A case study of the design related aspects of the introduction of a new turbocharger

by

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It should be noted that changes have been made to much of the detailed data in order to protect propietary information.

ABSTRACT

The design related aspects of the introduction of a new turbocharger - the Napier NA355

Napier Turbochargers Ltd. manufacture a range of turbochargers suitable for most medium and slow speed diesel engines above about 1000 hp. This work describes the many and varied aspects associated with the design of a new turbocharger during the early 1980's. The company was producing two similarly sized products designated the SA105 and the NA350, intended for engines developing about 3000 hp. A replacement for these two turbochargers was required.

The demands of the engines in the market-place were determined together with the extractable performance of the major competitor's products. The performance levels of the two Napier models was examined, and was found wanting, particularly with respect to the compressor efficiency. Also the mechanical limitations of the two models were less than desirable.

A number of technical proposals were considered, with the aim of deciding how best the needed improvements could be introduced. The SA105 offered more items that could be utilised in a new turbocharger, although both compressor and turbine required improvement. It was necessary to avoid lengthy and expensive development wherever possible, therefore only proven technology was employed. The major casings and rotor forgings were to be retained, if possible, due to the lengthy, expensive, and often troublesome procurement cycles. Only those components that had a lack of performance or had a mechanical limitation would definitely be replaced.

A new backswept compressor and an improved turbine stage were designed. A new rotor assembly incorporating the new impeller, together with many detail improvements, was also designed. The work describes the analytical techniques that were used to carry out these designs.

The turbocharger was rig tested, as an open cycle gas turbine, to simulate service duty and environment. The first turbocharger was built and tested with encouraging results. A satisfactory number of turbochargers have been manufactured and are now in service. Although some problems occurred that necessitated modifications the basic concept was clearly satisfactory. The NA355 can be considered as a particularly successful turbocharger.

KEYWORDS - TURBOCHARGER,TURBOCHARGING,DESIGN,DIESELENGINE,
COMPRESSOR,TURBINE,TURBOMACHINERY

CONTENTS

CHAPTER EIGHT DESIGN AND DETAIL

NOMENCLATURE

A	Area	m^2
C_o	Exit gas velocity for isentropic expansion through a turbine total-static	m/s
c	Chord	m
c_p	Specific heat at constant pressure	kJ/kg K
c_v	Specific heat at constant volume	kJ/kg K
D	Diameter	m
h	Specific enthalpy	kJ/kg K
IMEP	Indicated mean effective pressure	bar
m	Mass flow rate	kg/s
M	Mach number	
N	Number of vanes; rotational speed	; rev/min
P	Pressure	bar
Q	Volume flow	m^3/s
r	Radius	m
R	Gas constant	kJ/kg K
R_C	Compressor pressure ratio	
R_{WD}	Diffuser flow coefficient	
s	Specific entropy; pitch	kJ/kg K; m
T	Temperature	K
U	Blade speed	m/s
V	Velocity; volume	m/s; m^3
W	Power	W
z	Number of blades	
α	Gas angle	degree
β	Blade angle	degree
γ	c_p/c_v	
δ	Deviation	degree
ε	Deflection	degree

η	Efficiency	
θ	Blade turning angle; meridional angle	degree; degree
ρ	Density	kg/m^3
σ	Compressor slip factor	
τ	Power input factor	
ϕ	Impeller flow parameter	
ω	Angular velocity	rad/s

SUBSCRIPTS

1-3	Stations eg. Compressor shown in figure 2.7
a	Axial
C	Compressor
D	Diffuser
E	Equivalent
h	Hub
M	Mean; mechanical; maximum
N	Nozzle
P	Peak
R	Relative; rotor
S	Isentropic; stall
t	Turbine; tip
T	Throat; turbine
W	Whirl
CRIT	Sonic

SUPERSCRIPTS

*	nominal

CHAPTER 1

INTRODUCTION

1.1 Aims of the work

In the late 1970s Napier were producing two ranges of a particular size of turbocharger. The earlier range was coming to the end of its useful life and a more efficient unit capable of operating to higher pressure ratios was required. A later design was being manufactured However, this was not as successful as the previous design. The company had a dilemma, but for a time effort was concentrated on the later design until it became clear this was not going to be successful. Eventually the entire picture had to be reviewed in order to identify the best way forward. It is the aim of this work to report the strategic review process, the technical choices, and upon the detailed design, testing, and early field experience of this size of turbocharger.

The thesis chooses a particular turbocharger as the means of providing a deeper view of the subject. Included is a description of the applied technology and background material to enable turbocharging to be understood in the wider context of the company and the associated industry.

1.2 Layout of the thesis

It is intended that each chapter is as self-contained as possible, with the relevant references, figures, and tables included at the end of each chapter. Chapter 1 provides an introduction both to turbocharging and to the Napier company. The thesis may be seen to contain background theory in chapters 2 and 3 which provide a theoretical basis for the remaining work. The work described in chapters 4-7 provide the essential information upon which the design could be based. Chapter 8 describes the detailed design work that enabled the turbocharger to be manufactured, and chapter 9 covers the validation of the design. Chapter 10 draws concluding remarks from the main chapters.

1.2 History of the Napier company (1.1)*

The history of the company and its products are considered so that it may be understood how the manufacture of turbochargers initially became a natural progression and eventually the sole remaining product.

David Napier was born in Argyll in 1785. He travelled to London and worked for Henry Maudslay for a brief period before setting up in business on his own account specialising in printing machinery. Napier's business in Lambeth, London, flourished between 1820-1850 by invention, industry, and innovation. He produced such delightfully named machines as the 'Nay-Peer', 'Desideratum', and 'Double Imperial'. The company diversified into products such as bullet-making and coin-weighing machines.

James Murdoch Napier formally succeeded his father in 1867 and continued with banking machinery, including minting machines, and also stamp perforating machines. He had four sons, the youngest of whom was Montague Stanley Napier born in 1870.

M S Napier was not involved in the business at the time of his father's death. He purchased the business from the executors of his father's will. In addition to the established range of products of coin-classifying and printing machines he began experimenting with other possibilities, particularly the growth industry of transport. In 1898 or 1899 he designed a motor car and set to building it in the Lambeth works. Napier had been a racing cyclist and was acquainted with another cyclist called Selwyn Francis Edge, who was an extraordinary entrepreneur. Edge bought a Panhard-Levassor which had been placed second in the Paris-Marseilles race in 1896 He thought it could be improved and asked Napier to do so. The car was converted to wheel steering and pneumatic tyres. Napier also designed and produced a new engine which Edge made sure received wide acclaim. A business was arranged whereby all the Napier car production would be sold by Edge, an arrangement of great similarity with the later Royce and Rolls venture.

* references are given at the end of each chapter

The business expanded by Edge's business acumen and Napier's technical expertise. By 1906 the Napier works at Acton employed 1000 people and more than two hundred cars a year were being built. The firm had outgrown the capital capabilities of Napier and the company was incorporated as D Napier & Son Limited. Napier held all 5000 ordinary capital shares, apart from the seven £1 shares legally required by the nominees. The car manufacturing business was not large in output but it was extremely important in the formative years early in the 20th century. In general Napier made cars for the wealthy and powerful both in the UK and abroad, and competed on equal terms with the most respected companies such as Rolls-Royce. In 1911 Napier and Edge quarrelled with the result that Edge left the scene in 1912 and a new public company was formed under the old name.

The first world war swept Napier out of the luxury car business. However, the company continued in the commercial market. During this war period Napier made nearly 2000 vans, lorries, and ambulances nearly all to the order of governments. Napier undertook the production of three 200 hp aero-engines, intending it to be a stop-gap arrangement. However, it was not long before the Acton works became a government 'controlled establishment'. The effect was that the government took into its own hands the ultimate responsibility for deciding what the firm should do. All work on motor vehicles stopped for a time and the firm's entire effort was directed towards air-frames and aero-engines. The first engine to be made was the Royal Aircraft Factory 3a twelve-cylinder water-cooled engine. Later there came the eight-cylinder Sunbeam 'Arab'. Three thousand of these latter engines were ordered before a prototype had been properly tested. The 'Arab' as originally conceived was a failure, however, Napier and Rolls-Royce managed to salvage the situation. Napier became convinced that he could design a much better engine. He recruited A J Rowledge from Wolseley Motors Limited to design the new engine to meet the Ministry specification. The engine had to be capable of developing 300 hp at an altitude of 10000 ft. The resulting design was a 12 cylinder broad-arrow formation which became known as the Napier 'Lion'. Although produced too late for the First World War the engine became one of the most widely used aero-engines of all time. It won many awards and honours, including two Schneider seaplane trophies, and an abundance of world records on land, sea, and air. Napier realised that the aero-engine business was more suited to his company's abilities than car manufacturing.

In 1925, with Rowledge firmly established at Rolls-Royce revitalising their V-engine design, Napier made an agreement for the services of a designer not directly employed by the firm. The man whom Napier engaged was F B Halford, an eminent aero-engine designer since 1914. He had two outstanding designs to his credit: the Cirrus 'Hermes', and the 'Gypsy' which powered that most successful of light planes, the DH 'Moth'. Under an agreement which allowed him to work for other firms he worked on a number of unorthodox designs for Napier. In particular these included the 'Dagger' and eventually the 'Sabre' which became one of the most powerful piston-engines for fighter aircraft ever produced. It was also realised that the diesel engine might have a future in flying, and this led to the building under licence of the Junkers 'Jumo' engine, this was known as the Napier 'Culverin'. Napier died in 1931 although he had controlled the business from his home in Cannes for over ten years. Sir H Snagge, a banker and a director of several companies was, in 1932, appointed as chairman by the City finance house to whom Napier's shares had been sold. However, thoughts of diversification, for example to re-enter the car market, were swept aside and aero-engine design and manufacture continued as if under Napier. In 1935 Halford became a director of Napier when the Ministry was buying the 'Dagger' in quantity. By 1938 Napier was discussing with the Ministry the possibility of building 1000 'Sabre' engines a year. This led to the move of the production facility to Liverpool where the plant became productive in 1942. At that time Napier employed 10000 people. This rapid expansion required for the Second World War effort was too much for Napier alone and the Government funded the Liverpool factory. English Electric was asked by the Government to reorganise the Acton works whilst the Standard Motor Company was asked to do the same at Liverpool. Later the same year the company came to the end of its independent life by the share buy-out by English Electric.

The Government sponsored a gas turbine research station at the Liverpool factory in the mid-1940s. One of the most innovative engines to emerge from this period was the 'Nomad'. This was the Napier answer to the Admiralty requirement for a lightweight compression-ignition engine capable of long-range flying by virtue of its fuel economy. The 'Nomad' was a compound engine utilising the technologies of the piston-engine and gas turbine. For very good reasons at the time the demand proved to be insufficient for its continuing development. In 1947 Napier had the first axial-flow compressor turbo-prop engine running, this was called the

'Naiad'. This was followed by the 'Eland' which eventually went into airline service in 1958. Napier was involved with the particular requirements of helicopter gas turbines and probable the most successful of these was the 'Gazelle'. Napier continued their design of piston-engines, the most well known was the result of work for the Admiralty. This engine was called the 'Deltic'. The basic configuration is a triangulated arrangement of three crankshafts with three banks of cylinders. The engine is a 2-stroke with opposed pistons. First tested in 1952, although, originally conceived for high-speed defence craft the engine continues to be manufactured for a range of high-specific power requirements. One of the most well known applications for this engine was in the class of British Rail locomotives named after the engine. This supercharged 18-cylinder version (two per locomotive) went into service in 1958 and was withdrawn only recently. In the mid-1960s it became apparent that there was insufficient Government finances to support all the gas turbine research available in the country. The Napier aero-engine business was sold off to Rolls-Royce in this period of rationalisation.

In the mid-1940's Napier was concurrently working on highly-rated piston-engines and developing the technology for the gas turbine. It was inevitable that Napier began to combine this expertise and to commence the production of turbochargers. In 1946 the first Napier turbocharger was produced (Fig 1.1) and in 1947 this model was applied to rail-traction diesel engines. A whole range of turbochargers was produced for use with engines around 1000 kw up to the largest diesel engines produced. In the late 1960's Napier finally relinquished the Acton site and in 1970 the company moved from the Liverpool factory to Lincoln. This move coincided with the remaining Napier interest in diesel engines, the 'Deltic' engine, moving to Paxman in Colchester. The moves were part of the GEC reorganisation and brought the turbocharger business under the control of Ruston Gas Turbines. Shortly afterwards the name of the company changed from the old D Napier & Son Limited to the more appropriate Napier Turbochargers Limited. At the present time the number of people employed by the company is around 200.

1.4 Turbocharging

It is a commonly held view that turbocharging is a recent innovation in the internal combustion engine field. This may well be the case in the automotive industry. However, in engines of larger size, turbochargers have been a major influence for a number of decades.

The principles of turbocharging were established early in the history of engine development. The credit for the idea is generally attributed to Buchi, in conjunction with Sulzer, and patents were granted from 1905 onwards (1.2). If supercharging alone is being considered, experiments by Rudolf Diesel were carried out as long ago as 1896 (1.2).

The object of turbocharging is to enable a particular size of internal combustion engine to develop more power. The amount of power that an engine can develop is potentially proportional to the amount of fuel that can be burned. Increasing the density of the air being supplied to the cylinders allows more fuel to be burned, and hence increases the power the engine develops. Achieving this part of the process by an increase in the air pressure is termed 'supercharging'. The density of the air can also be increased by reducing its temperature. As this is normally carried out after a compression stage this part of the process is called 'intercooling'. A common method of driving the supercharger compressor is through a direct mechanical drive from the engine. However, in the case of a turbocharger the compressor is driven by an engine exhaust-driven turbine.

Most turbochargers consist of a single stage radial compressor driven by a single stage axial or radial turbine, both rotor discs being mounted on a single shaft. Smaller turbochargers employ radial inflow turbines, whereas the larger machines have axial-flow turbines. In the turbocharger there is no mechanical link with the engine.

It may be seen that the elements of a turbocharger are essentially similar to those of a gas turbine. It is not surprising that the earlier experimenters in turbocharging were beset by some of the same problems associated with the development of the gas turbine, such as the lack of high-temperature-resistant materials. Buchi's earlier experiments were not particularly successful and it was not until 1923 that the first successful application of exhaust turbocharging was recorded. This application was on two MAN-designed passenger ships. The turbochargers allowed the engine power to be increased from 1750 to 2500 hp (1.2). Other experimenters such as Rateau were at work around the First World War period, and he is generally given the credit for developing the internal layout of the machine. In 1918 the General Electric Company of the United States produced a turbocharger similar to Rateau's. It was applied to a Liberty aero-engine with the major object of maintaining the sea-level power at altitude. This was achieved with some success. However, apart from prototype flying the system was not developed (1.3). In addition the Royal Aircraft Establishment applied a turbocharger to the Napier 'Lion' engine, with not a great deal of success, concluding that the engine was not sufficiently robust to 'withstand the additional loads of the supercharge'. Also, with particular reference to the Napier company history, Major Halford, later to be the principal involved in the design of the 'Sabre' aero-engine, was involved with turbocharging in 1925. He constructed a 1.5 litre turbocharged grand prix racing car engine developing 96 bhp at 5300 rpm (Fig 1.2). It has to be recorded that the turbocharger was replaced by a much less novel but more suitable Roots type compressor (1.4).

Piston-engined aero-engines developed rapidly during the Second World War. However, almost universally the mechanically-driven supercharger was fitted during this era. An exception to this rule was the 12-cylinder horizontally-opposed turbocharged Lycoming engine (1.5).

The development of high-temperature materials and the introduction of the technology required to design gas turbines provided the means whereby the turbocharger industry could develop. Particularly for the larger size engine the turbocharger has enabled the specific power output to be increased steadily since 1945 and whereas the bmep of a naturally aspirated engine may be around 10 bar, figures well in excess of 20 bar are now standard (1.6).

1.5 Diesel engine categories and principles of operation

Mention has already been made of a distinction in engine size and the level of applied technology. It is therefore constructive to define to a limited extent the differences in diesel engine types to be considered here. One method of distinguishing between types of engines is by their application, for example it is easy to recognise the term automotive engine. However, the larger diesel engine can be utilised in a great many different applications. Therefore to differentiate by application is not particularly appropriate. It has become customary to differentiate the various types by the terms 'high', 'medium', and 'low-speed'. Although the definitions can become blurred the term high-speed diesel engine generally refers to smaller engines with rotational speeds above 1000-1200 rev/min. At the other extreme the term low-speed generally refers to the large 2-stroke engine with rotational speeds below 500 rev/min, often used for marine main-propulsion. Between these two designations the medium-speed diesel engine is a size of engine used for a wide variety of applications. For example the range covers smaller marine propulsion, rail traction, auxiliary power units, and generator applications. The power that can be developed in this range can vary from about 1000 kw up to 30000 kw.

The development of the diesel engine has been spread over a considerable period. The originator of the diesel engine cycle has been the subject of a certain amount of debate. In 1890 Akroyd patented and produced an engine that operated by the compression of air alone, with the injection of liquid fuel at or about the end of the compression stroke (1.7). However, it was Diesel who presented slightly later his pamphlet on rational heat motors and who is credited with the progress of this type of engine in the 1890s (1.7).

It is instructive to consider the effects of supercharging from a cylinder pressure/volume (P-V) diagram. Figure 1.3 shows an ideal naturally-aspirated dual-combustion air-standard cycle. This is considered to represent most closely the operation of the diesel engine in its present form. The process 1-2 is the compression stroke from bottom- to top-dead-centre. Process 2-3 is the part of combustion occurring at constant volume and 3-4 is the remaining combustion occurring at constant pressure whilst the piston is moving away from top dead centre. 4-5 is the expansion process following the end of combustion. Point 5 signifies the opening of the exhaust valve, allowing the gases to be expelled from the cylinder and the pressure to fall back to the ambient level. The intake and exhaust processes have not been shown. The useful work during the cycle is represented by the area beneath process 3-4-5 whereas the work required to compress the air in the cylinder is represented by the process 1-2. The net work output per cycle is represented, therefore, by the area 1-2-3-4-5-1. In the case of a 4-stroke engine it follows that the area of the diagram (joule) multiplied by half the engine speed (rev/sec) and the number of cylinders gives the power (W) that theoretically is developed. It should be considered that not all of this indicated power can be transmitted; for example frictional losses have been ignored.

Figure 1.4 compares the supercharged and naturally-aspirated ideal air-standard cycles. The new starting point 1' represents the increase in the engine inlet pressure, commonly referred to as boost pressure, resulting from a supercharger. It may be seen that extra fuel can be burned between 2'-4' as a consequence of the additional amount of air, higher pressure/same volume, at point 2'. Immediately apparent is the increased diagram area and therefore power. Also the large increase in maximum pressure 3' compared with 3 should be noted. Clearly this increase in maximum pressure will cause a corresponding increase in the mechanical loads that the major components have to endure. However, the maximum pressure is greatly affected by the compression ratio (V_1/V_2). As boost pressures have increased over the years the compression ratio has been reduced to ensure the increase in maximum pressure can be withstood. Figure 1.5 compares the result of reducing the compression ratio of a supercharged, compared with the naturally-aspirated, cycle. It is instructive to compare these cycles with representative values and table 1.1 presents a set of results for comparison. The reduced value of compression ratio is rather lower than would be acceptable for starting and satisfactory idling, a more typical value would be 11:1. The values presented in table 1.1 are those plotted in figure 1.3, 1.4, and 1.5. Although the arguments have been presented with particular reference to the ideal air-standard applied to a 4-stroke cycle a similar argument can be put forward for the ideal 2-stroke engine cycle and for real diesel engines.

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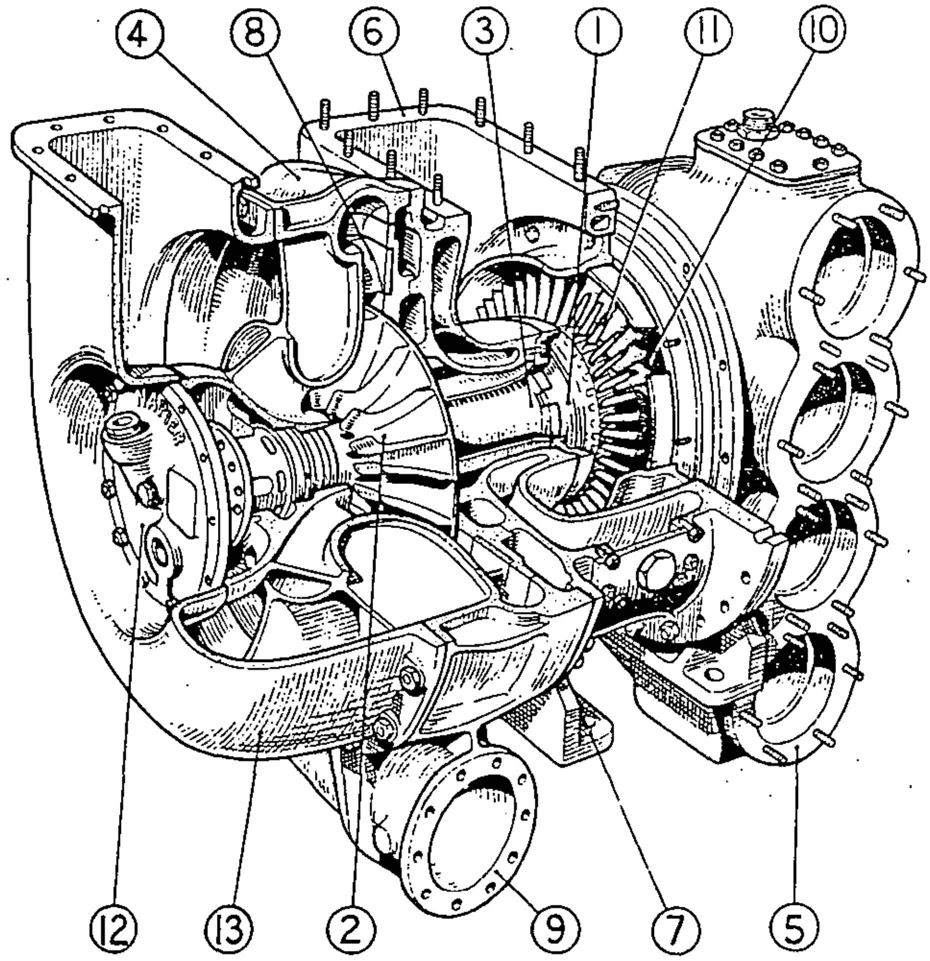
FIGURES

TABLES



1. Turbine wheel
2. Compressor impeller
3. Rotor shaft
4. Centre casing
5. Turbine inlet casing - 4-entry
6. Turbine outlet casing
7. Mounting feet
8. Compressor diffuser
9. Compressor delivery volute casing
10. Turbine nozzle
11. Turbine blade
12. Bearing housing end-cover
13. Compressor inlet casing

Figure 1.1:Napier TS200 turbocharger (1946)

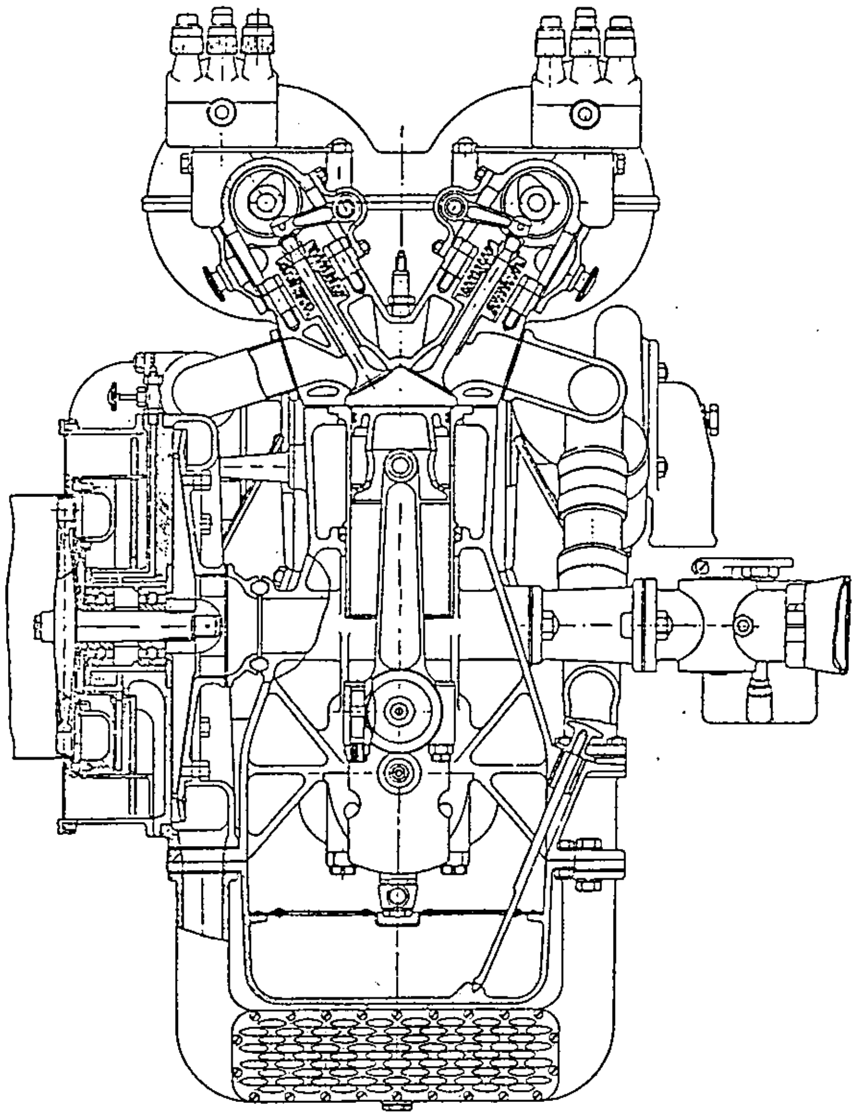


Figure 1.2:Halford 1.5 litre grand-prix racing engine

Cycle processes

1-2 Adiabatic compression
2-3 Combustion at constant volume
3-4 Combustion at constant pressure
4-5 Adiabatic expansion
5-1 Exhaust valve open

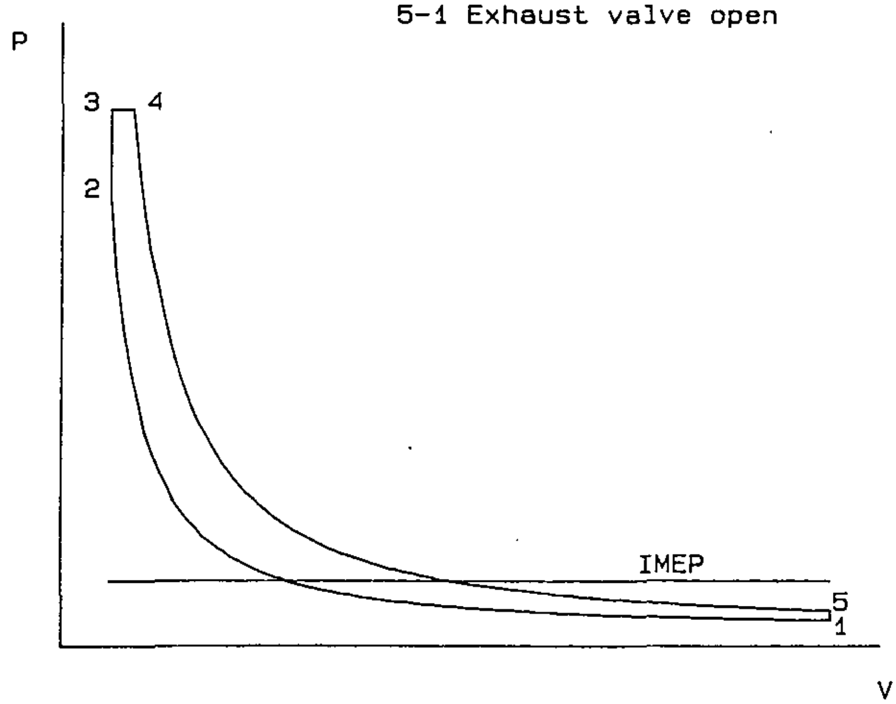


Figure 1.3: Ideal dual-combustion (air-standard) cycle
Naturally-aspirated

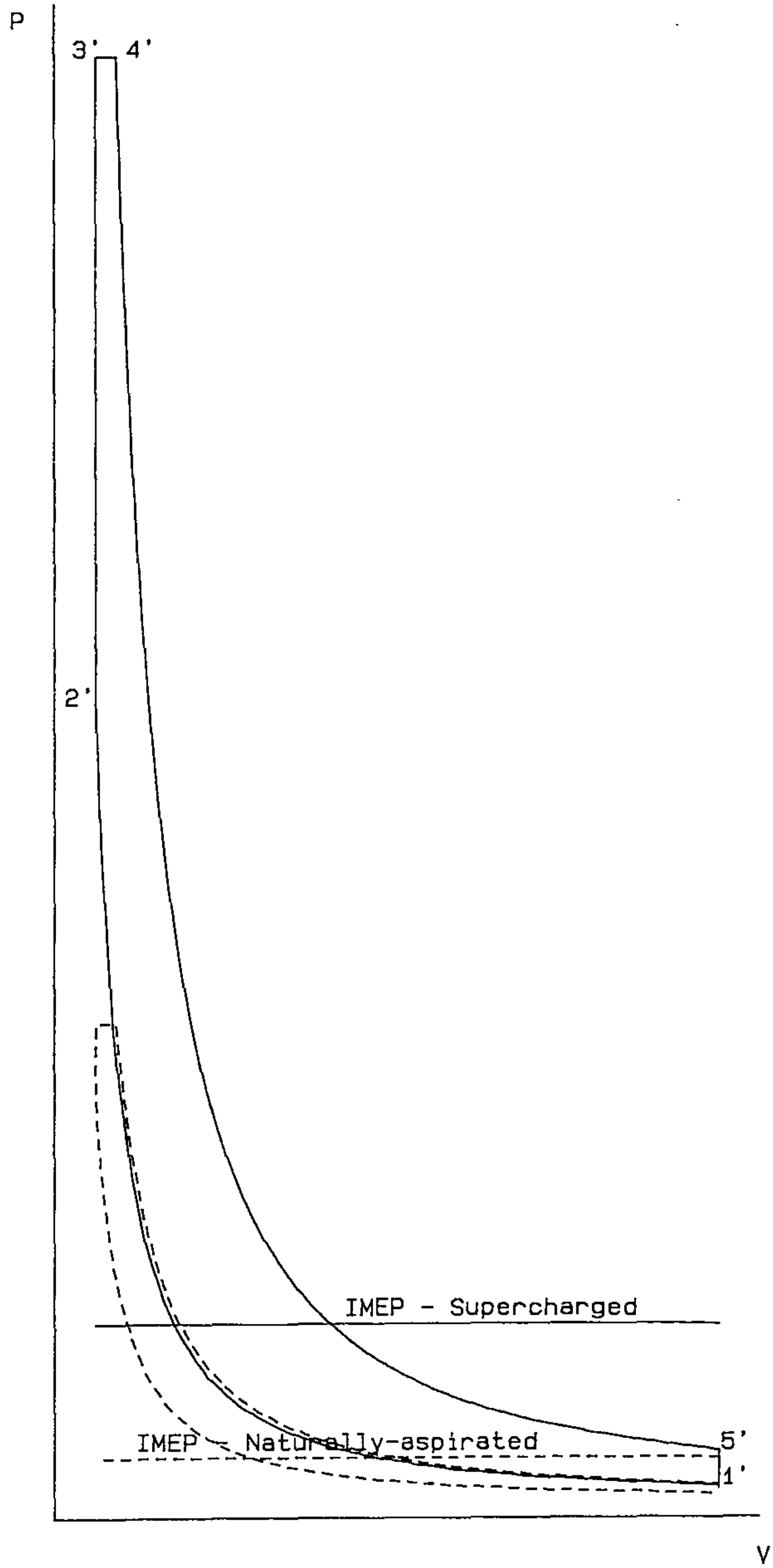


Figure 1.4: Ideal dual-combustion (air-standard) cycle
Naturally-aspirated and supercharged

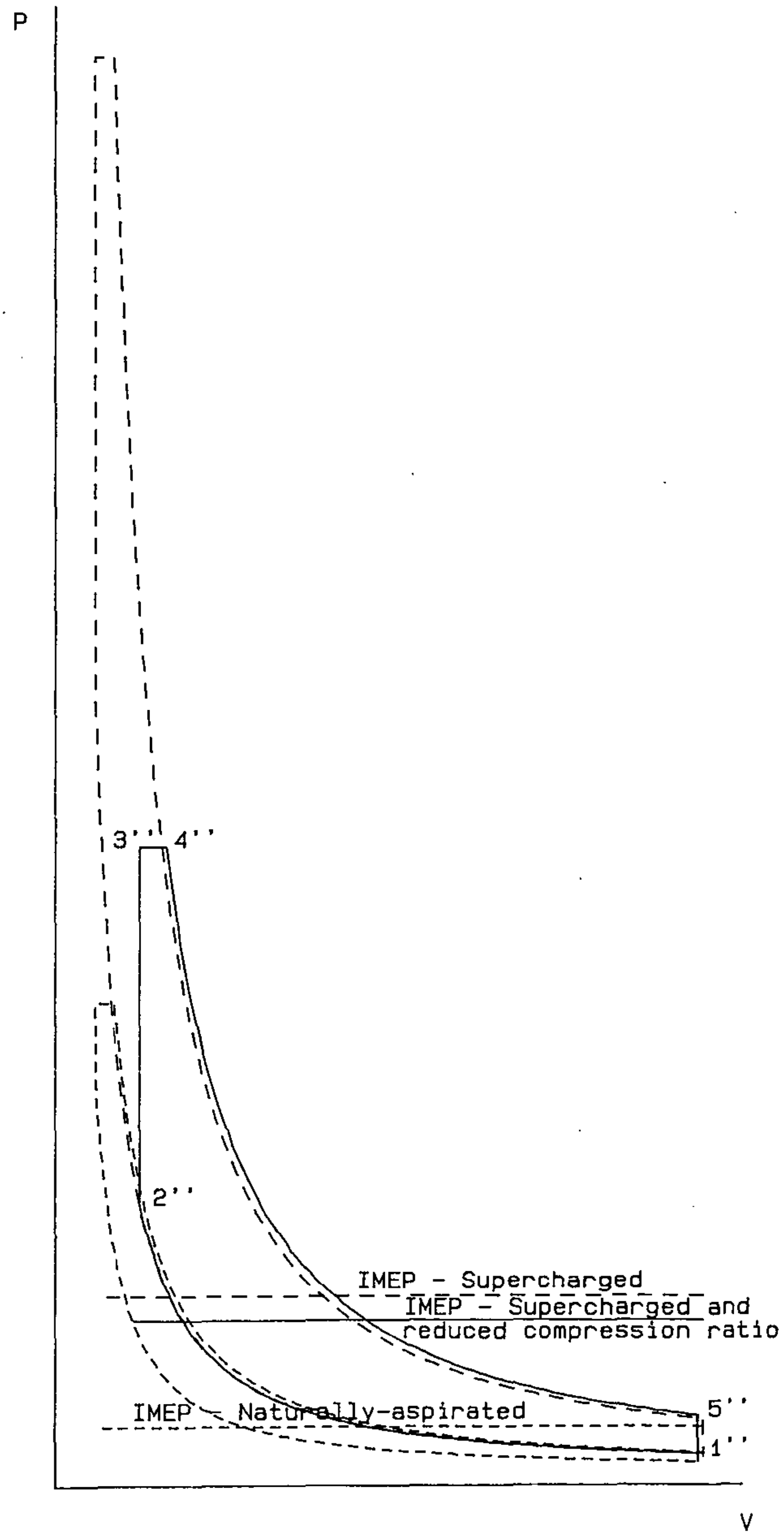


Figure 1.5: Ideal dual-combustion (air-standard) cycle
Effect of reduced compression ratio

	Naturally-aspirated	Supercharged	Supercharged lower compression ratio
Compression ratio	16:1	16:1	8:1
P_1 bar; T_1 K	1;288	2;288	2;288
P_2 bar; T_2 K	50;901	100;901	38;677
P_3 bar; T_3 K	60;1080	181;1632	80;1440
P_4 bar; T_4 K	60;1620	181;2448	80;2233
P_5 bar; T_5 K	2;612	6;612	8;1137
Cycle work	100%	200%	167%
IMEP bar	5.6	22.4	19.9

TABLE 1.1 Comparison of naturally-aspirated, supercharged and supercharged with reduced compression ratio using dual-combustion (air-standard) cycle

CHAPTER 2

ELEMENTARY THEORY

2.1 Turbocharging principles

Consider first of all the compression part of the turbocharger cycle. Figure 2.1 shows in simplified form the turbocharging process on a specific enthalpy/entropy (h-s) diagram. It is customary to neglect the heat transfer in turbomachinery processes, as these are small and difficult to measure. Therefore the compression and expansion processes are considered to be adiabatic. Process 1-2 signifies the compression part of the cycle, and process 1-2s shows the equivalent ideal compression process. The definition of the isentropic compressor efficiency is the work required for ideal compression divided by the actual work to achieve the same pressure ratio. This follows from the second law of thermodynamics from which it can be shown that, for a reversible adiabatic process, the entropy is unchanged. For a perfect gas the specific heats at constant pressure and volume do not change. For a semi-perfect gas the specific heats are functions of temperature only. It follows that the change in enthalpy is equal to the specific heat at constant pressure times the change in temperature. Hence the following equations are obtained:

η_c = specific isentropic work/specific actual work

$$= (h_{2S}-h_1)/(h_2-h_1)$$

$$c_P = dh/dT$$

$$\eta_C = (T_{2S}-T_1)/(T_2-T_1)$$

therefore the actual compressor power (absorbed) is:

$$W_C = m_C\, c_{PC}\, (T_2-T_1)$$

The expansion process 3-4 is also shown in figure 2.1 with the ideal reversible adiabatic expansion shown as 3-4s. The definition of the isentropic turbine efficiency is the actual work divided by that obtained from the reversible adiabatic expansion to the same pressure. Therefore the following expressions are obtained:

η_T = specific actual work/specific isentropic work

$= (h_3-h_4)/(h_3-h_{4S})$

$\eta_T = (T_3-T_4)/(T_3-T_{4S})$

As the turbocharger is a single-shaft machine it is straightforward to calculate the turbocharger cycle conditions by considering the energy balance between the turbine and compressor power. Therefore the compressor power absorbed is equal to the turbine power supplied plus the mechanical losses. Defining the mechanical efficiency as the ratio of useful to theoretical turbine work gives:

$W_C = W_T \ \eta_M$

and the theoretical turbine power is:

$W_T = m_T \ c_{PT} \ (T_3-T_4)$

using:

$K_C = (P_2 / P_1)^A$ where $A = (\gamma_C - 1)/\gamma_C$

$(T_2-T_1) = T_1 \ K_C / \eta_C$

and similarly:

$K_T = 1 - (P_4/P_3)^B$ where $B = (\gamma_T - 1)/\gamma_T$

$$(T_3-T_4) = T_3 \; K_T \; \eta_T$$

and rewriting the power balance equation in a more convenient manner results in:

$$K_C/K_T = T_3/T_1 \; m_T/m_C \; c_{PT}/c_{PC} \quad \eta_M \; \eta_T \; \eta_C$$

It may be seen that K_C/K_T is primarily a function of P_2/P_3 especially when P_1 and P_4 equal the ambient pressure. This ratio is effectively the pressure ratio across the engine and is a convenient way of judging the turbocharging process. It is desirable that the ratio is in excess of 1, that is P_2 exceeds P_3 by a small but positive amount. This ensures that there is a positive scavenge when both the inlet and exhaust valves are open. In addition the ratio may be used to indicate the overall efficiency of the turbocharger. In figure 2.2 various efficiencies are plotted for a temperature ratio of 3:1, using the relationship given above. A temperature ratio of 3:1 with a compressor inlet temperature of 288 K gives a typical turbine inlet temperature of 864 K (591°C). Figure 2.3 shows a similar picture for a temperature ratio of 2.7 (505°C). It should be noted that the importance of high turbocharger efficiency increases with higher boost pressures, and also with lower turbine inlet temperatures. For example, for a temperature ratio of 3:1 and for a engine pressure ratio of 1.08:1 and compressor pressure ratio of 1.75:1 an overall turbocharger efficiency of .45 is acceptable. If the pressure ratio is increased to 3.5:1 this implies an increased turbocharger efficiency of .5 is required to keep the other parameters constant. A temperature ratio of 2.7:1, for the other parameters to remain constant, implies the efficiency would have to increase from .5 at 1.75 up to .56 at a compressor pressure ratio of 3.5:1.

The above description is most applicable to constant pressure turbocharging. The early, and unsuccessful, attempts at turbocharging by Buchi utilised this system. All the cylinder exhaust outlets were connected to a large chamber and this then led to the turbocharger turbine. The object was to smooth out large changes in flow and pressure throughout the cycle. Theoretically the high pressure (point 5'' in figure 1.5) for a small part of the cycle from each cylinder is smoothed down to give a uniform pressure (P_3, figure 2.1) and flow to the turbine. Although many engines today operate with the constant pressure system, as shown above the turbocharger efficiency has to be sufficiently high to allow the operation to be successful.

A significant disadvantage of the constant pressure system is that the high kinetic energy of the exhaust-gases leaving the cylinder is largely not recovered in the lower velocity manifold. The requirement for high turbocharger efficiency, not achievable until more recently, coupled with the normal loss of energy led to the development of the pulse turbocharging system. In this arrangement the exhaust manifold is kept small enough to maintain the exhaust velocity leaving the cylinder. The turbine then receives large variations in pressure and flow. It may be appreciated that if all of the cylinders were joined to one exhaust manifold then some cylinders would discharge into other cylinders during part of the cycle. This is avoided by having multi-manifolded arrangements whereby the opening and closing of one cylinder exhaust valve is (largely) completed before another cylinder exhaust-valve opens. Generally for a 4-stroke engine, with 720° crank angles per cycle and an exhaust opening period of 240°, it may be seen that the maximum number of cylinders that can be connected to one manifold is 3. Figure 2.4 shows a number of engine cylinder groupings and it may be seen that the system, in these cases, demands turbine inlet casings with 2, 3, and 4 entries. In the pulse system the turbine operates at widely different conditions throughout its circumference, and therefore generally operates at a lower efficiency than under constant pressure conditions. However, the overall efficiency is often higher due to the retention of the pulse energy.

Although pulse turbocharging, as described above, was the most popular system a number of improvements have been introduced. The most significant of these has been the pulse-converter. In the model case two cylinders that would normally interfere are joined together by means of a junction that by virtue of its shape prevents interference. A later development has been the modular-pulse-converter in which the same pulse-converter junction is used throughout the manifold. Much detailed work has been carried out on the subject (eg. 2.1).

As mentioned the turbocharger cycle shown in figure 2.1 is a simplified view of the process. As previously calculated it is implied that the overall turbocharger efficiency accounts for the full recovery of the kinetic energy at the exit from each stage. After the compressor stage it is normal to have a restricted dump into an intercooler. This may be seen to have little regard for recovery of the kinetic energy. Therefore it was customary Napier practice for the isentropic compressor efficiency to be based upon the stage inlet total pressure and the stage outlet static pressure. In a similar manner the turbocharger exhaust exit is normally not diffused as would be required to recover the kinetic energy. Again it was Napier practice to base the isentropic turbine efficiency upon the stage total inlet pressure and the stage exit static pressure.

2.2 Radial Compressors

The turbocharger compressor stage consists of four basic components (Fig 2.5):

a Inlet casing

This can be a simple piece of ductwork, but generally consists of some type of combined filter and silencer, which is often an integral part of the turbocharger. Inevitably there will be a pressure loss in this system, often significant if there is a filter-silencer. The object of this component is to accelerate the air up to the velocity required by the next component in the stage. Although inlet guide vanes have been adopted on a number of occasions, they are not generally fitted and will not be considered here.

b Rotating impeller

This is the only part of the stage where energy transfer takes place and where the stage enthalpy is increased. The front curved entry section is called the inducer; in some impellers this is a separate component as shown in figure 2.5. At outlet from the impeller the vane is either radial or back-swept (ie. leaning backwards relative to the direction of rotation) as shown in figure 2.6.

c Diffuser

This stationary component can be vaned or vaneless. If vaned it is inevitably preceded by a vaneless space. The object of this component is to transfer the kinetic energy imparted by the impeller to the air into pressure energy by means of increase of flow area.

d Compressor outlet casing

This component transforms the flow from the separated radial direction at exit from the diffuser into a developed pipe flow. There is normally great scope for diffusion either into or within this component. The casing is usually in the form of a volute.

Velocity triangles can be used to determine the specific energy transfer within the impeller (Fig 2.6). The inlet triangle may be seen to vary with the impeller velocity which varies with the diameter. Experience has shown that it is desirable to base the inlet triangle on the root-mean-square diameter, the diameter that splits the impeller eye into two annuli of equal areas (2.2).

$$D_1^2 = (D_{t1}^2+D_{h1}^2)/2$$

At this diameter the impeller tangential velocity is:

$$U_1 = \omega D_1/2$$

$$= \pi \; N \; D_1/60$$

The basic Euler equation (2.3) describes the motion of a particle of ideal inviscid fluid along a streamline. In this case the equation represents the work, per unit mass, done by the pressure forces in changing the kinetic energy of the fluid. In words this can be expressed as: the sum of the moments of external forces is equal to the rate of change of angular momentum. This leads to (2.4):

$$W = \omega \; (U_2 \; V_{W2} \;) \quad \text{as } V_{W1} = 0 \text{ for zero pre-whirl}$$

For the ideal case the exit whirl velocity is a direct function of the exit absolute velocity and the exit blade angle. However, the air leaving the impeller normally does so at a lower angle than the blade angle. This is due to the fluid inertia and its resistance to turning. The reduction in whirl velocity, the tangential component of the absolute velocity, is reconciled by the use of the term 'slip factor'. A correlation of slip factors from many sources has been obtained by Wiesner (2.5). This results in the following definition of slip factor:

$$\sigma = V_{W2} \; / \; U_2$$

$$= 1 - (\cos \beta_2)^{\,0.5} / \; z^{0.7} \; - V_{R2} \tan \beta_2 \; / \; U_2$$

The theoretical work given above is normally smaller than the actual applied work. This is due to such losses as that caused by friction between the impeller disc and the air. A term called 'power input factor' is introduced to account for that proportion of the shaft power which does not appear as temperature rise or as momentum change. This leads to the following definition.

$$W = \tau \; V_{W2} \; U_2$$

$$= \tau \sigma U_2^2$$

The diffusers fitted to the Napier range of turbochargers are all of the vaned type. Between the impeller and the diffuser there is a vaneless space, and typically the inner diameter of the diffuser is at 1.1 times the impeller diameter. The reason for this gap is to reduce the interaction between the diffuser and the impeller. The most externally apparent effect is to reduce the noise emission. However, lesser values of this gap can reduce the mechanical integrity of the impeller (2.6). Although the action of the diffuser is complex, it is normal to consider that it behaves as a channel diffuser, and therefore the shape is generally governed by parameters such as equivalent cone-angle and area-ratio (2.7).

