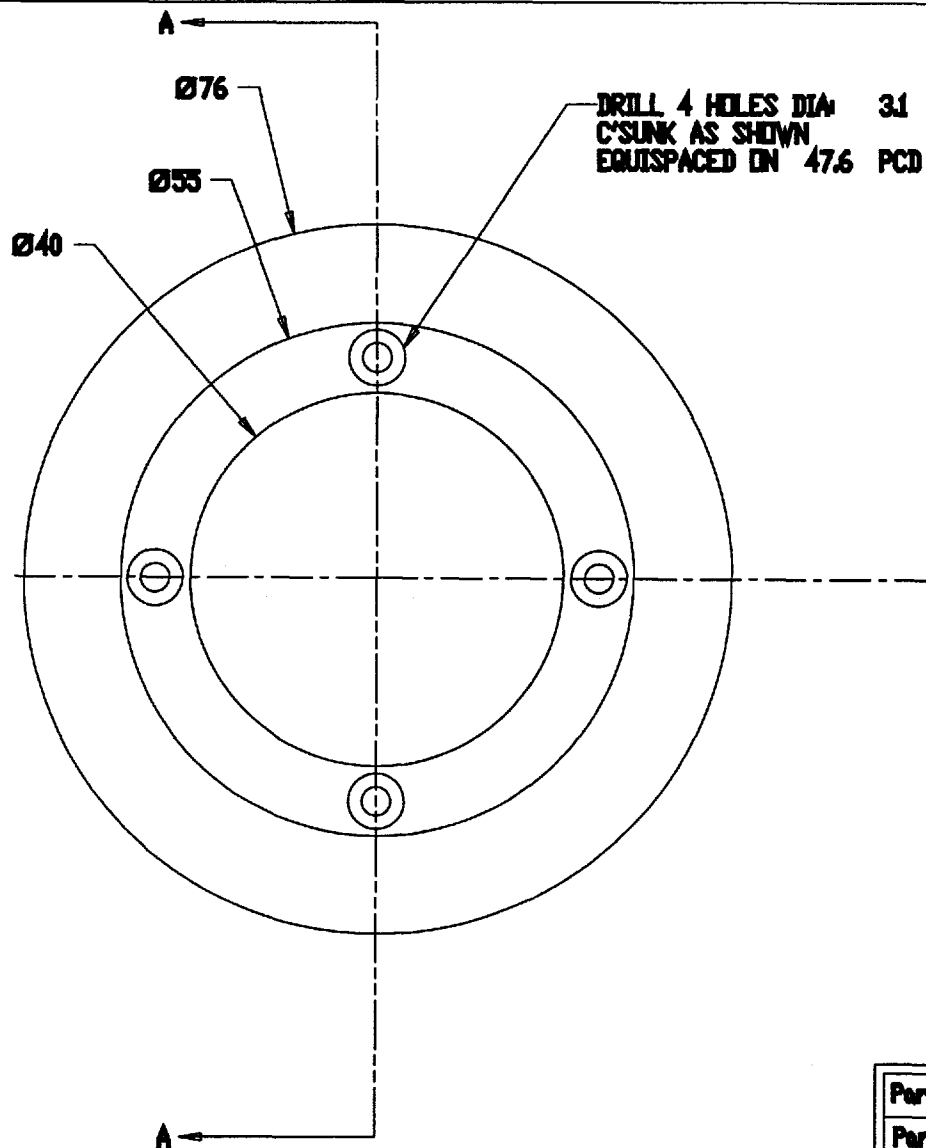


SECTION A-A

Part Name: DIFFUSER 3 VERSION 2	No. Off: 1
Part No: IC-GCMS-323	Sheet: 1/1
Material: EN 24T	Version: 2
General Tolerance: +/- 0.1mm	Finish:
All dimensions in mm	Edge Chamfers: 0.2 mm
DRN by: V E Thornton (v.thornton@icac.uk)	Corner Chamfers: 0.1mm
CHK by:	Date:

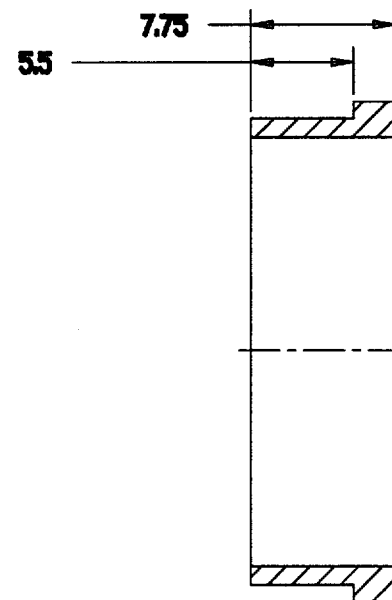
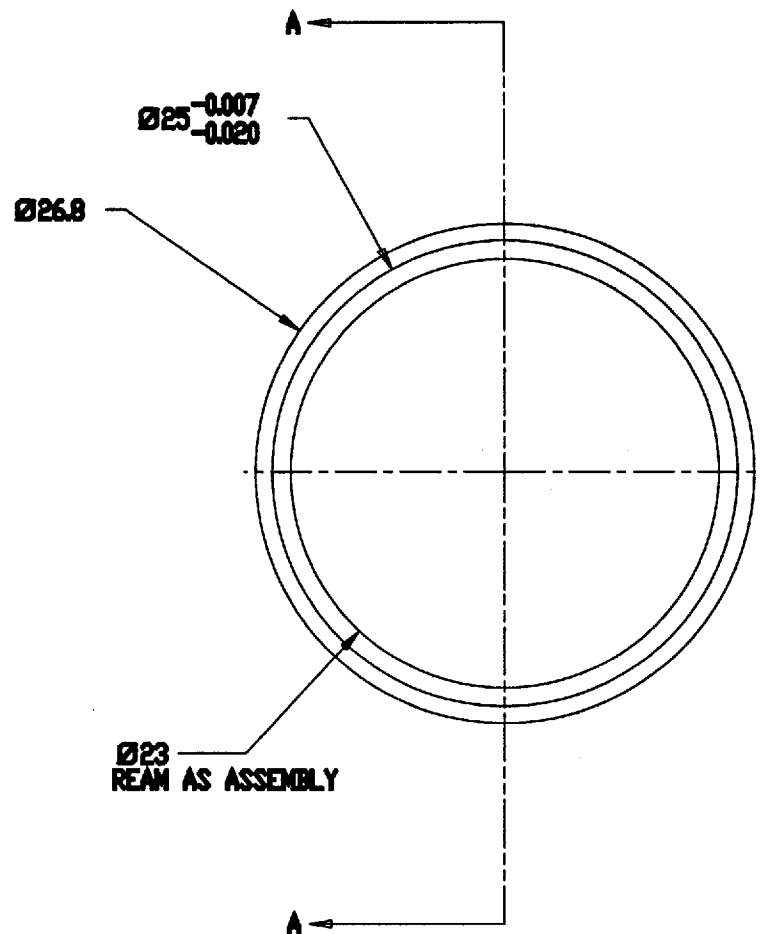
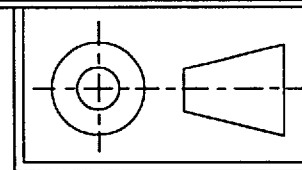


No. Off: 1	
Part Name: DIFFUSER 4 VERSION 2	Sheet: 1/1
Part No: IC-GCMS-324	Version: 2
Material: EN 24T	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by W E Thornton (w.thornton@icacuk)	Date:
CHK by:	Date:



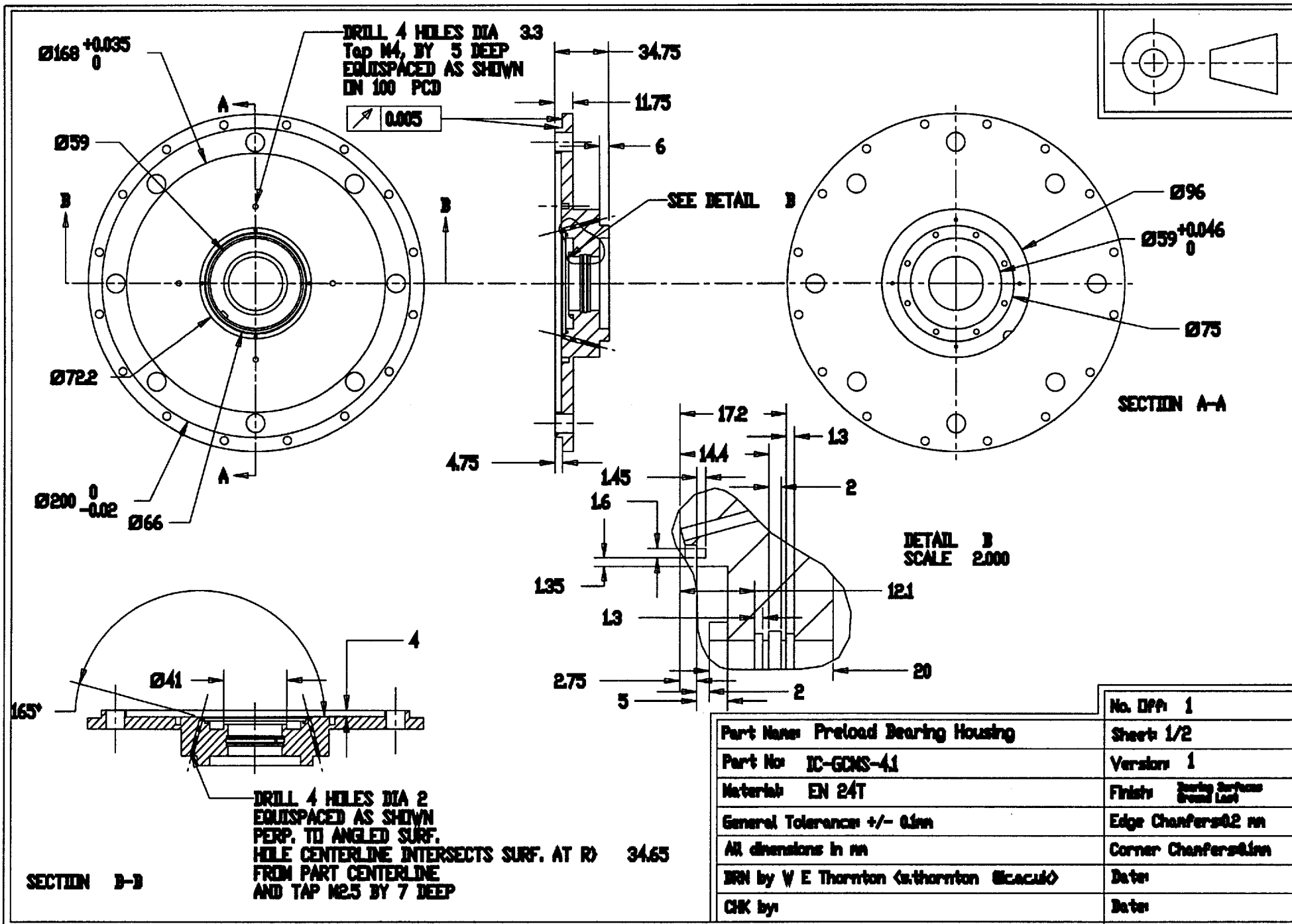
SECTION A-A

Part Name: ABRADABLE ROTOR SEAL		No. Diff: 5 (1 spare)
Part No: IC-GCMS-3.4		Sheet: 1/1
Material: PEEK 450G		Version: 1
General Tolerance: +/- 0.1mm		Finish:
All dimensions in mm		Edge Chamfers: 0.2 mm
DRN by V E Thornton (v.thornton@cacuk)		Corner Chamfers: 0.1mm
CHK by:		Date:
		Date:



SECTION ON A-A

No. Diff 8 (4 spare)	
Part Name: Shaft Abradable Seal	Sheet: 1/1
Part No: IC-GCMS-3.5	Version: 1
Material: PEEK 450G	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by W E Thornton (s.thornton@cacuk)	Date:
CHK by:	Date:



DRILL HOLES DIA 12
EQUISPACED ON 182 PCD
TAP 1/4" BSPT

$\varnothing 35^{+0.006}_0$
SUBTRACT 0.2mm FOR GRINDING
GRIND WHILE ASSEMBLED
WITH MAIN HOUSING
(IC-GCMS-2.1)

DRILL HOLE DIA 2
C-BORE DIA 8.75
TAP 1/8" BSPT, BY 14 DEEP
Approx Tap DIA: 9.728

DRILL DIA 5 AT
55° FROM PART CENTERLINE
TAP 1/8" BSPT, 17 DEEP
(Approx Tap Diameter: 9.725)

DRILL 16 HOLES DIA 5.2 THRU
EQUISPACED ON 208 PCD

DRILL 8 HOLES DIA 3.3
TAP M4, 8 DEEP
EQUISPACED ON 68 PCD

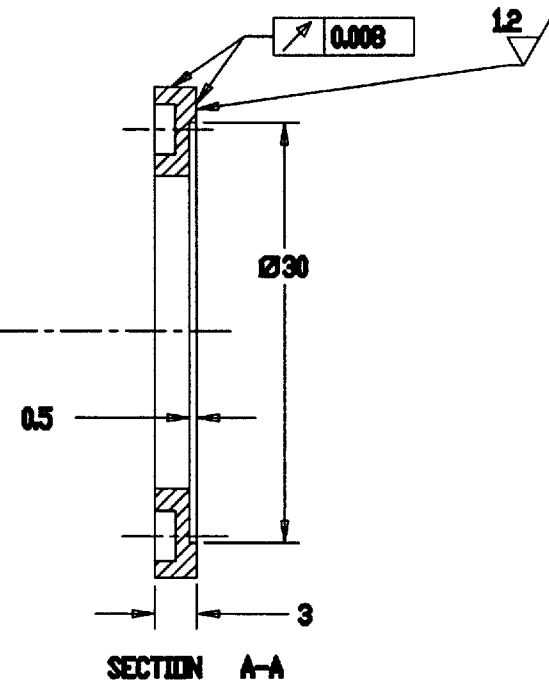
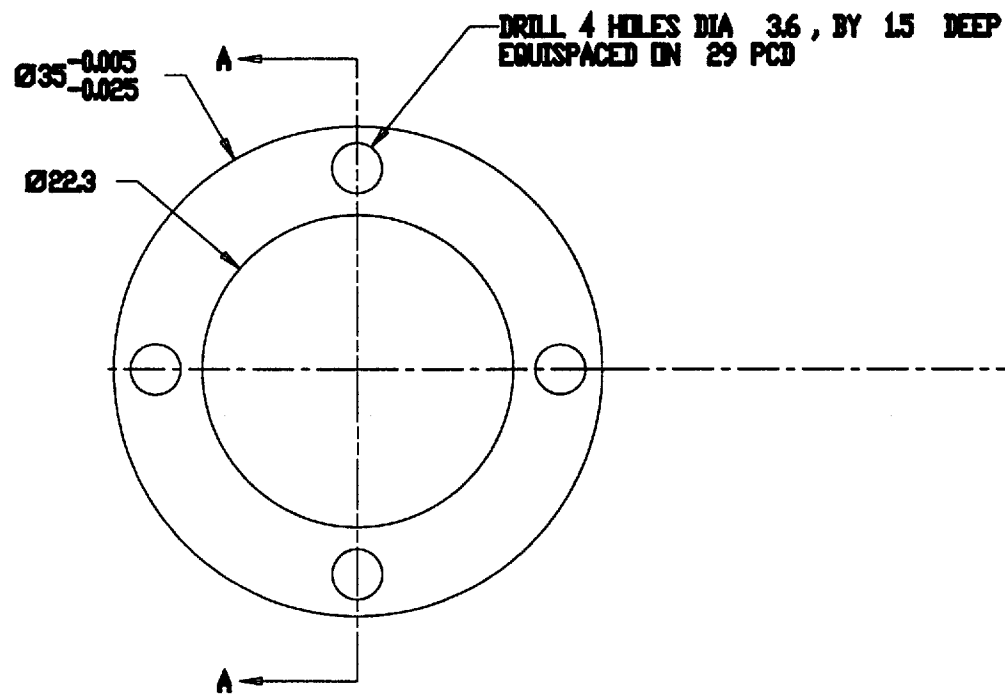
DRILL DIA 2
BY 30.1 APPROXIMATE DEPTH
DRILL DIA 8.75
BY 23 DEEP
DO NOT BREAK THRU