The elementary theory, based on the concept of one-dimensional flow compared with the ideal isentropic flow can be developed to describe all the sections within the compressor stage. Refs 2.8, 2.9, 2.10 for example present this in detail.

In order that a particular size of turbocharger can be fitted to a number of different engines it is necessary to be able to adjust the capacity of the machine by a significant amount. This is generally carried out by having different capacity sizes of the same basic impeller and within these sizes to have a range of diffuser sizes. The different sizes of impeller are achieved by having different vane lengths, this affecting the compressor channel (Fig 2.7). For example, a particular compressor may have four impeller sizes, each with ten diffusers. The number of outlets of the compressor can vary. Although there is normally only one some engines require two outlets. This is usually only necessary when a Vee engine has only one turbocharger, the two outlets feeding the two manifolds of the cylinder banks.

2.3 Axial Turbines

The efficiency of axial turbines reduce as their size becomes smaller. This is because the leakage spaces tend to be fixed and therefore become proportionally larger. At the smaller sizes the radial turbine can be far more efficient; this is why automotive turbochargers use radial turbines. As radial turbines are increased in size inertia becomes a significant problem; also manufacturing problems increase. The diameter at which the radial gives way to the axial for these reasons is of the order of 300 mm.

The turbocharger axial turbine consists of five components (Fig 2.5).

a Turbine inlet casing

As mentioned this can incorporate 1, 2, 3, and 4 entries. In addition even 6, and 8 entry casings have been manufactured. The object of this casing is to duct the exhaust gases into the turbine maintaining the separation of the exhaust pulses into the nozzle.

b Nozzle

This is the stator device for introducing the correct velocity and angle of the gas into the turbine wheel.

c Turbine wheel

This is the part of the turbine stage that extracts work from the exhaust gases. Although the majority of axial turbines have separate blades, a number of bliscs (integrally-cast blades and disc) have been introduced. The blade length is small compared with the rotor diameter, therefore it is normal to consider the conditions at the mean blade diameter.

d Exhaust diffuser

This item often incorporates the rotor shroud. In addition the component provides a diffusing channel to assist in the efficient transfer of kinetic energy at the rotor exit into pressure energy.

e Turbine outlet casing

This casing, unlike that fitted to the average gas turbine, can be a very poor diffuser. The shape of the casing is often limited so that there is insufficient space from the exhaust diffuser exit to turn the gas through 90° and mix effectively. Introduction of losses at this point can be extremely wasteful.

The velocity triangles at inlet and outlet from the rotor can be conveniently drawn together as shown in figure 2.8. This is only strictly true when the inlet and outlet mean diameters are the same. However, in most turbocharger turbine rotors this is the case. The angle α_3 is the exit 'swirl angle' and for the zero swirl condition, that is $\alpha_3 = 0°$, the exit velocity is at a minimum. The higher this velocity the more important is the efficient recovery of the kinetic energy, as this effectively increases the pressure ratio across the turbine. By considering the energy transfer to be equal to the rate of change of momentum, from the Euler equation, the following result is obtained:

$$W = \omega U_M (V_{W3} + V_{W2}) \quad \text{as } r_2 = r_3$$

This can be expressed in terms of the known velocities from the geometry of the velocity triangles if desired.

An important parameter for judging the turbine is the degree of reaction. Although the definition can vary, the term is intended to indicate the pressure drop across the rotor as a fraction of the total pressure drop across the stage. The value of reaction can be chosen and typically in turbochargers a value of 50% at the mean-diameter is used. It may be seen that the reaction will vary from hub to tip, the blade speed will vary linearly with the radius. This has led to the use of zero reaction at the hub as an alternative, or additional requirement. Although expressions can be derived for the degree of reaction (2.9, 2.10) the ratio of nozzle to rotor area can be used as an effective alternative (2.11).

In order to account for the variation in blade speed, and the required gas angles, with blade height it is desirable to use twisted blades. It is normal to consider radial equilibrium conditions apply and this has led to the use of free vortex design where the whirl velocity varies inversely with the radius (2.9). In practice the requirement for constant axial velocity across the stage may be sacrificed for ease of manufacture of the nozzle, this dictating a constant outlet angle.

2.4 Finite Element Analysis

The finite element method (FEM) was developed in the 1950's, particularly for the analysis of aircraft structures and buildings. The essential utility of FEM is the ability to model structures of complex geometry as an assemblage of simple elements. The main requirement is to have, for a range of different geometric elements, exact or approximate solutions of the governing differential equations for arbitrary boundary conditions. The most common elements used to define a structure are either of a simple triangular or quadrilateral shape. Solution of the plate and beam equations within the elements provide all the necessary capability to determine the behaviour of the whole structure. Many computer codes exist with large element libraries to enable even the most complex structure to be analysed.

The points at the extremities of the elements are called nodes and the co-ordinates of these nodes fully define the geometry of the structure. Elements are described by the nodes they connect. This data together with the element properties fully define the structure. By specifying the loads acting at particular nodes the computer code is provided with all of the necessary information to solve the problem. The type of elements, in most computer code libraries, include two-dimensional, axisymmetric, and three-dimensional triangular and quadrilateral elements. A range of analyses usually can be carried out including stressing, natural frequencies, and thermal.

Prior to the introduction of the FEM many of the complex items in a turbocharger that are required to be stressed were difficult to analyse. In particular the stresses that would exist in the rotor discs at the normal operating speeds had to be determined. Traditionally the Donath method (2.12) was used. For discs of symmetric cross-section that varied in thickness by relatively small amounts this method gave reasonably adequate results. However, for rotor discs that did vary significantly and were not symmetrical a better analysis method was used (2.13, 2.14). This method was used during the 1960's until the FEM became the standard means of analysing rotor discs. Even with the general introduction of FEM the methods of analysing such complex items as the compressor impeller were not standardised. It may be seen the the impeller consists of a hub which can be simply modelled as a two-dimensional section and analysed using axisymmetric elements. However, the effect of the vanes are far more difficult to model. In addition to their centrifugal loading on the hub, which can be easily modelled, they play a significant part in the structure of the disc. A method was proposed (2.15) to reduce the problem into two parts. Firstly the vane was modelled as a series of plate elements and the outermost boundary of the hub as part of a ring. Secondly the hub was modelled with axisymmetric elements with a series of loadings to simulate the vanes. The object was then to adjust the loadings on the second model until the deflections of the two models were the same. It then being assumed that the hub stressing would be predicted with sufficient accuracy. The main assumption was that the vane was dictating the final deformation of the disc. The simpler alternative was to assume that the vanes did not contribute to the impeller structure, and use a axisymmetric analysis. Another approach was to model the vanes as a solid continuation of the hub but with a very low modulus of elasticity in the circumferential direction. Essentially the best method to be adopted depends upon the particular shape of impeller and the FEM codes available. However, almost all of the ways adopted to stress radial compressor impellers can be seen to have limitations.

2.5 Market Research

Market research is a particular activity within the marketing process. Marketing is a much misunderstood term, often considered as an activity of selling. In actual fact selling is also part of the marketing process. A 'market' can be defined as where goods are bought, sold, or exchanged. This is as true of the turbocharger market as any other. Marketing can be defined as the 'management process responsible for identifying, anticipating, and satisfying customer requirements profitably' (2.16). In order for a market to exist it is necessary to have a buyer, with the money to buy, and a manufacturer with a suitable product.

The market covers the group of people who are potential customers. The more a company knows about its market the better it can plan how it should be organised. In order to improve its understanding of the market it is necessary to carry out market research. The object is to arrive at an accurate assessment of the present and future market for the product. The sort of questions that have to be answered include:

a. How many buyers of the product are there?

b. Where are the buyers located?

c. When do they decide to buy the product?

d. What are the buyers most important requirements?

e. What is the nature of the competition?

f. What improvements to the existing product are needed?

g. How large is the market of each buyer?

h. What is the life cycle of the buyers product?

Successful market research depends upon the right people asking the right questions of the right people. In addition the company itself has much useful information available as a result of previous sales figures, and information relating to the customers and competitors' products. As a result of market research a company can work out what its share of the potential market might be and from this a sales forecast and sales targets can be produced. These figures can then be used to determine the immediate policies of the company. In addition, dependent upon the design cycle of the product, the information can be used to determine the forward technological improvements that will have to be incorporated into the product.

The purpose of the market research as far as the designer is concerned is to enable him to know what product should be designed. The functional requirements of the product must be clear. For example, it must be known how efficiently it should perform, and how long it should survive. In addition the position within the market into which the product should fit should be well understood . A 'Rolls-Royce' product may perform admirably; however, the product will not flourish if the market requires a 'Mini'. The product life is a term used to denote the transition of the product from introduction to obsolescence, and should not be confused with the service life. If the introduction of the product takes a large percentage of the product life the chances of success will not be high.

The estimation of the stages of the product life are most important in deciding the profitability of the product. A company makes a profit over a period of time if it earns more than it spends. The expenses can be divided into those that vary with the level of production and those which are incurred for the company to exist. In the introduction stage of a product such as a turbocharger there will be a considerable amount of expense on such items as design, tooling, patterns, and materials. Only by making accurate estimates of all of the sales and expenses will a company know if it will make a profit.

2.6 Project Control

One of the most well-known methods of project control is PERT, program evaluation and review technique. PERT may be defined as a manager's tool for defining and coordinating what must be done to accomplish successfully the objectives of a project on time. It provides a means of presenting knowledge about the uncertainties in the many diverse activities required to complete a program to a pre-determined schedule (2.17). An event is described as the start or completion of an activity. The sequence is the activities that properly follow one another. A network is a combination of all of the events and activities in their correct sequences. Estimates of times can be broken down into optimistic, most likely, and pessimistic. The slack in the project can be positive, zero, and negative. PERT techniques can be applied to a wide range of projects from digging holes to building battleships.

Critical path network analysis is also used to control projects, generally by the use of computer programs. In this technique, it is general to have activity on node rather than as above with an activity between nodes. The network is built up by consideration of 'immediately preceding activities'. Also the earliest and latest start times throughout the network are calculated. A critical path is defined as when the earliest start time coincides with the latest start time for an activity. Thus any delay in the start of the particular activity will cause a delay to the project. Activities not on the critical path will have earliest start times which are earlier than the latest start times. The difference between the two is called the float, and is an indication of the time that the activity may be delayed without affecting the overall timescale.

Resource-limited scheduling introduces the concept that a project is often limited more by resources than by planning and control. On many occasions there are far more activities than people to complete them. This introduces extra complexity into the methods outlined above. In addition to an activity being dependent on the preceding activity being completed it becomes additionally dependent on somebody being available to carry it out. An algorithm was developed that allows this feature to be accommodated. This algorithm will be described in more detail at a later time.

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FIGURES

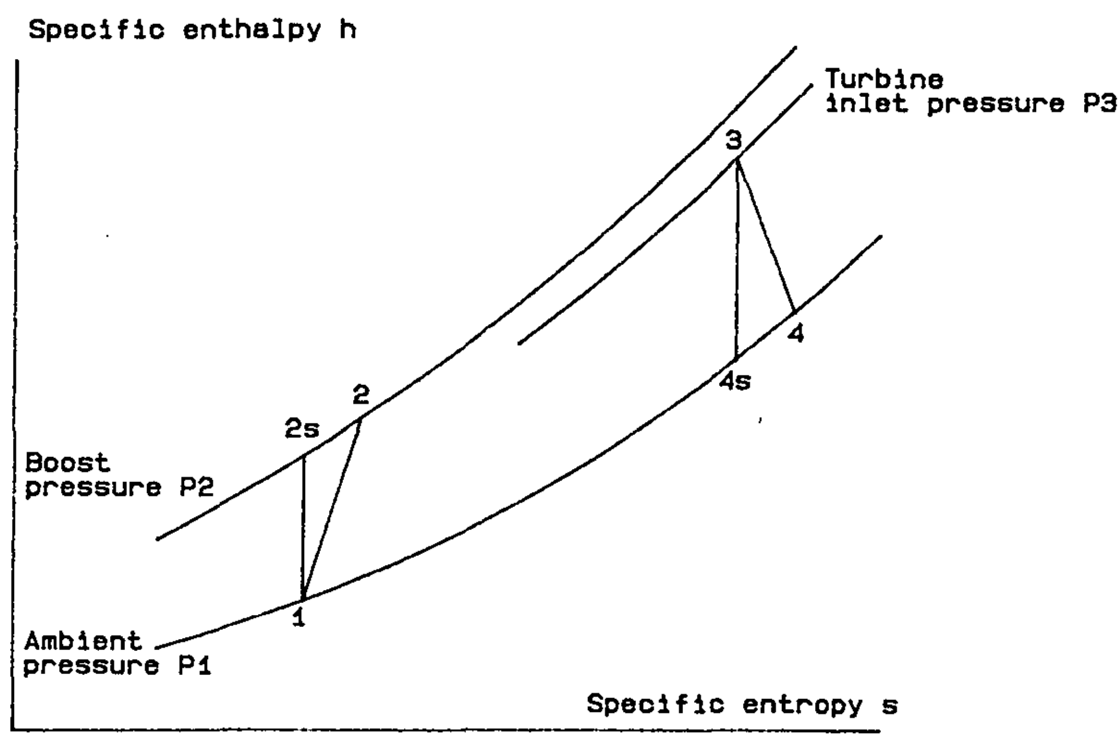

Figure 2.1: Turbocharger cycle plotted on h-s diagram

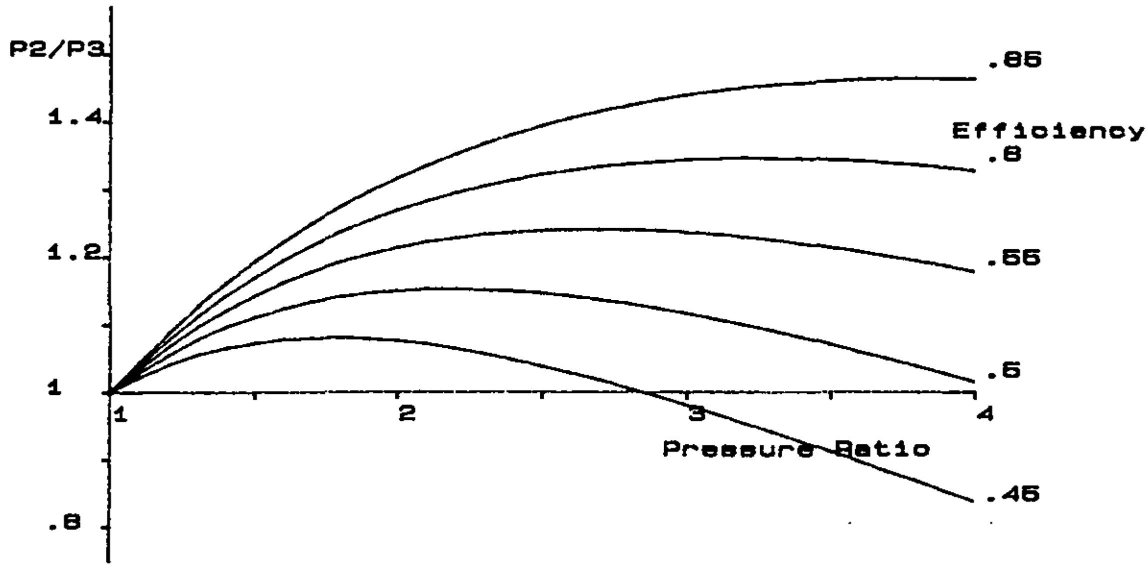

Figure 2.2: P2/P3 against compressor pressure ratio for T3/T1=3

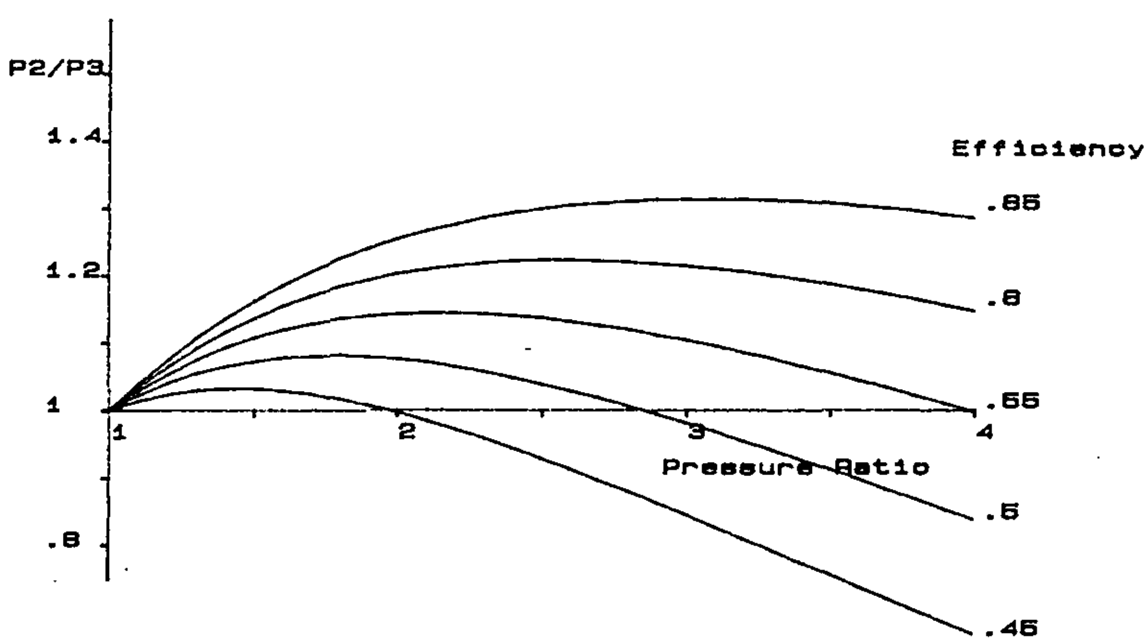

Figure 2.3: P2/P3 against compressor pressure ratio for T3/T1=2.7

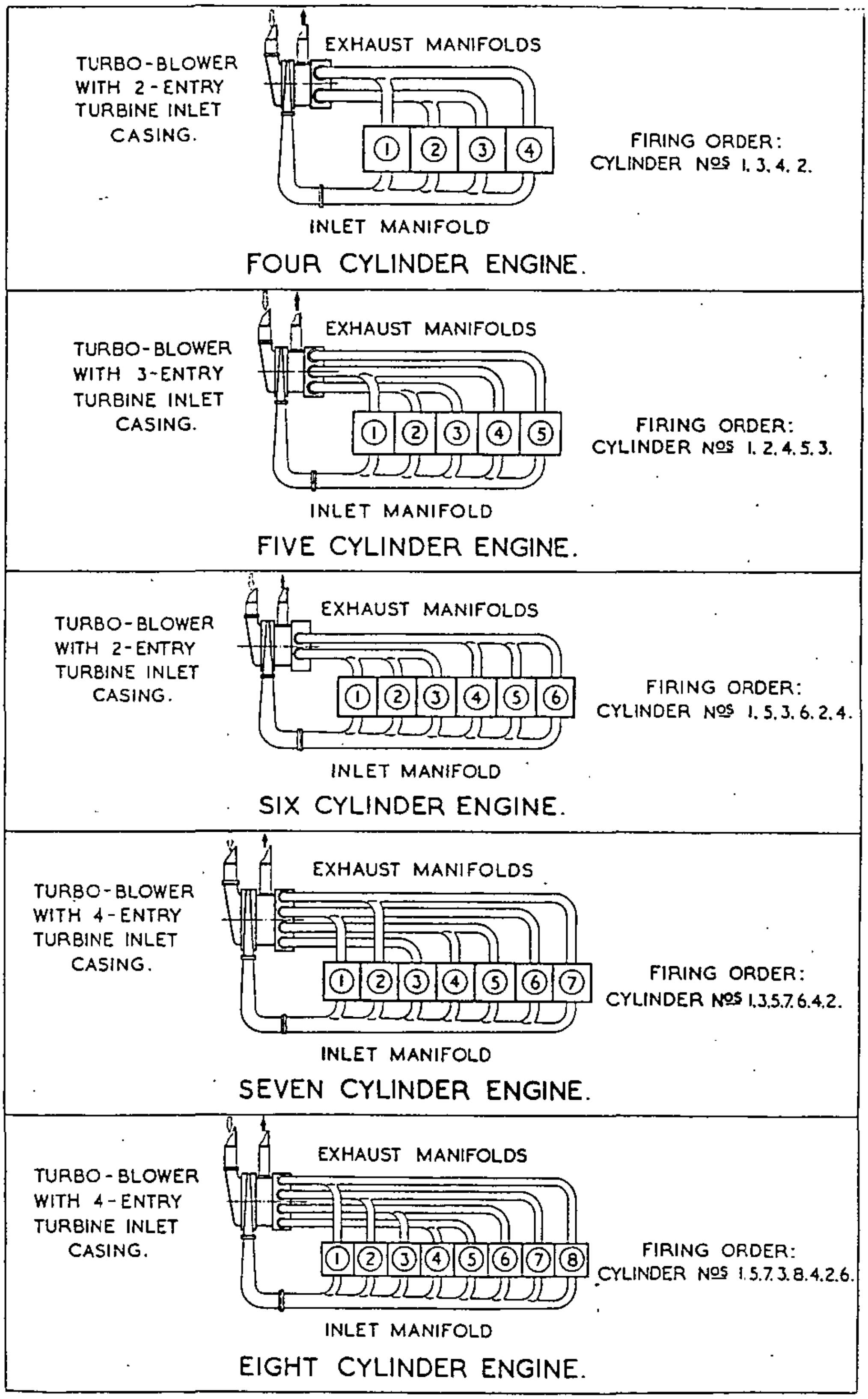

Figure 2.4: Pulse turbocharging

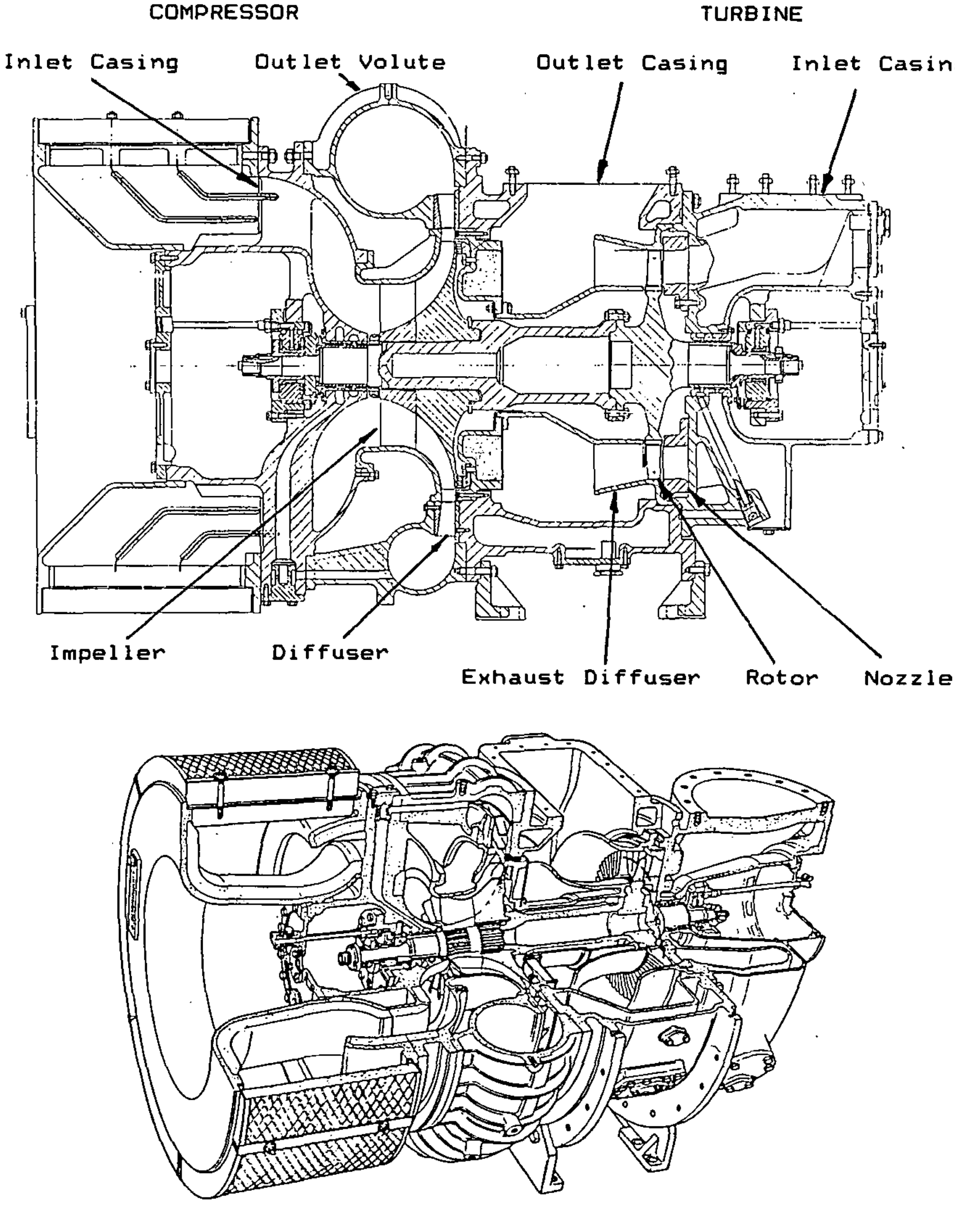

Figure 2.5:Napier Large Turbocharger

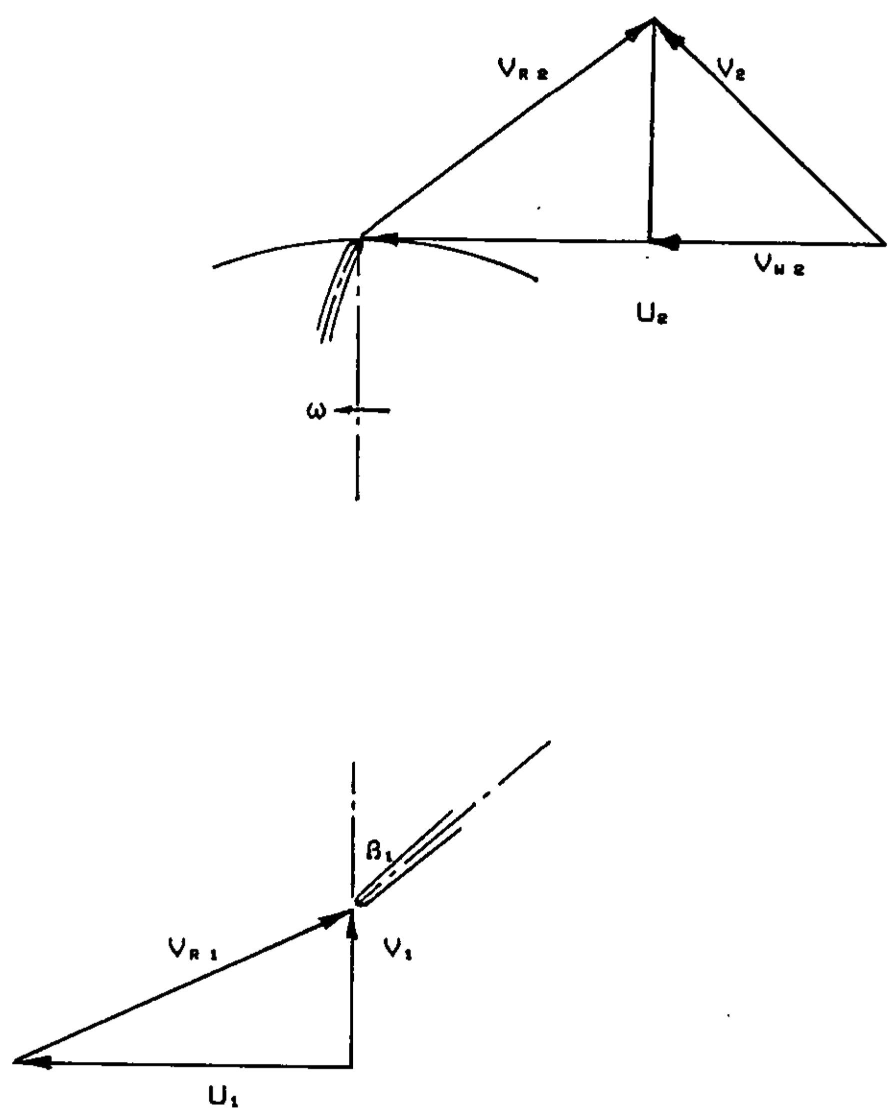

Figure 2.6:Compressor inlet and outlet velocity triangles

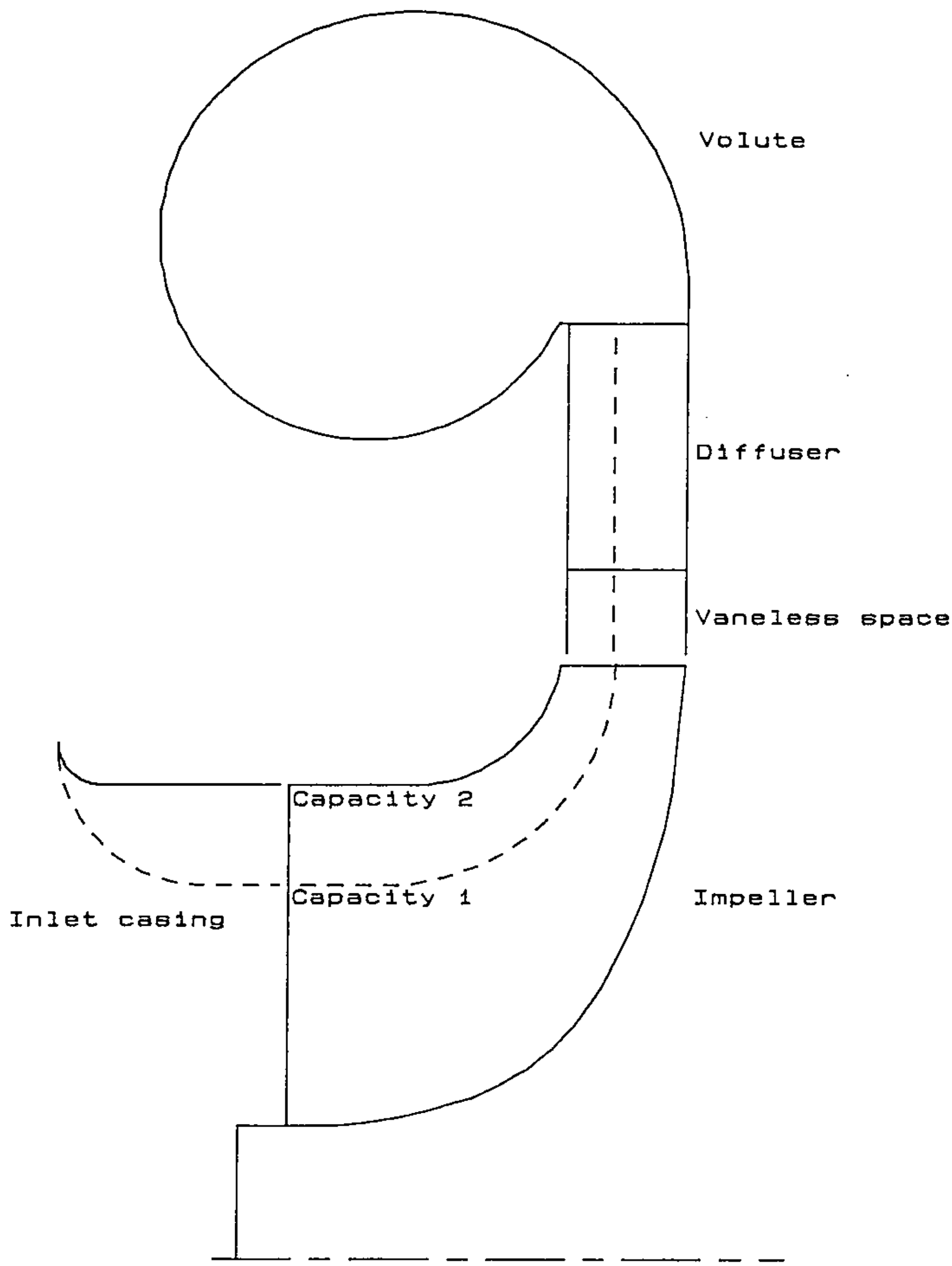

Figure 2.7: Compressor channel

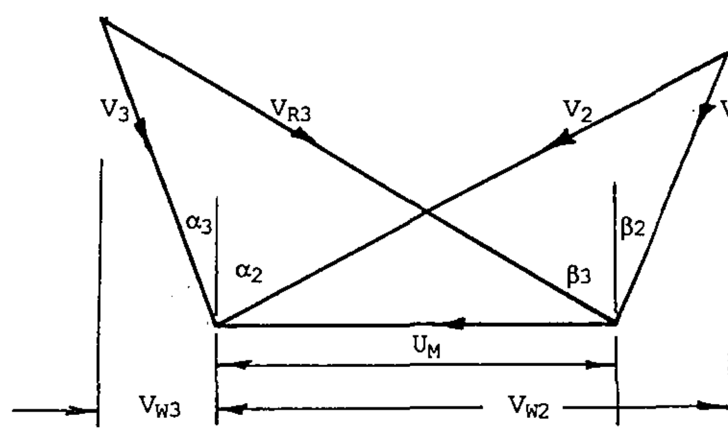

Figure 2.8: Turbine rotor inlet and outlet velocity diagrams

CHAPTER 3

PERFORMANCE AND DESIGN

3.1 Compressor Performance

3.1.1 Compressor performance representation

The performance of a centrifugal compressor may be specified by lines of constant rotational speed plotted against boost pressure and mass flow. However, this would be inconvenient as the curves so produced would vary with the temperature and pressure at entry to the compressor and the physical properties of the fluid. By considering the various quantities that will influence the behaviour of the compressor the following solution can be obtained:

$$\text{function}(D, N, m, P_1, P_2, RT_1, RT_2) = 0$$

The technique known as dimensional analysis is used to combine some of the variables into non-dimensional groups. This results in the following list of combined quantities (3.1):

$$P_2/P_1,\ T_2/T_1,\ m(RT_1)^{0.5}/D_2{}^2P_1,\ ND/(RT_1)^{0.5}$$

Using these quantities the performance of a compressor can be expressed in non-dimensional terms. The expressions are normally simplified as the performance of a particular size of compressor of a specified gas is generally required. The following resulting equation is obtained:

$$\text{function}\ (P_2/P_1, T_2/T_1,\ m\sqrt{T_1}/P_1,\ N/\sqrt{T_1}\ ) = 0$$

Although the pressure ratio term is non-dimensional, the capacity function $m\sqrt{T_1}/P_1$ and the speed function $N/\sqrt{T_1}$ are not truly non-dimensional.

However, these parameters have become the most useful and generally accepted way, at least within Napier, of expressing the performance of a centrifugal compressor. It has been shown that the isentropic efficiency is a function of P_2/P_1 and T_2/T_1 (section 2.1) and this parameter is used to replace the two non-dimensional quantities. Lines of constant efficiency are generally added to the lines of constant speed function using axes of pressure ratio and capacity (function). Figure 3.1 shows a typical centrifugal compressor characteristic expressed in terms of the above parameters.

The way in which the compressor operates along a constant speed function line may be described as follows. When the delivery is not restricted the compressor is operating in 'choke'. This means that the compressor channel is operating at a maximum flow condition and the gas at some station is at sonic velocity. In figure 3.1 the compressor will be operating at the lowest pressure ratio and maximum flow on the speed line. The position in the compressor channel that controls the flow is generally either the inducer or the diffuser. If the compressor delivery is throttled the operating point will move up in pressure ratio and to a reduced capacity, as may be seen in figure 3.1. The pressure ratio will reach a peak and then start to reduce until the line called the surge line is reached. Compressor surge is a complex phenomenon. However, the effect is that the compressor cannot be operated at capacities below the surge point on a particular speed line. Surge is a reversal of flow caused by an adverse pressure gradient at one or more of a variety of positions in the system. The effect can be local, or can affect the stage as shown by the surge line in the figure. At this point the flow within the stage will abruptly reverse, often accompanied by an audible signal. The characteristic can be used for judging and comparing the performance of a centrifugal compressor in a convenient manner.

3.1.2 Compressor performance prediction

For two main reasons it is desirable to be able to predict the performance of a turbocharger compressor. Firstly, it is undesirable to commit the design to manufacture before the expected performance is estimated. Secondly, even when the major components are available, testing the whole range of possible compressors would be extremely time consuming and expensive. As previously mentioned a particular turbocharger frame-size may have forty combinations of impellers and diffusers and a choice of two delivery casings. Testing every compressor combination is clearly an impracticable proposition. The most well-known method of compressor performance prediction method is that proposed by Rodgers (3.2), and this was used at Napier particularly for assisting with the matching of the turbocharger to an engine.

The method of radial compressor performance prediction adopted is based on the assumption that the losses within the stage can be assigned to two sections. These sections are the inducer and the diffuser, and the functions are related to the section area and peak efficiency. In addition the inducer root-mean-square section geometry characterizes the inducer section. The compressor characteristic can be constructed using one-dimensional flow analysis with the combination of these two loss models. The loss models for the inducer and diffuser were determined using extensive test data available within Napier. These loss models allow the shape of the constant speed lines to be predicted. Although the loss models produced could be used for all the Napier compressors some knowledge of the expected peak efficiency has to be determined before the correct level of performance could be predicted.

As the inducer loss model was determined empirically by comparison with a great number of actual characteristics, a form of expression that was convenient to modify was chosen. Firstly the expression for the flow parameter at the peak efficiency was determined:

$$\phi_{1P}/\phi_{1M} = (U_2/\sqrt{T_1} + 5.64)/36.98$$

where:

ϕ_{1P} = the impeller flow parameter corresponding to the peak efficiency

and

ϕ_{1M} = ϕ_1 critical at inducer throat

where:

$\phi_1 \quad = \quad (V_1/\sqrt{T_1}) \ / \ (U_2/\sqrt{T_1})$

The values of efficiency above and below the flow corresponding to the peak efficiency were calculated by the following expressions:

Above the peak efficiency flow;

$\eta_i/\eta_{IP} \quad = 1 - 0.4 \ ((\phi_1/\phi_{1M} - \phi_{1P}/\phi_{1M}) \ / \ (1 - \phi_{1P}/\phi_{1M}))^P$

where

$P \quad = 6.08 \ \phi_{1P}/\phi_{1M}$

Below the peak efficiency flow;

$\eta_I/\eta_{IP} = 1 - (\phi_{1P}/\phi_{1M} - \phi_1/\phi_{1M})^2 \ / \ 5$

Typical values of the impeller characteristic are plotted in figure 3.2.

The diffuser characteristic was determined, in conjunction with the impeller characteristic, from the wealth of Napier compressor test data. Figure 3.3 shows the derived characteristic where the flow parameter is given by:

$R_{WD} = m\sqrt{T_2}/A_D P_2 \ / \ (m\sqrt{T}/AP)_{CRIT} \ / \ B_{LD}$

As may be seen the inducer characteristic is directly related to the critical flow parameter at the inducer throat. For this to be determined the inducer throat area is required. In the case of an established design this should be available from the geometry. However, for a new compressor an estimate can be made as follows:

$A_{T1} \quad = \quad A_1 \cos \beta_1 \ B_{LI}$

Two methods were used to determine the choking mass flow. Firstly assuming the flow to be one-dimensional and no losses occur between the inducer inlet and the throat gives:

$$T_{R1} = T_1 + U_1^2/2c_P$$

$$P_{R1} = P_1(T_{R1}/T_1)^{((\gamma-1)/\gamma))}$$

$$(m\sqrt{T}/AP)_{CRIT} = (\gamma/R)^{0.5}/((\gamma+1)/2)^{((\gamma+1)/2(\gamma-1))}$$

$$m_{CRIT} = (m\sqrt{T}/AP)_{CRIT}\, A_{T1}\, P_{R1} \,/\, \sqrt{T_{R1}}$$

Secondly, experimental two-dimensional cascade data (3.3) were used to calculate the relative Mach number directly. Figure 3.4 shows the experimentally determined relationship between the choking Mach number with throat width/inlet width.

The inducer stall follows the method set out in Ref 3.4 using two-dimensional cascade work to relate deflection and incidence to nominal deflection in a dimensionless manner(3.5). The non-dimensional curve of deflection against incidence parameter gives at the peak deflection a value of incidence parameter of 0.4. This value is often used as a design value of stall, however, a higher value of the incidence parameter has been used to give a stall limit:

$$i_S = 0.52\,\varepsilon^* + i^*$$

therefore

$$i_S = 1.52\;\varepsilon^* + \alpha_2^* - \beta_1$$

where

$$\varepsilon^* = \alpha_1^* - \alpha_2^*$$

and

$$i^* = \alpha_1^* - \beta_1$$

using the nominal deviation relationship in Ref 3.3:

$$\delta^* = m\,\theta\,(s/c)^{0.5}$$

where

$$m = .23\,(2a/c)^2 + \alpha_2^*/500$$

and noting that when the outlet angle is zero:

$$\delta^* = \alpha_2^*$$

$$\theta = \beta_1$$

$$\alpha_2^* = (.23\,\beta_1\,(s/c)^{0.5}\,(2a/c)^2)\ /\ (1 - \beta_1\,(s/c)^{0.5}/500)$$

For a typical value of pitch chord ratio and the value of maximum camber applicable to a parabolic camberline the above expression simplifies to:

$$\alpha_2^* = 0.234\,\beta_1/(1 - 0.0014\,\beta_1)$$

The curve of nominal deflection against outlet angle from Ref. 3.5 is shown in figure 3.5. Thus, having calculated the nominal outlet angle, this figure can be used to determine the nominal deflection and therefore the limiting stall incidence. The value of stall incidence is reduced with increased velocity and the correlation in Ref 3.5 was used to obtain the following approximation:

$$\Delta i_S = (M_{R1}{}^4 - 0.026)/0.13$$

Combining the value of limiting stall incidence and the reduction due to increased velocity enables the stall incidence to be obtained. This value enables all of the stall parameters, stall flow for example, to be determined.

A surge correlation was determined using, in the main, characteristics from non-backswept compressors. The range of the characteristic between surge and choke was considered with respect to the relative Mach number at the impeller tip. The result is shown in figure 3.6. In order to define choke in a uniform manner a value of choke efficiency was chosen as a fixed proportion (0.86) of the highest attained efficiency. It was intended that the most damaging effects of inducer choke were not included in this correlation. This would have reduced the surge-to-choke range by an apparently arbitrary amount. However, it was considered desirable to account for this reduction in the prediction. An adjustment was introduced as a function of the impeller efficiency at flow parameters above that for the peak efficiency value. The majority of measured characteristics available at the time of this work related to non-backswept compressors. However, by considering a relatively small number of backswept compressor characteristics a simple means of increasing the range as a function of the backsweep angle and number of diffuser vanes was chosen. The resulting surge prediction is as follows:

surge capacity/choke capacity

$$= K1 \, . \, K2$$

where $K1 = -0.333 \, M_2{}^2 + 1.233 \, M_2 - 0.04$

and $K2 = (0.11 \, \beta_2{}^2 - 0.14 \, \beta_2 + 1) \, (0.0007 \, N_D + 0.975)$

A computer program in Fortran was written with an added plotting program. Figure 3.7 shows the resulting prediction for the compressor to which the test results shown in figure 3.1 relate.

Although newer and more elegant prediction techniques have been presented (eg. Ref 3.6) the method described has the merits of simplicity and of being easily tuned to give reasonable accuracy.

3.2 Compressor Design

In designing a new turbocharger to suit a variety of target engines, certain relevant data are obtained from a knowledge of these engines. For example, the expected minimum to maximum range of the compressor will be determined from a knowledge of the specific air consumption of the target engines. In most cases this will mean that the largest capacity compressor will be known. The preceding section presents a method for determining the performance of the compressor. The computer program was used as a rapid method of assessing the ranges that a particular compressor family would achieve. Figure 3.8 shows the range of compressor capacities for the four different sizes of the same compressor. The prediction program required only the basic scantlings of the compressor to enable the preliminary design of the compressor to be carried out. In order to design the impeller channel it is necessary to have an adequate understanding of the aerodynamics of the compressor wheel and diffuser.

3.2.1 Impeller vane design

The backswept vane is a complex three-dimensional shape, unsuited to representation on a two-dimensional drawing. Aerodynamic analysis of the channel would require a great deal of complicated and tedious geometry manipulation if the vane was represented by a drawing. A computer-based procedure was therefore used to provide a complete definition of the vane (3.7). The vane geometry is described by a family of three-dimensional surfaces or 'patches' as shown in figure 3.9 (3.8). Each patch consist of an infinite number of cubic curves running from leading to trailing edge of this patch, and a series of straight lines from hub to tip. The geometric model required the meridional dimensions of the hub and shroud and the patch corner co-ordinates (Fig 3.10). In addition the vane angles at inlet and exit and assumed meridional angles at hub and shroud were required. Initially a vane camber angle, and circumferential vane thickness distributions were specified based upon previous experience. Also the vane taper ratios at leading and trailing edges were chosen using values that had proved to be satisfactory. With these data the computer program was able to produce a fully three-dimensional representation of the vane.

The method adopted to analyse the flow within the impeller was based upon that proposed by Katsanis (3.9). The vane definition described above was manipulated to provide the data for the analysis. The flow is assumed to be axisymmetric. It is further assumed that the axisymmetric flow may be represented by a mean stream surface that is a stream surface being conceptually the average of the flow across a vane passage. A two-dimensional solution of the equations of motion for the flow within the mean stream surface is obtained. A correction for a loss in relative total pressure was included in the continuity equation to account for blade losses. In general the mean flow surface is assumed to be parallel to the mean blade surface. From this stream surface solution an approximate calculation of the velocity at the blade surface is made. The calculated relative velocities over the vane surfaces were considered with respect to various criteria. Violation of the criteria necessitates revision of the vane shape and an iterative process is followed until an acceptable compromise is obtained.

When the iterations involving the vane-geometry and the aerodynamic analysis had produced a satisfactory vane-shape the data was manipulated to enable a finite element (FE) mesh to be defined. The stresses calculated by an FE program could necessitate further iterations, involving thickness and taper ratio schedules, to obtain satisfactory stresses. Although this may mean that the aerodynamic analysis had to be repeated further iterations involving changes to the vane geometry were not usually necessary.

The vane geometry was manipulated to provide cutter-path definitions using the straight-lines between the hub and tip. The latter ensures that the vane can be flank-milled with a straight-sided cutter. A computer controlled 5-axis milling machine produces the vane shape from a solid billet of forged aluminium. The workpiece can be rotated, tilted, raised, moved sideways, and backwards and forwards. The vane was completely machined with roughing and finishing programs thus eliminating costly hand-finishing. Computer controlled machining also ensured that the vane shape was repeatable. The space between the vanes was removed by clearance routines. However, these could not be the most efficient in terms of metal removal. Instead the machining had to follow the meridional path thereby approximately following the streamlines at the hub. By this means the cusps produced by the milling process could be left, again ensuring that hand-finishing was avoided (Fig

3.11). The straight-sided cutter was coned to ensure that cutter deflection, whilst machining, was reduced as much as possible. The maximum cutter dimension had to be limited by interference with the next vane during machining.

3.2.2 Diffuser design

The vaned diffuser is an important element in the compressor stage. It has to achieve a high proportion of static pressure recovery and also largely control the efficient operating-flow range. The design has to be a compromise between good aerodynamic performance over a wide range of throat areas and ease of manufacture. The pressure recovery after the impeller is achieved in three areas:

i The vaneless space - the radial gap between the impeller outside diameter and the diffuser vane leading edge.

ii The semi-vaneless space - the space between the diffuser vane leading edge and the diffuser channel throat.

iii The diffuser channel - the channel formed between every throat and the diffuser trailing edge.

Recovery in the vaneless space is a function of the area change due to the increasing annulus area. The channel performance can be obtained by reference to extensive empirical data (3.10). Greater pressure recovery is achieved from larger area ratios through the channel, within the limits of satisfactory divergence angle. This directs the designer towards the use of higher vane numbers, as these can more easily obtain higher area ratios. However, two factors oppose this direction. Firstly, the semi-vaneless space has a great effect upon the tendency to promote local stall and subsequent surge. Generally the surge flow is reduced with lower numbers of vanes; this is extremely advantageous as the operating range of the compressor is therefore increased. It has been shown that low values of vane numbers have a beneficial effect on the geometry of the vaneless space (3.11). Secondly, high vane numbers imply lengthy and expensive manufacturing times. These factors necessitate a move towards low vane number diffusers that have large area ratios whilst maintaining optimum divergence angles. The design of diffuser families was accomplished on a desk-top microcomputer using the three-dimensional channel optimization technique recommended in reference 3.11. Starting with the inner and outer diameter constraints and the area range required, the program calculates the vane number and shape for the required range of diffusers.

3.3 Turbine Performance

The performance of an axial turbine can be represented using the same pressure ratio against mass flow parameters as the centrifugal compressor. However, the characteristic reduces to almost a single curve, often called the 'swallowing capacity curve'. In figure 3.12 seven of these curves are shown for the different turbine sizes for the Napier SA105 turbocharger. At higher pressure ratios the mass flow parameter becomes almost constant as it reaches the choking flow due to the gas attaining sonic velocity. Generally the nozzle provides the main flow control, the gas being sonic at the nozzle throat. If the nozzle was the only flow control then the curve would clearly be totally independent of rotor speed. The turbine can be considered as two nozzles in series and the two throat areas can then be combined to give the 'equivalent area'. This can be used to scale the swallowing capacity curves for the range of nozzle areas. To a first order of accuracy:

$$A_E \quad = \quad (A_N\ A_R)\ /\ (A_N^2 + A_R^2)^{\ 0.5}$$

Clearly as the swallowing capacity curve is almost a single line it is necessary to plot the efficiency on a separate diagram. This can be represented as shown in figure 3.13 where the efficiency is plotted against velocity ratio. This latter is the rotor blade speed, at mean blade height, divided by the velocity equivalent of the isentropic enthalpy drop across the turbine stage (3.1). Lines of constant pressure ratio are drawn on the map. This method of representation is useful when considering the matching of the turbine to the compressor wheel sizes, to ensure that the turbine will operate near the peak efficiency. Unfortunately, when pulse turbocharging is considered the operating point will move through the peak efficiency. The blade speed will remain substantially constant whilst the enthalpy drop will be high when the cylinder is exhausting into the turbine and will fall during the interval between pulses. For a particular turbocharger when tested under constant pressure conditions the variation in the parameters such as velocity ratio are usually so small that the performance is represented in a different manner (Fig 3.14). It is convenient to use the expansion ratio as the common axis, with the swallowing capacity, non-dimensional speed, and efficiency plotted as shown.

3.4 Turbine Design

The starting point for a turbine design is a mean-radius calculation of the gas properties and flow angles, derived from the power requirements. The next step is to determine the expected performance. Both the design and off-design performance of an axial turbine can be predicted by the use of a well established method(3.12). This is based upon a one-dimensional mean-radius technique and the correlation of the various aerodynamic losses that will be incurred. These include profile, secondary, and tip clearance losses. The method has limitations, and the lack of hub and tip information can be a serious drawback. For this reason two-dimensional axisymmetric-flow calculation methods have been introduced. The most widely-used method is the streamline-curvature throughflow method (3.13). This calculates the flow conditions at a large number of radially spaced grid-points lying on stations both upstream and downstream of the nozzle and rotor. A simplified technique was used at Napier that combined the mean-radius and streamline-curvature approach (3.14). In this method the axisymmetric-flow equations at only three radial stations were solved instead of the customary nine or more (Fig 3.15). The object was to provide an adequate, and rapid, method of performance prediction over the whole range of possible turbine operating maps.

For blade-profile design the technique used at Napier was the Prescribed Velocity Distribution method (PVD) (3.15). This program, operating on a desk-top computer, allowed the velocity distribution around the blade profile to be chosen. The calculation then determined the blade profile that was required to obtain the velocity distribution. The flow conditions at hub, mean, and tip were chosen to enable a satisfactory blade to be obtained. Iteration, between the velocity distribution and blade profile, was necessary to obtain a satisfactory shape that would satisfy all of the criteria. However, this was found to be not too arduous as the process time to produce a new blade profile was only a few minutes. The parametric spanwise interpolation between the three sections used a quadratic form which was found to be sufficient for the requirements. This method of representation allowed the manufacturing geometry to be easily obtained. Off-design performance can be also be obtained, the calculated blade-profile can be analysed to determine how the velocity distribution is affected by variations in incidence (3.16).

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FIGURES

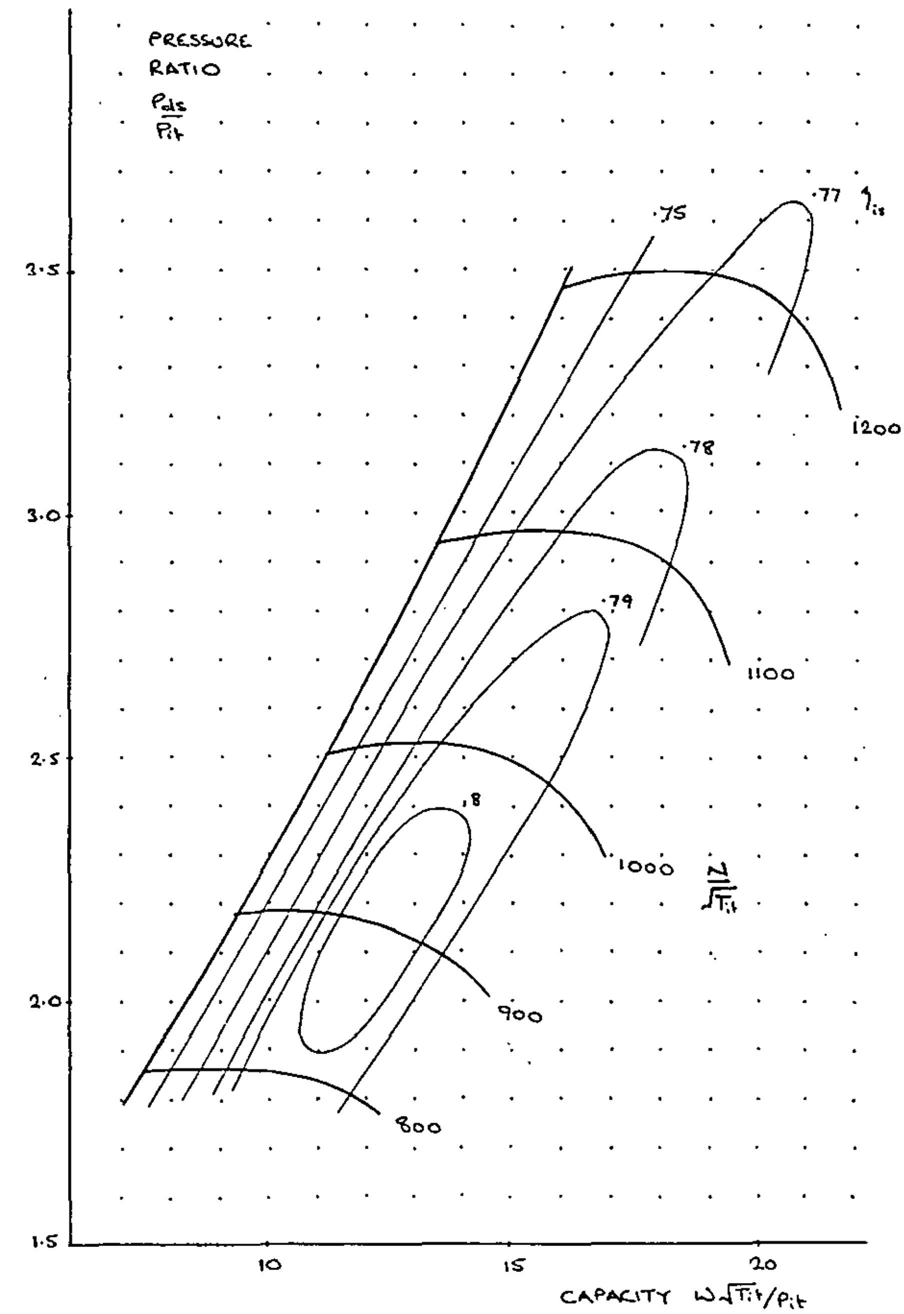

Figure 3.1: NA355 compressor characteristic

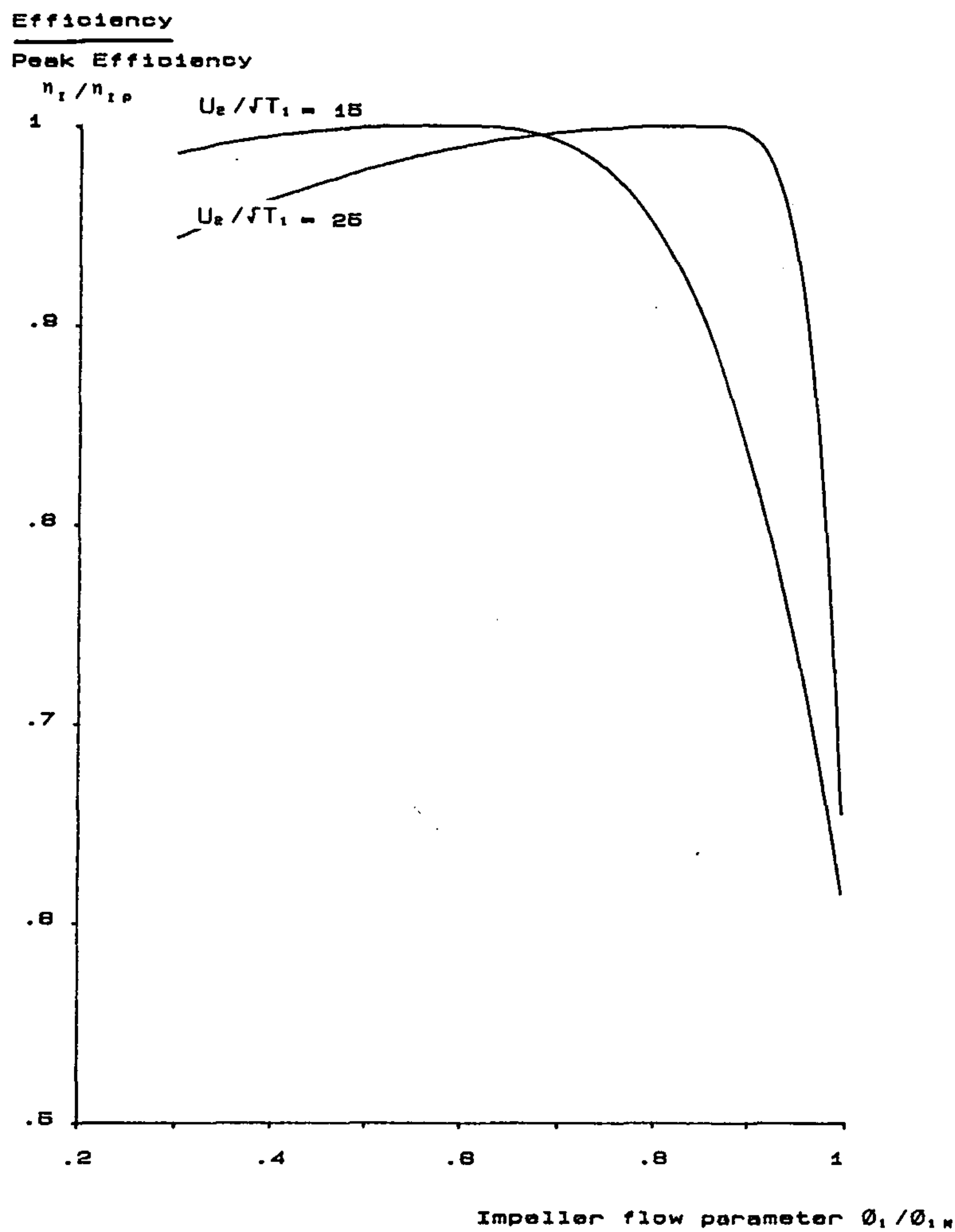

Figure 3.2: Impeller characteristic

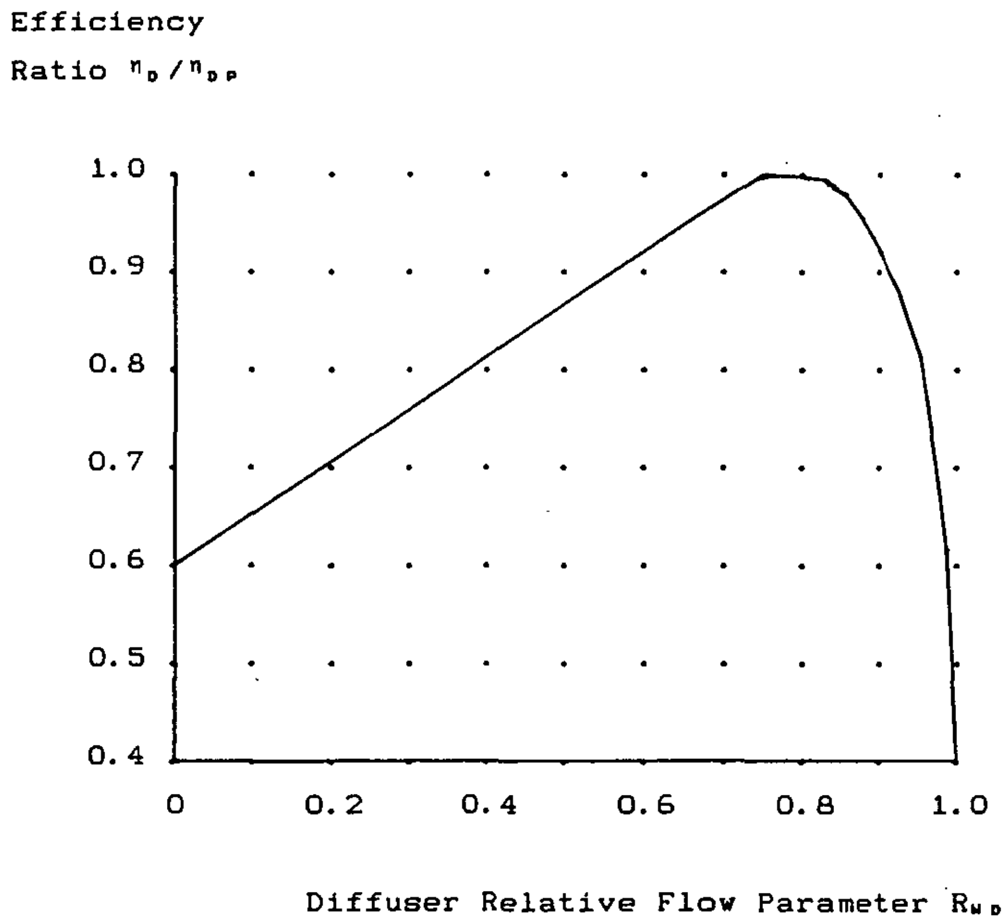

Figure 3.3: Diffuser characteristic

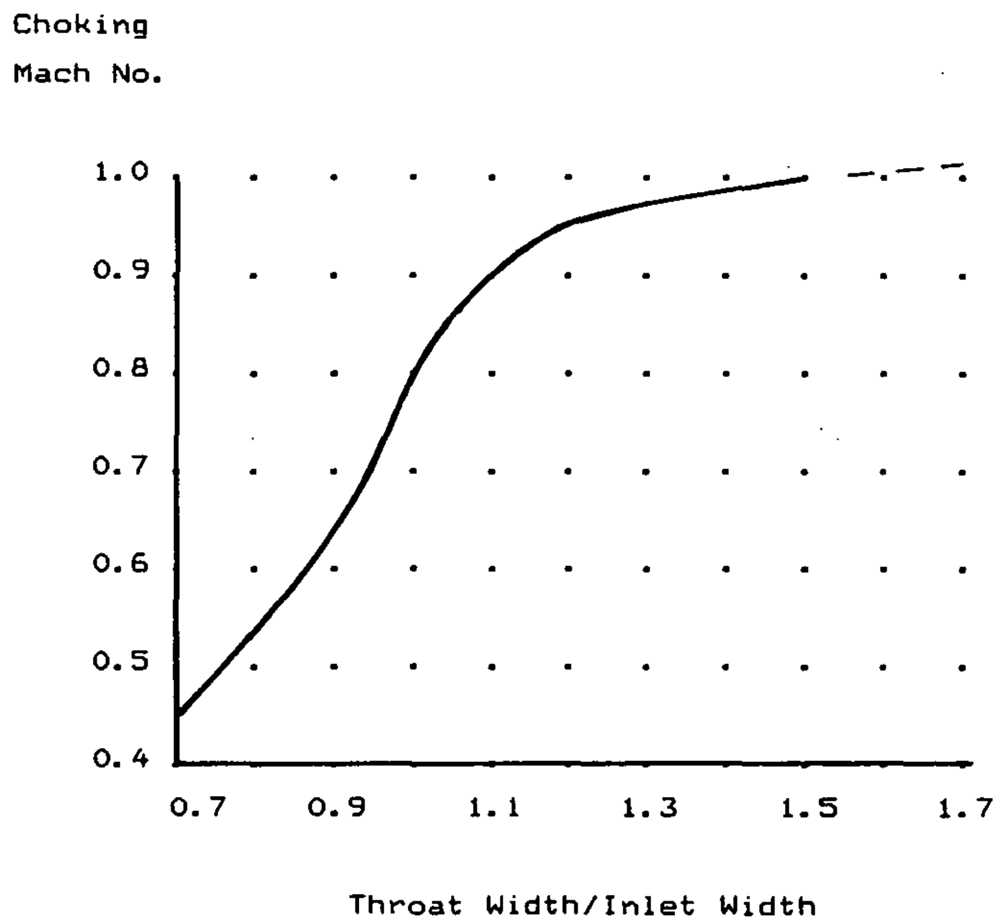

Figure 3.4: Howell cascade data

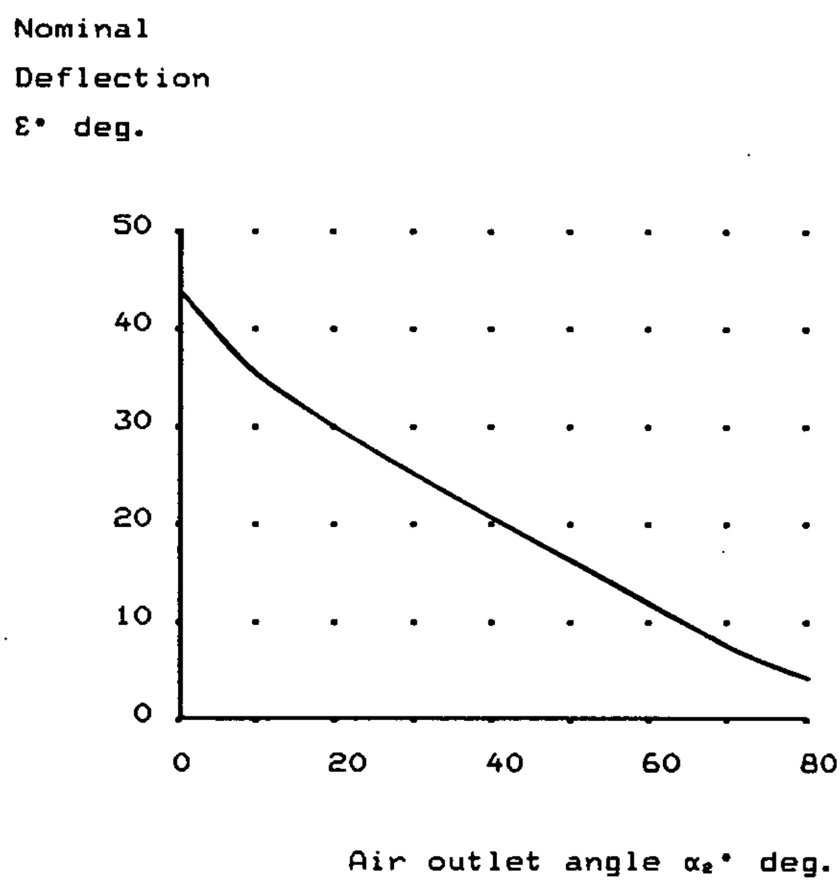

Figure 3.5: Nominal deflection against outlet angle

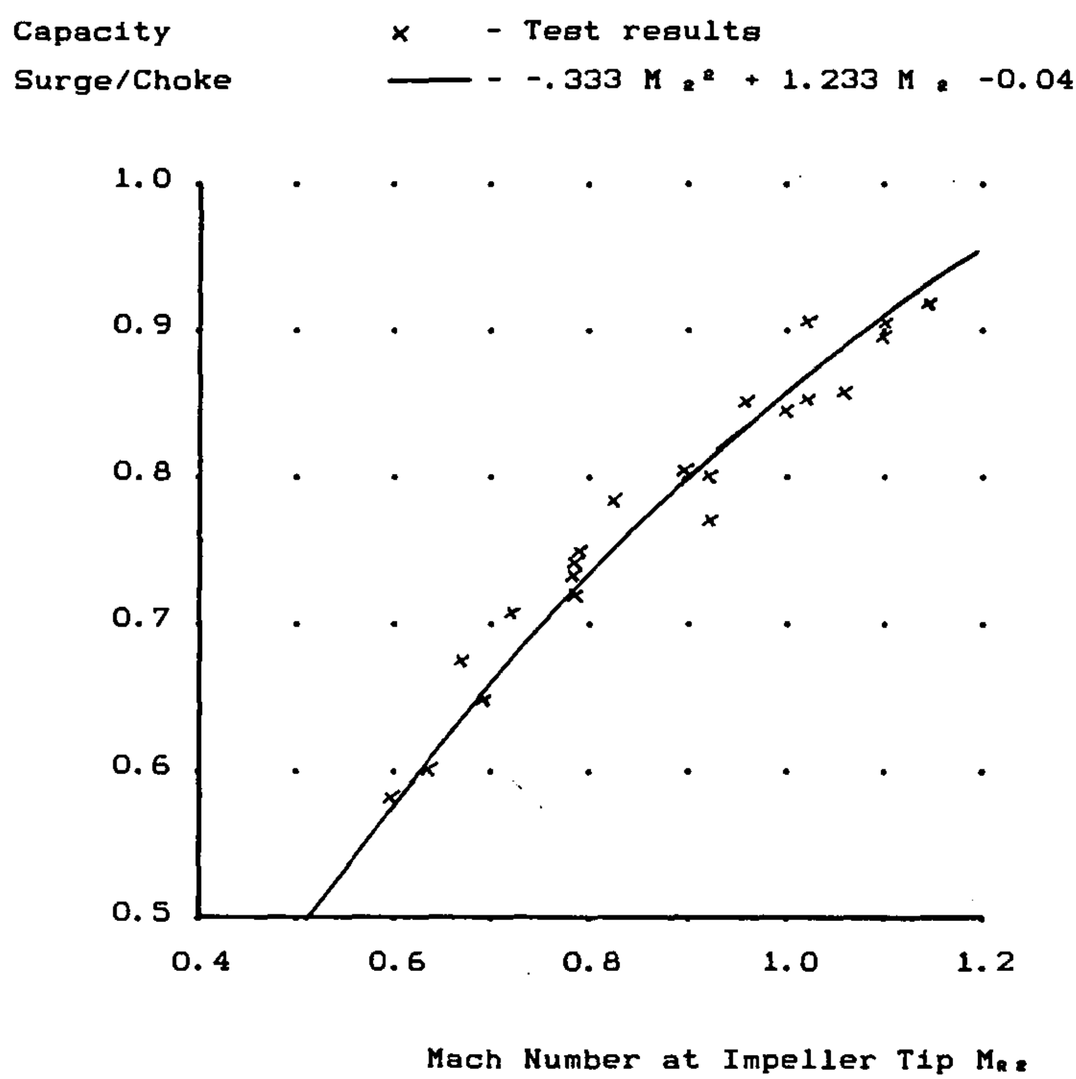

Figure 3.6: Surge correlation

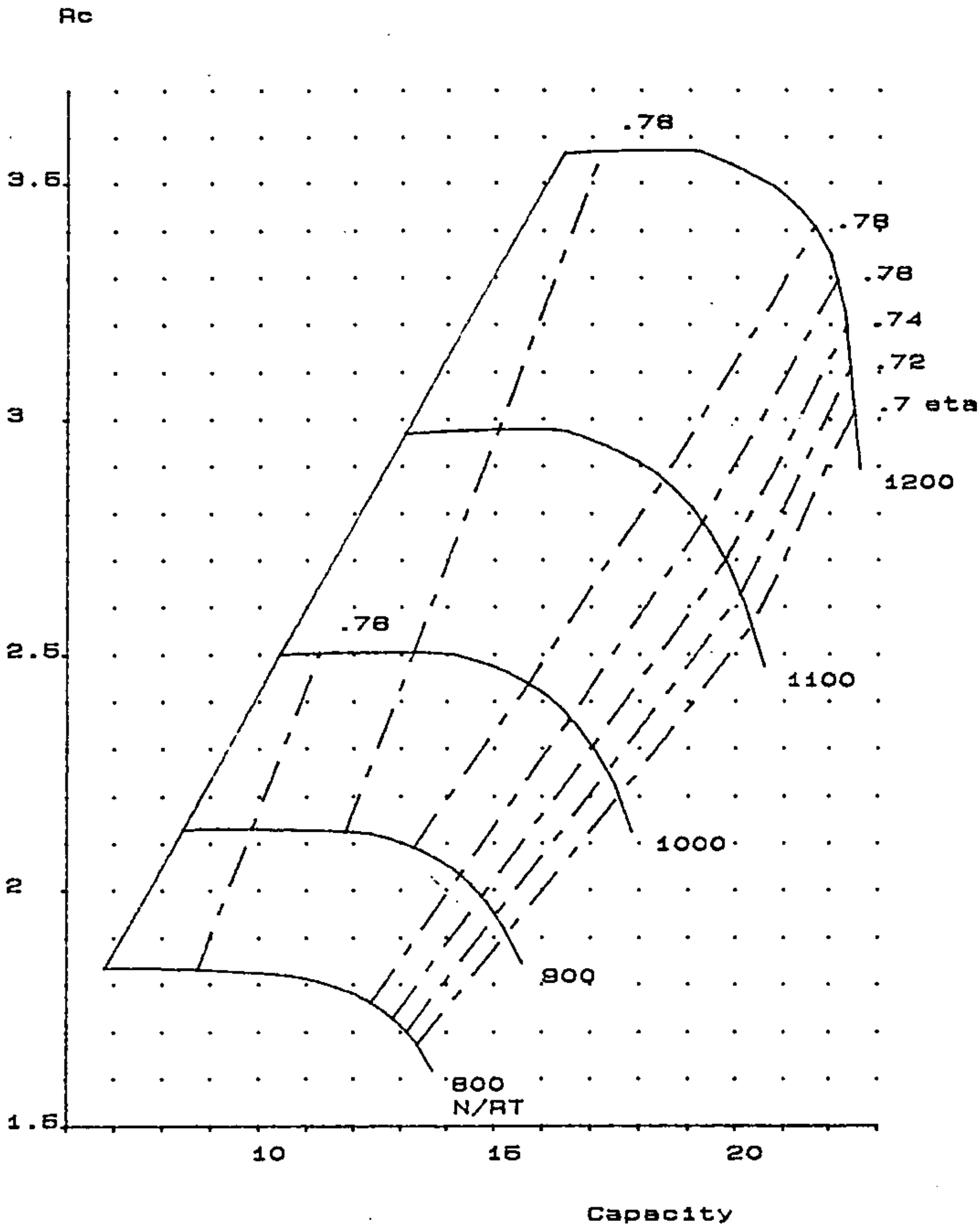

Figure 3.7: Predicted compressor characteristic

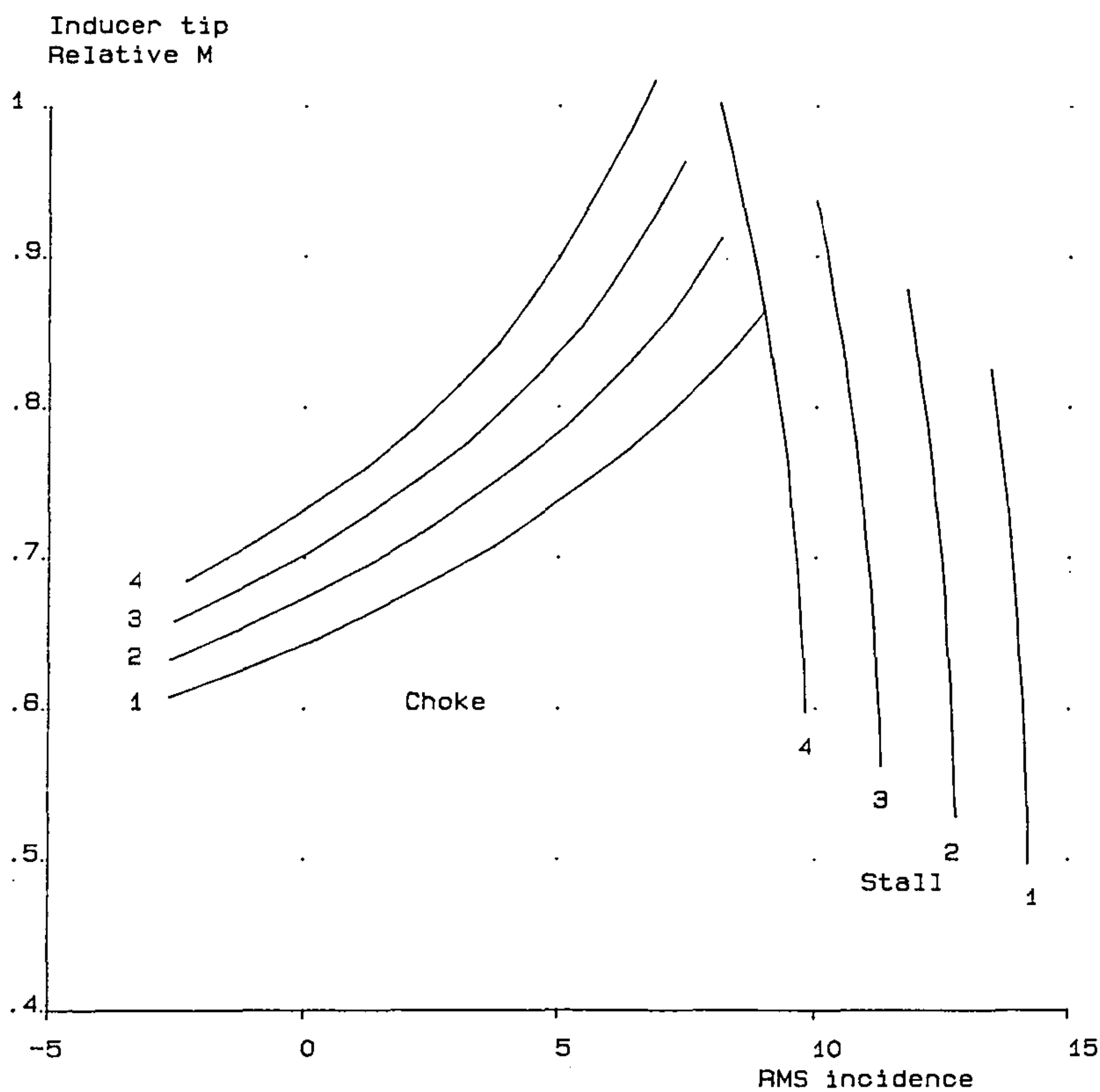

Figure 3.8: Inducer flow range - capacities 1-4

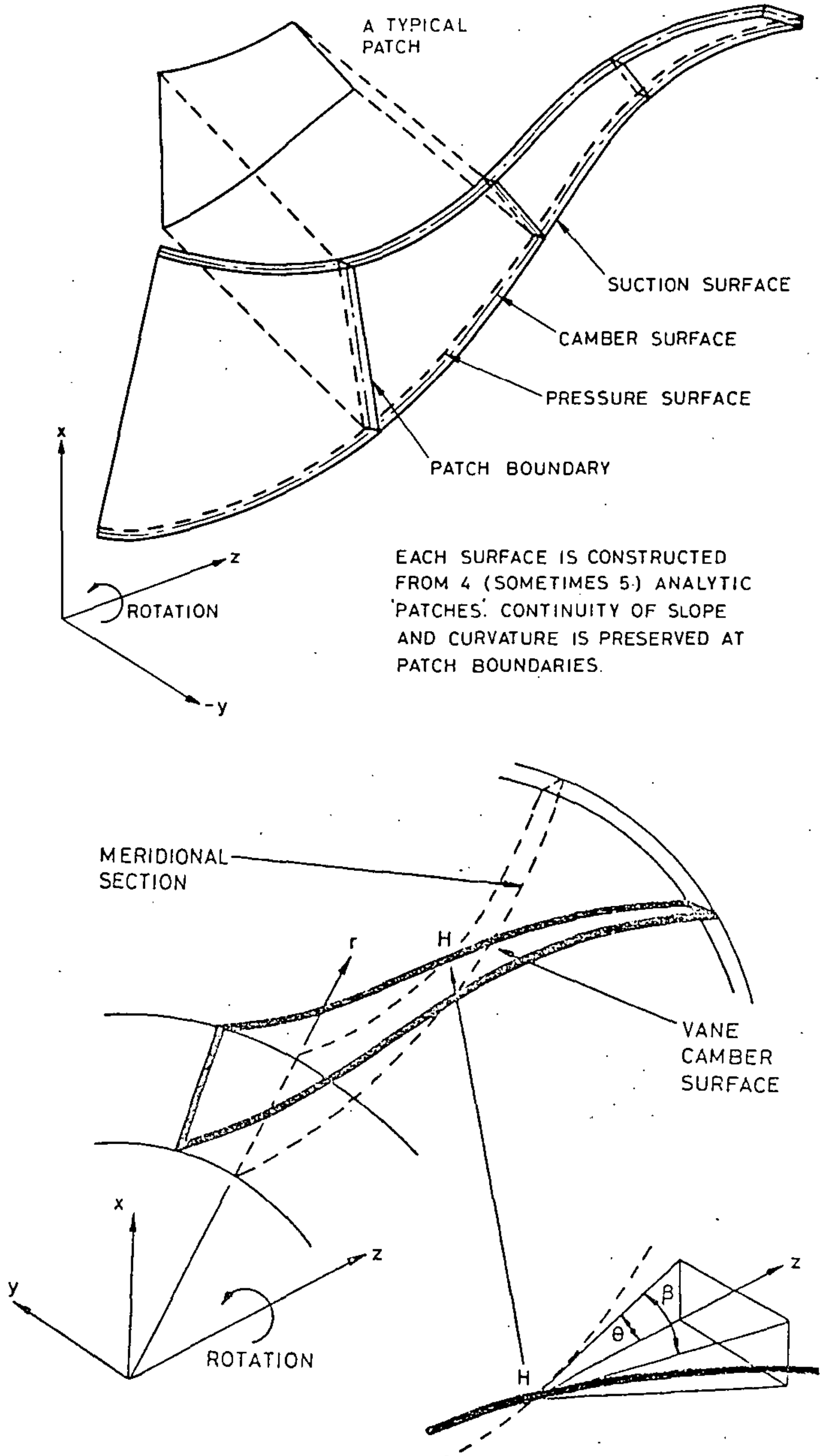

Figure 3.9: Analytical patch representation of the impeller vane surfaces

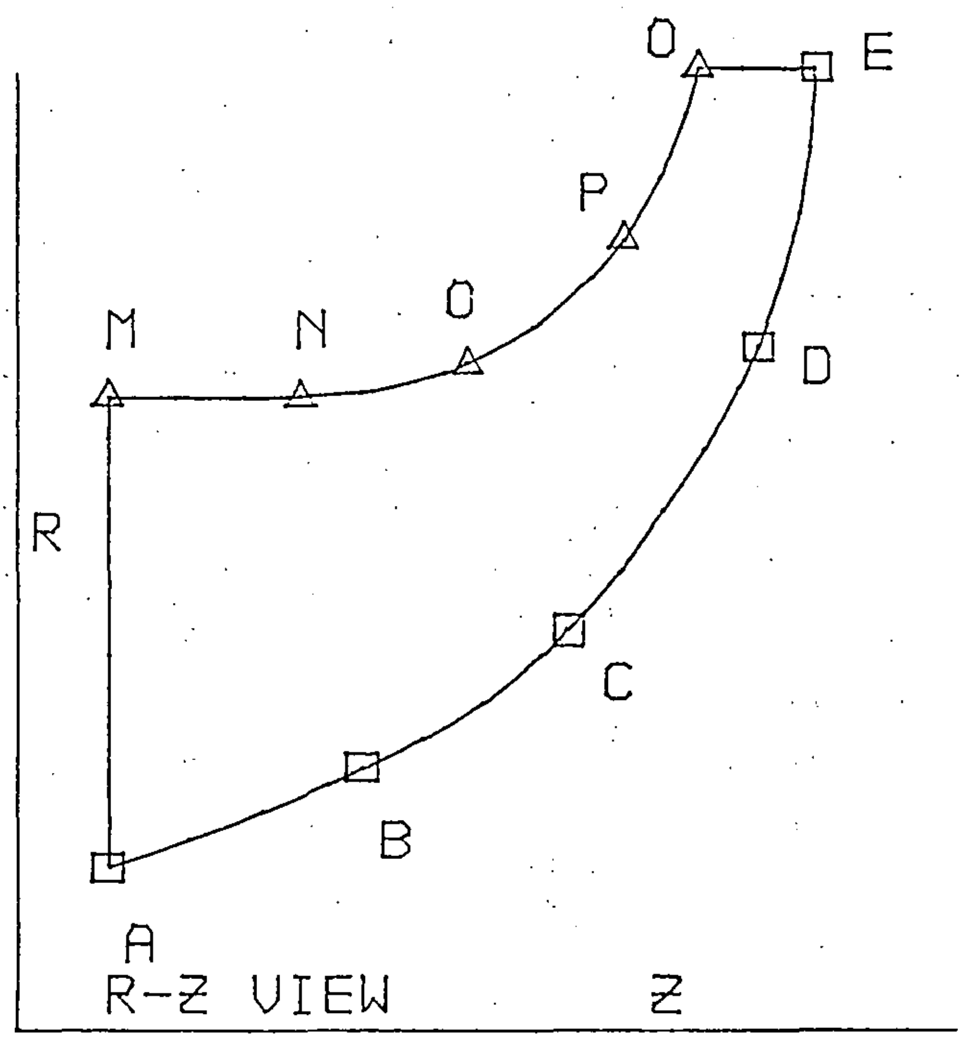

Figure 3.10: Vane patch corner co-ordinates

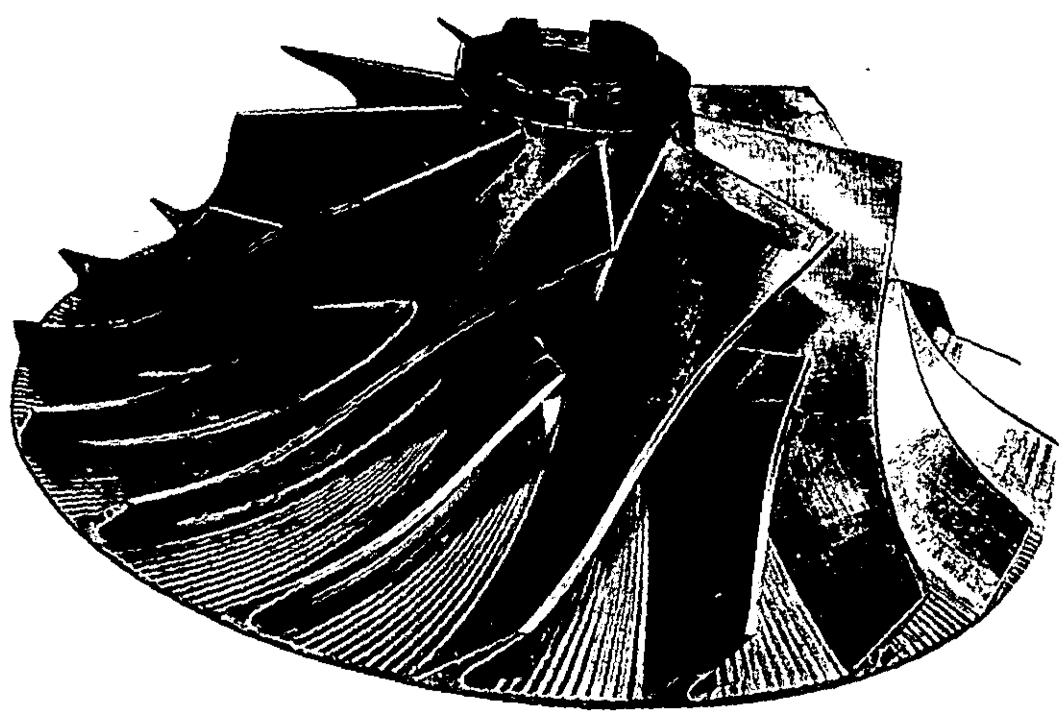

Figure 3.11: Impeller

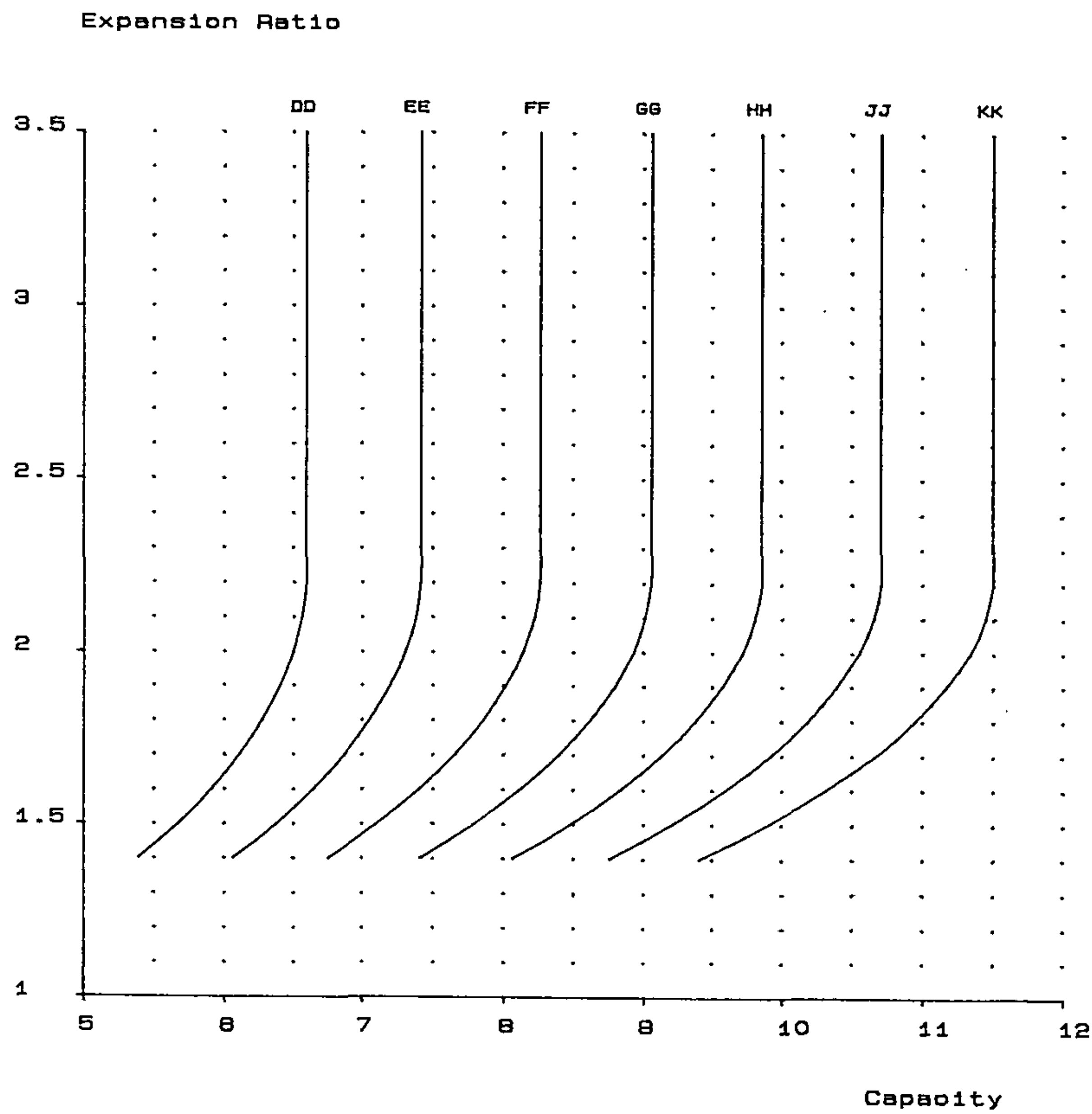

Figure 3.12: SA105 Turbine swallowing capacity

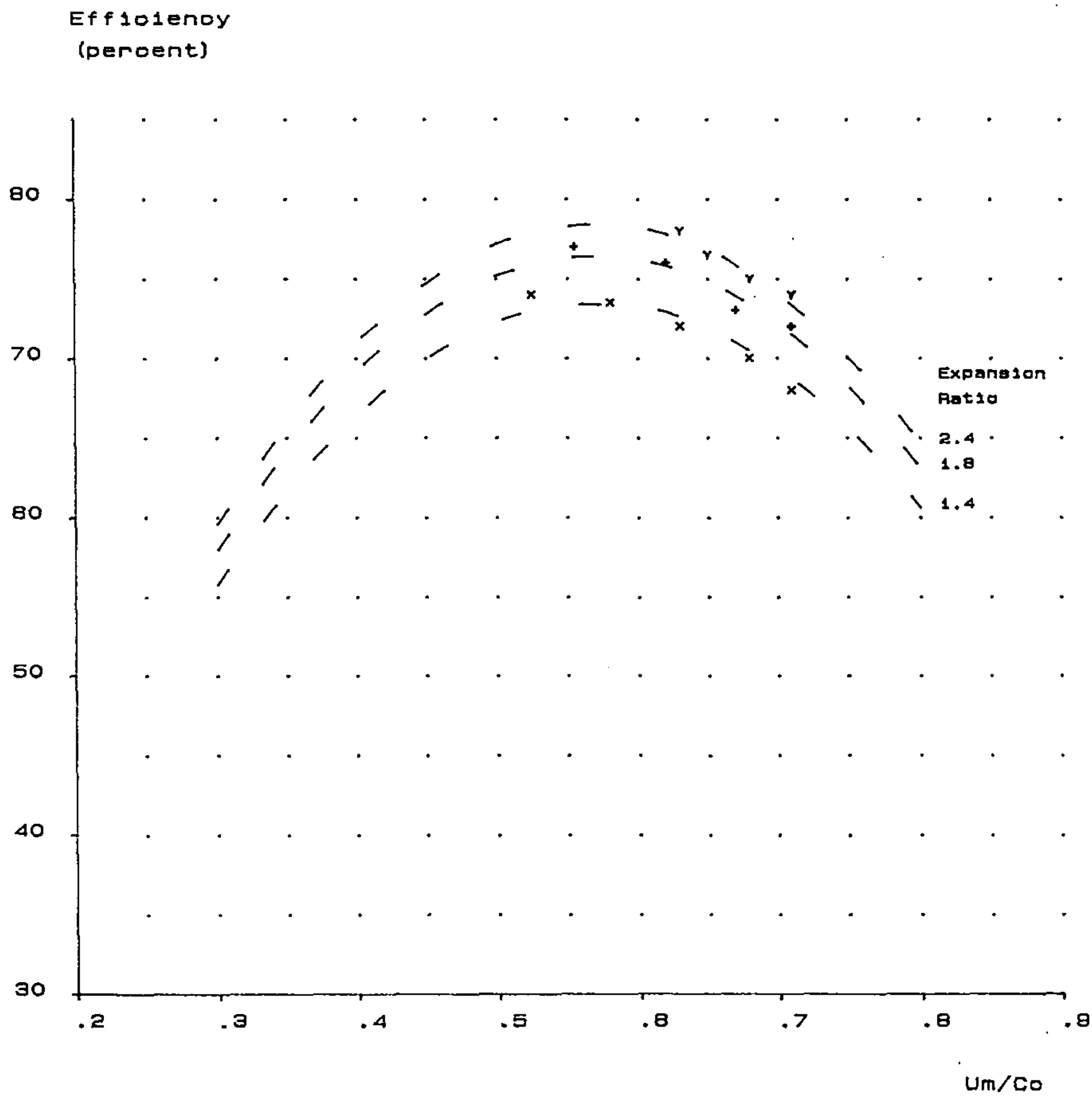

Figure 3.13: Efficiency versus velocity ratio

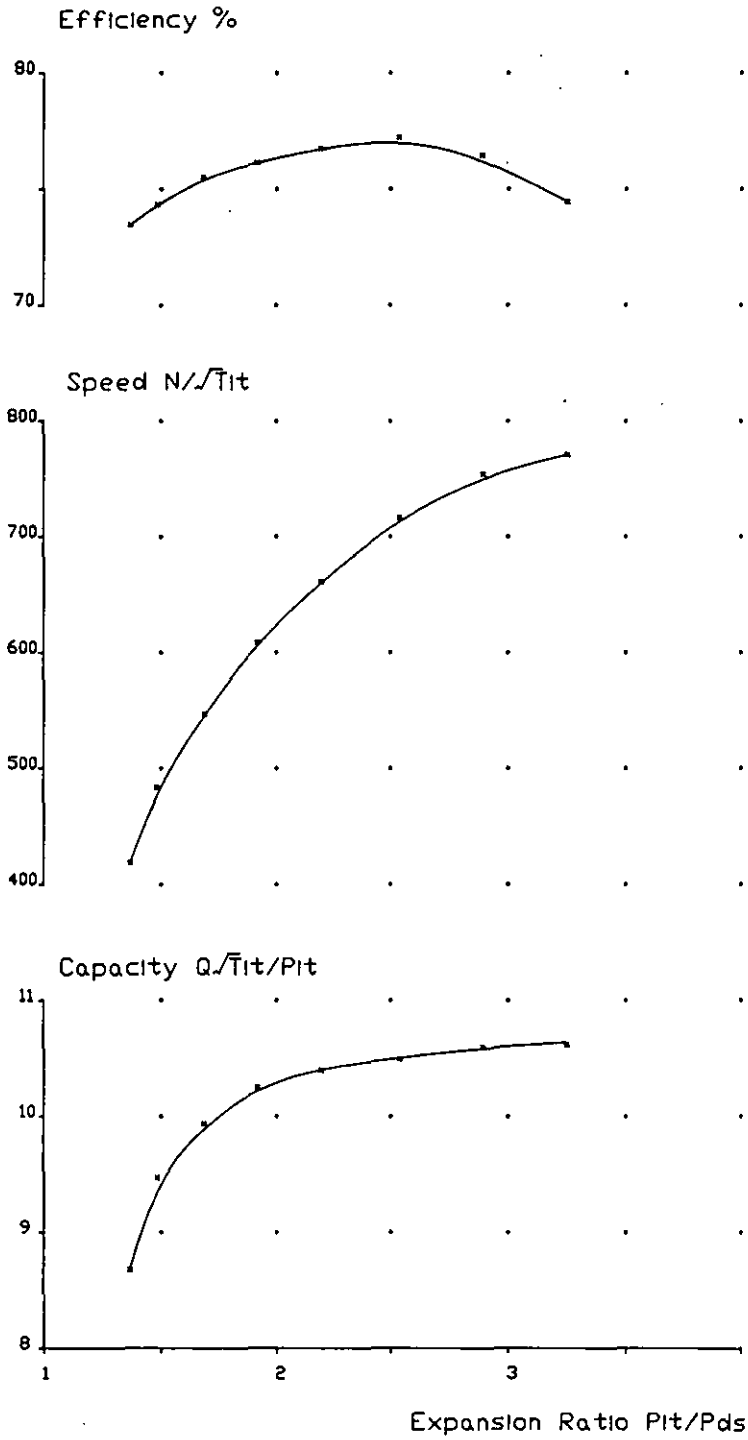

Figure 3.14: Turbine performance test results

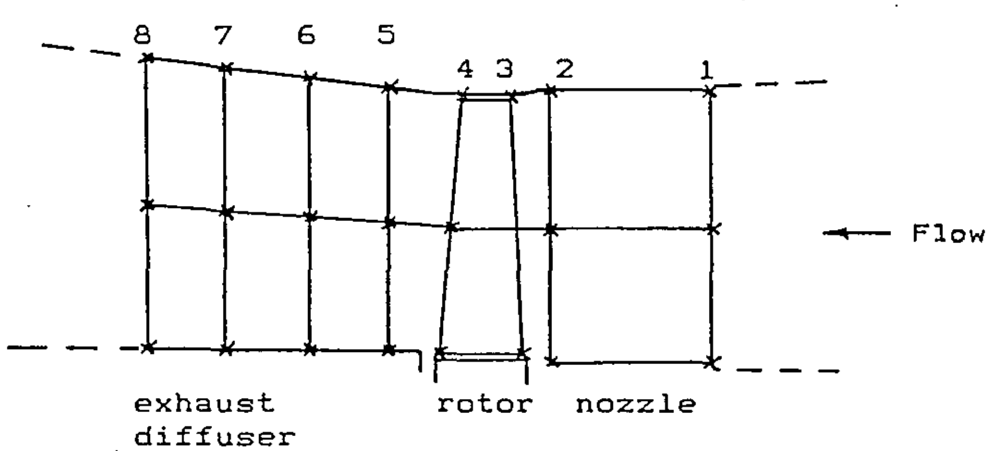

Figure 3.15: Calculation grid for simplified streamline curvature

CHAPTER 4

MARKET INTELLIGENCE

4.1 Determination of the Market Requirements

In a manufacturing company such as Napier it is important that the person who is responsible for deciding the marketing policy is clearly identified. In a consumer industry the importance of market research is often sufficiently apparent to warrant a marketing director. However, in a small manufacturing company, such as Napier, this position could not be justified. It is also important to decide who should carry out the market assessment. The obvious choice was to use the salesmen as they had regular contact with customers. However, the function of a salesmen is basically to get the customer to buy the current product whereas market analysis is basically a tool for the future and it has to provide the information upon which the policy for new product is based. Within Napier the General Manager contributed most towards the marketing policy.

The salesmen were crucially important in providing the customer contact. Their most important function in the marketing role was to maintain and strengthen the customer, or possible customer, relationship. As mentioned it was appreciated that the UK customer output had reduced, although two customers, Paxman and Ruston, remained particularly important as they were also GEC-owned companies. Therefore, the move towards Europe was more by necessity than by policy. Engineering department staff often visited customers to discuss with their opposites the direction the product was moving and the expected timescales. The natural split of responsibility for collation of data between the commercial and engineering departments was generally maintained. Progress with a new customer generally would follow a pattern established by experience. Firstly, the possible customer would be identified. Such sources as reference 4.1 would be consulted to assist this if original equipment was the market. Secondly, sales visits to the area would be planned to economise on travel particularly if in far off places in the world. This visit would have a number of objectives:

a Present the Napier range of products to the potential customer
b Obtain an understanding of the customer's product range and market
c Establish the most promising customer target application/s
d Ensure that the customer's pricing policies were understood
e Obtain sufficient technical data for initial requirements and ensure the technical contacts were known

The sales force were supplied with a proforma (Fig 4.1) for the technical details, so that sufficient data for an initial matching could be carried out. (Matching was the name given to the activity by means of which the particular build and size of turbocharger appropriate to a customer's engine is determined). For obvious reasons the sales visit was often primarily concerned with any possible immediate prospects. However, experience showed that a number of exploratory meetings by the salesmen would be required before orders became a reality. Following the sales visits generally there would be technical meetings, where the expected benefits of fitting the particular Napier product would be presented. Next would follow the possibility of carrying out a test on the customer's engine. As it was in the interest of Napier for this test to take place the turbocharger was often supplied on a sale-or-return basis. On some occasions the test would be carried out on a production engine. However, this was often not satisfactory as the timescale rarely permitted sufficient testing for reasonable evaluation. In some cases, by far the most desirable from the technical viewpoint, this testing was incorporated into an on-going engine development test programme. Following successful evaluation the particular Napier turbocharger would be cleared technically by the customer and possible further orders would become a matter of routine. This process to achieve quantity orders could take up to ten years to achieve, although in exceptional circumstances a turbocharger could be in service after about two years from the initial contact. The larger and more prestigious the customer, the lengthier this process could be. For licensee manufacturers is technical approval would have to be given by the licensor. However, this did not preclude testing on the licensee's engine. Relationships with existing customers ensured that the technical policies and future requirements could be determined much more easily. In this case it was essential that the technical liaison was maintained so that the Napier products would continue to be included in their on-going testing programmes. Additionally, established customers were used to ensure that the future Napier product requirements and timescales were in-line with the market demands.

A turbocharger can only be fitted as ancillary equipment to a diesel engine. The market is restricted to the provision of either original or replacement equipment. In the original equipment market (OEM) the customer is a diesel engine manufacturer, whereas in the replacement market the customer is normally a diesel engine operator. Within the original equipment diesel engine market there are two basic groups of manufacturer, firstly the original design group, and secondly the licensee group. Within the UK the number of diesel engine manufacturers, for the size of turbocharger to be considered, is limited. In addition the diesel engine cylinder horsepower being produced has declined at a steady rate for some years. Rail-traction has moved away from large fleets of diesel engines by electrification programmes and because of the increase of road-transport. Marine fleets have been reduced, even though trade has increased; the UK and European shipbuilding industry has been greatly reduced. Shipbuilding has become concentrated in the Far-East and Iron-Curtain countries. Restricted by the market demands, the UK diesel engine producers have suffered from reducing output and the consequential lack of sufficient development funding to retain a high position in the world market.

The large diesel engine industry is well established and the OEM markets are well-known at least in terms of their description. The total market can be broken down into the following applications sectors (proportion of total quantity ordered in 1980)(4.2):

i	marine propulsion	37%
ii	marine auxiliary	27%
iii	stationary power generation	14%
iv	emergency stand-by power	9%
v	traction	9%
vi	others	4%

However, the market information has to be analysed on a regular basis to enable the future trends to be predicted. The Napier range of turbocharger can be conveniently separated into a number of size-related groups. Firstly, there are the smallest types with centripetal turbines (Fig 4.2). These are used in auxiliary power generation, or small marine engines. The market is limited because this size of turbocharger tends to be in competition with the larger automotive unit, particularly when a multiple turbocharger option is possible. Secondly, there are the small axial turbine range, distinguished by having in-board bearings. This group of three frame-sizes has applications in rail-traction, small marine propulsion, and auxiliary power generation. The largest three frame-sizes have out-board bearings (Fig 2.5), and these have application in marine main propulsion, and general industrial power generation. The published statistical trade data can be analysed with respect to the type of applications describe above to obtain a broad understanding of the trends. For example, table 4.1 presents data (4.3) relating to piston engine generator sets for the UK and Europe. This shows that the UK maintained a more constant level of import of this commodity and slightly worse export rate than the rest of Europe over the period considered. However, the value of exports divided by imports clearly shows that the UK maintained at least twice the rate pertaining to Europe as a whole. In this particular market it would appear reasonable to expect the same level of requirement for turbochargers as had been seen over the preceding years. Table 4.2 shows a not too dissimilar picture in the marine piston engine market, although there is a definite downward trend in UK exports. As marine main propulsion and auxiliary power generation is a significant eventual market for large turbochargers the trends in this industry can be examined (4.4). Table 4.3 shows the trends in gross tonnage orders and completions over the period 1973-1982. The large downturn in orders with the lag of completed tonnage can be clearly seen. Table 4.4 shows how the new orders were dominated by Japan, a totally self-contained market from the diesel engine point of view. The only way that a European manufacturer can invade the Japanese market is by selling licences. During the period considered the Japanese company Niigata manufactured Napier products under licence. It is interesting to see that on the whole the UK retained its very small percentage of the world market whereas Western Europe shows a clear decline. The largest sizes of turbochargers were greatly affected by this market and it was expected that this group would continue with a reducing requirement into the future. As the smaller in-board bearing units also had applications in the marine market the effects would also be seen in this range. Rail-traction market figures in table 4.5 do not show the true picture. This table shows that the ratio of goods moved by road and rail did not vary a great deal over the period, although the totals reduced. More significantly are the British Rail (BR) diesel building programmes. During the period 1957-1968 a total of 3096 main-line locomotives were built, and a relatively large proportion of these are still in service. There are currently refit programmes to ensure some of these locomotives continue in service into the 2000's. The largest fleet of freightliner locomotives was Class 56; a total of 135 were built between 1976 and 1984. These were fitted with two Napier SA085 turbochargers to each Ruston 16RKC engine. The next additional freightliner locomotives were Class 58. It was envisaged that 20-25 of this class would be built, entering service in 1983. This locomotive is powered by a Ruston 12RKC with a single Napier SA105 turbocharger. The InterCity 125 (HST) passenger locomotive is fitted with a Paxman 12RP200 engine and a Napier SA084 turbocharger. The fleet of 200 locomotives went into service in 1976. The intention of BR is to maintain and refit the HST fleet and they currently do not envisage any more major diesel passenger locomotive build programmes this century. It is clear that the BR turbocharger requirements have progressively changed from original equipment to spares and repair.

The type of information presented above provides a general picture of the turbocharger market direction. This enables the individual customer information to be structured, and fitted into the general pattern. It also provides valuable information upon which to base the future product strategy. The large turbocharger market may be seen to be reducing in size in most of the existing application areas.

4.2 Market Size

The market demands for a specific turbocharger are more difficult to determine. There are two main sources of information. Firstly, it is necessary to ask the customer. The future turbocharger requirement information is often nearer to what is expected rather than what can be achieved. Secondly, the market trends for existing turbochargers can be obtained from within the company. The quantity of sales over preceding years should enable predictions to be made with respect to possible future sales. Table 4.6 presents the sales information for the SA105 and for the NA350 from 1977 until 1981. Three important facts may be observed from these results. Firstly, the quantity of total sales over this period is small; the average is 50 units per year. Secondly, the figures are not smooth - a good year may be followed by a bad year. This was often due to the product being purchased in batches. In particular the figures for 1981 include forward purchase for the British Rail Class 58 contract. Thirdly, the extremely low sales for the NA350 tends to indicate that the market for this turbocharger was less than significant.

The product life-cycle can be estimated from the data above at least for the SA105. Apart from the distortion in 1981 caused by one very large order the decline in numbers can be clearly seen. This decline is the result of the product coming to the end of its life-cycle. When the reasons were analysed it was considered probable that they would include insufficiently high overall efficiency and a lack of sufficiently high pressure rating capability.

4.3 **Assessment of main competitors**

There are relatively few manufacturers of large turbochargers throughout the world. By far the largest, accounting for some 60% of the European market was Brown Boveri & Company (BBC). The Napier share during the period considered amounted to about 20%. The remaining 20% was made up by companies such as MAN and Hispano-Suiza. The Far-Eastern market was mainly covered by Japanese companies manufacturing European products under licence and this generally ensured that the European market was not available to them. One exception was the Mitsubishi range of turbochargers, a Japanese designed and manufactured product. Discussions with this company eventually led to Napier agreeing to manufacture this product under licence. The United States market was mainly covered by the indigenous engine manufacturer producing its own turbocharger. This was particularly the situation for the smaller medium-speed engine used extensively for rail traction applications. Larger engines are generally made under licence. For instance the Colt company manufacture Pielstick engines. Therefore, in terms of assessing the main competitors it was inevitable that the centre of attention was BBC.

During the late 1970's it became apparent BBC were working on a new range of turbochargers, which was to be known as the VTR..4 series. In 1981 BBC published a series of articles describing the product range and strategy (4.5, 4.6, 4.7). This series of articles was invaluable in the assessment of the competition. The previous range of BBC product was the VTR..1 series, and comparative testing with the Napier product had been carried out on a number of engines. The message of BBC was clearly set down. They believed that the diesel engine industry needed higher efficiency turbochargers that were capable of operating up to 4.5:1 pressure ratio. This the 4 series was able to do and the stated improvement in turbocharger operating efficiency was up to 5 percentage points, from 62 to 67% (total/total). The articles focused on improvements in fuel consumption and on how this could be used to offset the increased turbocharger cost of about 34%. The way that the increased efficiency had been achieved was explained. For example a new back-swept compressor and a new turbine design had been incorporated. In an attempt to reduce the velocity-associated losses in the casings there had been a conscious decision to ensure these were adequately proportioned. This meant that, in general, the VTR..4 series could not be fitted where the previous VTR..1 had been fitted. For the larger three of the most recent Napier range this was particularly fortunate as the Napier product was mostly interchangeable with the previous BBC offering. Therefore a customer could fit the Napier latest product in place of the old standard BBC product. For the smaller three of the Napier axial turbine range this was not as significant as the two products were not interchangeable. However, in general for the same mass flow it appeared that the Napier product could have a size advantage.

In summary, the new BBC product was of high efficiency, was capable of high pressure ratios, was slightly bulkier, and was considerably higher priced than its predecessor.

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FIGURES

TABLES

NAPIER TURBOCHARGERS

TURBOCHARGER SALES ENQUIRY DATA

To enable the NAPIER engineers to optimise the design of turbocharger offered for any particular application, the client should provide certain basic details of the engine design and should also define the operating parameters of the installation.
In general terms the client should provide data under the following headings:

Bore Stroke No. of Cylinders In-line or V	Engine
Stroke cycle. Loop or Uniflow Scavenge	Designation

Rating	Cont.	Max.	O/Load	
B.H.P.				Manufacturer
R.P.M.				
B.M.E.P.				Altitude
Specific Fuel Consumption				
Engine Exhaust Temperature				Ambient Temperature
Turbine Inlet Temperature				
Inlet Air Temperature				Humidity
Specific Air Consumption				
Depression at Compressor Inlet				Inter-coolers
Exhaust Back Pressure				
Pressure Drop across Cooler				Pulse Convertors
Compressor Outlet Pressure				
Inlet Manifold Pressure				Application
Exhaust Manifold Pressure				

No. of blowers per engine

Constant pressure or pulse exhaust system

Firing order

Direct injection or pre-combustion chamber

Scavenge pumps series or parallel

Valve or port timing AO AC EO EC

Exhaust configuration

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Napier Turbochargers Ltd. P.O. Box 1 LINCOLN ENGLAND

Figure 4.1: Enquiry technical proforma

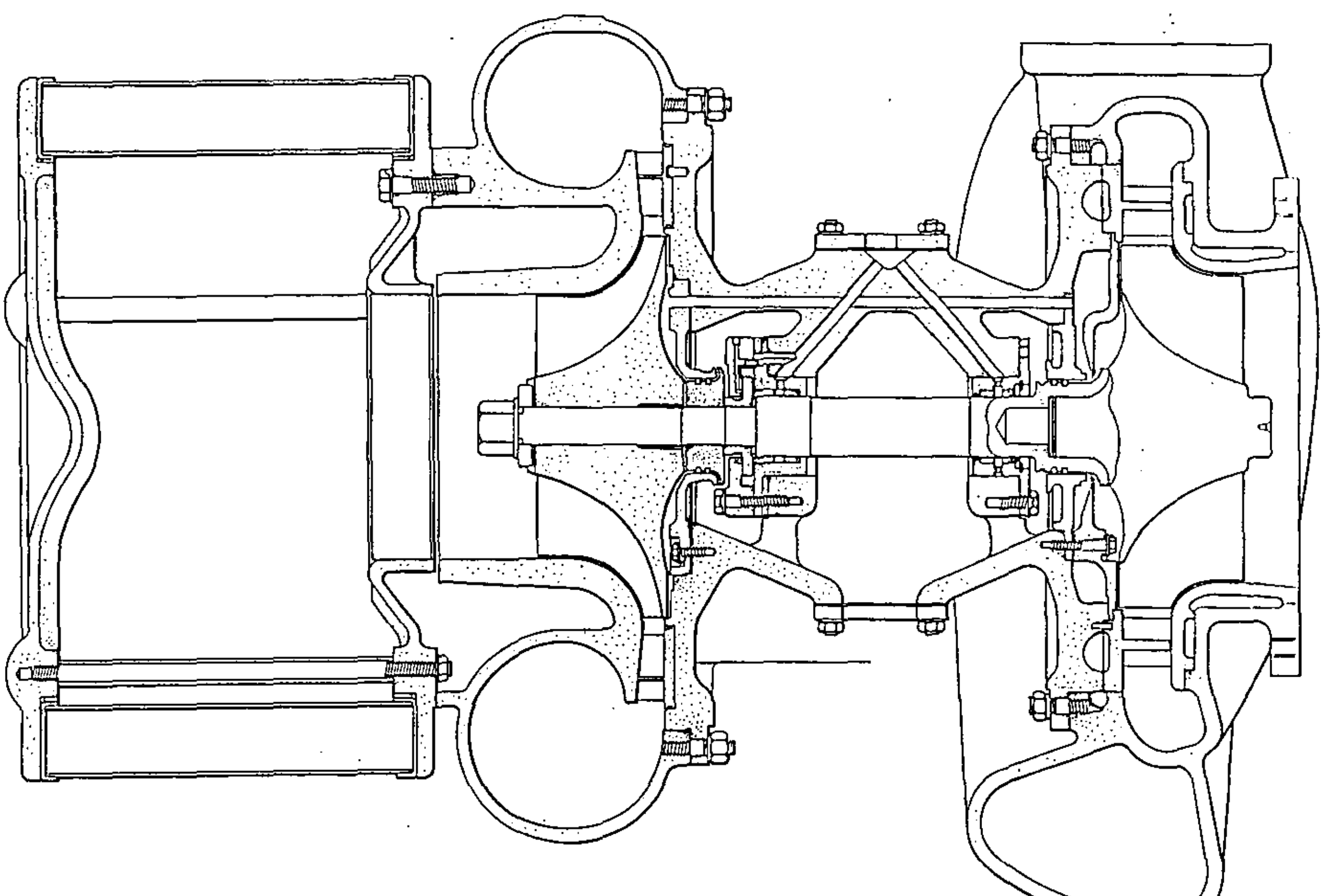

Figure 4.2: In-board bearing turbocharger

		'77	'78	'79	'80	'81	'82
Europe	imports	1	1.23	1.53	1.71	1.58	1.52
	exports	5.81	7.68	6.98	7.2	8.94	8.45
	exp/imp	5.81	4.7	4.6	4.2	5.66	5.56
UK	imports	0.16	0.18	0.21	0.21	0.31	0.22
	exports	2.21	2.89	2.05	2.22	3.02	2.53
	exp/imp	13.8	16.1	9.76	10.6	9.74	11.5

TABLE 4.1: Analysis of generator sets with engines in normalised values using 1977 imports to Europe as a datum

		'77	'78	'79	'80	'81	'82
Europe	imports	1	1.09	1.18	1.38	1.16	1.22
	exports	1.32	1.26	1.51	1.51	1.29	1.41
	exp/imp	1.32	1.16	1.28	1.24	1.11	1.16
UK	imports	0.07	0.09	0.11	0.08	0.09	0.07
	exports	0.21	0.18	0.19	0.18	0.18	0.12
	exp/imp	3	2	1.73	2.25	2	1.71

TABLE 4.2: Analysis of marine piston engines in normalised values using 1977 imports to Europe as a datum

	'73	'74	'75	'76	'77	'78	'79	'80	'81	'82
World trade										
orders	74	28	14	13	11	8	17	19	17	12
completed	30	34	34	34	28	18	14	13	17	17

TABLE 4.3: World ship-building orders and completion
(million gross tonnage)

	'73	'74	'75	'76	'77	'78	'79	'80	'81	'82
Japan	49	38	49	56	52	43	49	53	48	49
West Europe	42	39	22	24	27	26	27	24	25	21
Rest of world	9	22	29	20	21	31	23	23	27	30
UK					3.9	1.9	1.4	2.7	2.9	2.5

TABLE 4.4: World ship-building percentage of new orders

	'73	'74	'75	'76	'77	'78	'79	'80	'81	'82
Rail%	10	10	10	10	10	9	9	9	9	9
Road%	85	85	85	84	82	83	82	82	82	83

TABLE 4.5: Goods moved in UK

	'77	'78	'79	'80	'81
SA105	100%	62%	66%	42%	90%
NA350	0	2.5%	2.5%	0	18%

TABLE 4.6: Turbocharger sales (using 1977 sales of SA105 as a datum) (Published with permission, courtesy of Napier Turbochargers Ltd.)

CHAPTER 5

CURRENT PRODUCT EVALUATION

5.1 Comparison with the market requirements

The Napier SA105 was designed in the late 1960s, and was mainly a scaled-up version of the SA085, the prototype of which first ran in 1966. The product strategy at that time was primarily to reduce cost whilst maintaining a reasonably high overall efficiency. Fuel costs at that time were extremely low in comparison to more recent times. The previous range of turbocharger of this size had outboard bearings, similar to the large range shown in figure 2.5. The inboard arrangement allowed a dramatic reduction in the number of parts (about half) and therefore this allowed the cost of manufacture to be reduced. The decision was made to air-cool the turbine-inlet casing, instead of water-cooling. This also allowed a significant reduction in cost. At that time a great amount of work had been carried out within Napier to improve the turbine efficiency. The SA085, and hence the SA105 benefitted from this improvement. The compressor was not improved, at least not initially, the previous range of compressor having been modified to fit the new inboard bearing layout. The SA range had been designed with the Ruston engine in mind. The space and weight restrictions were not too severe, the efficiency requirement was not too arduous but the purchase price had to be low.

The Paxman range of diesel engine had generally resulted in special Napier products, at least in terms of casings. For instance the MS/HP210 became the HP210DP (DP for Davey Paxman). This latter had the rotating assembly of the 210 but had a completely new set of casings. The Paxman requirement was primarily determined by space and weight considerations, with performance and cost very much a secondary criteria. It was inevitable that when the InterCity 125 (HST) philosophy was being developed in the late 1960s incorporating Paxman 12RP200 engines a special Napier product became necessary. However, for the 12RP200, it was decided to manufacture a completely new turbocharger. This became known as the SA084 as the unit was a similar capacity to the SA085 but was significantly more compact. This design, incorporating what was considered to be the most up-to-date thinking of the time was then scaled-up to produce the NA350. The change of style of nomenclature was to fit the turbocharger into the same series as the larger group that had been called the large NA series. The NA350 was then the largest of the in-board bearing axial units followed by the NA250, and NA150 in descending order.

The main competitor for the SA105 and NA350 had been the Brown Boveri (BBC) VTR321. This latter unit was being replaced by the more advanced VTR354. Details of all four turbochargers were compared.

Initially the size and capacity range for the four turbochargers were considered, together with the requirements of typical target engines. Figure 5.1 presents the compressor capacity comparison. The BBC data were extracted from Reference 5.1. It may be seen that the SA105 and VTR354 have a similar capacity envelope, up to the pressure ratio limit of the SA105. Table 5.1 shows how the sizes of the four turbochargers vary. The volume has been determined assuming that the units will fit into a cylinder with the width as a diameter. This volume is expressed as a percentage of the VTR354 volume to make comparison more convenient. A similar approach is used for the specific throughput, or capacity divided by volume comparison. Apparent is the dramatic increase in size of the VTR354 compared with the slightly lower capacity VTR321. Also apparent is the similarity of the parameters VTR354 with SA105, and VTR321 with NA350. The specific throughput parameter shows how significantly different in philosophy these two types of unit are. The larger this value the higher will be the velocities within the casings and the greater the potential for increased losses. BBC's market intelligence was that the diesel engine manufacturers wanted higher efficiency turbochargers and this was in agreement with the market intelligence gathered by Napier. One part of BBC's approach to this requirement appeared to be to decrease the specific throughput of the product. This new direction of BBC appeared to bring its product into line with the SA105. Also plotted on the capacity envelope (Fig 5.1) are a number of Ruston engine requirements, with the

12RK270 operational envelope. This shows that both the VTR354 and SA105 frame-sizes would have a satisfactory capacity range for these engines.

5.2 The product strengths and weaknesses

The relative merits of the SA105 and the NA350 will be discussed. The SA105 had been sold in reasonably large numbers and therefore the manufacture had become well established. In general the components had been through the process of modification to make them as suitable as possible for manufacture. Sound castings were being received from sub-contract foundries on a routine basis. This iterative process involving re-design and revised foundry process can take a considerable period, particularly for the more complicated castings. The supply of other bought-in components was also being achieved on a regular basis, and therefore problems would only occur on a relatively small number of components. The NA350 was, however, in a different position. The sales of this turbocharger had been disappointing, and the sales forecasts had rarely been achieved. This had meant that some major items had become extremely "slow-moving". This expression was used for items that had been in the finished part or raw stock stores for more than 1-2 years and was associated with the inventory control process. So the capital tied up in hardware was higher than it should have been, and therefore there was an incentive to reduce the number of these items in stock. However, if subsequently these items had to be replaced further supplies would be treated as new because of the elapsed time between the previous deliveries. Other items such as turbine discs and blades were in a similar position. In simple terms the SA105 could be supplied relatively easily whereas selling the NA350 was likely to provide sufficient problems to make supply of a small number uneconomical.

The rotors of the two turbochargers were similar. However, the differences made a significant effect upon the manufacture and assembly costs. The SA105 had "floating sleeve" journal bearings. This means that the shaft rotates inside a ring which itself rotates inside a housing (Fig 5.2). There are, therefore, two oil-films - the first between the shaft and the ring and the second between the ring and the housing. This type of bearing is good at avoiding sub-synchronous instability in the bearing oil-film. The disadvantage of the design of the ring in the SA105 was that the ring radial thickness was insufficient. This had two effects, firstly the ring distorted under the bearing pressure loads and secondly the ring speed was high. The result of the ring distortion was that the clearances had to be increased. This in turn meant that rotor shroud clearances had to be increased. As leakage losses are greatly affected by clearance this was obviously undesirable. The high ring speed meant that the oil supply through the ring was limited by the centrifugal force. This meant that the maximum speed of this type of bearing arrangement was limited. The thrust bearing in the SA105 had originally been a tilting pad whitemetal design. This was situated close to the turbine wheel and, not unnaturally, there had been problems in service particularly when the engine and oil-supply had been cut-off from high-loads. In these cases the turbine disc would conduct heat into the shaft and as there would be no oil to remove the heat the bearing would overheat resulting in the bearing melting. The bearing had been improved by changing the pad material to aluminium-tin which behaved in a similar manner to the whitemetal, but could withstand much greater temperatures. In spite of these disadvantages, the SA105 rotor assembly was extremely tolerant. The assembly was simple and the out-of-balance requirements could generally be met with little problem.

The NA350 used "lobed" bearings. This means that the bearing surface is formed by a larger radius than the inner radius of the bearing (Fig 5.3). The thrust bearing had been relocated from the turbine end to the compressor end - away from the hostile environment. However, to do this a thrust collar and journal "bobbin" was fitted to the shaft and the compressor-end combined thrust and journal bearing was horizontally split into two parts to allow assembly (Fig 5.4). In this arrangement the thrust bearing was a taper-land design. This type of journal bearing was reasonably good from the stability point of view. However, the out-of-balance tolerance had to be about 1/3 of the SA105 figure to ensure satisfactory operation. Assembly was extremely difficult. The bobbin was an interference fit on the shaft and had to be jacked on and off during balancing and assembly. Cleanliness, choice of materials and relative hardness was of vital importance in order to avoid scrap at the assembly stage. Damage due to fretting between the shaft and the bobbin did cause problems in service. However, the arrangement of a solid and substantial bearing with a taper-land thrust bearing at the compressor end was most desirable.

The nozzle assembly in both turbochargers was similar. The nozzle vanes were formed by bent sheet steel cast into inner and outer rings. This style of nozzle had been introduced in order to reduce costs. The aerodynamic losses due to this thin plate design would be high. An estimate of between 1-2% of turbine efficiency improvement could be obtained by improving the aerodynamic design of the nozzle.

It was considered that the bearing arrangement of the SA105 would provide a lower maximum speed limit than that of the NA350. However, this was considered to be slightly academic as both turbochargers were limited more by the stresses in the rotors than the bearings. The dimensions of the NA350 turbine wheel were some 10% larger than the SA105. This meant that, in simplified terms, the limiting stresses would be approximately 20% greater for the NA350 than the SA105 at the same speed, or conversely the maximum speed of the SA105 was likely to be 10% more than the NA350. Both turbochargers were limited by the mechanical design of the compressor impellers. The shape of the rear of this style of impeller was described as "flat-backed" (Fig 5.5). As may be seen by the stress contours plotted in this figure the stresses increase sharply towards the bore at the rear corner of the impeller. This particular feature imposed the lowest operating parameter of both turbochargers, especially when a cyclic duty was being considered. A cyclic duty had been defined as one in which the turbocharger rotor would have to survive at least 5×10^6 excursions from stationary to maximum speed during its life. The limitations of this type of impeller design for duties such as rail-traction have received a great deal of attention (5.2). The compressors in both of these turbochargers had radial-vaned impellers. The type of impeller has the advantage that the rotational speed for a required pressure ratio is lower than a backswept wheel, other parameters being equal. This means that the radial impeller may have a distinct stress advantage. However, the compressor with a backswept impeller has now almost completely replaced the compressor with a radial impeller for new designs of turbochargers because of its superior performance. Figure 5.6 shows a compressor characteristic for the SA105. Comparison with figure 3.1, a characteristic for a similarly-sized backswept compressor , indicates clearly the reason for this. The backswept characteristic has a much wider operating range and the peak efficiencies are in an area of the characteristic that can be fully utilised.

5.3 Performance levels

The compressor performance has been mentioned, and it was clear that in this respect the NA350 and SA105 were similar, both units having radial impellers. From figure 5.6 it may be seen that the peak total/static isentropic efficiency is of the order of 80% for this typical characteristic of both units. This peak efficiency is at surge and therefore this is not an area of the characteristic that may be utilised. The necessary margin between the engine operating line and surge under steady-state running depends on a number of factors. Account must be taken, for example, of transient operation and of turbine nozzle fouling in service. Generally during the matching process of the turbocharger to the engine a series of surge checks will be carried out. These may include running the engine at abnormal conditions that would tend to promote surge (5.3). As a general rule it is desirable to ensure the operating-line is at least 10% away from surge. Using the characteristic in figure 5.6 as an example, this gives 76% as the maximum efficiency operating. This is a representative value for this type of compressor.

The turbines of the two units were of different designs and considerable efforts were made to compare the performance. Included in this work was the effect of the relative size of casings. It was considered that the two parameters that most influenced the stage performance were the axial velocity to rotor speed parameter (V_A/U) and the stage reaction. The simplest way of considering these parameters for two specific similarly sized turbine was to use blade length (Dtip/Dhub) and nozzle area/rotor area (An/Ar) respectively. A great number of test results were considered in order to correlate the two turbines performance data. The trends obtained by this means were then theoretically substantiated using the analysis methods outlined in chapter 3. Figure 5.7 and 5.8 show some of the test results for the SA105 and NA350 turbine, at an expansion ratio of 2.5:1. One significant difference between the two designs was the relationship between the range of nozzle area to rotor area. The SA105 used seven blade heights, designated D-K as shown in figure

3.12. However, the NA350 used only three, designated 7-9. The effect of this was that the SA105 used only three nozzles per blade height whereas the NA350 used approximately seven. Therefore, the range of nozzle-area/rotor- area for the NA350 was considerably greater than for the SA105. From figure 5.7 it does not appear to be desirable to have a large range of nozzle-area/rotor-area. The test data shows that there is a great similarity between the performance level of the two turbines, when operation at the same condition is considered. It appeared clear that the NA350 had too few blade heights requiring too large a range of nozzle-area/rotor-area to achieve the same capacity coverage as the SA105.

The performance of the two turbochargers was similar. Both had radial compressors with flat-backed impellers, with consequent limitations on operating efficiency and maximum speed respectively. The turbine design philosophy was somewhat different, the result being that the SA105 was, in most applications, able to achieve a far better efficiency. In addition, the larger diameter of the NA350 meant that the SA105 could operate up to higher speeds for the same stresses. The design of the rotor and bearing system in both turbochargers had advantages and serious disadvantages, neither unit being entirely satisfactory. The capacity envelope of the SA105 was larger than the NA350 and this compared more favourably with the opposition and the requirements of possible customers.

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FIGURES

TABLES

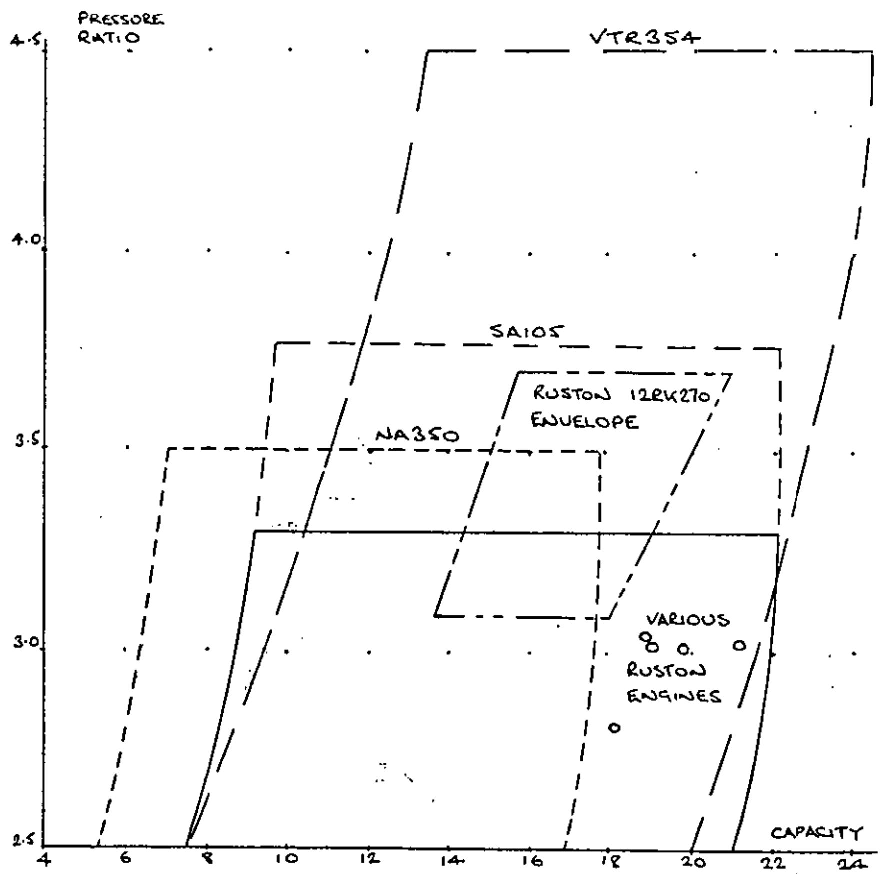
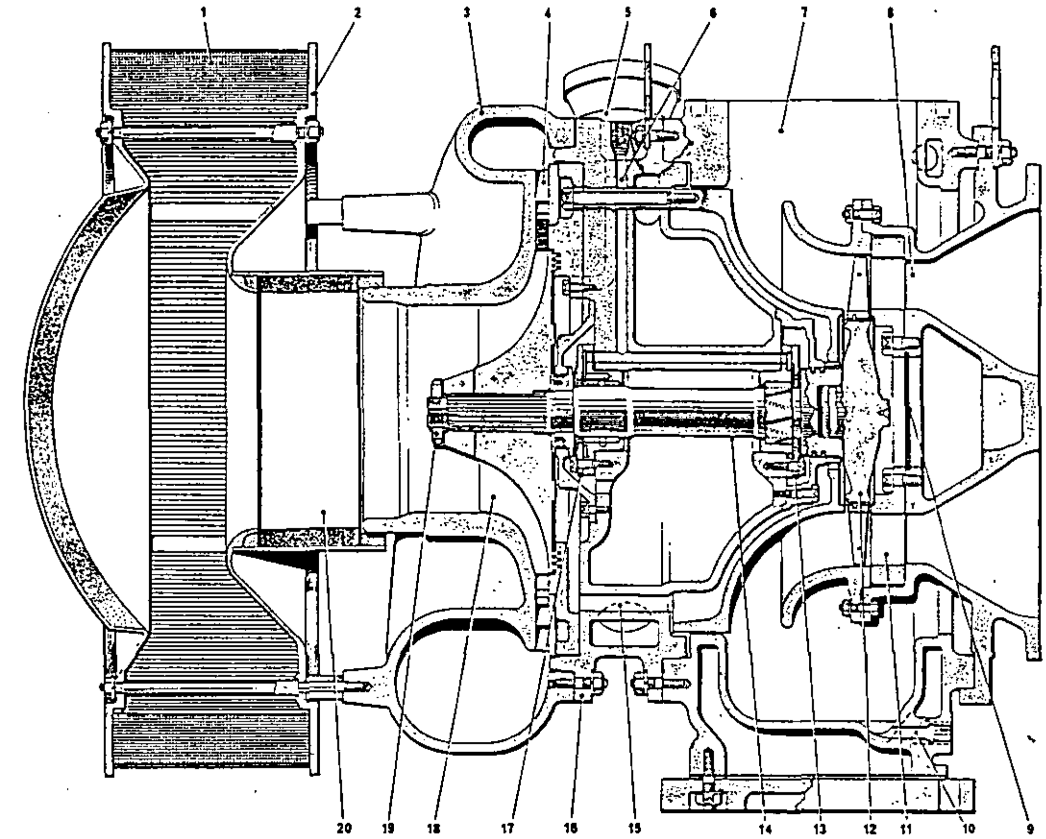


Figure 5.1: Capacity envelopes



1. Air filter
2. Air filter casing
3. Compressor casing
4. Diffuser
5. Breather
6. Oil inlet
7. Turbine outlet casing
8. Turbine inlet casing
9. Gas barrier plate
10. Gas passage drain
11. Nozzle ring
12. Turbine wheel
13. Bearing & sleeve housing (turbine end)
14. Rotor shaft
15. Bled air duct
16. Centre casing
17. Bearing & sleeve housing (comp. end)
18. Impeller
19. Ring nut
20. Sound absorbent lining

Figure 5.2: SA105 turbocharger

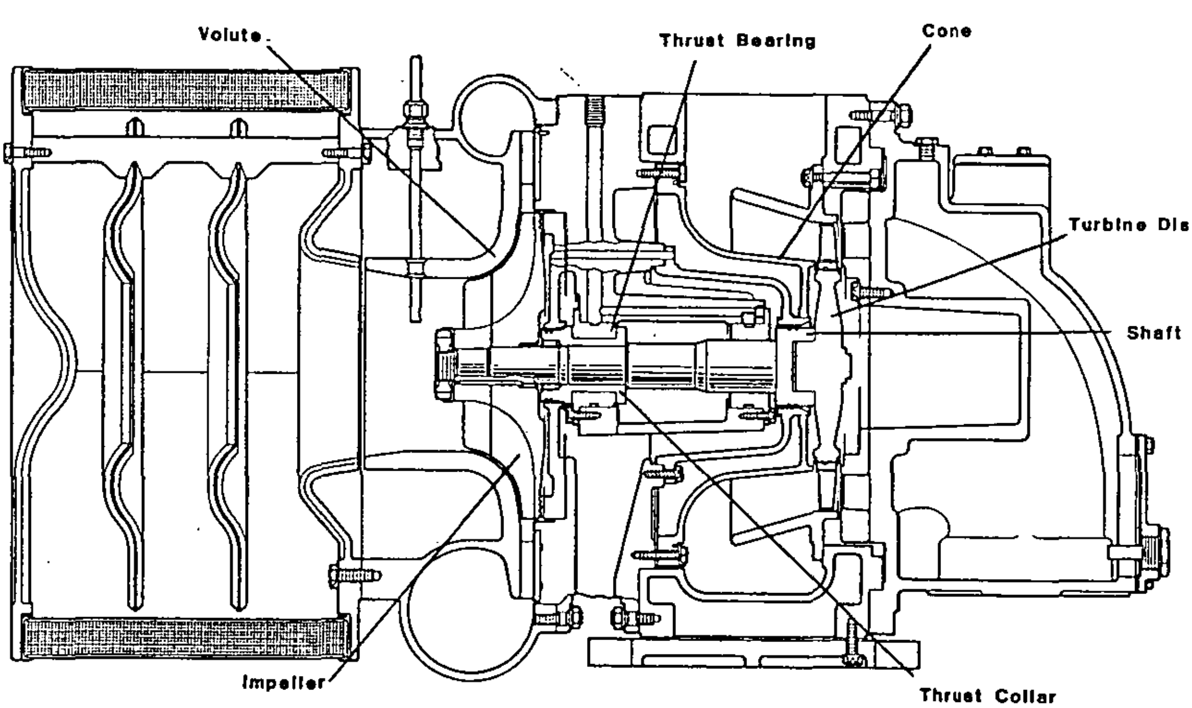


Figure 5.3: NA350 turbocharger

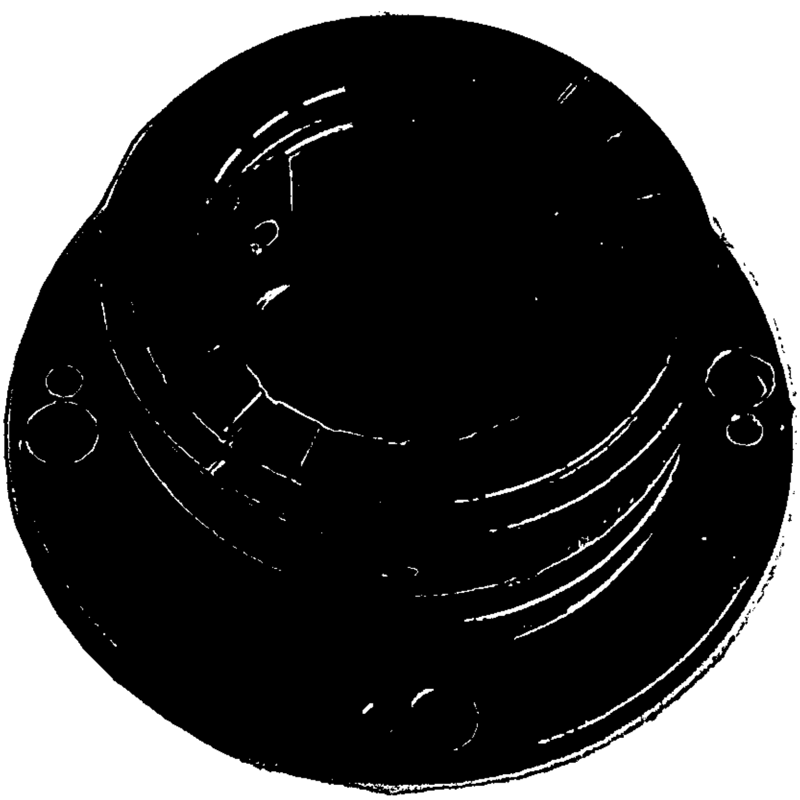


Figure 5.4: NA355 combined thrust and journal bearing

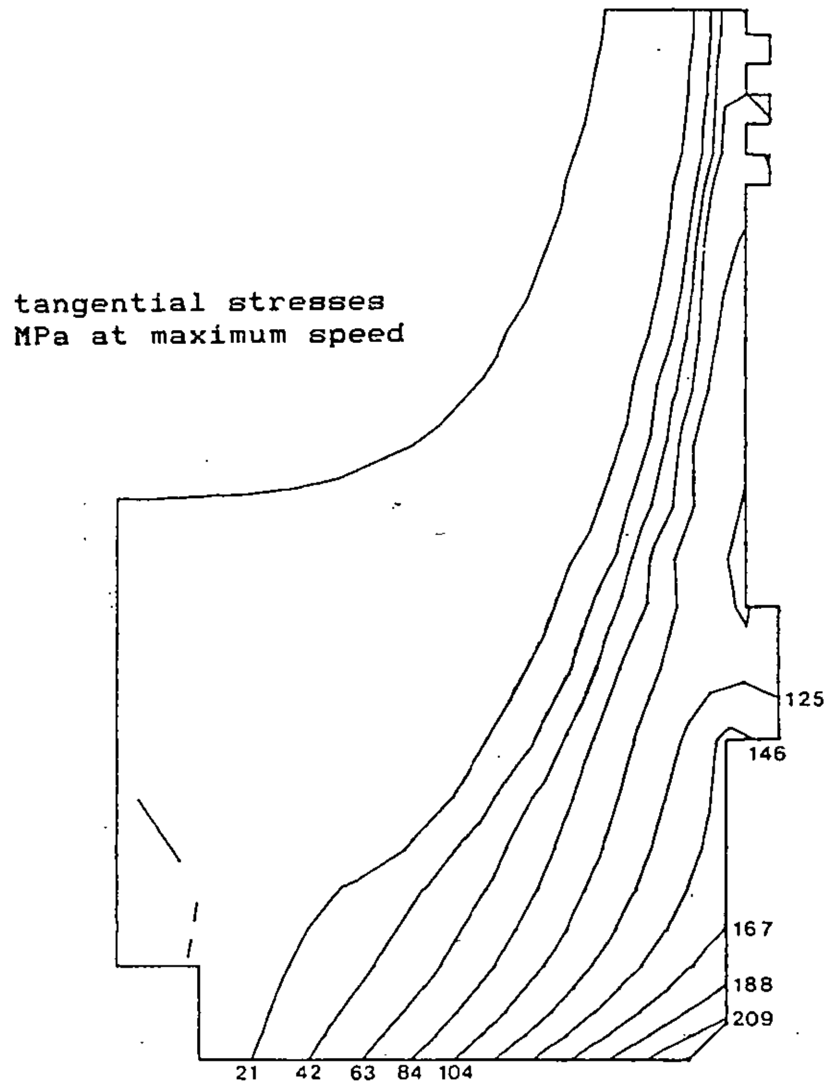


Figure 5.5: SA105 flat-backed impeller stresses

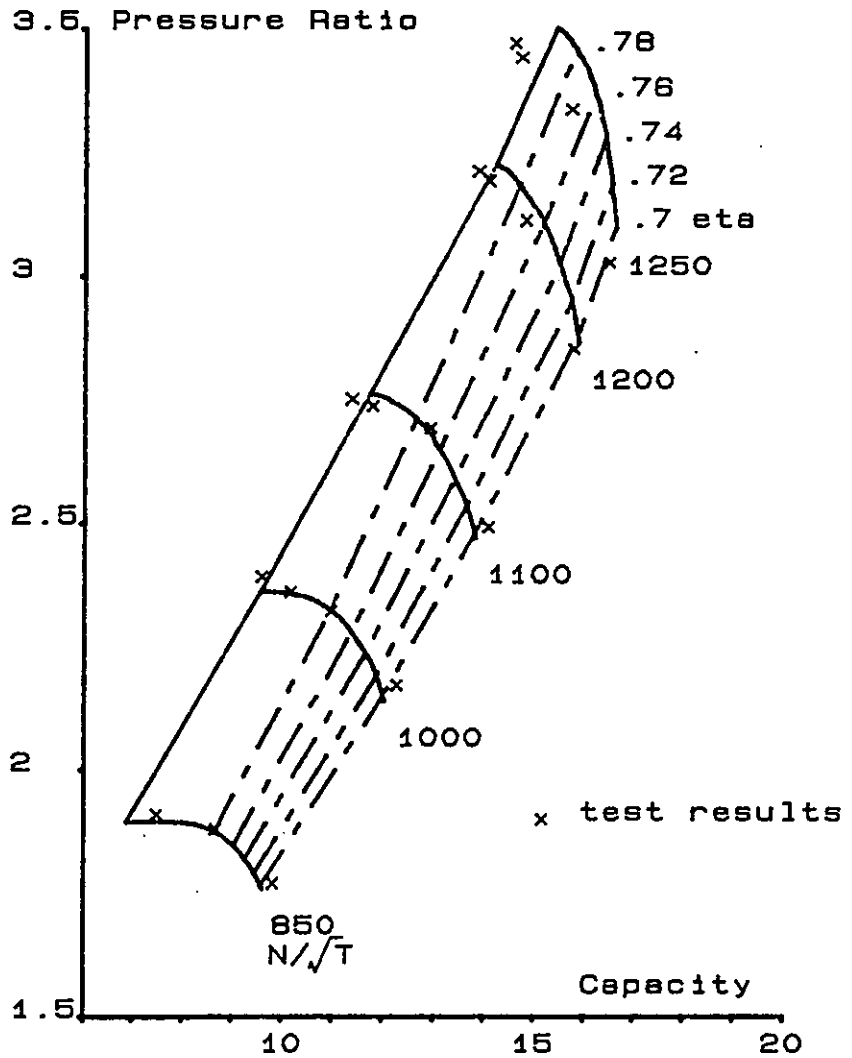


Figure 5.6: SA105 compressor characteristic

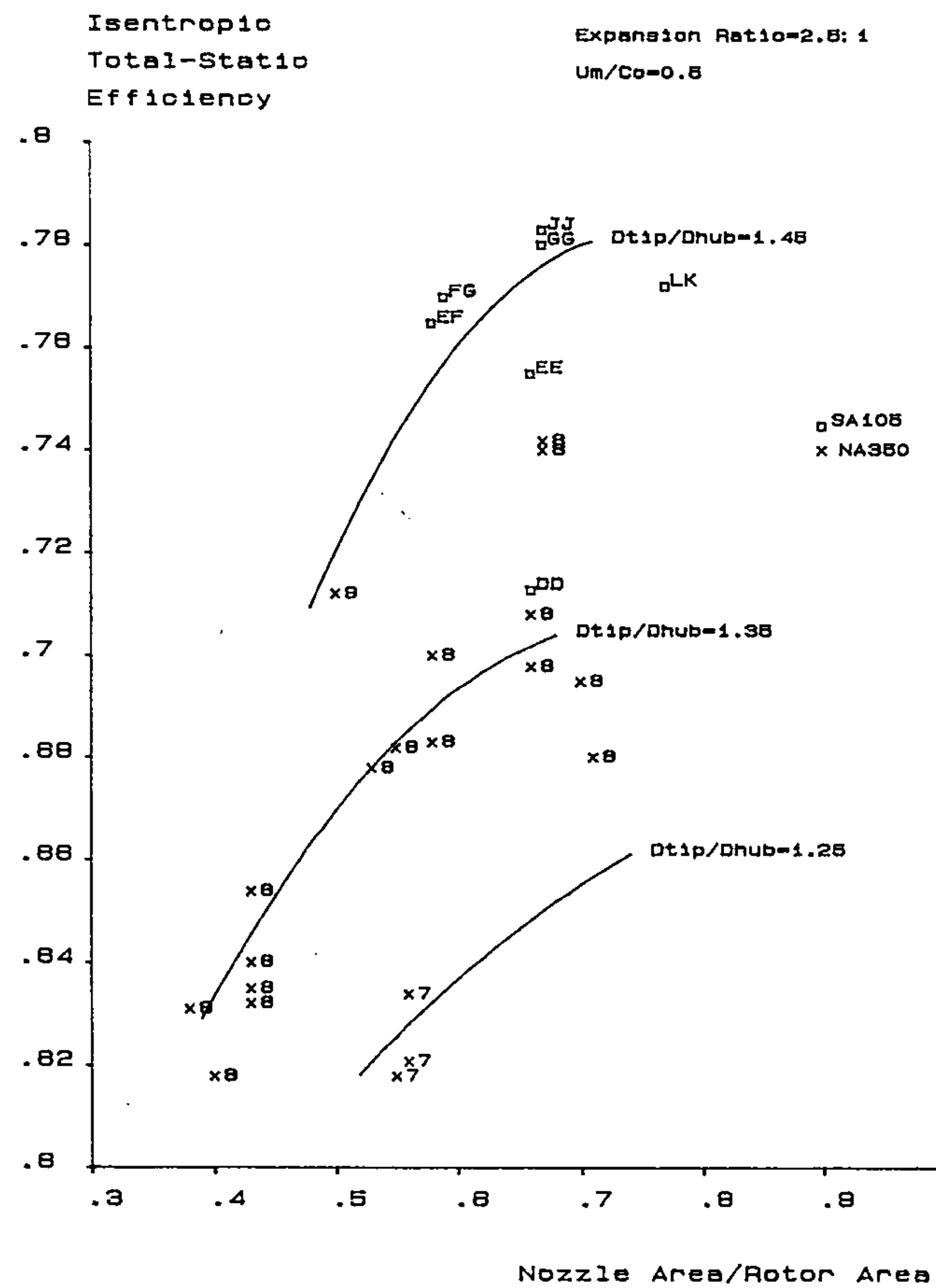


Figure 5.7: Correlation of turbine efficiency with An/Ar

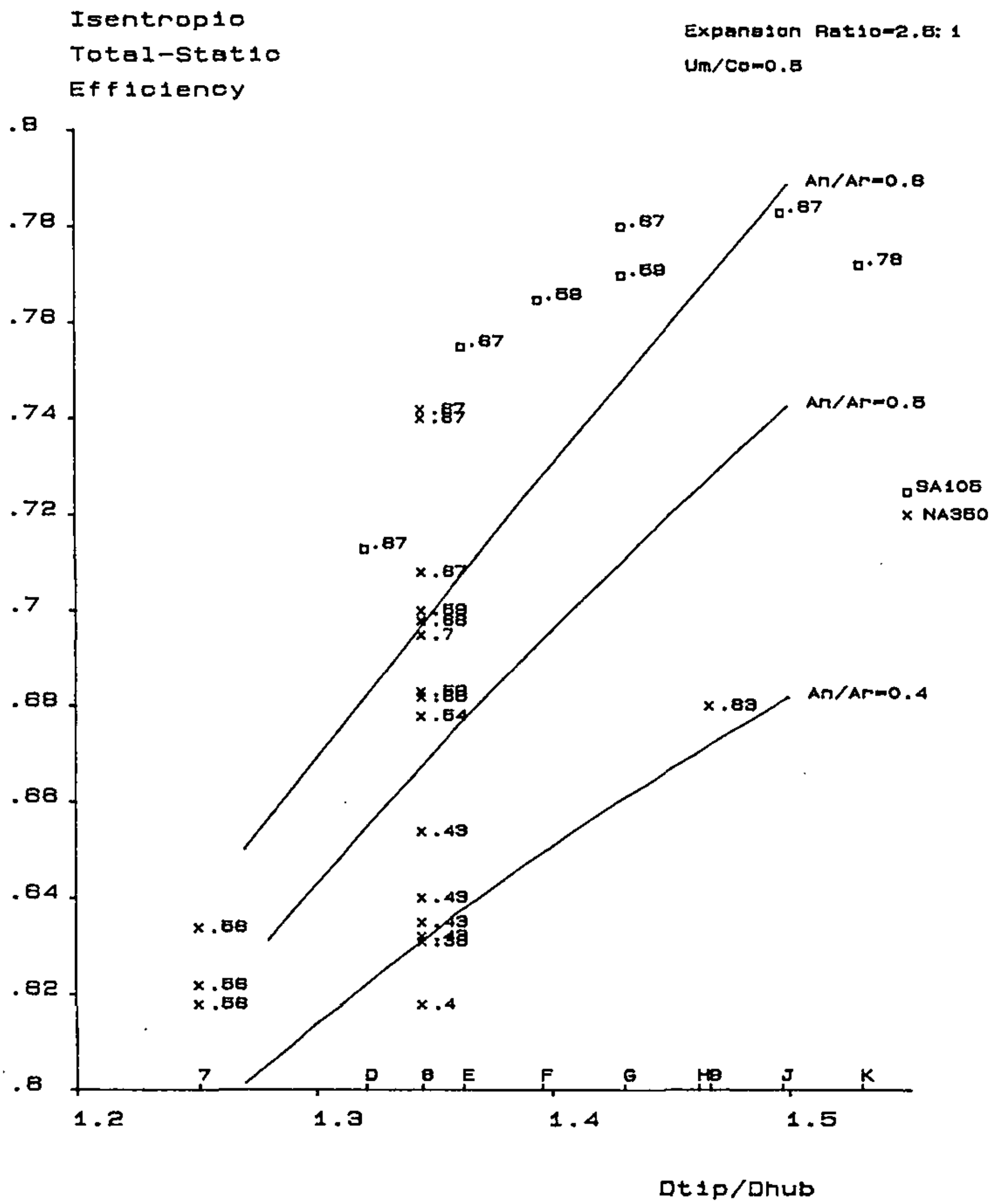


Figure 5.8: Correlation of turbine efficiency with Dtip/Dhub

	Length m *	Width m	Volume **	R_C	$m\sqrt{T_1}/P$ ***	Capacity/volume **
NA350	1.47	0.76	60	3.5	17.9	132
SA105	1.43	0.97	93	3.5	22.0	105
VTR321	1.42	0.76	58	3.0	18.1	138
VTR354	1.63	0.94	100	4.5	22.6	100

Nb. * Length adjusted for difference between axial and side entry casing

** Percentage of VTR354 value

*** Imperial pseudo non-dimensional units

TABLE 5.1: Turbocharger size and specific throughput comparison

CHAPTER 6

PROPOSALS

6.1 Assessment of company available technology

6.1.1 Compressor technology

The flatbacked design of impeller disc was known to be highly stressed. The design of the SA105 and NA350 had occurred during the late 1960's and early 1970's when pressure ratios and therefore speed requirements were lower than in the early 1980's. Also, as these impellers were both radially-vaned designs the pressure ratio to speed relationship was higher than the more modern backswept design. With the acceptance of backswept impellers and higher ratings it became necessary to be able to design the wheels for much greater tip velocities. A smaller frame-size called the C065 was the first turbocharger within Napier to incorporate a one-piece backswept impeller design. The rear of this impeller was profiled to reduce the peak stresses and therefore make better use of the material. This impeller had been stressed, using the finite element method, and this indicated that an impeller of the same type for the larger frame-size could be tackled with confidence. The experience gained with understanding the behaviour of the most commonly used material, forged aluminium alloy (BS1472 16TF), in highly cyclic operation provided further confidence.

A material that could have provided significant improvements was titanium alloy. However, use of this material presented a number of problems. Firstly, the material is notoriously difficult and time-consuming to machine. Secondly, the material was considerably more expensive than the aluminium alloy. Estimates of final component costs appeared to indicate that a titanium impeller would be at least five times as expensive as the equivalent aluminium alloy impeller. A comparison of the material and machining cost for the SA105 is given in table 6.1. Thirdly, Napier had no experience of machining titanium and so it was to be expected that a learning curve would have to be traversed before a new design in this material could be tackled. Therefore, introduction of a titanium impeller was to be avoided if at all possible. Other materials, mostly high-strength steel alloys, had been used for impellers for large air and gas compressors. However, these were considered to be generally undesirable from the rotor inertia point of view.

The aero-thermodynamic design technique for the design of a backswept compressor has been described in some detail in Chapter 3. This shows that the design of new compressor could be undertaken with confidence.

6.1.2 Turbine technology

At the time of the introduction of the replacement turbocharger for the SA105 and NA350 the technology for designing and assessing the turbine stage was available. In the case of a new nozzle design to replace an existing flat-plate design this was a well proven technique. The NA555 and NA455 had achieved notable improvements in turbine performance with the introduction of a prescribed velocity distribution (PVD) nozzle design. A proven figure between 1-2% improvement in turbine efficiency had been achieved by this means. However, in order to carry out an entirely new stage additional time for confirmation and validation of the design process would be necessary.

Stressing of the main turbine components was not considered to be a problem, although some geometric manipulation routines would be required. Either the mechanical design of a new stage or re-assessment of the limits of an existing design was considered to be within the in-house capabilities. It was appreciated that as the depth of understanding of the design was increased the time and cost of the analysis would increase. The intrinsic value of the manufacturing process also would provide constraints, particularly for the turbine design. For example, all Napier fir-tree roots were broached at that time. Each fir-tree design would require a number of broaches, so that sharpening could be carried out. The delivery of new broaches would take up to six months and would require time for process proving to ensure that the design of the broach was adequate. Therefore the introduction of a new design of fir-tree might take at least a year and could cost many tens of thousands of pounds. As a consequence of these constraints new designs of fir-tree roots were only carried only out when absolutely essential. This could mean that an undesirable design would be tolerated, rather than incur the delay and cost of a theoretically superior design.

6.1.3 Rotordynamics

At that time the available technology within the Company to predict the rotordynamic behaviour was limited. The Company had an analysis method for determining the critical speeds of rotor systems. The analysis technique was based upon the well-known Myklestad-Prohl method (6.1, 6.2) but using transfer matrices to model the rotordynamic system. The analysis of turbomachinery rotor systems by the use of these techniques is well established (6.3,6.4). The rotor is considered as a number of interconnected elements, each idealising a section of the rotor. The bearings have a great influence upon the dynamic behaviour of the rotor, and the fluid film is represented by four springs and four dampers. Determination of these oil film forces can be considered simply as in Reference 6.5 or in considerable detail as in Reference 6.6.

Analytical methods for determining the stability of the rotor system were not used on a routine basis and were not available within Napier, although such techniques had previously been used. It was hoped that the experience gained with the SA105 and NA350 rotor designs would provide an adequate platform upon which to base a new design.

6.1.4 Bearings

Hydrodynamic taper-land thrust bearing design capability within the company was considered to be adequate. In addition to a considerable amount of practical experience with this type of design analytical methods had been developed with company support (6.6). This analytical technique was based upon the solution of the Reynold's and energy equations using finite difference techniques. Account was also taken of the variation in viscosity of the lubricant with the developed temperature and hot-oil carry-over from one pad to the next was also considered. This work had been verified by extensive experimental work within the company as well as with other analytical methods available outside the company and with manufacturers data.

6.2 Achievable levels of performance

6.2.1 Turbine performance

Comparing the experimental turbine overall efficiency with the expected blading efficiency showed that there was a shortfall. Clearly, some losses could be attributed to the turbine inlet casing. This casing was, in most cases, shorter than desirable due to installational requirements. The effect of the length limitation was to create excessive incidence onto the nozzle leading edge, particularly at the end-of-sector position. The result of extensive air-flow work on a typical turbocharger turbine inlet casing (6.7) indicated that the incidence varied up to $\pm 30^{\circ}$ across the sector of a two-entry casing. The corresponding pressure loss coefficient - total pressure drop/inlet kinetic head - was shown to be as high as ± 2 also at the end of the sectors. The averaged value across the casing appeared to be much lower, of the order of 0.3. This suggests that the turbine inlet casing accounted for only a small proportion of the difference. This work also attempted to correlate the losses in the outlet casing, but this aspect of their work appeared to be less convincing, probably due to the significant effect that the simulated rotor exit swirl had upon the results. At this stage in the project it was realised that the absolute value of achievable turbine efficiency was not well understood when compared with the existing test results. This resulted in the formation of a collaborative research programme of work to investigate and understand the losses within the turbine for the purpose of effecting improvements (6.8,6.9).

The design of a turbine for a specific steady-flow duty can be difficult. However, the design of a non-steady flow turbine for high efficiency when significantly reduced in size to achieve a smaller capacity and over a range of operating expansion ratios can become extremely complex. In the case of a turbocharger turbine the peak efficiency at one operating point is only one of the many significant parameters that have to be considered.

Although the basic means of analysing typical designs were available, it was clear at the start of this project that this work would take a considerable amount of time and effort to become fruitful. The study of the comparison between the SA105 and NA350 turbine results had well supported the superior performance of the SA105. However, insufficient work had been carried out to indicate how the level of turbine performance that was being achieved by the SA105 could be significantly increased over the whole range. By considering results such as those shown in Figures 5.7 and 5.8 the build of SA105 turbine could be chosen to enable an optimum efficiency to be obtained for that design of turbine. Figure 6.1 was produced from figures 5.7 and 5.8 and shows how the efficiency becomes related to the capacity function. In the example shown in figure 6.1, if the vertical dotted line represents the desired capacity requirement this can be achieved by all of the blade heights shown. However, the longest blade is clearly the most efficient being nearly four points in efficiency better than the smallest blade. This result is complicated by the effect of pressure ratio.

At low expansion ratios the efficiencies of large An/Ar turbine builds can become greater than the efficiencies of low An/Ar turbine builds, for the same capacity function. This type of correlation of turbine efficiency with Dtip/Dhub and An/Ar ensured that the most suitable build was adopted for a particular application. The test results were from specific turbochargers, tested as open cycle gas turbines, and therefore the velocity ratio parameter (Um/Co) was virtually constant.

The position of the peak efficiency can be shown to be dependent upon the degree of reaction. For example, the efficiency of a low reaction turbine around 5% peaks at a Um/Co of 0.4, whereas the 50% reaction turbine will peak at a Um/Co of 0.7. For the SA105 design the degree of reaction was about 50% at the mean diameter of the largest blade reducing to almost impulse at the root. Therefore the degree of reaction may be seen to vary with blade length or Dtip/Dhub. Some tests were carried out to determine the efficiency peak position of one blade height for the SA105 design by the use of different impeller diameters. The result is shown in figure 6.2. Unfortunately these results do not truly indicate the position of the peak efficiency. However, by considering other test data it was concluded that the peak efficiency occurred at a Um/Co of 0.55 for a mid-size blade. As mentioned, the theoretical work tended to support the performance achievement of the SA105 without providing significant ways of improvement. It was therefore concluded that, at least for a initial period, it was desirable to base the possible performance achievement of the new turbochargers upon the SA105. Additional improvements were considered to be possible by the introduction of a profiled nozzle and by the use of the techniques for determining the optimum efficiency build as described above. For the purpose of performance estimation it was considered that an overall total-static isentropic efficiency of 80% should be used as a target value.

6.2.2 Compressor performance

The level of performance that the compressor might achieve was far more readily available than for the turbine and could be based on published work carried out for high performance centrifugal compressors for gas turbines. The standard compressor stage of a turbocharger is quite closely related geometrically to that of a gas turbine centrifugal compressor. Published results presented of the research, design, and development of the backswept centrifugal compressor showed that a total-static isentropic efficiency of 80% was achievable with an acceptable surge margin(eg 6.10,6.11,6.12).

The operating range of a turbocharger compressor can be very different to that for a gas turbine. Typically, this can mean that the largest compressor mass flow can be three times the smallest for basically the same unit (6.13). The adjustment in flow was achieved, for economic reasons, by cropping the impeller channel, and by a range of diffuser sizes, as described in Chapter 2. The peak diffuser efficiency would vary over the range. However, this was not considered to be a major effect and for the purposes of design a constant value of the peak diffuser efficiency was considered appropriate. The value of impeller peak efficiency also would change as a result of the cropping. In this case published work (6.14) indicated that the impeller efficiency could be correlated with specific speed. To a first order of accuracy the specific speed parameter may be seen to vary directly with the impeller eye tip diameter, for a particular wheel. The data shown in figure 6.3 (6.14) indicated that significant losses can be incurred at extremes of specific speed. However, for typical eye tip diameter/impeller diameter ratios it was considered that the reduction in efficiency could be small, particularly if values of specific speed close to optimum were maintained.

6.2.3 Impeller mechanical design

The impeller disc stresses were considered to be a likely limitation on the achievable rotor speed. As mentioned above, a smaller backswept design with an improved rear surface profile had been stressed. This enabled a relationship to be obtained for the maximum wheel tip velocity, thereby enabling the maximum speed of a particular impeller diameter to be determined. This maximum speed, together with the backsweep angle, would allow the maximum pressure ratio to be determined. Using the same criteria for both designs it was calculated that a figure of 464 m/s was a typical maximum operating speed for a flat-backed design and the corresponding value for a profiled disc was 542 m/s. This represents an increase of over 17% in maximum speed achievable. In terms of maximum achievable pressure ratio this represents a very significant increase. Assuming that the tip speed of 464 m/s represents a pressure ratio of 3.0:1, the following calculation shows the effect of an increase to 542 m/s.

For a constant slip factor and isentropic efficiency :-

$\Delta T \quad = \quad \text{constant}.U^2$

$k \quad = \quad \text{constant}.\Delta T$

where $k = R_C^{((\gamma-1)/\gamma)} - 1$

$k \quad = .3687$ when $R_C = 3$ and $\gamma = 1.4$

$k' \quad = .503$ corresponding to an increase to 542 m/s

$R_C' \quad = 4.163$

6.3 Proposed turbocharger configuration

The previous sections indicate that the SA105 turbine , with minor modifications, represented a level of performance that would be difficult to improve. A programme of research work had been instigated that would provide the means of improving the level of turbine performance. It was therefore proposed that the SA105 turbine, with the additional modifications, was used until the information became available that would allow a new stage to be designed and introduced. This would also allow the proposed turbocharger to be used as a development vehicle, at a future date, to validate thoroughly any possible new turbine stage.

The compressor and turbine diameters are related because the machine is a single shaft unit and the turbine power produced is absorbed by the compressor. The relationship is given by:

$$Um/Co = \left(\left(\eta_T \ (Dm/Dc)^2 \right) / \left(2 \ \Delta Hc/Uc^2 \right) \right)^{0.5}$$

In the case of the radial compressor the work parameter ΔH_C remains substantially the same value at constant speed. However, the work parameter for a backswept impeller will vary at a constant speed and be dependent upon the compressor map position. However, for the purposes of determining the relative diameter relationship a nominal value of 0.7 for $\Delta Hc/Uc^2$ was used. This value corresponds to a backsweep angle of approximately 30°. The optimum choice of backsweep is a compromise between increasing map width and tip speed. It was considered that a backsweep of 30° would provide an adequate surge-to-choke range whilst allowing a sufficiently high pressure ratio to be achieved. In addition, this value of backsweep coupled with a tip-speed limit of 542 m/s, with profiled rear surface, corresponded to a maximum pressure ratio of over 4.5:1 provided the compressor efficiency was maintained. Thus, using these values together with a turbine efficiency of 0.8 and Um/Co of 0.55 gives:

Dm/Dc = 0.743 and for a Dm = 308.5mm Dc = 425mm

For the range of turbine diameters the range of Um/Co would then be 0.58 for the largest blade height to 0.52 for the smallest. Figure 6.2 tends to indicate that reducing the impeller diameter, thereby increasing the nominal Um/Co may well have been beneficial. However, the values chosen were considered to represent a reasonable compromise particularly for the higher pressure ratio requirement of the proposed turbocharger.

The previously designed backswept compressor for a smaller turbocharger was considered with respect to the required capacity envelope, figure 5.1. Scaling the maximum flow range for the new compressor from the smaller design resulted in a capacity value of 23.7 at a pressure ratio of 3.0:1. This compared well with the capacity value of the VTR354 of 24 at a pressure ratio of 4.0:1. This confirmed that the impeller diameter choice was close to an optimum value as any reduction would have made the required maximum capacity more difficult to achieve.

As previously discussed it was considered necessary to introduce an improved rotating assembly and bearing system in order to achieve the speed requirements. This was in addition to the modifications that would be necessary simply to incorporate a new compressor impeller.

It was appreciated that some new casings would be required although the introduction cost of these items would have to be considered. Up-to-date estimates for the pattern and tooling costs for the major SA105 casings were obtained, thereby providing indications of the order of the cost of a newly-designed casing so that the introduction of these items could be justified:

Casing	Pattern & Tooling (in £'s 1981)
Turbine outlet casing	4000
Turbine inlet casing (ref. 2 entry axial)	2000
Centre casing	1500
Compressor delivery casing	1800

For a turbocharger with an expected yearly quantity of the order of 20 and a selling price little more than £10k these figures represent a fairly modest investment. However, the major casings only represent a small part of the total introduction cost.

In brief it was proposed that the new turbocharger should look externally similar to the SA105, having substantially the same turbine and casings wherever possible. The compressor should incorporate a backswept impeller and the turbocharger should be capable of operating to the identified higher level in terms of pressure ratio and performance. To achieve the higher rating in terms of speed meant that a number of changes in the rotating assembly should be introduced. The new range of turbochargers, at that time replacing one smaller and one larger unit that had incorporated backswept compressors had been designated the "5 series", and so the appropriate designation for this new unit was the NA355.

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FIGURES

TABLES

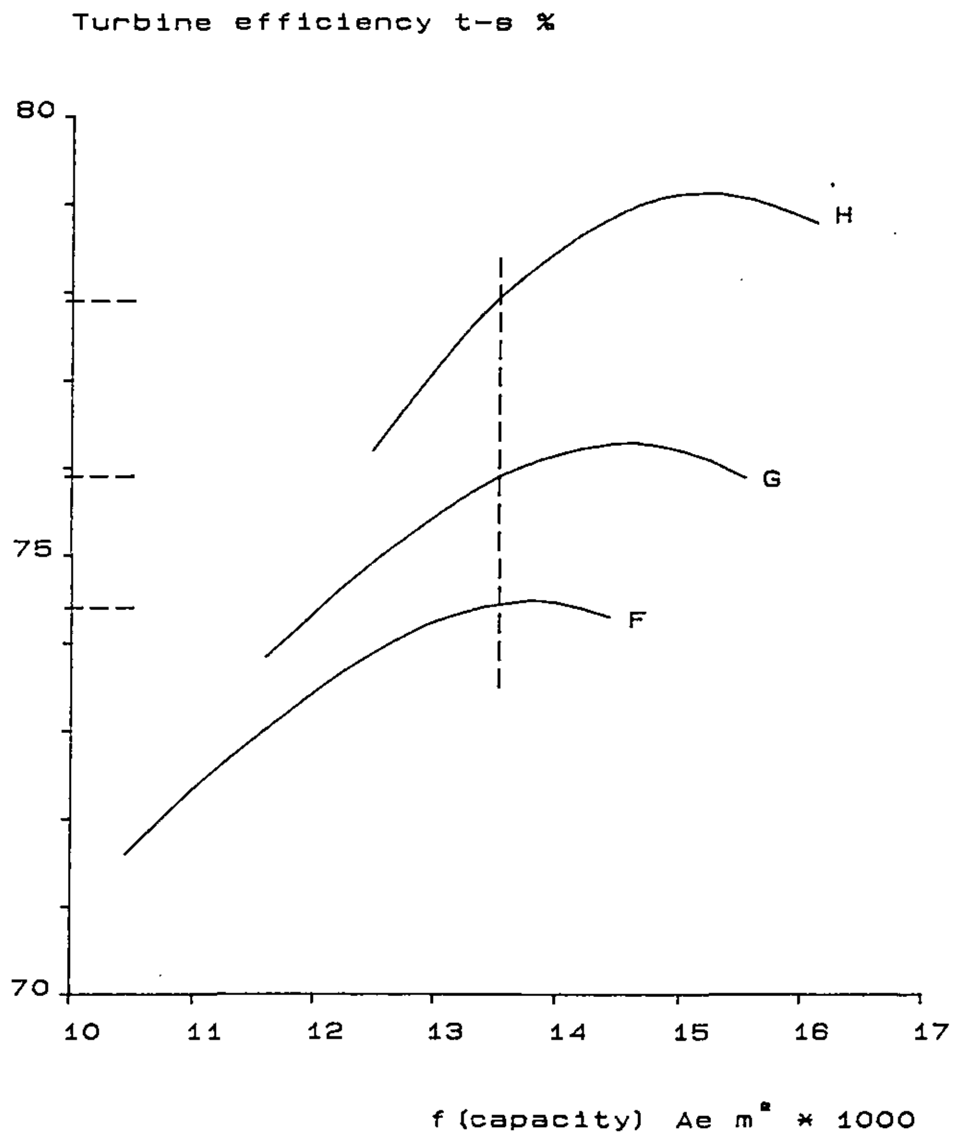


Figure 6.1: Turbine efficiency versus flow relationship

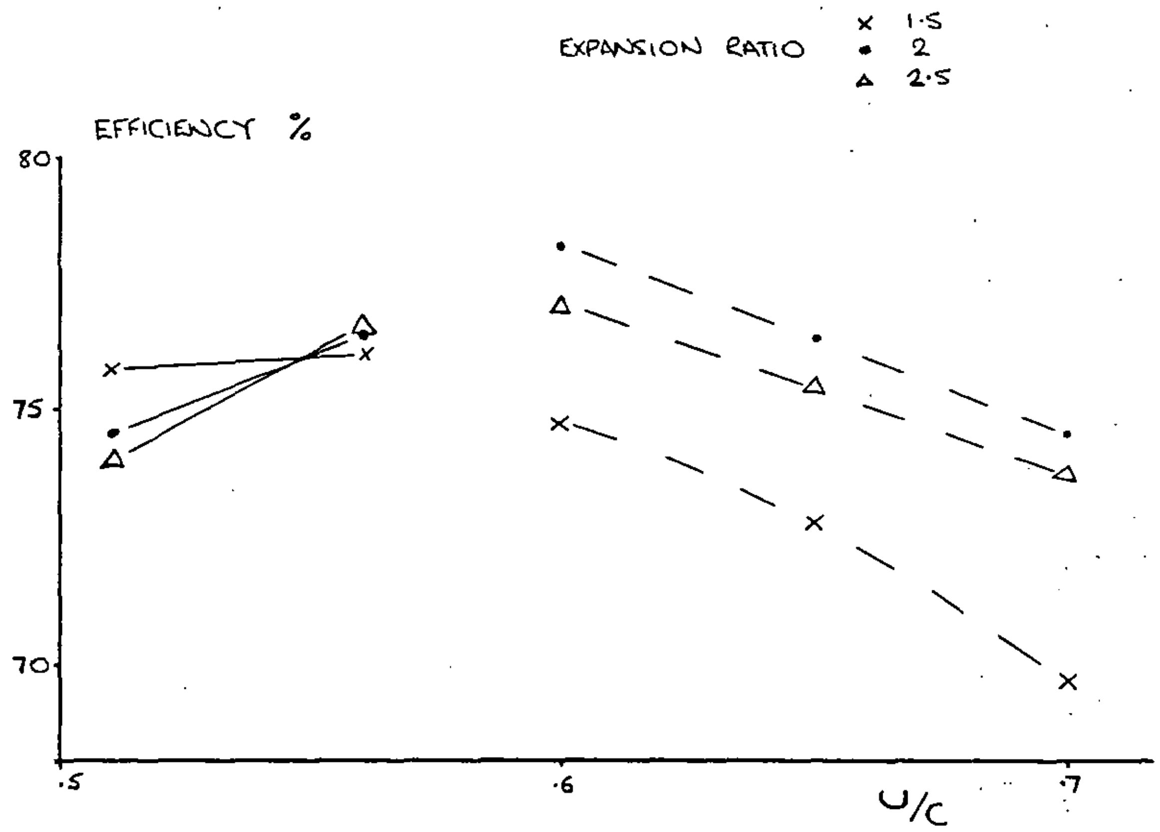
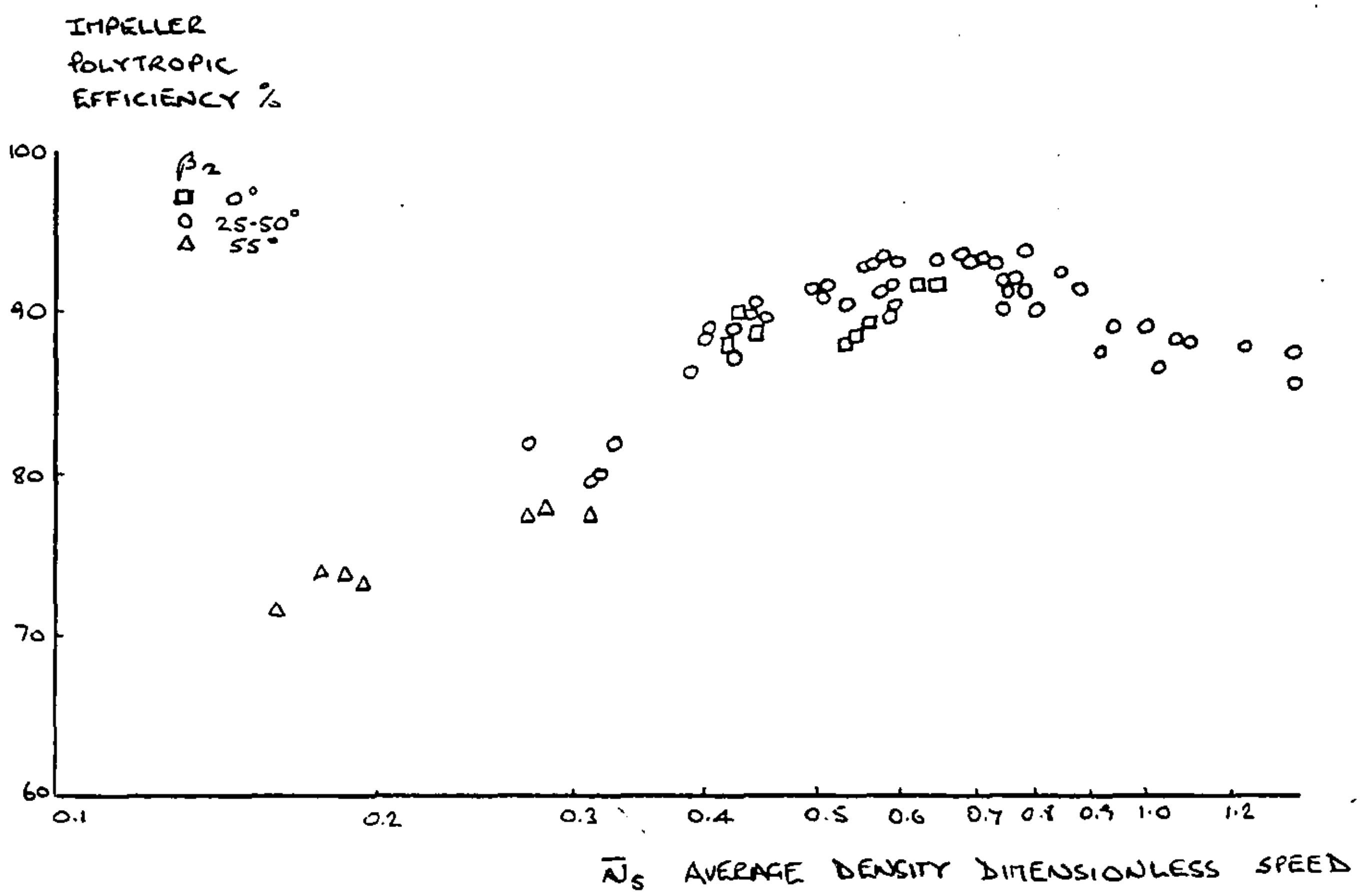


Figure 6.2: Turbine efficiency trends with U_M / C_O



$$Q_{1-2} = (w / \rho_1 + w / \rho_2) / 2$$

$$\Delta h_2 = c_{p2} T_2 - c_{p1} T_1$$

$$\overline{N}_s = \frac{(2 \pi N / 60)(Q_{1-2})^{0.5}}{(\Delta h_2)^{0.75}}$$

$$\eta = \frac{Rc^{A} - 1}{Rc^{B} - 1}$$

where $A = (\gamma - 1) / \gamma$

$B = (\gamma - 1) / \gamma \, \eta_{poly}$

Figure 6.3: Impeller efficiency versus average specific speed

	forged aluminium alloy	forged titanium alloy
Material cost £/kg	4	21.21
Material cost (based on 73.3 kg cheese)	293	1555
Machining cost (based on SA105 impeller)	416	2100 min.
Total	709	3655

TABLE 6.1: Cost implication of use of titanium for impellers

CHAPTER 7

COMPANY TARGETS

7.1 Company objectives

During the period when the NA355 configuration proposals were being prepared the Company was developing a number of new products. The means that the Company chose to direct these projects was by a meeting called the "New Products Meeting". This was a gathering of senior and middle management across the Company. Nominally the meetings were held every month. The purpose of the meetings was to ensure that the Company was directed along an agreed path that was generally in accord with the Company policy decided at the Management Meeting - a monthly meeting of departmental heads. The Company policy was to ensure that all "5 series" turbochargers progressed as quickly as possible within the limitations of the resources available and the customer requirements. The Management Meeting then delegated the responsibility for the determination of specific targets to the new products meeting. One of the major objectives of the meeting was to ensure that the lines of communication between departments were as clear and open as necessary.

The minutes of the new products meetings during this period briefly record the history of the NA355 as the turbocharger slowly developed. The conflict between the NA355 and many other projects created many changes and slippages to the original requirements. The relevant part of the minutes of these meetings is included in appendix 7.1.

7.2 Design reviews

At the start of this project there were a large number of meetings within Engineering Department. The majority of these were informal, with the object of resolving minor difficulties or communicating decisions made by members of different areas. The formal meetings that were held mainly addressed the design concepts of the new turbocharger as covered in Chapter 6. For example, five meetings were held during 1982, with various proposals and work being reported as a result of these meetings. Those involved in this project were a clearly identified group of people. Although this group worked on other projects during the period, as can be seen by the slippage that occurred, they remained involved in it at all times. The purpose of the formal meetings was to review the design and to ensure that the project proceeded in a technically competent manner. The meetings also provided a means of reminding those involved of the aims of the project and also communicating a knowledge of overall activity to the group.

7.3 **Project control**

A requirement was stated at the first of the New Product Meetings dealing with the NA355 for a project bar-chart (Gantt chart). The data were collected from the individual departments. The result is shown in table 7.1. It is not too surprising that by the time this was issued the information was out-of-date, as the work involved was not inconsiderable. Also the exercise was not repeated on a regular basis owing to the lack of resources. To have controlled the project in this manner would have taken a great deal of tedious effort throughout the project timescale. The number of new components was, at this early stage, identified as 23.

The major advantage of using numerical management analysis techniques to control a project is that the individual departments have to make estimates of the activities and times, for all of the components. A major criticism of a manually completed bar-chart is that updating the chart is only likely to be carried out at lengthy intervals due to the effort involved. This then largely negates the advantage of providing the information. It may also be seen that completion of the bar-chart in this case was rather arbitrary in terms of manpower resources. For example, from table 7.1 it is possible to work out the manpower resources week by week for all the different areas to complete the schedule. This is shown for the design function in figure 7.1. It may be clearly seen that the number of designers working on the project fluctuates widely throughout the period. Also, there is no distinction between the various functions that have been called design, and process. The activity called design was in reality the drawing activity. A better breakdown of the activities was needed that was more aligned to the company separated functions.

Some consideration was given to means of improving the control of this, and similar, projects. Firstly, the activities undertaken within the company were rationalised:

Specification
Design
Drawing
Planning
Equipment design
Equipment procurement
NC programming
Material procurement
Machining

This list provided the maximum number of steps through which a component would have to pass from specification to completion. Clearly some components would not have to pass through all of these activities. For example bought-out-finished items would probably pass through specification, design, drawing and material procurement. The identified aim of the project control technique was to provide a flexible and easily updated means of determining the project completion date throughout the project consistent with the resource constraints.

Further consideration was given to the order in which a component would pass through these activities. In critical path analysis terms, the activities were considered from an "immediately preceding activity" point of view. In the main a component would pass sequentially through the list, with the exception that the material procurement may follow the drawing, planning, or other activity in exceptional cases. This appeared to be an extremely simple network.

The amount of available manpower was one of the major resource limitations of the project. Also the fluctuation of manpower shown in figure 7.1 was unacceptable. A more efficient way of using the available manpower was required. The technique that was considered to be most appropriate was "resource-limited scheduling". In addition, for the exercise to be meaningful, a far more flexible means of presenting the reduced data was needed.

A number of available software packages was considered and compared with the requirements. None seemed suitable. Either the particular package was too simple to deal adequately with resource-limited scheduling, or else was far too demanding in its data requirement to allow data to be easily entered and modified. Most of the available software was intended to deal with the solution of complex networks with variable activities, and was not ideally suited to solving the problems of this project.

It was decided therefore that a resource-limited scheduling computer program would be written in-house. First the available literature was studied(eg. 7.1,7.2,7.3). An appropriate course of action was then determined. The most important point was to ensure that the estimates of all of the various activities for all components were reasonably accurate. Possibly the best method is the "three-time estimating" technique (7.4). This technique uses three estimates of the times:

1 optimistic time
2 pessimistic time
3 most likely time

The calculated time then combines the three estimates and weights the most likely time:

expected time = (optimistic + 4 * most likely + pessimistic)/6

This technique can be adjusted when the tendencies of those providing the time estimate become known. However, it is important that the confidence of those providing the data is maintained, and therefore any adjustment of the data has to be carried out with extreme caution.

Therefore it was possible to provide:

1 An estimate of times for all of the activities for all of the components. Noting that some activities for some components would have zero times.

2 A list of the immediately preceding activities for all activities and for all components. It should be noted that a decision was made to have only one preceding activity. This was considered to provide an adequate level of network complexity for this particular problem.

It was necessary to devise the rules that governed the allocation of the limited resources. These are as follows:

1 Calculate the total times for each component taking the network into account, assuming unlimited resources.

2 Determine the descending order of the total times of all the components

3 Search for all completed activities for all components

4 Search for all activities for all components available to start

5 Determine the available resources for all activities

6 Assign the available resources in descending order of total times to the activities available to start

7 Move to next increment in time

8 Add 1 to all of the identified activities available at the previous time step to start without allocated resources

9 Repeat 1-8 until all activities for all components have been completed.

Appendix 7.2 provides a simple numerical example of the algorithm used to determine the scheduling of this type of project, incrementing through the project timescale. The required software was developed with unsophisticated input and output routines tailored to this specific application.

The data presented in table 7.1 was re-arranged to suit this program as shown in table 7.2. The output from the program, initially for unlimited resources is shown in table 7.3. This provides a datum result to show the minimum completion time for all components. Table 7.4 presents the output for the actual resources available. The material and equipment procurement were external to the company activities and were considered to be independent of resources. Also, as the manufacturing system was computer controlled the component machining times provided were, for all practical purposes, independent of available resources. It should be noted that the overall completion date does not change from table 7.3 to 7.4, the lead item being the impeller. The float, that is the time within the overall timescale that any activity may be delayed without affecting the end date, has clearly been reduced significantly.

Table 7.5 presents the result of a "what if?" situation. In this case the process resource was reduced from 2 to 1. It may be seen that the overall timescale was extended by 3 weeks. Figure 7.2 shows how the drawing activity has been scheduled for comparison with that produced manually shown in figure 7.1. This clearly shows the benefit of this technique taking the scheduling from a greatly variable requirement that exceeded the total resources to a virtually level requirement over the period.

Unfortunately, the software development was too slow to be of great benefit in this project. However, the problems associated with trying to predict the completion date for this project provided the incentive to complete this work. Later projects were able to be controlled in a much better fashion, largely as a result of the lessons learnt in this project.

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FIGURES

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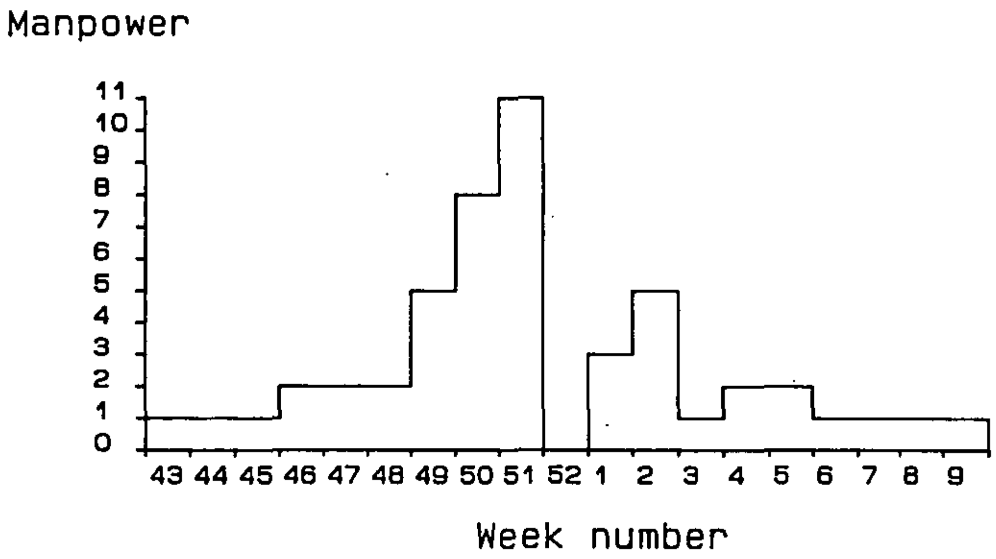

Figure 7.1 Manually scheduled drawing activity

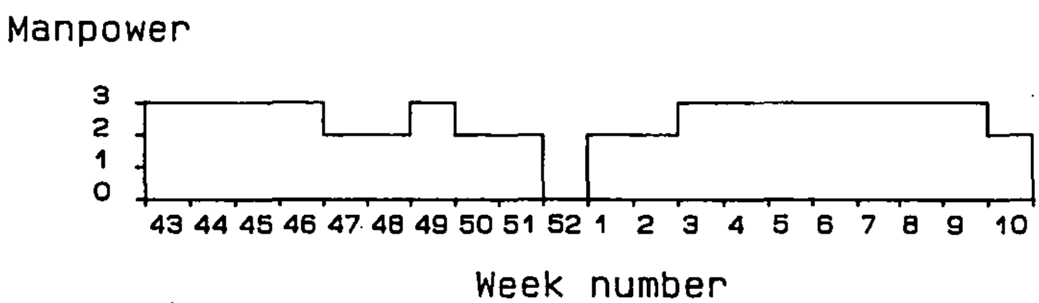

Figure 7.2 Resource limited scheduling drawing activity

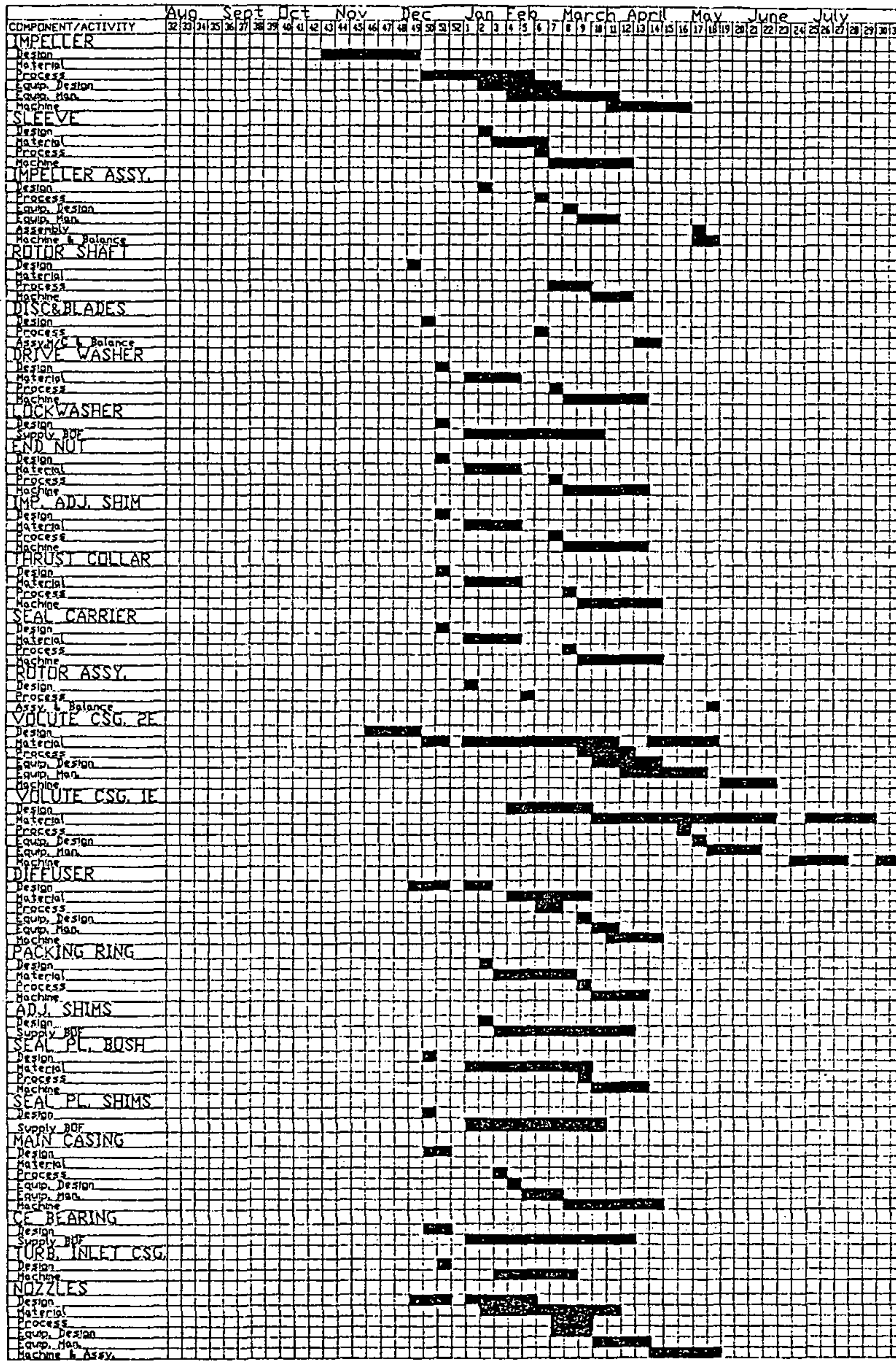

Table 7.1 : NA355 project programme

Resource Limited Scheduling Program
******** ****** ********** ******

INPUT DATA

		Drg 1	Mat 2	Proc 3	Equ D 4	Equ P 5	M/C 6
Impeller	time	7	0	8	6	8	6
	ipa	0	0	1	3	4	5
Sleeve	time	1	4	1	0	0	6
	ipa	0	1	1	0	0	2
Imp. assy.	time	1	0	1	1	3	1
	ipa	0	0	1	3	4	5
Rotor shaft	time	1	0	3	0	0	3
	ipa	0	0	1	0	0	3
Disc+blades	time	1	0	1	0	0	2
	ipa	0	0	1	0	0	3
Drive washer	time	1	4	1	0	0	6
	ipa	0	1	1	0	0	2
Lockwasher	time	1	10	0	0	0	0
	ipa	0	1	0	0	0	0
End nut	time	1	4	1	0	0	0
	ipa	0	1	1	0	0	0
Imp adj shim	time	1	4	1	0	0	6
	ipa	0	1	1	0	0	2
Thrust coll	time	1	4	1	0	0	2
	ipa	0	1	1	0	0	2
Seal carrier	time	1	4	1	0	0	6
	ipa	0	1	1	0	0	2
Rotor assy	time	1	0	1	0	0	1
	ipa	0	0	1	0	0	3
Volute 2E	time	4	20	4	5	6	4
	ipa	0	1	1	3	4	2
Volute 1E	time	6	20	1	1	4	8
	ipa	0	1	1	3	4	2
Diffuser	time	5	6	2	1	2	4
	ipa	0	1	1	3	4	2
Packing ring	time	1	6	1	0	0	4
	ipa	0	1	1	0	0	2
Adj shims	time	1	10	0	0	0	0
	ipa	0	1	0	0	0	0
Seal pl bush	time	1	9	1	0	0	4
	ipa	0	1	1	0	0	2
Seal pl shim	time	1	10	0	0	0	0
	ipa	0	1	0	0	0	0
Main casing	time	2	0	1	1	3	7
	ipa	0	0	1	3	4	5
CE brg	time	2	12	0	0	0	0
	ipa	0	1	0	0	0	0
T I csg	time	1	0	0	0	0	6
	ipa	0	0	0	0	0	1
Nozzles	time	8	10	3	3	4	5
	ipa	0	1	1	1	4	5

Nb ipa - immediately preceding activity

Table 7.2 : Resource-limited schedule program - Input data

Elapsed Time Data
******* **** ****

(start of wk/end of wk)

Holiday week nos. 52,

Activity -	Drg	Mat	Proc	Equ D	Equ P	M/C
Manpower -	23	23	23	23	23	23
Impeller	43/49	0/ 0	50/ 6	7/12	13/20	21/26
Sleeve	43/43	44/47	44/44	0/ 0	0/ 0	48/ 2
Imp. assy.	43/43	0/ 0	44/44	45/45	46/48	49/49
Rotor shaft	43/43	0/ 0	44/46	0/ 0	0/ 0	47/49
Disc+blades	43/43	0/ 0	44/44	0/ 0	0/ 0	45/46
Drive washer	43/43	44/47	44/44	0/ 0	0/ 0	48/ 2
Lockwasher	43/43	44/ 2	0/ 0	0/ 0	0/ 0	0/ 0
End nut	43/43	44/47	44/44	0/ 0	0/ 0	0/ 0
Imp adj shim	43/43	44/47	44/44	0/ 0	0/ 0	48/ 2
Thrust coll	43/43	44/47	44/44	0/ 0	0/ 0	48/49
Seal carrier	43/43	44/47	44/44	0/ 0	0/ 0	48/ 2
Rotor assy	43/43	0/ 0	44/44	0/ 0	0/ 0	45/45
Volute 2E	43/46	47/15	47/50	51/ 4	5/10	16/19
Volute 1E	43/48	49/17	49/49	50/50	51/ 3	18/25
Diffuser	43/47	48/ 2	48/49	50/50	51/ 1	3/ 6
Packing ring	43/43	44/49	44/44	0/ 0	0/ 0	50/ 2
Adj shims	43/43	44/ 2	0/ 0	0/ 0	0/ 0	0/ 0
Seal pl bush	43/43	44/ 1	44/44	0/ 0	0/ 0	2/ 5
Seal pl shim	43/43	44/ 2	0/ 0	0/ 0	0/ 0	0/ 0
Main casing	43/44	0/ 0	45/45	46/46	47/49	50/ 5
CE brg	43/44	45/ 5	0/ 0	0/ 0	0/ 0	0/ 0
T I csg	43/43	0/ 0	0/ 0	0/ 0	0/ 0	44/49
Nozzles	43/50	51/ 9	51/ 2	51/ 2	3/ 6	7/11

Table 7.3: Resource-limited schedule program -
Unlimited resources output data

Elapsed Time Data
******* **** ****

(start of wk/end of wk)

Holiday week nos. 52,

Activity -	Drg	Mat	Proc	Equ D	Equ P	M/C
Manpower -	3	23	2	3	23	23
Impeller	43/49	0/ 0	50/ 6	7/12	13/20	21/26
Sleeve	8/ 8	9/12	10/10	0/ 0	0/ 0	14/19
Imp. assy.	9/ 9	0/ 0	12/12	13/13	14/16	17/17
Rotor shaft	9/ 9	0/ 0	11/13	0/ 0	0/ 0	14/16
Disc+blades	10/10	0/ 0	13/13	0/ 0	0/ 0	14/15
Drive washer	7/ 7	8/11	10/10	0/ 0	0/ 0	14/19
Lockwasher	7/ 7	8/17	0/ 0	0/ 0	0/ 0	0/ 0
End nut	9/ 9	10/13	14/14	0/ 0	0/ 0	0/ 0
Imp adj shim	7/ 7	8/11	9/ 9	0/ 0	0/ 0	13/18
Thrust coll	8/ 8	9/12	11/11	0/ 0	0/ 0	15/16
Seal carrier	6/ 6	7/10	8/ 8	0/ 0	0/ 0	12/17
Rotor assy	10/10	0/ 0	14/14	0/ 0	0/ 0	15/15
Volute 2E	43/46	47/15	47/50	51/ 4	5/10	16/19
Volute 1E	43/48	49/17	49/49	50/50	51/ 3	18/25
Diffuser	49/ 2	3/ 8	3/ 4	5/ 5	6/ 7	9/12
Packing ring	6/ 6	7/12	7/ 7	0/ 0	0/ 0	13/16
Adj shims	6/ 6	7/16	0/ 0	0/ 0	0/ 0	0/ 0
Seal pl bush	5/ 5	6/14	6/ 6	0/ 0	0/ 0	15/18
Seal pl shim	5/ 5	6/15	0/ 0	0/ 0	0/ 0	0/ 0
Main casing	3/ 4	0/ 0	5/ 5	6/ 6	7/ 9	10/16
CE brg	3/ 4	5/16	0/ 0	0/ 0	0/ 0	0/ 0
T I csg	8/ 8	0/ 0	0/ 0	0/ 0	0/ 0	9/14
Nozzles	49/ 5	6/15	7/ 9	6/ 8	10/13	14/18

Table 7.4: Resource-limited schedule program - Actual resources output data

Elapsed Time Data
******* **** ****

(start of wk/end of wk)

Holiday week nos. 52,

Activity -	Drg	Mat	Proc	Equ D	Equ P	M/C
Manpower -	3	23	1	3	23	23
Impeller	43/49	0/ 0	1/ 8	9/14	15/22	23/28
Sleeve	8/ 8	9/12	20/20	0/ 0	0/ 0	24/29
Imp. assy.	9/ 9	0/ 0	25/25	26/26	27/29	30/30
Rotor shaft	9/ 9	0/ 0	22/24	0/ 0	0/ 0	25/27
Disc+blades	10/10	0/ 0	26/26	0/ 0	0/ 0	27/28
Drive washer	7/ 7	8/11	19/19	0/ 0	0/ 0	23/28
Lockwasher	7/ 7	8/17	0/ 0	0/ 0	0/ 0	0/ 0
End nut	9/ 9	10/13	28/28	0/ 0	0/ 0	0/ 0
Imp adj shim	7/ 7	8/11	18/18	0/ 0	0/ 0	22/27
Thrust coll	8/ 8	9/12	21/21	0/ 0	0/ 0	25/26
Seal carrier	6/ 6	7/10	17/17	0/ 0	0/ 0	21/26
Rotor assy	10/10	0/ 0	27/27	0/ 0	0/ 0	28/28
Volute 2E	43/46	47/15	47/50	51/ 4	5/10	16/19
Volute 1E	43/48	49/17	51/51	1/ 1	2/ 5	20/27
Diffuser	49/ 2	3/ 8	15/16	17/17	18/19	21/24
Packing ring	6/ 6	7/12	14/14	0/ 0	0/ 0	20/23
Adj shims	6/ 6	7/16	0/ 0	0/ 0	0/ 0	0/ 0
Seal pl bush	5/ 5	6/14	9/ 9	0/ 0	0/ 0	18/21
Seal pl shim	5/ 5	6/15	0/ 0	0/ 0	0/ 0	0/ 0
Main casing	3/ 4	0/ 0	13/13	14/14	15/17	18/24
CE brg	3/ 4	5/16	0/ 0	0/ 0	0/ 0	0/ 0
T I csg	8/ 8	0/ 0	0/ 0	0/ 0	0/ 0	9/14
Nozzles	49/ 5	6/15	10/12	6/ 8	13/16	17/21

Table 7.5: Resource-limited schedule program - What if? output data

APPENDIX 7.1

NA355 records from "New Products Meeting" minutes

Meeting No.155 9th February 1982

6.0 NA355/NA255

Engineering department currently re-specifying these units. Wheel sizes to be decided as soon as possible; final design required in order to define "Rigid" activity. Drawing programme to be fed into the project programme bar-chart when available

Meeting No.157 1st April 1982

1.0 NA355

The concept and preliminary design of the NA355 was presented. Primary requirements are for high performance, attractiveness in general market, attractiveness to GEC Diesels, and compatible with BBC and involving minimal investment. Solution of these requirements lies in use of SA105 turbine end - exhaust casing, centre casing, inlet casing, rotor disc and blades.

New components required are profiled nozzle, new compressor wheel, diffuser, volute, shaft and bearings (NA350).

The new shaft and bearing arrangement will be introduced at an early date in the SA105 mk5 with the dual objectives a) improving the current SA105 bearing problems, b) obtaining advance evidence of the rotating assembly behaviour of the NA355. Availability of these units is expected to be July (SA105 mk5) and October for the NA355 (first test).

Manufacturing requested that consideration be given to a separate insert and volute.

Sales stated that a 90° turbine inlet bend is very likely to be required in the general market.

Meeting No.158 6th May 1982

1.0 NA355

Engineering design effort has concentrated on the rotating assembly and drawings are almost complete. The rotating assembly will be tested in the form of SA105 mk5 as planned and first test is intended for July. Engineering department will call up a unit following issue of new drawings.

Several schemes for separate volute and insert are being studied. Engineering and estimating to arrange the optimum design from design, manufacturing and customer view points.

Design of the compressor wheel has not yet started and is behind schedule due to priority on NA455. Preliminary design to be commenced in 2 weeks and overall dimensions of compressor wheel to be given to buying department as soon as possible.

Production expressed concern about available effort for pursuit of NA355 programme according to schedule. A meeting is to be arranged to discuss this critical situation.

Meeting No.159 17th June 1982

1.0 NA355

As planned, the new rotating assembly will be first built and tested in the form of the SA105 mk5. Drawings for this turbocharger have been issued and a unit for test called up. Planning going according to schedule and it is aimed to have a unit available for tests before the annual holidays July/August.

Two schemes for separate volute and insert have been submitted for quotation for manufacture. Quotations expected this week. Decision then needed on type of volute to be adopted.

Compressor wheel design started four weeks late due to priority work on the NA455. Compressor aerodynamic design well advanced and manufacture now is expected to be available for mid-September. Availability of Rambaudi five-axis machine will not coincide and so interim supplier for the compressor wheel must be identified.

Design of 90° turbine inlet bend not started. This must take second place to scheming major lead items.

Meeting No.160 13th July 1982

2.0 NA355

The new rotating assembly will be built and tested in the form of SA105 mk5 - unit available in August.

Two schemes for separate volute and insert have been costed and assessed. Cost of two-piece and three-piece arrangements are about 1000 and 1100 respectively compared to the cost of a one-piece design of 700. Also two-piece and three-piece do not give sufficient advantage in manufacturing. Planning of the two-piece offers no advantage in stripping and servicing. Hence the component will remain as a one-piece component.

Design of aerodynamic components generally progressing well but about four weeks behind schedule. Compressor wheel design to be completed by end of August. Diffuser design also to be completed by end of August. Work to commence on profiled nozzle design as soon as effort is available.

Meeting No.161 29th September 1982

3.0 NA355

Programme is generally slipping by about six weeks due to pressure of other projects. Compressor design is complete; NC data will now be produced for trial manufacture by the sub-contractor.

Drawings of two capacity types will be completed.

Profiled nozzle design is now stopped due to lack of engineering effort. This will be started in four weeks time. Drawing office design progressing on volute casing and other components.

The new rotating assembly will be tested in the form of SA105 mk5 during October.

Manufacturing asked what were the commercial forces driving the NA355. Should effort be concentrated on the NA355 rather than the 305 or 255, for example. It was agreed that more sales information will be needed to determine order of priority. However, it was noted that the last meeting recorded that the NA255 was a higher priority than the NA305.

Meeting No.162 4th November 1982

1.0 Small "5 Series"

NA355 - in advanced design phase. Rotating assembly trial has led to revised design. Programme has slipped due to pressure of other projects. A revised programme is necessary.

Meeting No.163 2nd December 1982

1.0 Small "5 Series"

The meeting centred on the NA355 which is currently the subject of a concentrated effort in engineering department to complete the design by January. Two engines have been identified by Sales as being the best target for first tests of this turbocharger. These are the Ruston 12RK270 which requires a higher performance turbocharger particularly in its higher ratings, and the Mirrlees 6K Major engine.

Critical items coming between now and the projected rig test in April are the improved volute and the turbine nozzle. Progress on the improvements hinges on the success of the Rambaudi and also on software proving trials yet to take place at the sub-contractors for machining the impeller. This is urgently required.

The supplier for the volute casting has yet to be selected. The design of the turbine nozzle must be completed before Christmas, and the supply of vane castings arranged by that date.

For the Mirrlees application there are likely to be special requirements for the volute (single delivery) and for the turbine inlet casing (90° water cooled). This will not affect the rig test.

Meeting No.164 3rd February 1983

1.0 Small "5 Series"

Production engineering questioned whether the whole schedule could be achieved in view of the growing bottle-neck in production engineering department which was now threatening the NA355 schedule.

The result of a long discussion of new product schedules was as follows:

1 The first test date of a new turbocharger is the most important milestone in the life of the product. At this point teething problems if any can be remedied, test data are made available to sales department and general confidence can be built up in the product.

Hence the objective should be to bring the first unit to test by using short-cut tactics but in no way reducing the commercial viability of the early units.

2 As a result of 1 long planning and tooling schedules may have to be sacrificed in the interests of achieving the first test.

3 Using the above guides we should now aim to achieve the following date(s):
June 1983 - first test of NA355

Meeting No.165 17th March 1983

1.0 Small "5 Series"

The NA355 was also briefly discussed. It was agreed that the first turbocharger will be available for test in July; thereafter production standard turbochargers could be made available within 16 weeks of a successful test. Requirements for water-cooled inlet bends are being investigated by sales department.

Meeting held 19th May 1983

2.0 NA355

Orders expected from Wichmann and Colt as well as probable requirements from Ruston and Alco have accelerated the requirements for completion of design, manufacture, and rig test of the first unit. Latter is planned for end of

July and every effort must be made to keep to this date. Lead items are impeller, volute, bearings and nozzle.

To achieve October deliveries for Wichmann and Colt a new single-entry turbine inlet casing and a new single-delivery volute are required.

Further customers require a water-cooled 90° turbine inlet casing. The sales policy was to persuade customers to adopt air-cooled casings but the need to meet surface temperature requirements created a demand for water-cooling.

Meeting held 7th July 1983

1.0 Small "5 Series"

NA355 - Manufacture of first two units proceeding on schedule for test in Week 19 (mid-August). Essential this date for test is maintained.

Single-entry turbine inlet and single-delivery volute casing for Wichmann project will be on time.

Meeting held 29th September 1983

1.4 NA355 First unit tested successfully with very good compressor performance and very satisfactory mechanical behaviour.

Wichmann units threatened by slipping of the schedule. Nozzles are greatest supply problem.

Ruston now strongly interested in delivery of NA355 for 12RK270 test in October. Momentum therefore to be maintained on these units.

Meeting held 24th November 1983

1.3 NA355

Further tests have taken place on compressor and turbine all giving good results.

Nozzle ring dimensions outside tolerance are giving cause for concern. New supplier urgently required.

Test on Wichmann 7AXAG engine are imminent

Tests on Ruston 12RK270 engine scheduled for December

APPENDIX 7.2

Limited Resource Scheduling example

1 Resources

Activity	max. availability
1	1
2	1
3	1

2 Input Data

act.	1	2	3
item	times/IPA		
1	1/0	2/1	1/2
2	1/0	0/0	1/1
3	0/0	1/0	1/2
4	2/0	2/1	1/2

3 Incrementing time calculation

Total times (unlimited resources from particular time)

time=0

act.	1	2	3	descending
item				order
1	1	3	4	2
2	1	0	2	3=
3	0	1	2	3=
4	2	4	5	1

Total times (unlimited resources from particular time)

time=1

act.	1	2	3	descending
item				order
1	2	4	5	1=
2	2	0	3	3
3	0	1	2	4
4	2	4	5	1=

Total times (unlimited resources from particular time)
time=2

act. item	1	2	3	descending order
1	3	5	6	1
2	3	0	4	3
3	0	1	2	4
4	2	4	5	2

Total times (unlimited resources from particular time)
time=3

act. item	1	2	3	descending order
1	3	5	6	1
2	4	0	5	2=
3	0	1	2	4
4	2	4	5	2=

Total times (unlimited resources from particular time)
time=4

act. item	1	2	3	descending order
1	3	6	7	1
2	4	0	5	2=
3	0	1	2	4
4	2	4	5	2=

Total times (unlimited resources from particular time)
time=5

act. item	1	2	3	descending order
1	3	6	7	1
2	4	0	5	3
3	0	1	2	4
4	2	4	6	2

Total times (unlimited resources from particular time)

time=6

act. item	1	2	3	descending order
1	3	6	7	1
2	4	0	5	3
3	0	1	2	4
4	2	4	6	2

Total times (unlimited resources from particular time)

time=7

act. item	1	2	3	descending order
1	3	6	7	1
2	4	0	5	3
3	0	1	2	4
4	2	4	6	2

Nb 1 circled times indicate those activities that are available to start.

2 underlined times are those activities that have been allocated resources at that time increment.

3 ticked times are those activities that have been completed at that time increment.

CHAPTER 8

DESIGN AND DETAIL

8.1 **Impeller**

8.1.1 Impeller geometry

The impeller design method has been discussed in some detail in section 3.2 Figure 8.1 presents this in a diagrammatic form (8.1,8.2). The work described in section 5.1 and 6.2 provided some of the more important parameters required for the preliminary design. The main details are:

i	Impeller diameter	425 mm
ii	Vane backsweep at impeller OD	30°
iii	Maximum capacity at R_C = 3.0:1	22
iv	Maximum impeller tip velocity	542 m/s

It should be noted that the units of capacity are not truly non-dimensional as discussed in section 3.1. Combination of 425 mm and 542 m/s results in a maximum rotational speed of 24356 rev/min. The choice of squareness ratio (eye tip diameter/impeller diameter) was made so that the specific speed would be around the peak of the curve shown in figure 6.3. This would allow for a) optimum efficiency in a popular flow regime for the turbocharger and b) an increase in impeller size if a larger flow machine was required. By using optimising techniques (eg. 8.3,8.4) and a number of starting values that had resulted in satisfactory impellers the following geometry details were obtained.

i	squareness ratio	.65
ii	eye tip diameter	276.25 mm
iii	hub ratio/hub diameter	.227/96.6 mm

iv	tip width ratio/tip width	1.0/27.3 mm
v	vane axial length ratio/length	.3/127.5 mm
vi	length ratio/length	.36/153 mm
vii	eye tip inlet angle	62.5°

where :-

squareness ratio	= eye tip diameter/impeller diameter
hub ratio	= eye hub diameter/eye tip diameter
tip width ratio	= impeller exit area/inducer throat area
vane axial length ratio	= vane axial length/impeller diameter
length ratio	= impeller length/impeller diameter

8.1.2 Compressor range prediction

The prediction method described in section 3.1 was used to provide the inducer range. This is shown in figure 8.2, together with two outline characteristics. Stall, in this context is related to minimum losses, so there is therefore no automatic association with stage surge. However, many compressors do exhibit problems around this region of flow, especially when the rest of the stage is operating in areas of low stability (8.5). It is necessary to have a large range of diffuser throat sizes available so that every eventuality that may occur in service will be covered. For example, for a lower rated engine requiring, say, 2.0:1 pressure ratio the largest diffuser size that would be satisfactory is much larger than if an engine required say 4.5:1 pressure ratio. This may be seen by the relative slopes of the compressor surge line and the choke limit shown in figure 8.2. The smaller of the two outline characteristics shown in figure 8.2 may be seen to be smaller than desired as slightly larger diffusers may operate at undesirable areas around the predicted stall values. The range of diffuser areas to the inducer throat area shown is from 0.33 to 0.57.

The range of capacity required to be covered by the turbocharger was roughly from 6.5 at a pressure ratio of 3.0:1 up to the maximum specified above, see figure 5.1. This minimum value was chosen to match the NA350, so that the new turbocharger could be used to replace that unit. One of the most important decisions to be made was the number of capacity types required to cover the turbocharger flow range. Inevitably this is a compromise between a number of conflicting requirements. In order to ensure the highest efficiency the number of capacity types should be large. However, each capacity type will carry an overhead in terms of inventory and tooling associated with other components. These include the impeller, diffuser, and compressor casing.

Initially a smaller capacity version of the basic impeller was chosen so that the predicted stall value matched the required minimum capacity. This was achieved with a squareness ratio of 0.541. Using nominal values the ratio of specific speed compared with that for the basic impeller was 84% (.586 to .49). It was estimated that this would incur a reduction of impeller efficiency of approximately 5 percentage points. This was made up of 3% directly from the specific speed effect (Fig 6.3) and an additional 2% because the smaller impeller would be produced from the larger basic impeller and would not have optimised geometry. This relatively large reduction in efficiency was considered to require two intermediate sizes so that the effect would be reduced to an acceptable value. This implied that the larger basic impeller would become the fourth impeller, with the smaller one being the first. For convenience the intermediate sizes were chosen so that the inducer area range increased by a constant value from the capacity type 1 to 4.

The prediction program was used to produce the flow ranges of these four capacity types shown in figure 8.3. The techniques described in section 3.2 were used to provide the necessary vane geometry consistent with the aerodynamic and mechanical requirements. The computer stored geometry was manipulated to produce cutter-path definition files for use by the 5-axis CNC milling operation.

8.1.3 Impeller disc design

The vane geometry from the aerodynamic design provided the starting point for the impeller mechanical design. The information available allowed some preliminary scheming to be carried out. The impeller hub was drawn to conform with the profiled rear surface as described in section 6.1. The finite element analysis (FEA) code used for this was the well known ANSYS (ANSYS is a registered trade mark of Swanson Analysis Systems Inc) available world-wide at many bureau facilities. The method of analysis used an axisymmetric model of the vane and hub. The element chosen for both hub and vane was 'STIF 42' using the options for axisymmetric and plane stress with thickness respectively. Circumferential vane thicknesses were specified at each node on a per radian basis. Therefore, although the vane was modelled as a flat plate, the mass and stiffness distribution were approximated. The loading was that due to rotation, additional loads due to thermal gradients were ignored. Only one constraint was specified and this was in the axial direction. This was at the front, and at the bore of the hub. The final model, with the elements shown contracted for clarity, is shown in figure 8.4. The results of the analysis, for a rotational speed of 23000 rev/min are given in figure 8.5, showing the displacement, and figure 8.6, the tangential stress contours. The object of the iteration was to achieve a satisfactorily stressed impeller with as short as possible axial length. The maximum tangential stress was 241.7 MPa in the bore at about 1/3 of the axial length from the rear. Using fatigue data for the material and a 'cyclic duty' specified as a maximum of 5×10^6 cycles allowed a maximum speed of 23215 rev/min to be achieved. For other 'non-cyclic' applications, specified as a maximum of 2×10^5 cycles a speed of 25800 rev/min was obtained.

8.2 Compressor-end detail

With a completely new turbocharger, requiring all new casings, the designer would have had a great deal of freedom to accommodate the new compressor. However, as one of the objectives was to make use of as many parts of the existing SA105 turbocharger as possible there were far more constraints. The compressor-end detail of the SA105 is shown in figure 8.7 together with the terms used for the most relevant components. The way in which the detail fits into the whole assembly may be seen from figure 5.2.

8.2.1 Centre casing

The centre-casing may be seen to be a complicated casting containing the rotating assembly, bearing assemblies, and various sealing components. It was considered that the centre-casing for the new turbocharger should be assessed by using the following list, in descending order of preference:

a) The NA355 should use the existing SA105 centre-casing

b) The SA105 casting should be modified so that when machined the same casing would be suitable for both turbochargers

c) The finish machined SA105 centre-casing should be further machined to become the NA355 centre-casing

d) The NA355 centre-casing should be machined from the SA105 casting

e) The NA355 centre-casing should be machined from a modified SA105 casting, still allowing the SA105 casing to be produced

f) The NA355 centre-casing casting to be different from the SA105 casting

8.2.2 Initial scheming

Initially the SA105 impeller and diffuser were removed from the compressor end detail scheme and replaced by the NA355 components. Figure 8.8 shows this arrangement, and the rear profile of the impeller may be seen to have provided an immediate problem. The centre-casing outer diameter flange provided the location for the compressor delivery casing, partially shown dotted. The exit flow from the diffuser preferably had to line up with the original surface of the delivery casing. As may be seen the diffuser flow channel is considerably displaced axially, so that the above requirement was far from being achieved. The sealing items would have to change, as the diameters were different. However, if the impeller could be moved closer to the centre-casing the amount of additional material in these items could be reduced. Clearly it was possible to extend the compressor delivery casing flange so that the flow channels did become aligned. However, a more elegant solution was required.

8.2.3 Diffuser design

The preferred method of manufacture of the diffuser was to machine the vanes out of a solid blank of material leaving a backplate. The NA355 design of diffuser incorporated a 7° divergence of the outer wall. This sidewall divergence allowed a greater diffusion ratio to be achieved within a particular radius ratio, where radius ratio is the diffuser outer radius divided by the inner radius. All previous turbochargers had incorporated the diffuser arrangement as shown in figure 8.7. It may appear obvious to consider reversing the diffuser, thereby making use of the sidewall divergence to bring the flow channel back towards the centre-casing. Initially it was not at all obvious. However, this is an example of the designer having to examine the constraints upon a design and being able to distinguish between those that are real and those that are imaginary. The scheme showing the diffuser in the reversed position is given in figure 8.9.

8.2.4 Final scheming

The arrangement, described above, did not achieve all of the desired axial alignment. The compressor-end bearing was to be similar to the NA350 bearing, figure 5.4, and the axial distance from the casing to the minor thrust would be slightly less than the combined thickness of the minor thrust plate and bearing housing of the SA105. This also helped to move the impeller nearer to the centre-casing, in this case by a small amount. In order to align the flow channels of the diffuser and the compressor delivery casing the compressor-end bearing was recessed into the centre-casing as shown in figure 8.10. This clearly shows that the axial alignment had been achieved without compromising the axial space available for oil sealing.

In terms of assessing the arrangement against the preferential list above it was clear that the most desirable situation had not been achieved. However, it appeared feasible for the NA355 centre-casing to be machined from the finished SA105 casing. This appeared to be an acceptable compromise at this stage.

8.3 Rotating assembly

8.3.1 Rotating assembly philosophy

The rotating assembly had been identified as an item requiring some re-design, as those of both the SA105 and NA350 had design weaknesses. One of the most undesirable features of both designs of rotating assembly was that the impellers were clamped onto the rotor by the end-nuts. The FEA showed that the impeller reduced in length by some 0.14mm and radially distorted, in the bore, by 0.1mm at a rotational speed of 23000 rev/min. The shaft dimensions preclude this component acting as a strain member and so it was expected that the impeller would move unpredictably at operational rotating speeds. This is extremely undesirable, as the essential requirement of all high-speed rotating machinery design is to be able to predict the components behaviour at the operating conditions. In addition the highly stressed bore of these impellers provided a mating surface with the shaft. This meant that the most likely areas of the impeller to receive physical damage (eg. scoring) during assembly was the one area that required to remain free from damage. Another area of concern was the number of components comprising the rotating assembly. Every component that became clamped by the end-nut would affect the attitude that the shaft adopted, at least when stationary. The components could only be made to reasonable tolerances. For example, parallel and squareness tolerances of 0.008 mm were generally specified on rotating components. Wherever possible conflicting locations were avoided. For example, the design of a shim requires a parallel tolerance and a dimension in the bore that will ensure there will be a clearance for all tolerances of the relevant components. If all of the compressor-end rotor components shown in figure 8.7 have the maximum allowable out of squareness then the shaft may be seen to bend by approximately 0.06 mm at the estimated impeller centre of gravity position. Fortunately the statistical chance of this happening is not high. However, the number of interfaces affected by the end-nut clamping should be as low as possible.

8.3.2 Rotating assembly objectives

The objectives of the assembly at the compressor end of the rotor were listed.

a) To provide a reliable, and predictable location for the impeller

b) To ensure that any location faces were not in the most highly stressed regions of the impeller

c) To ensure that the most highly stressed parts of the impeller were protected from damage during assembly

d) To provide a means of driving the impeller

e) To enable the axial position of the impeller to be adjusted to accommodate the tolerances in surrounding items in order to maintain a minimum shroud clearance

f) To form the rotor part of the oil-sealing arrangement

g) To provide the minor thrust face

h) To allow the components to be assembled and disassembled with ease

8.3.3 Impeller

In order to protect the impeller bore from damage, other designs had incorporated a shrunk-in steel sleeve. The amount of interference would ensure that the two components would not become separated even at higher than operational speed. The advantage of using a sleeve was that many of the objectives listed above could be satisfied. For example a sleeve would protect the impeller bore, the point of location could be chosen to be in a suitable region of acceptable stress, and it could be arranged to form an acceptable means of driving the impeller. In addition the sleeve could be used to carry additional items, such as the minor thrust face and the rotor oil-sealing items. The sleeve options considered included a plain outside diameter or flanged at either end of the impeller. The plain outside diameter appeared to offer little merit. A number of compressor-end rotor schemes were produced in order to assess the various possibilities.

The advantages and disadvantages of the two flange options were considered, with the flange at the outboard end of the impeller showing a clear advantage. The number of interfaces in the previous designs were examined with the objective of reducing them to as few as possible. It was considered that the rotor oil-sealing function, the impeller drive arrangement, the minor thrust block, did not require to be part of the interface between the end-nut and the rotor shaft. All of these functions could be absorbed into other components. However, the axial adjustment feature proved to require another item, as it was considered undesirable to adjust the axial length of the major impeller assembly. Therefore the adjustment washer and the minor thrust block were combined. The resulting assembly is shown in figure 8.11, and a description of some of the more important features follows.

8.3.4 End-nut

The end-nut was shaped so that it could only be assembled in one position. This reflected the way in which the item could be made, in that the thread and only one endface could be machined at one setting. Previous designs had used a parallel nut with problems of achieving squareness, and verifying the squareness, of both faces to the thread. The face of the end-nut abutted directly with the end of the sleeve.

8.3.5 Impeller drive assembly

The impeller drive arrangement formed no part of the interface, there being a small axial clearance between the nut, lock washer, drive washer, and impeller sleeve flange. The drive arrangement is shown pictorially in figure 8.12.

The sleeve flange provided a useful balancing area at the front of the impeller. It was normal to carry out dynamic balancing of impellers, with the two planes at the front and rear of the disc. The front of the impeller often caused problems as the only area for balancing was at a low radius and therefore relatively large amounts of material distribution were sometimes needed. Often, in other designs, it had been necessary to provide for metal to be removed and heavy metal plugs to be added. However, the steel flange of the sleeve provided more than an adequate amount of material for balancing purposes.

8.3.6 Oil-sealing arrangement

The rotor oil-sealing block was shrunk onto the sleeve at the rear of the impeller, this was effected by heating the block until a clearance would be achieved. This component was then slid onto the sleeve until the rear of the impeller was touched. When the component had cooled, and had shrunk onto the sleeve, an axial gap between the two components was formed. The oil-sealing block overhung the end of the sleeve. This provided a means of protecting the most important end-face of the sleeve. The impeller assembly was a clearance fit on the shaft, thereby interfering with the rear end-face location of the impeller as little as possible.

8.3.7 Major thrust collar

As the major thrust bearing was in between the two journal bearings, a means of fitting a thrust face to the shaft was required. The NA350 turbocharger used split bearing with a separate 'bobbin' on the shaft. This was undesirable, as splitting a thrust face can never be considered as reliable, and adding radial locations with their resulting tolerances cannot be considered good practice. After considering a number of complicated arrangements a simple press fitted thrust collar was used, and calculation showed that the fit alone could accommodate the bearing torque. The diameter for the location was slightly larger than the journal, thereby allowing the thrust collar to be fitted onto the shaft and aligned before jacking the component into position.

8.4 Turbine nozzle

The Prescribed Velocity Distribution (PVD) technique described in section 3.4 was used for the design of the nozzle blade profile. An example of the input and output from program is given in figure 8.13. The input required includes the inlet and outlet gas angle, and flow function (Mach number ratio). The pitch between blades is required to enable the profile to be scaled to the relevant size, and is dependent upon the number of nozzle blades. The number of entries of the turbine inlet casing may be seen to affect the number of nozzle blades if the same nozzle is to be suitable for different casings. To ensure that every sector of the inlet casing has an integral number of nozzle passages the total number of nozzle blades have to be divisible by the number of inlet casing entries. As 1, 2, 3, and 4 entry casings are manufactured the number of nozzle blades should be 12, 24, or 36. Generally the use of 12 would mean that the axial space to accommodate the nozzle would be too long, since the chord would have to increase to give an acceptable pitch/chord ratio. Also 36 would be more blades than were desired, leaving 24 as a reasonable compromise. The velocity distribution is specified at some of the points around the profile on both the suction and pressure surface. The program then calculates the blade profile that is needed to achieve the velocity distribution. In figure 8.13 the blade profile together with the position of the next blade may be seen. Also the velocity around the profile is shown starting from the trailing edge on the left, along the pressure surface and through the stagnation point in the centre of the figure and along the suction surface to the trailing edge on the right. The program also allows the off-design performance to be judged, by the incidence tolerance of the profile. Shown in the figure are the velocity distribution for an incidence of 35° and -35°. The method chosen to manufacture the nozzle had been developed in a slightly larger turbocharger, and it was necessary to adjust the design to accommodate the different profile and sizes. The nozzle was formed from two spinnings that were then laser-cut to allow the nozzle profile to pass through the rings. The nozzle blades were produced from a thin tube subsequently pressed to the correct shape. Inside the nozzle blade was fitted a 'wiggle-strip' to provide support. The whole assembly was vacuum brazed to provide a nozzle of low thermal inertia. As may be appreciated the design required a great deal of co-ordinated effort to achieve a satisfactory nozzle. The nozzle assembly, and a cross-section of the blade showing the 'wiggle-strip' is given in figure 8.14.

8.5. General assembly

As indicated in section 7.3 quite a large quantity of components were required to be designed for the NA355. The more important items have been discussed. The general assembly arrangement is shown in figure 8.15.

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FIGURES

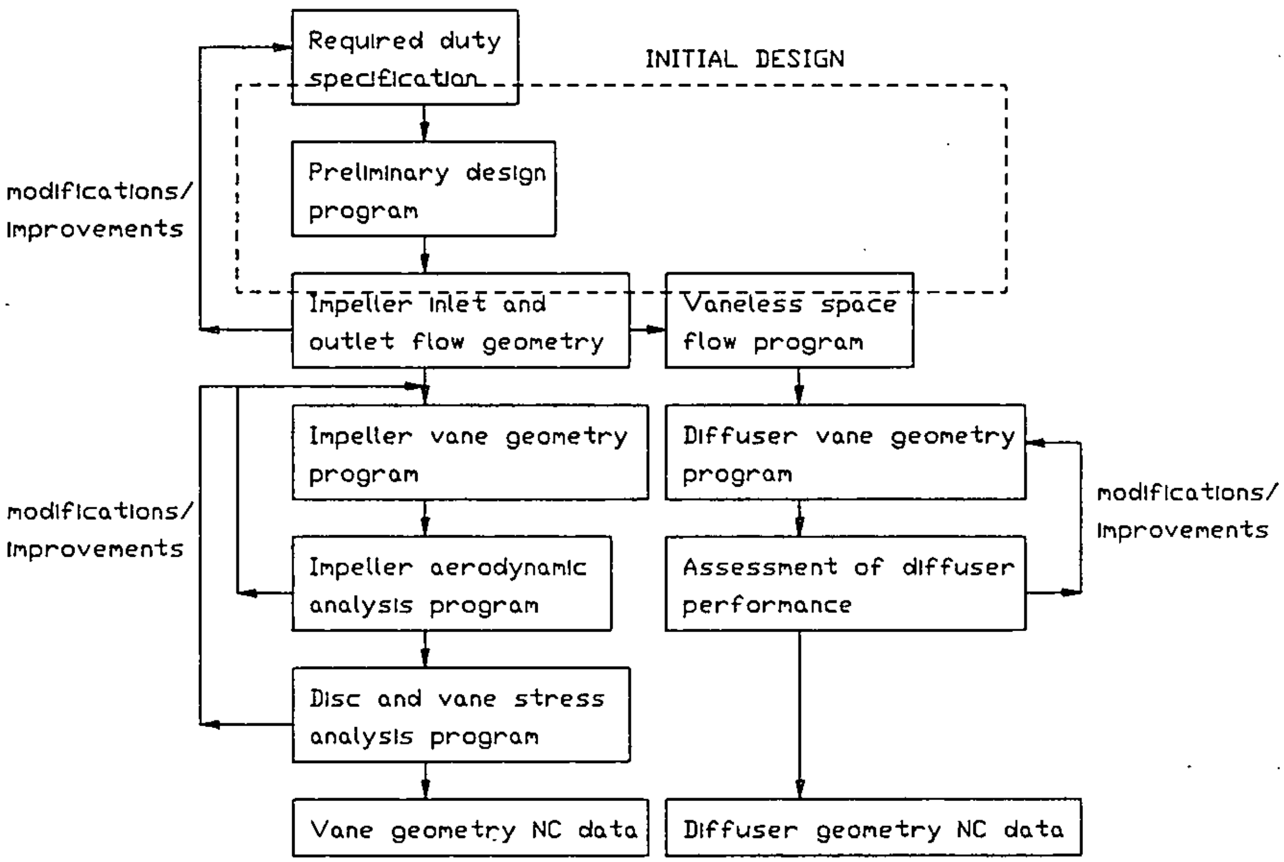


Figure 8.1: Centrifugal compressor design procedure

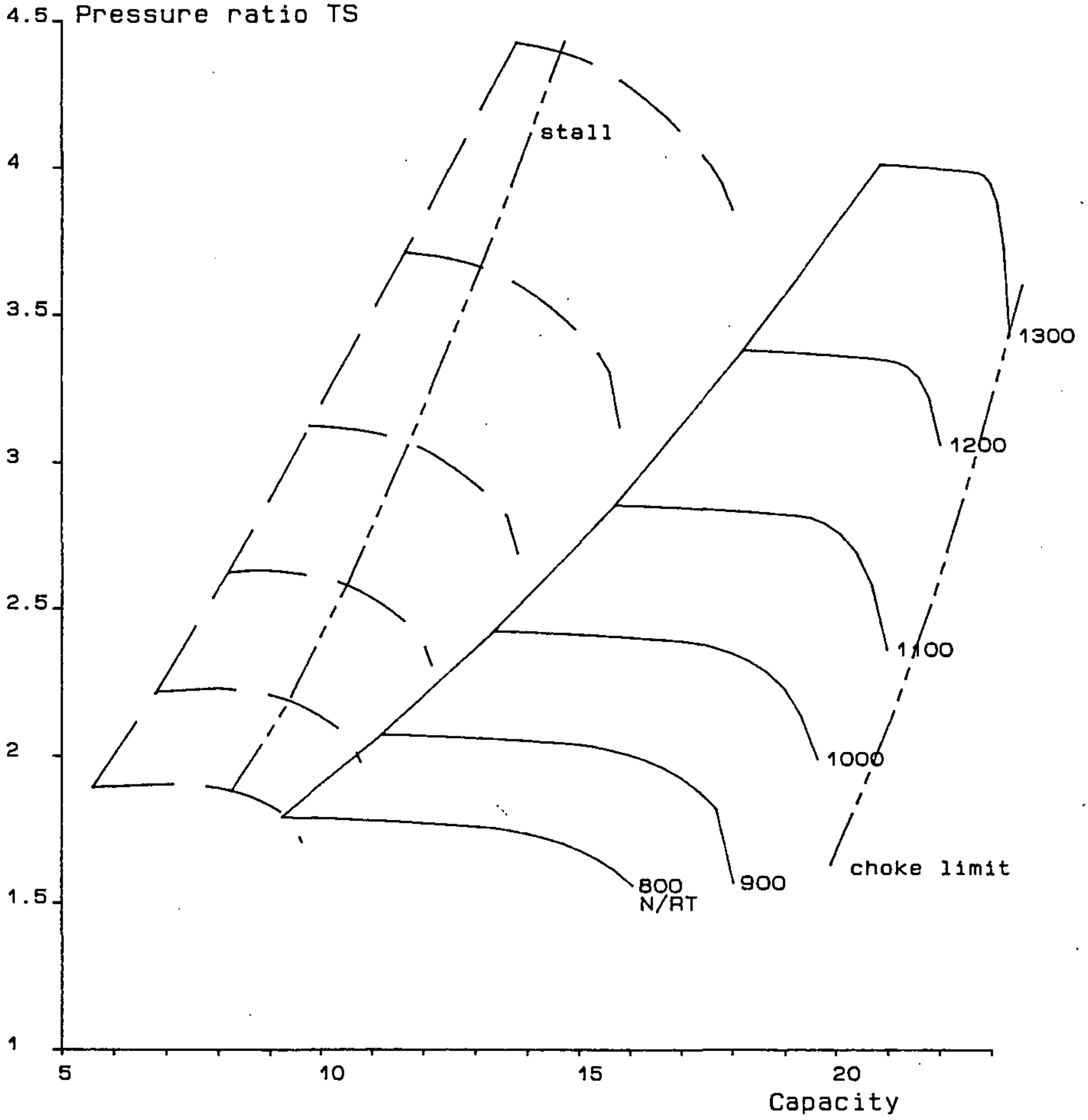


Figure 8.2: Capacity 4EI range

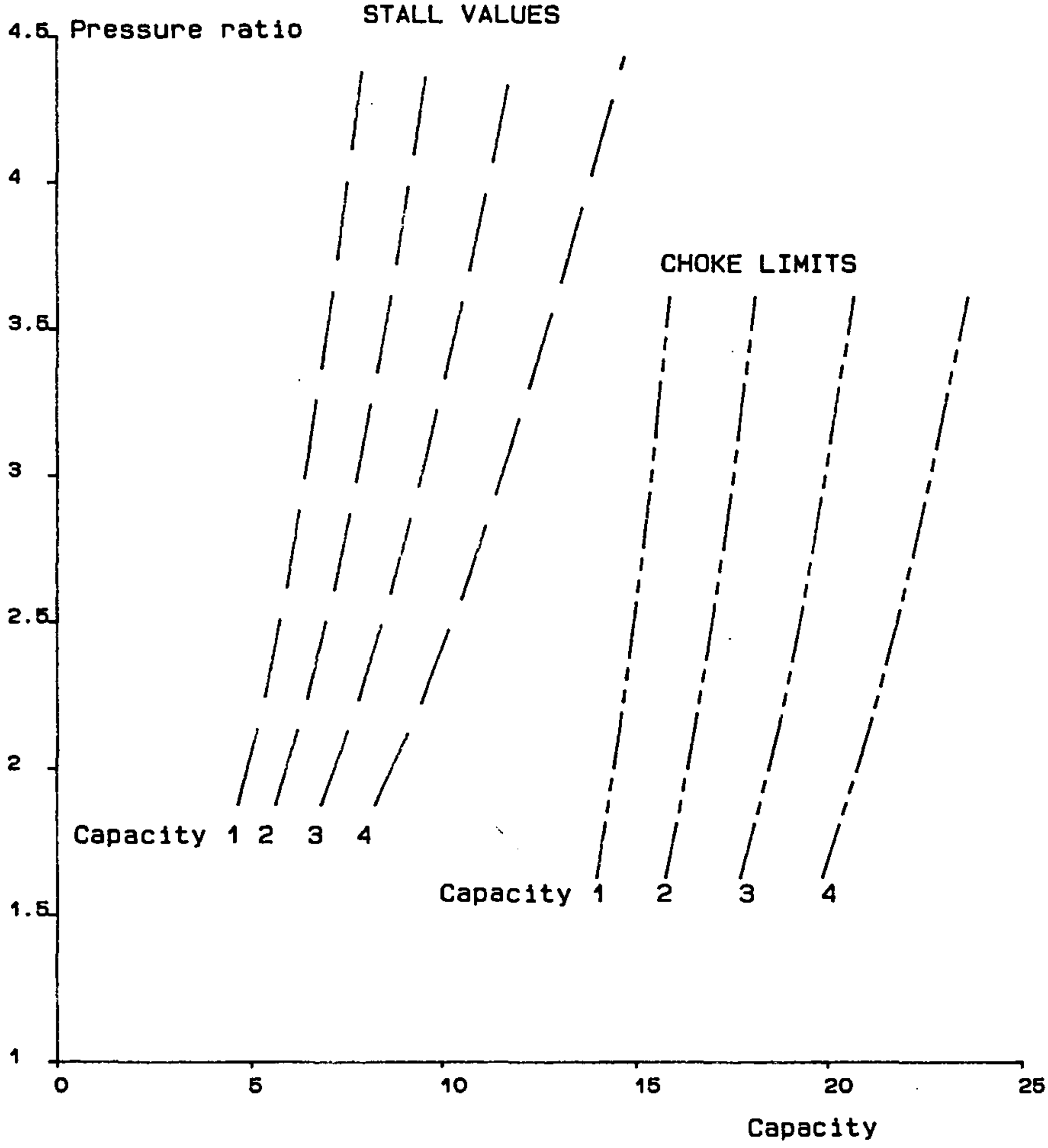


Figure 8.3: NA355 capacity ranges

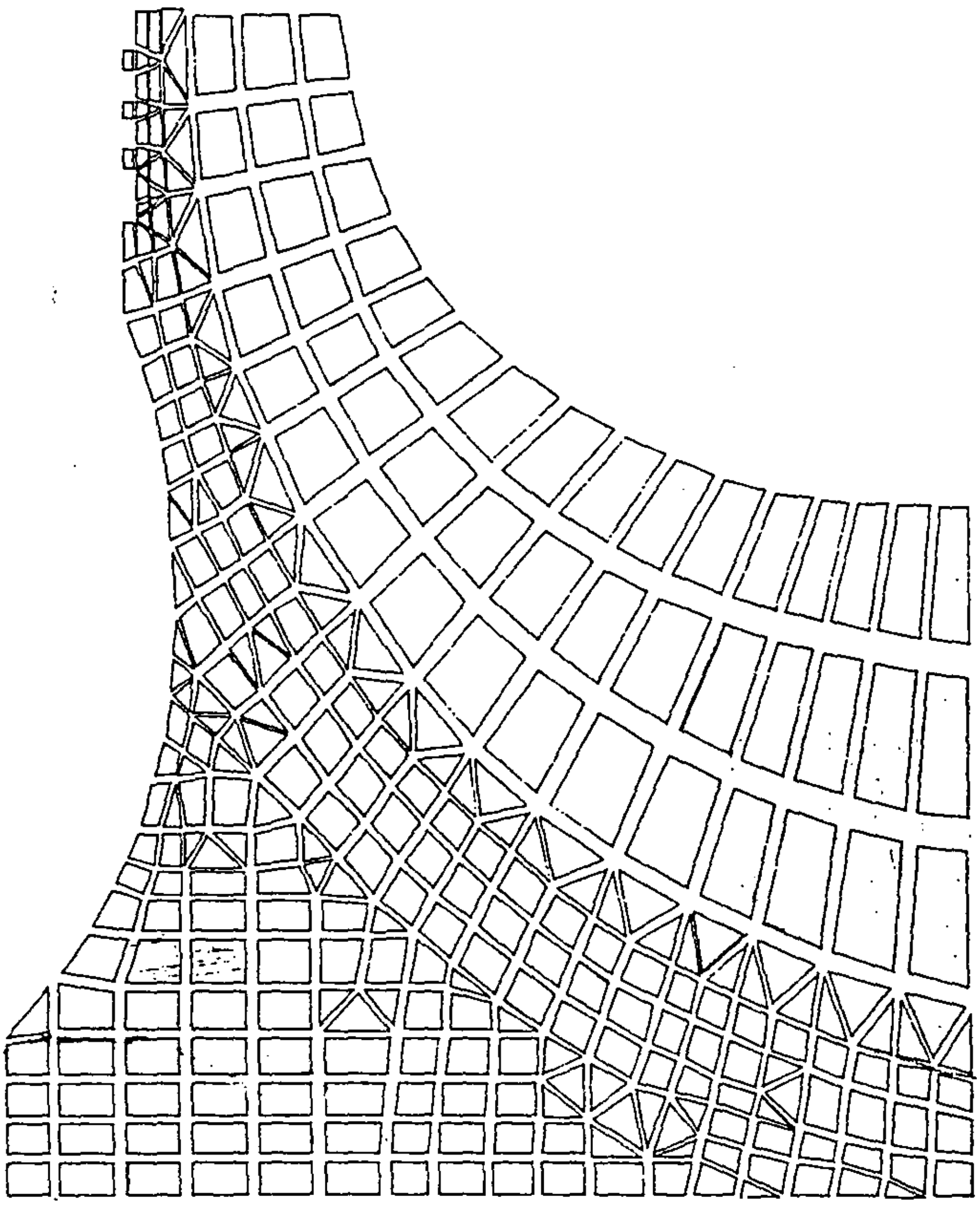


Figure 8.4: Compressor impeller finite element model

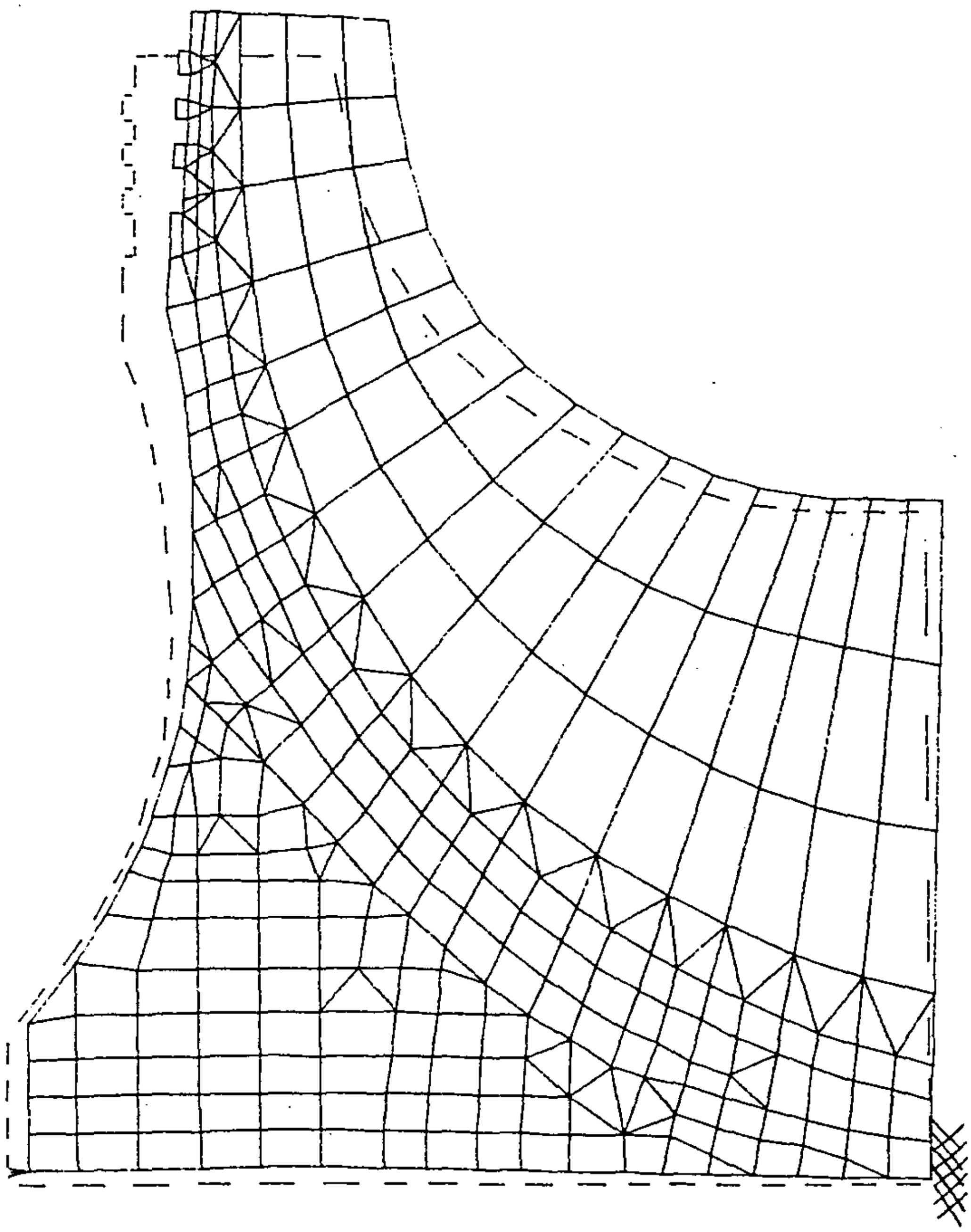


Figure 8.5: Compressor impeller displacement at 23000 rev/min

Contour	Value(MPa)
1	0
2	12.5
3	25
4	37.5
5	50
6	62.5
7	75
8	87.5
9	100
10	112.5
11	125
12	137.5
13	150
14	162.5
15	175
16	187.5
17	200
18	212.5
19	225
20	237.5

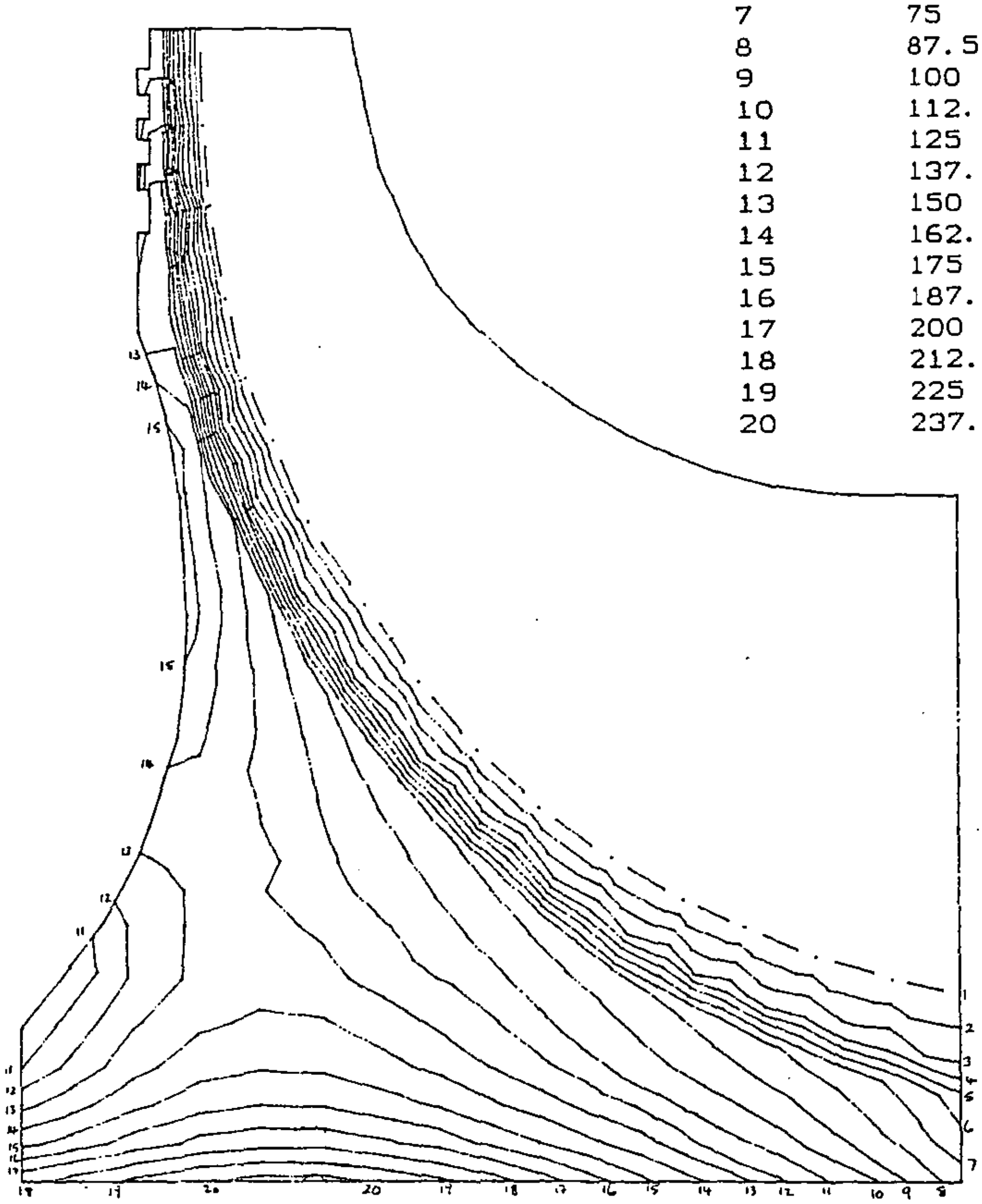


Figure 8.6: Compressor impeller tangential stress at 23000 rev/min

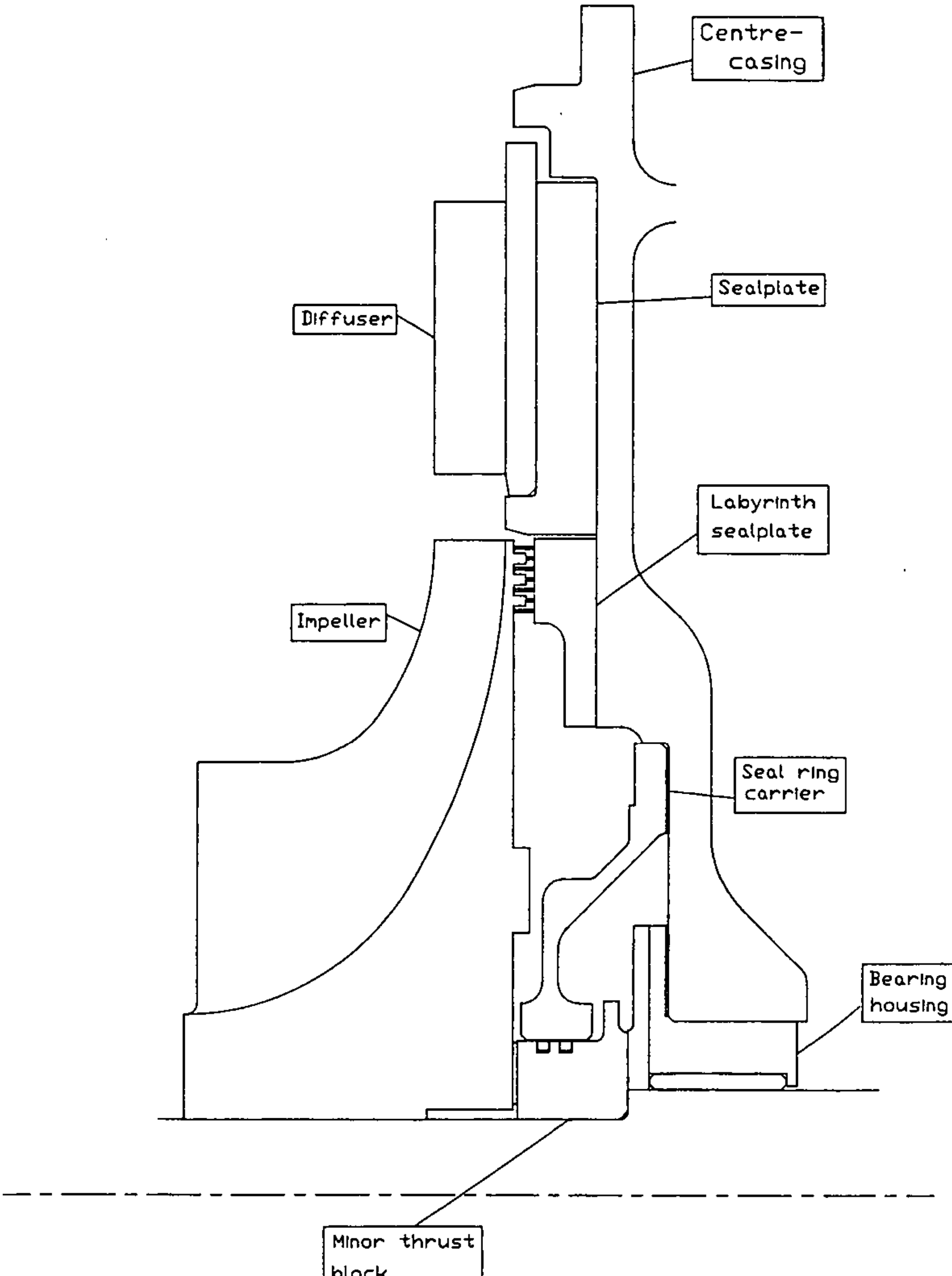


Figure 8.7: SA105 compressor end detail

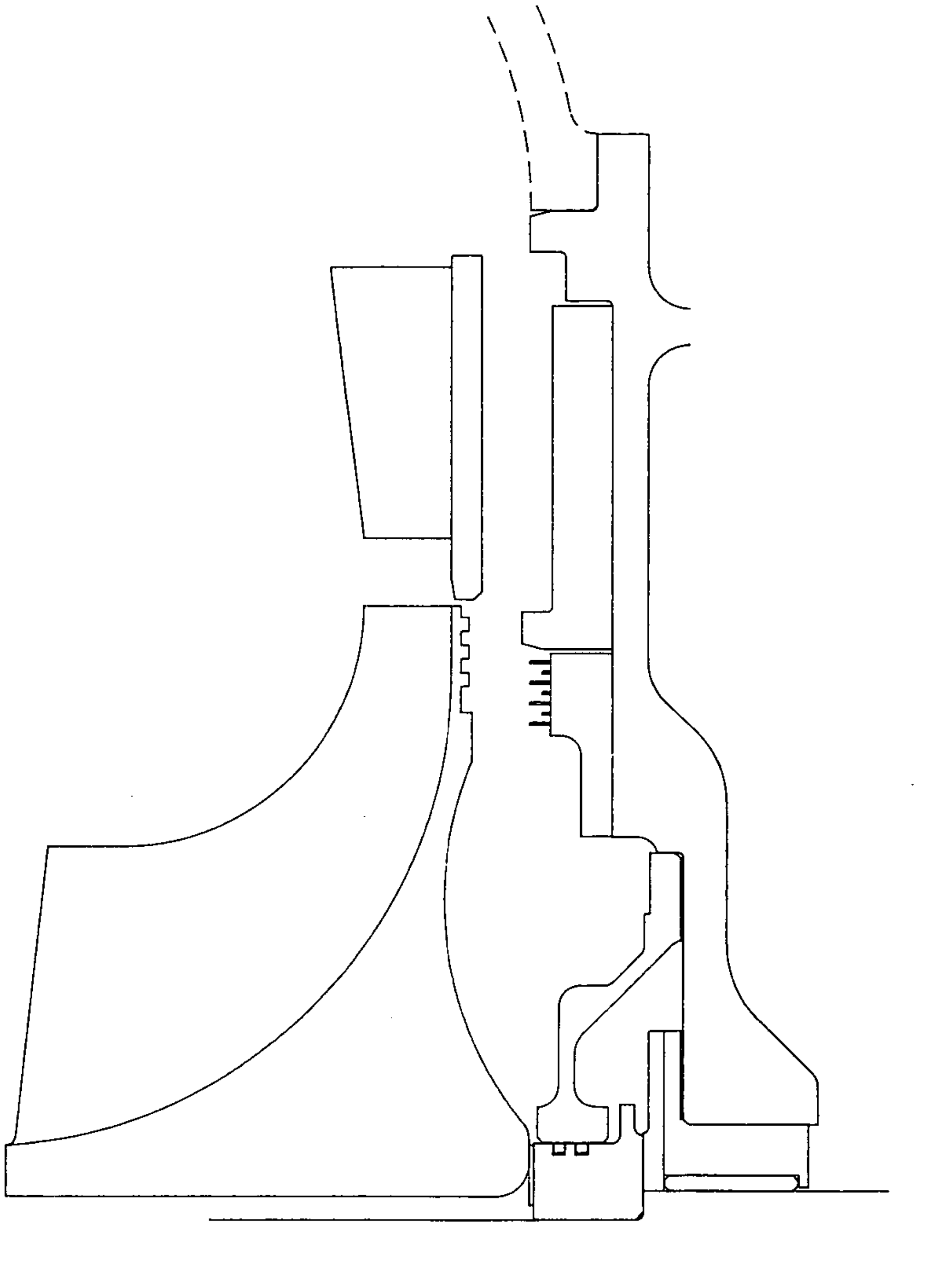


Figure 8.8: SA105 compressor detail with NA355 compressor

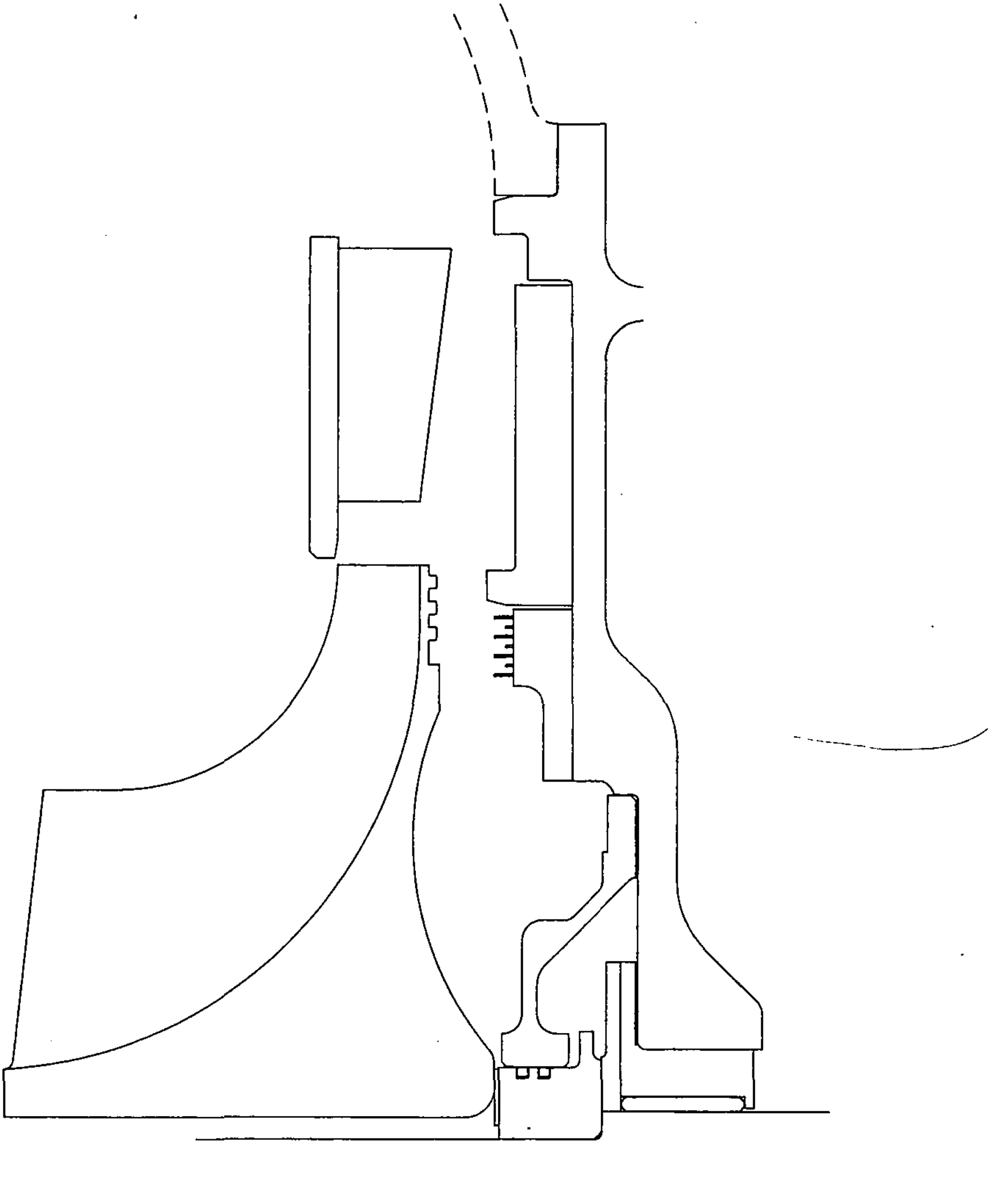


Figure 8.9: Scheme with reversed diffuser

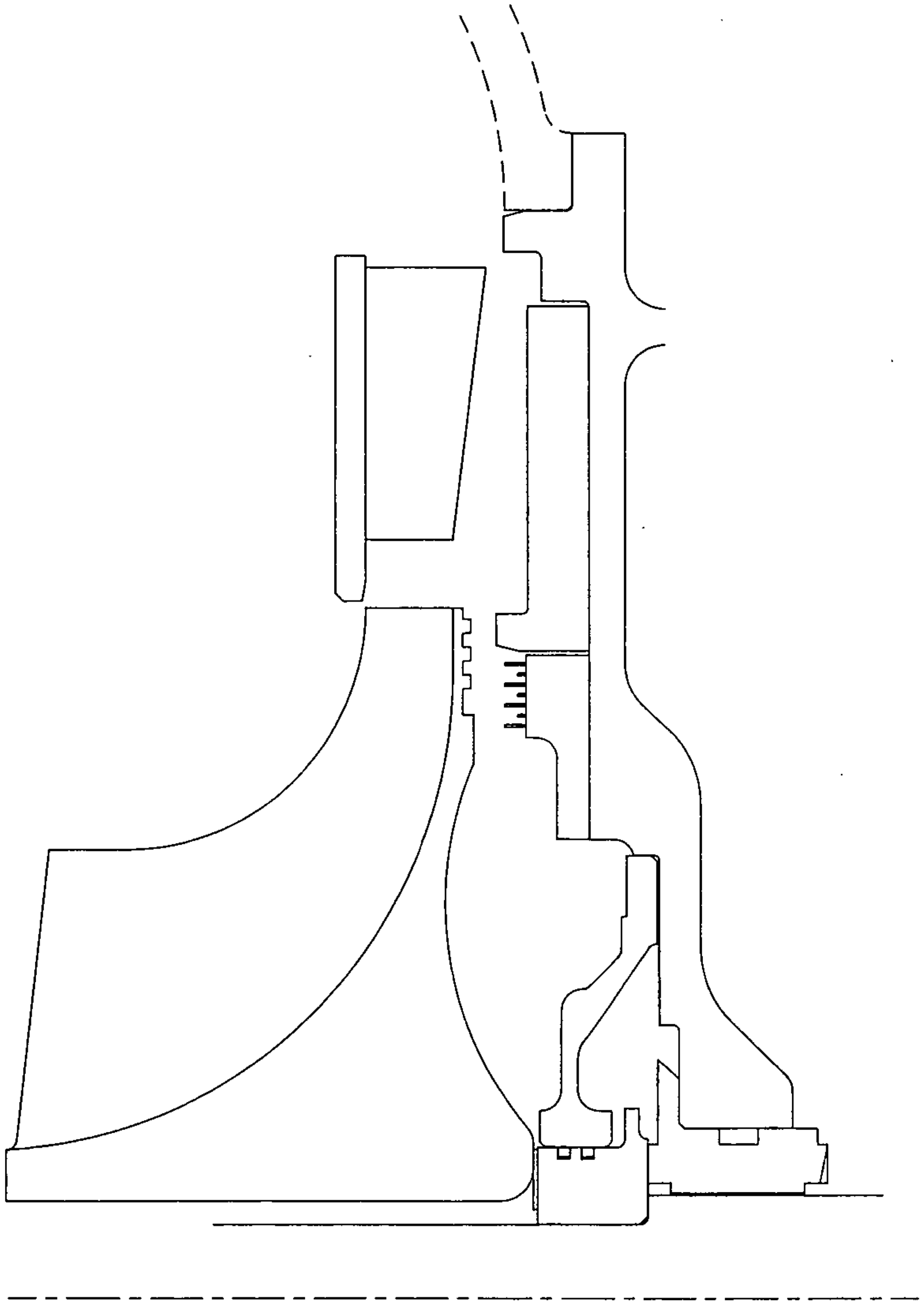


Figure 8.10: Scheme with reversed diffuser and recessed bearing

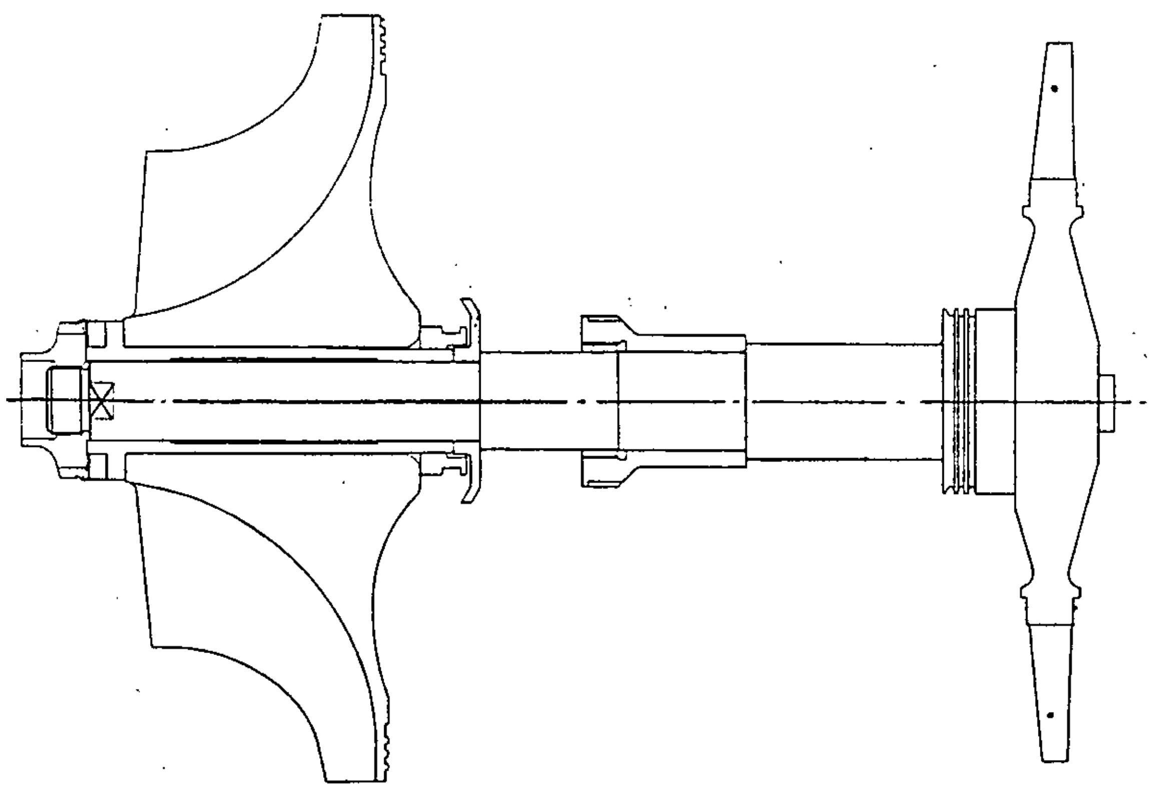


Figure 8.11: NA355 rotor assembly

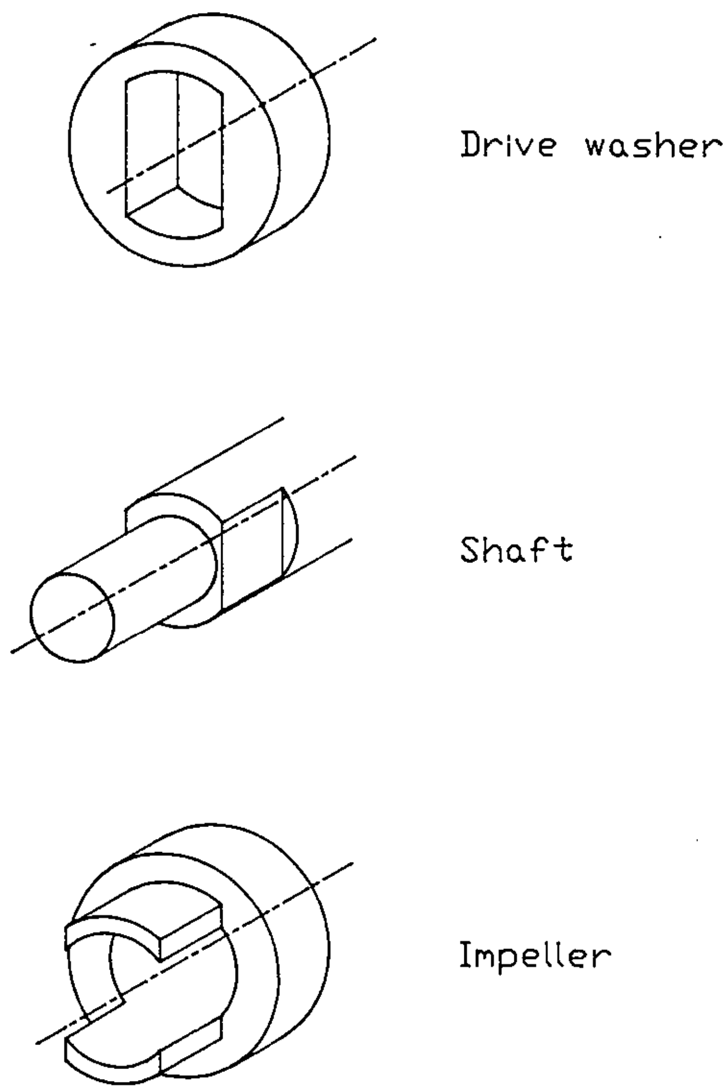


Figure 8.12: Impeller drive arrangement

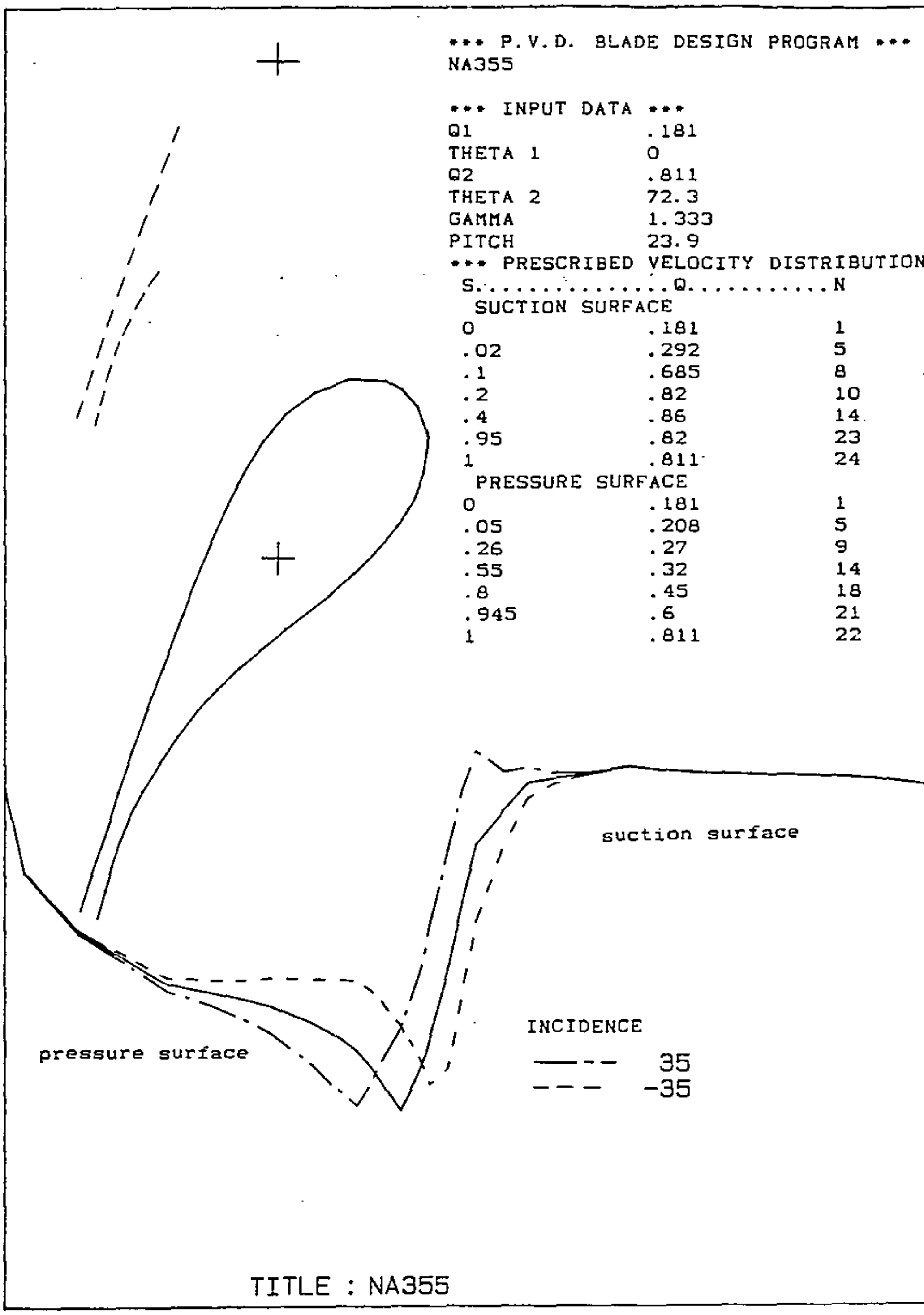
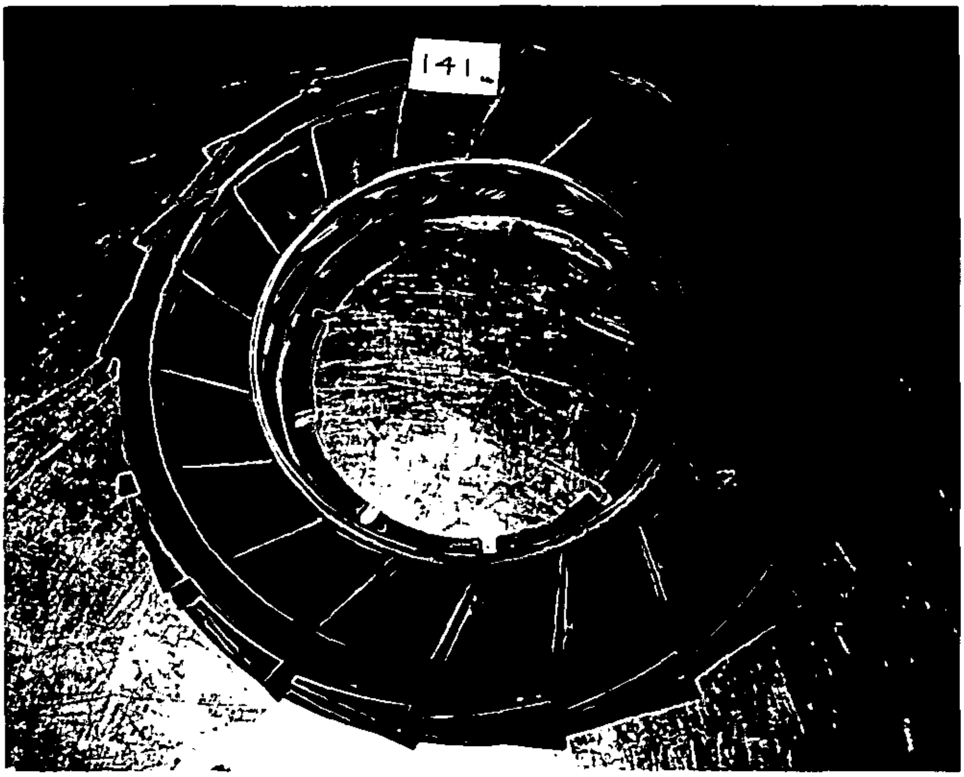
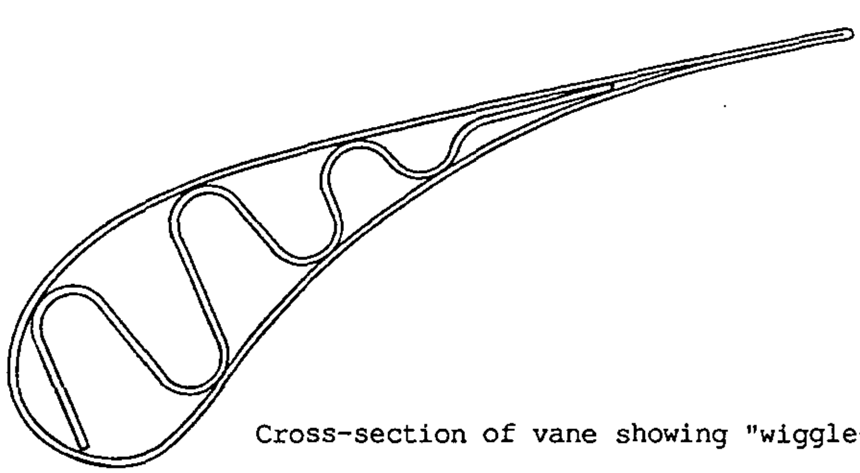


Figure 8.13: NA355 nozzle blade design by PVD



Complete nozzle after service experience



Cross-section of vane showing "wiggle-strip"

Figure 8.14: NA355 profiled nozzle

1 Air filter silencer
2 Compressor delivery casing
3 Centre casing
4 Turbine outlet casing
5 Turbine inlet casing
6 Impeller
7 Diffuser
8 Rotor shroud
9 Nozzle

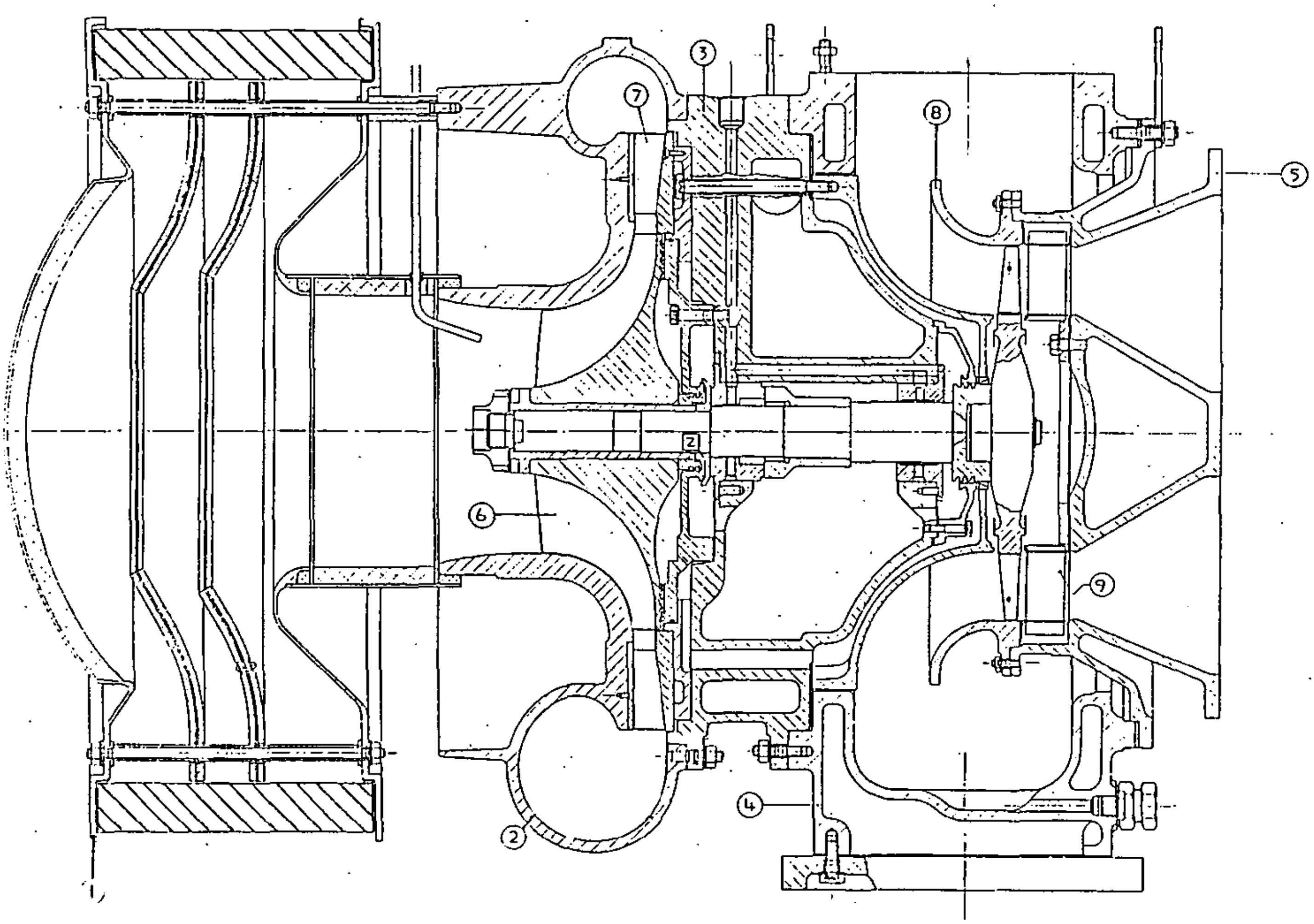


Figure 8.15: NA355 general assembly

CHAPTER 9

VALIDATION

9.1 Theoretical analysis

The new components required for the NA355 had been designed using techniques that were considered satisfactory. However, the turbocharger was to incorporate many components from the SA105. These items had been well-proven in service, but at operating conditions well below those required from the NA355. In addition some components, such as the rotating assembly, required work to be carried out to ensure that they would be satisfactory. The philosophy adopted was that much of the outstanding work had been considered at the turbocharger concept stage. This was primarily to ensure that the proposals made were viable. The intention was then to use the time between the end of the design stage and the first turbocharger manufacture to ensure that as thorough as possible knowledge of the limitations was obtained.

9.1.1 Rotordynamic analysis

Some of the limitations would be determined when the first turbochargers were operated. However, advance knowledge of possible problems was considered of great importance. In particular an understanding of the rotordynamics was considered essential. The theoretical analysis of even a relatively simple rotor such as the NA355 is far from an exact science. As described, the rotating assembly had been the result of a great deal of experience and the application of good practices. The problems associated with the NA350 rotating assembly mainly affected the repeatability of the behaviour at operational conditions. It was considered that the NA355 would not suffer from these problems. The available techniques were thought to be adequate for obtaining an initial understanding of the rotor behaviour. As discussed previously the techniques available allowed the undamped critical speeds to be determined using transfer matrices methods. The shaft was divided into a series of elements. Each element had a length, a stiffness, a moment of area, a mass, and a moment of inertia associated with it. The mass outer and inner diameters were used to calculate the element mass and moment of inertia. The stiffness outer and inner diameter may be less than the mass diameters to account for components added to the shaft that do not contribute to the stiffness of the element.

A view of the NA355 shaft model is given at the top of figure 9.1, showing both the mass and stiffness diameters. The mass and moment of inertia of an element are divided equally between each end. Additional masses and inertias, due to the compressor and turbine for example are also included at the ends of the relevant elements. The bearings are modelled as springs. In practice a range of bearing stiffnesses are chosen so that the tolerances of the bearing components can be included. Also in figure 9.1 are the particular critical speed shaft deflected shapes, shown grossly exaggerated, for specific values of bearing stiffnesses. In general the effects of shear and forward and backward whirl due to gyroscopic effects were considered. The resulting undamped critical speed map is shown in figure 9.2.

The areas of most interest are between the intersections of the bearing impedance range and the particular modes. The first and second critical speeds are rock and bounce modes respectively and are at a too low rotational speed to cause concern. The third critical speed is the first shaft bending mode and the areas of interest occur in the expected operational speed range. Backward modes are usually well damped and the intersection occurs at the lower end of the expected speed range. Therefore it was not expected that this mode would cause a problem. The third mode forward whirl intersection with the lower bearing stiffnesses appears to occur at a rotational speed close to the maximum.

The bearing stiffnesses were calculated for centralised journals. In practice the bearing stiffnesses are non-linear and they would become higher when the journal moved towards the bearing. It was considered that the resulting effective critical speeds would be higher than the maximum rotational speed required.

9.1.2 Thrust bearing analysis

The NA350 major thrust bearing had been retained as this had a good service record. It was considered that this could be subsequently increased in load carrying capacity if this was thought necessary. In order to determine the thrust bearing parameters it was first of all necessary to calculate the axial thrust load resulting from the pressures on the rotating components. The compressor aerodynamic design analysis provided the schedule of static pressure operating on the impeller. This enabled the axial thrust due to this pressure distribution to be calculated. In general the resultant axial lift due to the pressure difference on either side of the vane was ignored. Introduction of this effect was considered to be not worthwhile due to the level of accuracy of the whole thrust calculation and the additional complication that it would have meant.

The pressures acting on both sides of the turbine blades was also determined from the aerodynamic analysis. This enabled the resulting thrust across the turbine blades to be calculated. In addition, the static pressure at the root of the turbine acted over the whole upstream surface of the disc, enabling the thrust on that area to be calculated.

The static pressures, and temperatures at the impeller tip and at the downstream side of the blade root were also known from the compressor and turbine analysis. The flow of seal/cooling air was from the impeller tip through the impeller tip seals, through the centre casing air passages, into the space between the centre casing and the outer cone, through the holes in the turbine end oil sealing bush, and out through the space between the cone and the turbine wheel. The geometric details of the various restrictions in this seal air flow passage were known. A number of assumptions had to be made so that the mass flow of sealing air, and hence the pressures acting throughout, could be determined. It was assumed, for convenience, that the pressure drop through each restriction was a fixed percentage of the dynamic head. This percentage was varied to suit the type of restriction, the most common value being 100%. The process is iterative; an initial value of mass flow is assumed and the pressure after each restriction calculated and finally compared with the final value required. The initial guess is modified until the final pressures agree.

Determination of these pressures allows the thrust on the relevant parts of the rotating components to be calculated. Having obtained the axial thrust acting on all faces of the rotor the resulting net thrust could be determined. The accuracy of the technique described above is not good, there are many assumptions and simplifications. The net thrust is usually very small when compared with the individual calculations. Also in a particular engine installation the assumed operating conditions may be widely inaccurate. Figure 9.3 presents the thrust on either side of the impeller together with the net thrust.

The thrust bearing analysis technique previously described was used to determine the bearing parameters, and a comparison was made with the NA350. Although the thrust bearings were essentially the same in the two turbochargers the axial thrust relationship with rotational speed was somewhat different. The thrust bearing analysis program required the rotational speed and minimum oil-film thickness in addition to the geometric and oil property data. The output from the program was the axial load and general parameters such as the power loss and oil temperatures. Therefore, by running the program with a number of different values of minimum oil-film thickness, the basic relationship between the axial load and minimum oil-film thickness was obtained. Cross-plotting produced the results shown in figure 9.4. The minimum oil-film thickness at the highest speed was small; a value of 6.4 μm was predicted at 23000 rev/min. However, experience with other turbochargers where similar values had been predicted suggested that the NA355 would be satisfactory. At these sort of values the surface finish of the mating surfaces clearly becomes of vital significance.

The rotating part of the NA355 turbine was essentially the same as the SA105. The original work carried out to provide the blade and disc mechanical limitations was reviewed during the NA355 concept stage. This suggested that the rotor would not be limited by the turbine particularly if Nimonic was the material adopted for both the blade and disc. The SA105 had a stainless option for lower speeds and temperatures. However, Nimonic was used for the higher ratings. It was considered prudent to re-examine the work with respect to the expected duties of the NA355.

9.2 Rig testing

It was intended that the most important aspects of assessing the NA355 would be during the testing of the complete turbocharger. At this stage, particularly in regard to the mechanical features, the turbocharger would either perform well or problems would occur almost immediately. However, some features would develop into problems only after a considerable period, and a programme of testing had to be devised to ensure that any problems came to the surface as soon as possible.

A turbocharger is an addition to a diesel engine, but the cost is a small fraction of the total installation cost. Therefore, it was common to test the turbocharger within the Company on a self-contained rig and not on a diesel engine. In addition it was general for a new turbocharger to be fitted to a customers development engine, so that the customer could evaluate the product and also Napier could assess the likely service behaviour.

9.2.1 Napier test rig

The in-house facility was arranged so that the turbocharger operated as an open-cycle gas turbine. The combustion chamber effectively replaced the diesel engine cylinder and allowed the turbocharger to be operated over the whole speed range and at representative turbine inlet temperatures. The major competitor has published details of similar test facilities(9.1, 9.2, 9.3). A schematic of the Napier test rig is shown in figure 9.5. The mass flow measurement nozzle was a considerable distance upstream of the compressor inlet of the turbocharger, situated inside a large plenum chamber. After passing through the compressor the air was ducted into the combustion chamber and then to the turbine inlet.

The exhaust from the turbocharger turbine was led away through a silencer to the atmosphere. There was also a control valve in the closed-loop ducting to enable the working line to be adjusted. The starting arrangement employed a low-powered auxiliary fan, and a system of valves both in the main compressor inlet ducting and the starting fan ducting. The object of the starting fan was simply to make up the losses in the rig system and the disparity between the compressor and turbine efficiency at low speed. In order to start the turbocharger the valve in the compressor inlet ducting was closed. The starting fan was operated at a low rate sufficient to allow the turbocharger to rotate slowly and for the combustion chamber to be supplied with sufficient air for ignition of the fuel. The system was then in a position to accelerate the turbocharger slowly and with great control. The fuel flow and the starting fan speed were increased maintaining the turbine inlet temperature below the limit. The system could proceed until the turbocharger was above the minimum self-support speed. The fan speed could be adjusted so that the compressor inlet pressure would be similar to that through the normal compressor inlet ducting. The starting valve was then opened, the starting fan stopped and the starting fan valve closed.

This test rig was suitable for testing larger turbochargers than the NA355. However, a view of the effectiveness of this starting system can be obtained by the estimation that a starting fan of 10 hp would have been sufficient to start the NA355. This figure can be compared with component shaft powers of the order of 2000 hp. In order to be able to test the compressor with an air filter silencer a somewhat more elaborate arrangement of ducting was required. To overcome this difficulty a similar starting system placed in the turbine exhaust was successfully employed on a smaller turbocharger test rig.

9.2.2 Test rig instrumentation

The standard instrumentation at inlet and outlet flanges included at least four static pressure tappings, a total pressure rake to provide a mass averaged value, and at least two total temperature probes. The temperatures were measured by means of shrouded thermocouples. The dynamic head recovery was considered sufficiently good to use the temperature as total values. Mass flow measurement was in accordance with BS1042 for a nozzle with an large upstream space. The rotational speed was determined by measuring the frequency of the change of magnetic flux seen by a probe placed against a variable component on the rotor shaft. The shaft component had two notches placed on its periphery, this being sufficient to cause a detectable change in flux. At the time of the NA355 test evaluation data-logging had not been introduced. The pressures were measured by means of water and mercury manometers, and the temperatures were recorded manually. The reduced data from the test rig were used for three purposes:

a) To drive the turbocharger

b) To enable the component performance to be determined

c) To provide a check that the level of instrumentation was satisfactory.

The settings at which the readings were taken were based upon the compressor values for convenience. For example, the standard $N/\sqrt{T}$ values given in figure 3.1 of 800-1200 in steps of 100, and also including 700 and 1300 were the general rotational speeds used. In order to drive the turbocharger the rotational speed, compressor pressure ratio and capacity were required. This required the reduction of very few readings. For example, the readings required were the speed, compressor inlet total pressure and outlet static pressure, inlet total temperature, the pressure drop through the mass flow measurement device and temperature in addition to the barometric pressure. In order to obtain the compressor and turbine performance the basic readings at each measurement point and the relevant reductions were required. As mentioned in section 2 the turbine is debited with the mechanical efficiency for convenience.

The most important aspect of assessing the test results concerns the instrumentation checks. At this level of testing it is not acceptable to claim that truly absolute values are being determined, every reading has an associated tolerance. However, it is expected that the relative accuracy should be acceptable. For example, the nozzle used to measure mass flow may be inaccurate up to a figure of 4% in absolute value. However, it would be unacceptable if this sort of difference occurred from one test to another. The obvious check upon the instrumentation is a continuity check. The dynamic head at each measurement station must be related to and can be correlated with the mass flow. The pressure drop across the mass flow measurement nozzle and the recovery afterwards together with a particular inlet ducting means that the pressure drop, inlet flange static and total pressure are directly related together. Therefore, any differences in this relationship can be highlighted. The amount of fuel burnt, and the fuel details can allow the turbine inlet temperature to be estimated, again differences from the measured value can attract greater scrutiny. Although the readings were taken manually the data reduction was carried out with the aid of a micro-computer. Therefore, many and varied complex checking routines could be included.

9.2.3 Test rig control

The turbine swallowing capacity will generally dictate the operating position in this method of rig testing. The capacity versus speed relationship will be unique for a particular turbocharger build, provided the rig remains unaltered. There are a number of ways that this unique relationship can be changed. The most effective means of control is to introduce a pressure drop into the system. At Napier this was done by adjusting the control valve shown in figure 9.5 in the closed loop ducting. Closing this valve effectively reduces the available turbine expansion ratio for the same rotational speed, and the turbine will move to a lower point on the swallowing capacity thereby reducing the capacity. As the system has become less efficient more fuel will be required so that the turbine inlet temperature will rise. As the turbine capacity has been reduced, the turbine inlet temperature increased and the inlet pressure reduced then the turbine mass flow will be reduced. Reducing the turbine mass flow will naturally reduce the compressor mass flow, therefore the compressor operating point will move closer towards surge. Other features were used for special control, such as allowing air to escape to atmosphere in the closed loop ducting. This would artificially raise the turbine inlet temperature generally without changing the working line position by a great amount. Smaller rigs had the capability of adding compressed air into the closed loop ducting, as the amount of air was usually very small in comparison to the turbocharger flow this generally allowed the turbine to operate at a lower temperature level. By the means described above it was considered that any type of operating condition could be reproduced.

9.3 First turbocharger testing

The initial aim of the NA355 testing was to obtain as much information about all aspects of the turbocharger before carrying out more detailed investigations. A production test normally consisted of a set sequence of observations. Firstly there was a vibration run from the minimum stable speed up to the maximum speed in steps of 1000 rev/min. Secondly a full set of observations were taken at the maximum speed point. This was followed by reducing speed to 700 $N/\sqrt{T}$ and then carrying out points on increasing $N/\sqrt{T}$ values up to 1300. Finally, at 1200 $N/\sqrt{T}$, the closed loop ducting valve was adjusted and a surge point and point close to surge were observed. The purpose of the initial run up to maximum speed was to ensure that the unit was satisfactory from a vibration point of view, the rig was performing without problem and that the instrumentation was to an acceptable standard. For the first series of tests of a new turbocharger it would have been undesirable if the sequence just described was adopted. If the turbocharger developed a sufficiently serious problem, say whilst running at the maximum speed, to cause a failure, then the test programme could have been delayed for a number of weeks until the damage was repaired. In the meantime no performance information would have been available. However, a series of test were carried out, collecting all the required performance data at steadily increasing speeds until, eventually, the maximum speed was reached.

The first NA355 turbocharger produced was intended for a Norwegian engine, the Wichmann 7AXAG. This was viewed with some mixed feelings. On the beneficial side the rating was not too high, the boost pressure ratio was only 2.4:1 at full load. On the debit side, monitoring of the unit would not be too straightforward because of the remote location. In many ways it would have been preferable for the first unit to have gone to either Ruston or Paxman as they were both within GEC and in the UK. The initial series of rig tests were intended to provide sufficient performance data and to ensure that no unsatisfactory mechanical behaviour was observed. In the event, the testing was carried out without any particular incident. The performance was in accord with predictions, and the unit behaved mechanically without any problem. Subsequent disassembly also revealed no problems. The amount of testing had been kept to a minimum, mainly to achieve the required delivery date, although there was no particular reason for prolonging the test of that unit.

The first turbocharger was released to the customer, on the understanding that the second unit would be tested more rigourously. Testing of the engine with the NA355 was subsequently carried out and the performance was compared with the VTR304. The results showed some inconsistency and some disappointment as the NA355 performed better than the VTR304 only at 100% full load and 10% overload. The difference in exhaust temperature was within 10°C and the specific fuel consumption figures were within 4 g/kW/hr. The F turbine size was less than the most efficient for the NA355. However, the An/Ar was considered to be close to an optimum at a value of 0.704. With the size advantage, previously described, and the related cost advantage the customer was well satisfied with the turbocharger. Two problems were found on strip after the customer's engine test. After a great deal of discussion and investigation one of these problems was attributed to the engine. The other problem was associated with the downstream face of the fabricated nozzle. This had clearly not been in contact with the shroud, as was intended, and fatigue cracks had developed. Subsequently a new nozzle was fitted, and it was ensured that the downstream flange was in contact with the shroud. Although, any problem encountered particularly at the customer's site was undesirable this one was considered to be of a sufficiently minor nature.

9.4 Subsequent testing

Testing of the next turbochargers was split into two groups:

a) Performance

b) Mechanical

The performance was intended to be covered as and when a different build of turbocharger was tested. On an automatic basis, if the turbocharger build to be tested had not previously been tested then a full compressor calibration was determined. Carrying out this compressor calibration would mean that the performance of the particular turbine build would also be determined. In terms of overall performance checking this was considered sufficient. Specific performance programmes would then be intended to cover more particular requirements. For example, surge line determination at particularly critical areas for a specific customer might be carried out.

9.4.1 Endurance testing

For many reasons most of the investigative testing of subsequent turbochargers centred around obtaining a clearer understanding of the mechanical performance of the NA355. A series of endurance tests were planned, although endurance was not intended to mean to approach the creep-type predicted limits for the turbocharger. By endurance was meant a relatively short-term arduous-duty cyclic test. The schedule, although somewhat arbitrary, was intended to reflect a specific 20000 hr engine operation. The schedule determined was as follows:

a) 2 hours at 21400 rev/min

b) 1 hour at 20000 rev/min

c) 10 minutes at minimum self-support speed

d) 1 hour 50 minutes at 21400 rev/min

e) 10 minutes at minimum self-support speed

f) 30 minutes at 13600 rev/min

g) 10 minutes at minimum self-support speed

h) 10 minutes at 20000 rev/min

i) 1 hour at 23000 rev/min

j) 50 minutes at 21400 rev/min

k) 10 minutes shutdown

This was arranged to give a cycle of 8 hours, which was one working day. A total of fourteen cycles were planned , with the last test a standard production test so that at least 100 hours would be achieved. Some minor variations were also included, such as emergency shutdown at the end of the tenth cycle. All speed changes were to be achieved as quickly as possible, and within a maximum of 5 minutes. A great amount of inspection of various items before and after test were carried out, such as bearing and shaft diameters.

The result of the endurance testing on the Napier test rig was encouraging with virtually no problems encountered. This good result tended to suggest one of two situations was appropriate; either the NA355 would perform mechanically impeccably in the field or that it was impossible to imitate the real engine duty on a test rig. Clearly there was an element of truth in both interpretations. Experience with previous turbochargers had suggested that turbochargers in service would experience problems hardly imagined until they occurred.

9.5 Service problems

9.5.1 Piston ring wear

It was inevitable that some of the NA355 turbochargers experienced problems in service to a varying degree of severity. The first problem that came to light was piston-ring wear at the compressor end. These rings were used to provide a labyrinth so that the leakage of air past the seals into the oil-chamber was reduced to a minimum. The rings had to be free enough to move in the seal-ring bush to accommodate the thermal expansion of the shaft components. However, the increased NA355 ratings meant that the pressure drop across these rings was high enough to move the rings so that they rubbed on the shaft component. This caused severe and rapid wear. This problem may be considered somewhat minor in that the turbocharger could operate without rings, although seal-air leakage would be increased. What was introduced was called a positive stop, see figure 9.6. The way this operated was that the rings could move as before but were set so that when the air pressure moved them they would come up against a stop in the housing, thereby preventing wear.

9.5.2 Nozzle ring cracking

Mention has been made of the nozzle flange cracking which continued to be an irritating problem. The frequencies of the flange and nozzle were determined, and this suggested that the engine synchronous vibration was the most likely source of excitation. The conclusion was that the flange had to be incorrectly located and free of the shroud for this to be excited. Efforts were made to improve the integrity of the flange by ensuring that the radius was as large as the drawing specified, and care was taken to ensure that the flange was in contact with the shroud. A more serious fatigue failure of the nozzle vanes occurred. The frequencies of this nozzle assembly were determined, as normally clamped by the inner ring and also with the inner ring lightly clamped in a similar manner to the as found condition. In this case it was concluded that the vanes had cracked due to excitation of a 3rd diametral disc mode coupled with a first flap of the vane mode. The way that this mode might be excited was due to the 6th harmonic of the engine firing frequency interference around the 600 rev/min engine speed, the lower end of its speed range. The means of ensuring that the 6th harmonic was of sufficient energy to excite the nozzle was provided by the fact that the 18 cylinders were connected to a 6 entry turbine inlet casing. The mode of vibration could only be produced with a lower than standard inner ring clamping load. Whilst investigating the failed nozzle it was noticed that certain areas of the vane had not been brazed satisfactorily to the wigglestrips. This was mainly due to poorly fitting components. The quality of the components was improved and re-checking of the modes of vibration with adequate clamping proved that the problem would not re-occur.

9.5.3 Thrust bearing failure

The most serious problem encountered was a failure at high speed of the major thrust bearing. The effect of this type of failure is that a considerable amount of damage is caused before the turbocharger comes to rest. Irreparable damage is caused to the rotating assembly, and to the compressor delivery casing in the most fortunate of cases. In extreme cases, the whole of the turbocharger is damaged beyond repair. A certain amount of detective work is necessary to determine the prime item of failure. The work to determine why the bearing failed was considerable. A number of hypotheses were tested and found to be wanting. The cause of this problem was eventually attributed to oil-whirl in the journal bearings. Some levels of sub-synchronous vibration, in the region of .4-.5 times the synchronous frequency, had on occasions been seen on test, and some fretting between rotor components had been seen in service. These tend to be the outward signs of rotordynamic problems. The whirl orbits were determined with a total of six proximity probes measuring rotor displacement. The testing was carried out with a thrust bearing better able to withstand some contact rather than the standard bearing. The phenomena was induced and the mode of vibration was in a conical manner with extremely large orbits in relation to the bearing clearance. Clearly the orientation of the shaft when this mode of vibration occurred was sufficient to allow contact between the thrust bearing and thrust collar. The way that the resonance was avoided was to modify the bearings, and reduce the journal bearing clearance, when it was found that the phenomena could not be induced.

9.5.4 Sales record

Although some detail has been provided of the problems encountered in service with the NA355 , this should not divert attention from the numbers of units which have accumulated many thousands of hours without problem. As an indication of the success of the NA355 the numbers sold throughout the period are presented:

Year	Quantity
1983	5
1984	39
1985	19
1986	19
1987	14
1988	12
1989	42
total	150

Allowing for the natural swings expected in this type of business an average of 21 per year represents a reasonably satisfactory level of business.

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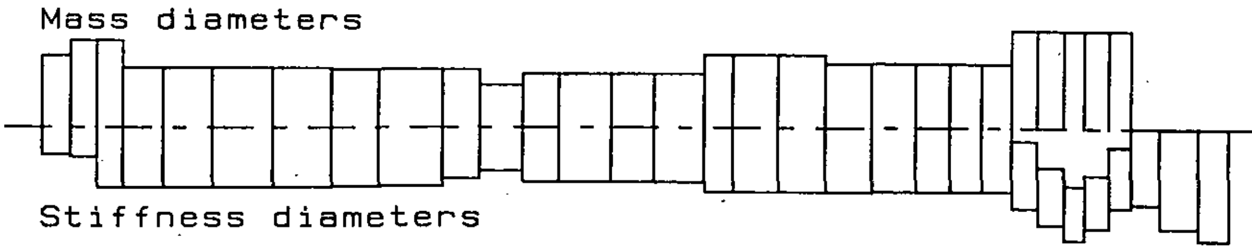
9.2 H Spaeti, Measurement,analysis and assessment of compressor characteristics in turbochargers. Brown Boveri review. (1977)

9.3 U Burkhard, Acceptance testing of turbochargers. Brown Boveri review. (1977)

FIGURES

SHAFT MODEL



SHAFT DEFLECTED SHAPES

critical speed = 6207 rev/min

critical speed = 16031 rev/min

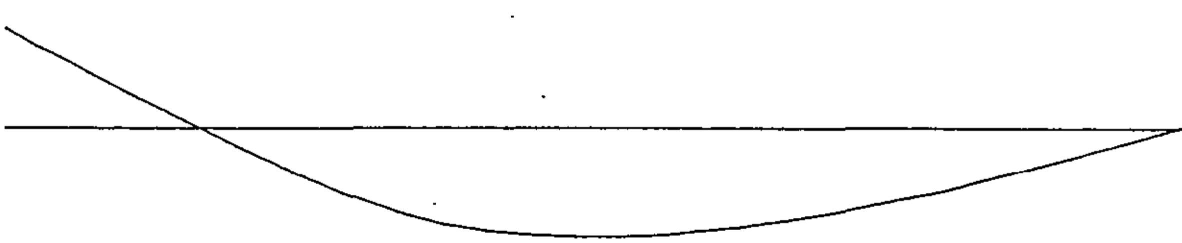


Figure 9.1: NA355 critical speeds
Bearing stiffnesses = 60000 N/mm

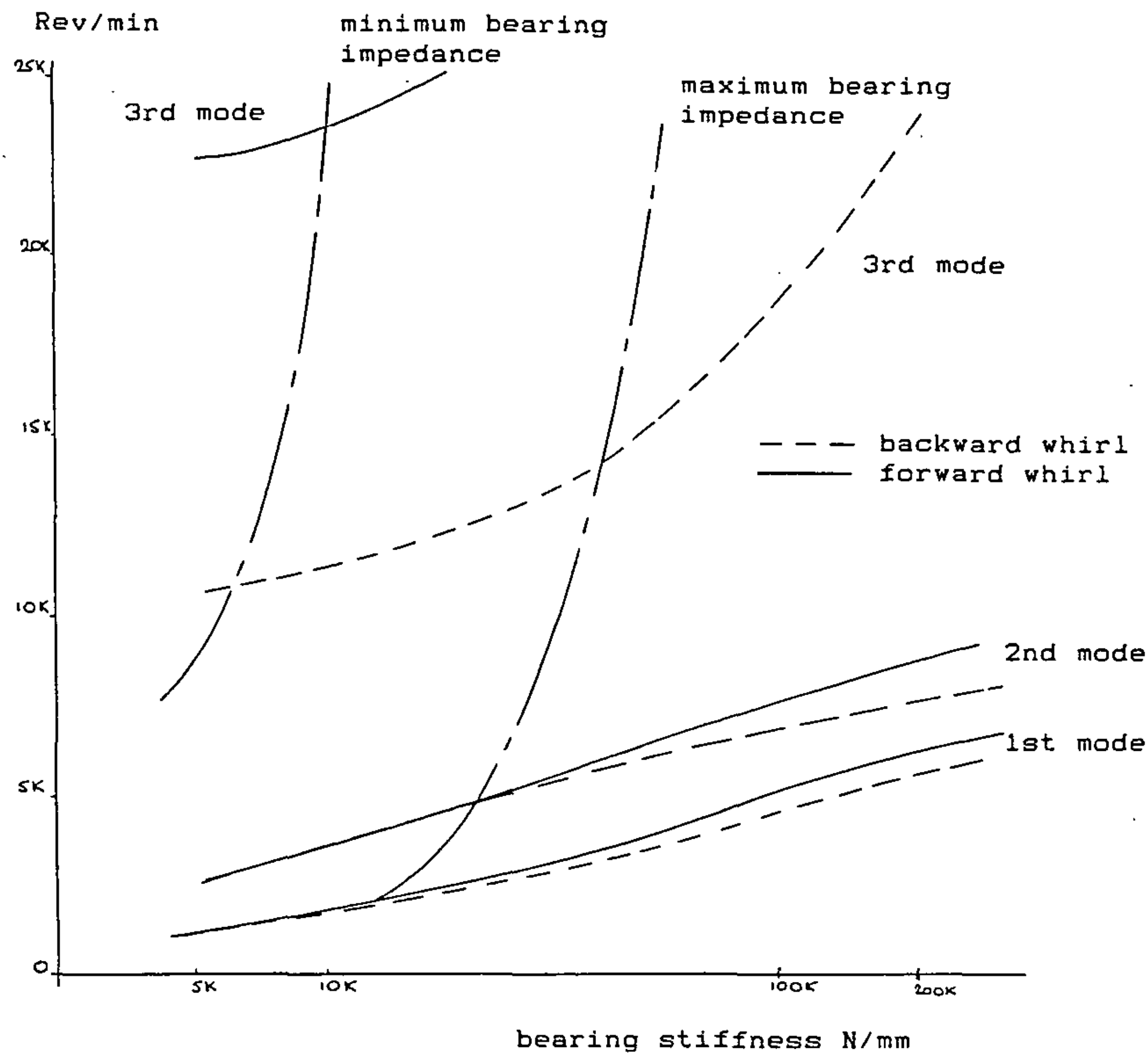
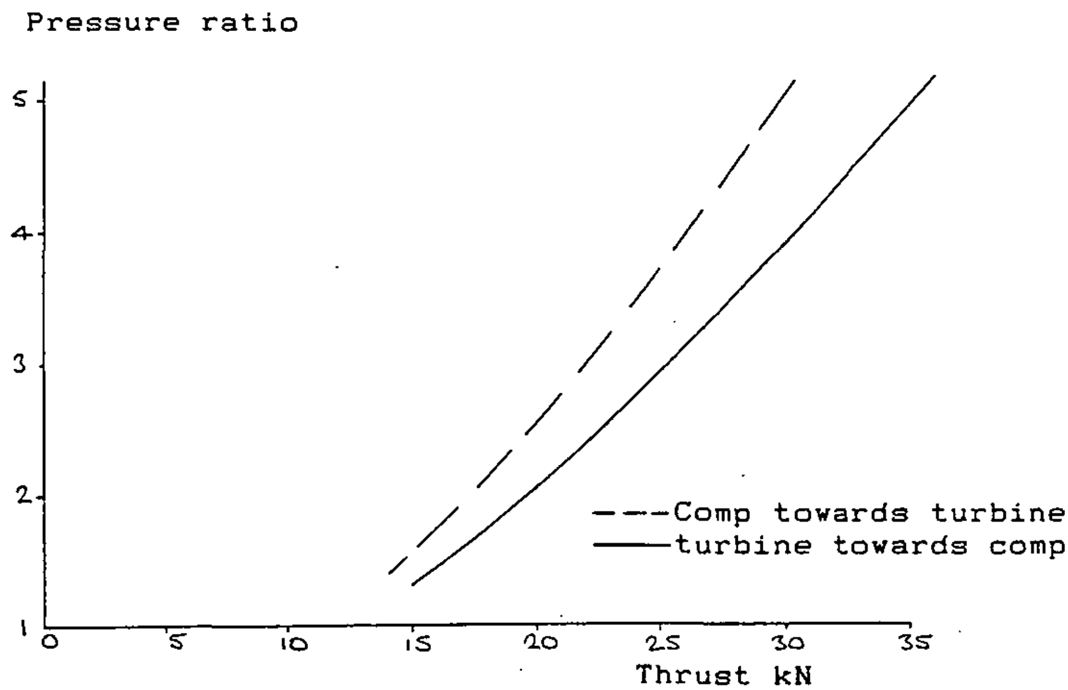
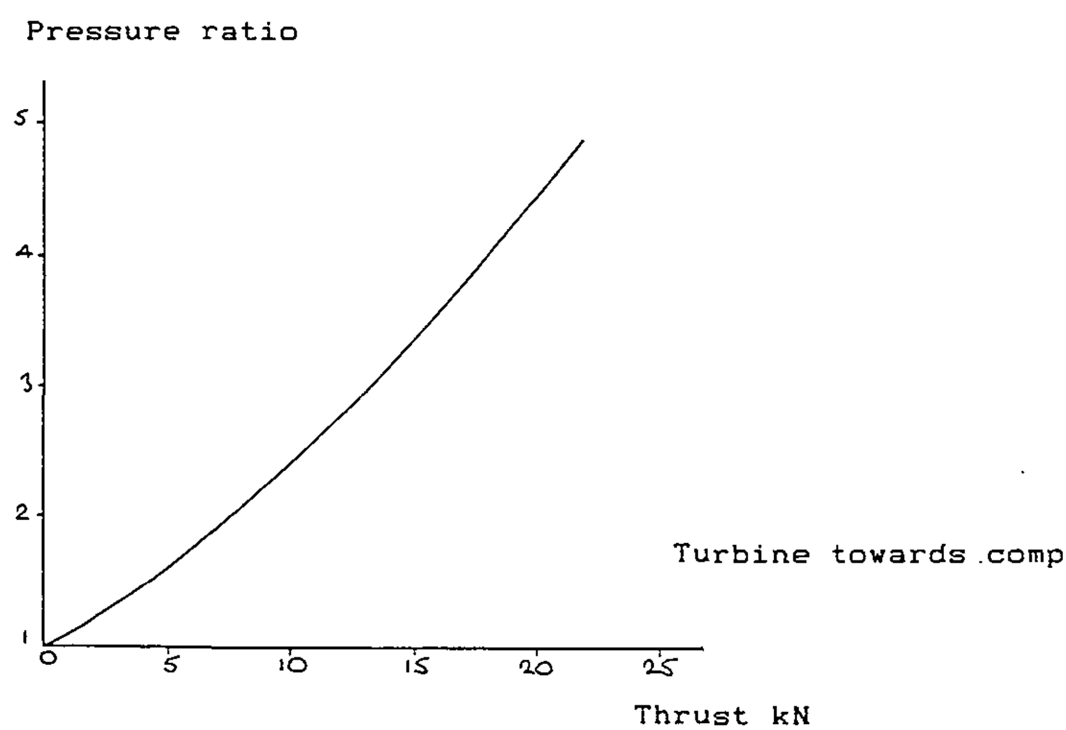


Figure 9.2: NA355 undamped critical speed map



IMPELLER THRUST



TOTAL THRUST

Figure 9.3: NA355 axial thrust

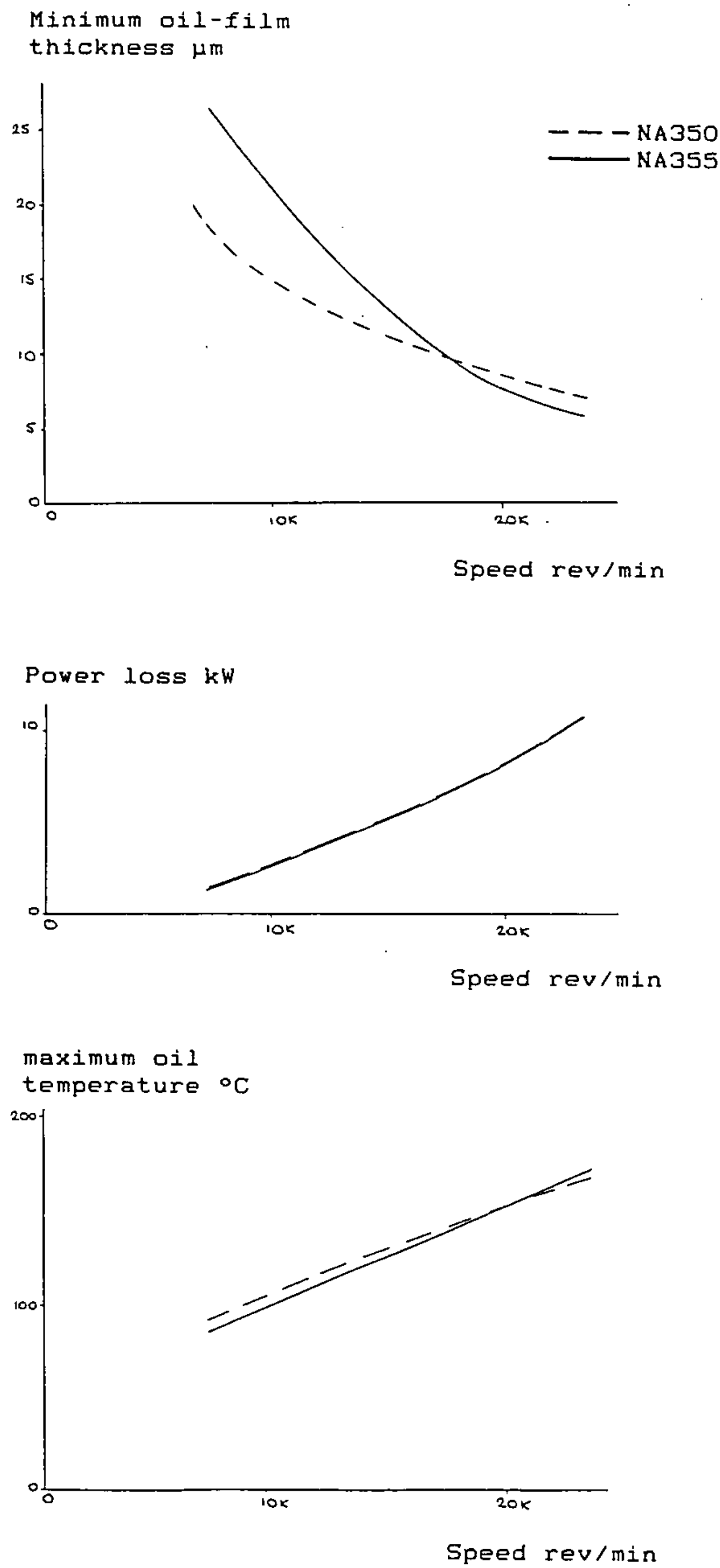


Figure 9.4: NA355 axial thrust bearing performance