DO NOT BREAK THRU
LEAVE AT LEAST
0.3mm MATERIAL

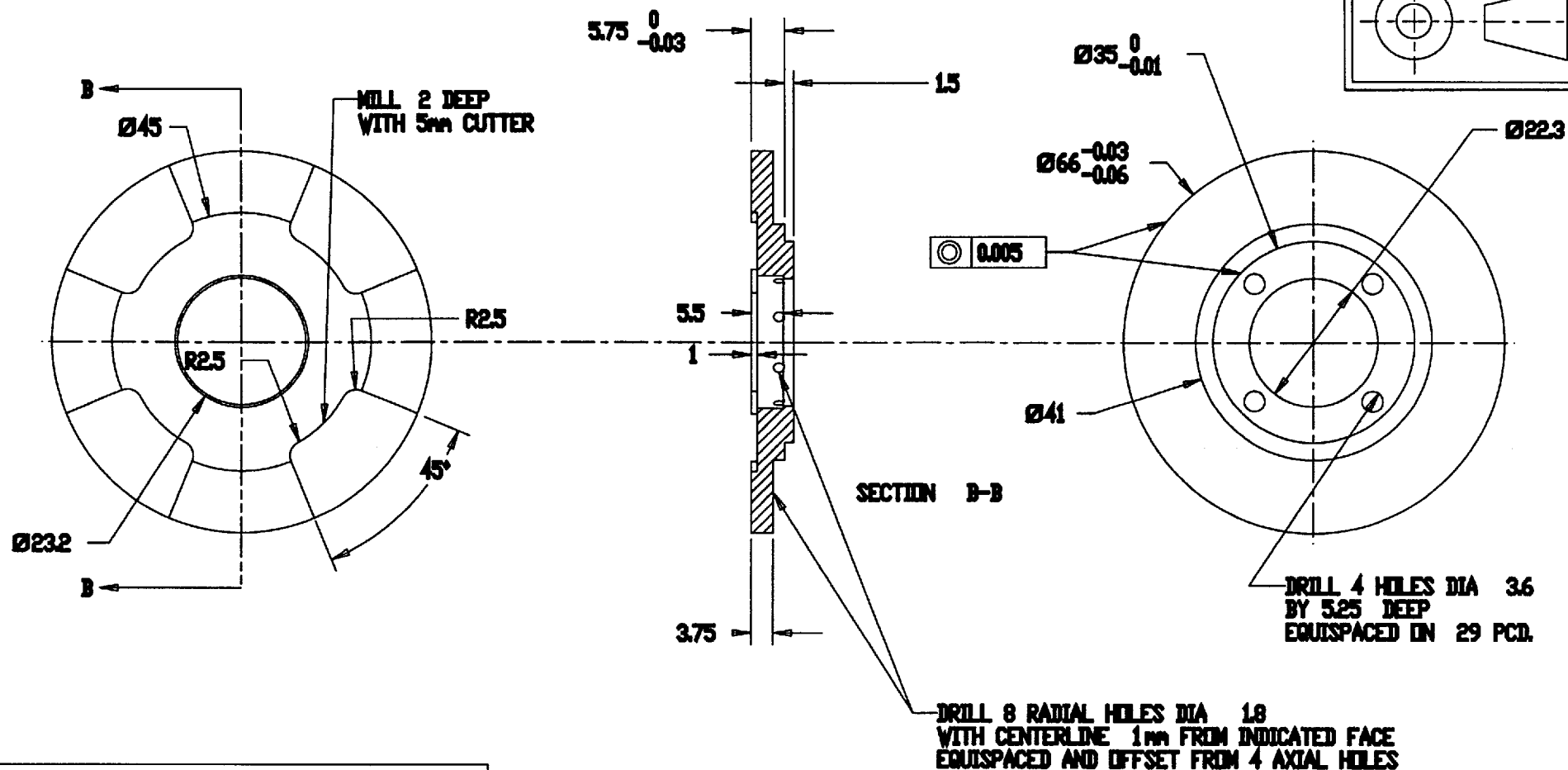
SECTION C-C

SECTION D-D

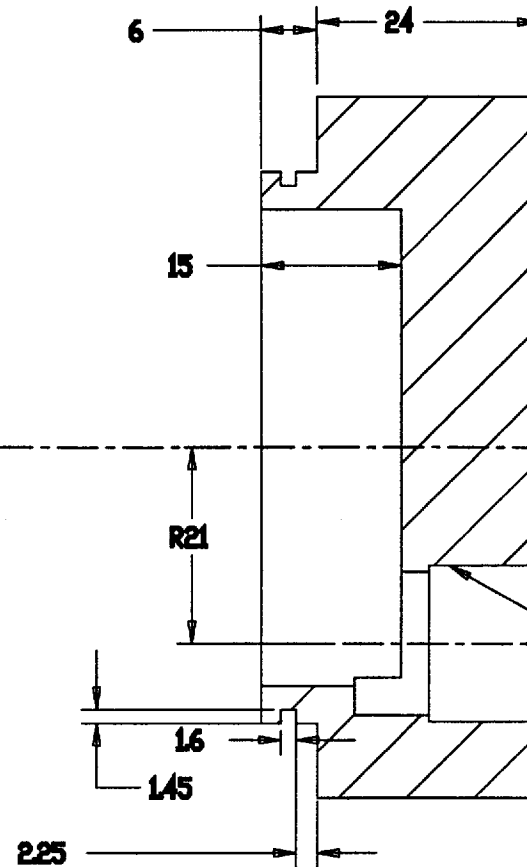
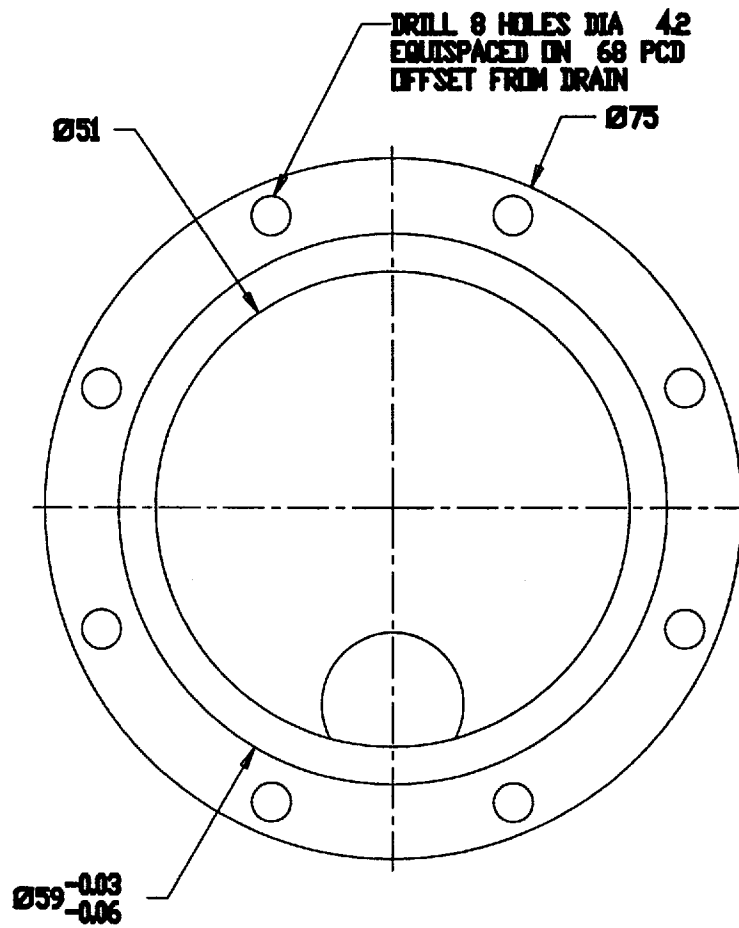
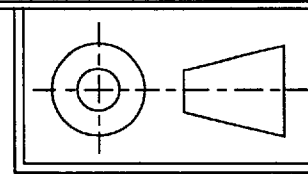
Part Name: Preload Bearing Housing	No. Diff: 1
Part No: IC-GCMS-4.1	Sheet: 2/2
Material: See 1/2	Version: 1
General Tolerance: +/- 0.1mm	Finish: See 1/2
All dimensions in mm	Edge Chamfers: 0.2 mm
DRN by W E Thornton (w.thornton @cacul)	Corner Chamfers: 0.1mm
CHK by:	Date:
	Date:



No. DPA 1	
Part Name: Spring Race	Sheet: 1/1
Part No: IC-GCMS-4, 2	Version: 1
Material: EN 1A	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@icacuk)	Date:
CHK by:	Date:



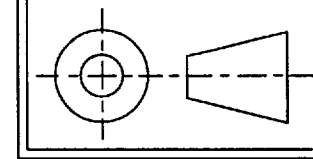
No. Off: 1	
Part Name: Spring Stop	Sheet: 1/1
Part No: IC-GCMS-4.3	Version: 1
Material: EN 1A	Finish:
General Tolerance: $\pm 0.1\text{mm}$	Edge Chamfers: 0.2mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (v.thornton@icacul)	Date:
CHK by:	Date:



DRILL HOLE 15.25 , BY 20 DEEP
AND TAP 3/8" BSPT, BY 12 DEEP
(approx tap OD 16.6)

SECTION A-A

No. Diff: 1	
Part Name: DR Cap	Sheet: 1/1
Part No: IC-GCMS-4.4	Version: 1
Material: EN 16T	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by V E Thornton (s.thornton @cacuk)	Date:
CHK by:	Date:



Ø240^{+0.046}₀

Ø328

Ø298^{+0.13}₀

A

B

B

A

10.75

29.1

12.5

4.1

10.75

DRILL 16 HOLES, DIA 6.2
EQUISPACED ON 318 PCD

DRILL 2 PAIRS OF RADIAL HOLES
DIA 3.5, THRU
C'BORE AND TAP 1/8" BSPT, 10.4 DEEP
APPROX TAP D/D 13

SECTION A-A

18.35

18.35

SECTION B-B

Part Name: InnerVolute

Part No: IC-GCMS-5.1

Material: EN 24T

General Tolerance: +/- 0.1mm

All dimensions in mm

DRN by W E Thornton (w.thornton@icacuk)

CHK by:

No. Of P: 1

Sheet: 1/1

Version: 1

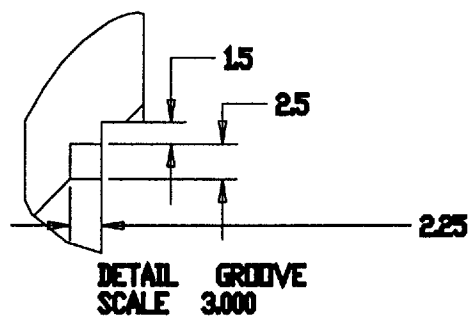
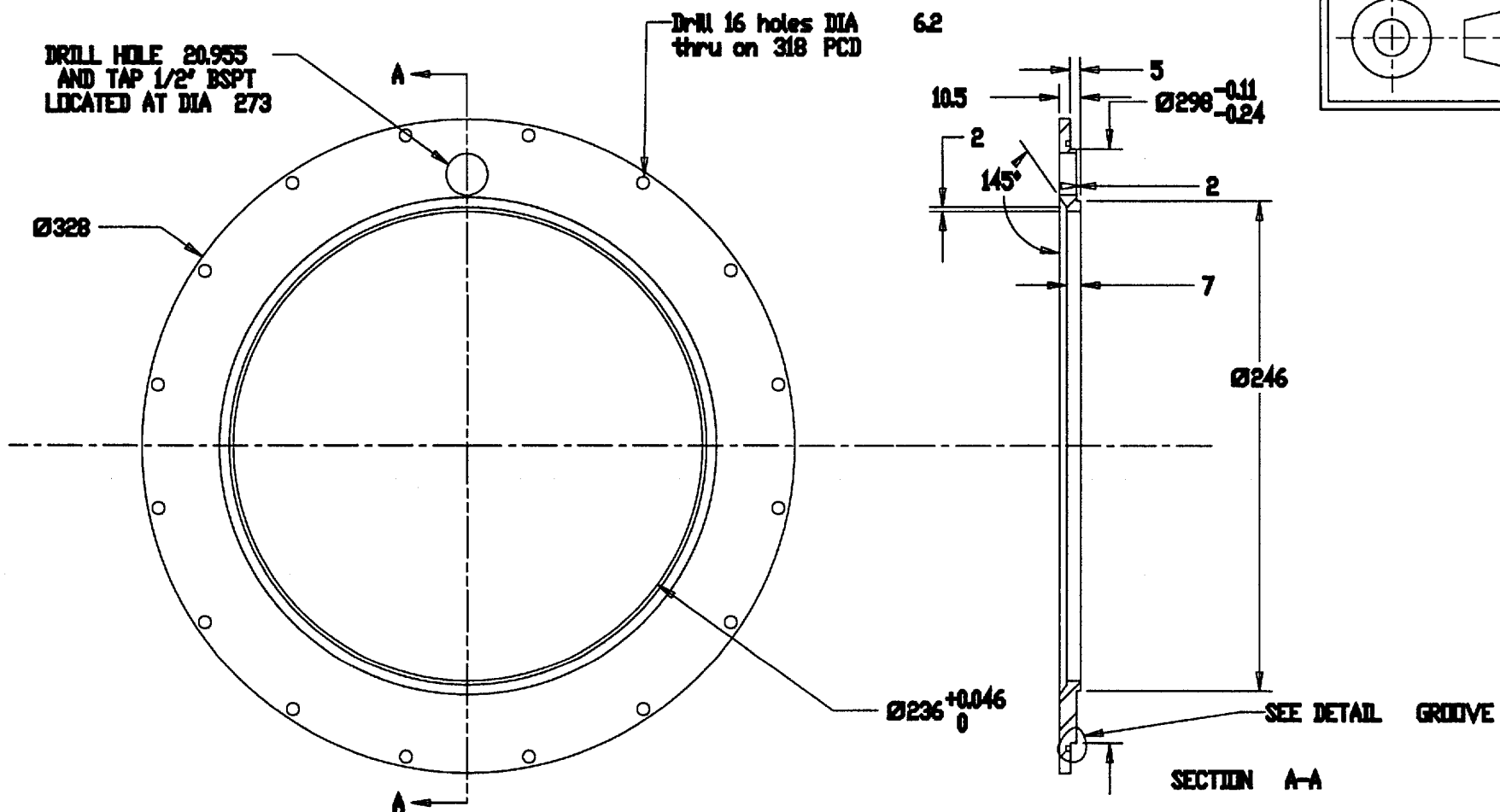
Finish:

Edge Chamfers: 0.2 mm

Corner Chamfers: 0.1mm

Date: 25.9.02

Date:



	No. Dft: 2
Part Name: Outer Volute	Sheet: 1/1
Part No: IC-GCMS-52	Version: 1
Material: EN 24T	Finish:
General Tolerance: +/- 0.1mm	Edge Chamfers: 0.2 mm
All dimensions in mm	Corner Chamfers: 0.1mm
DRN by W E Thornton (w.thornton @cacuk)	Date:
CHK by:	Date: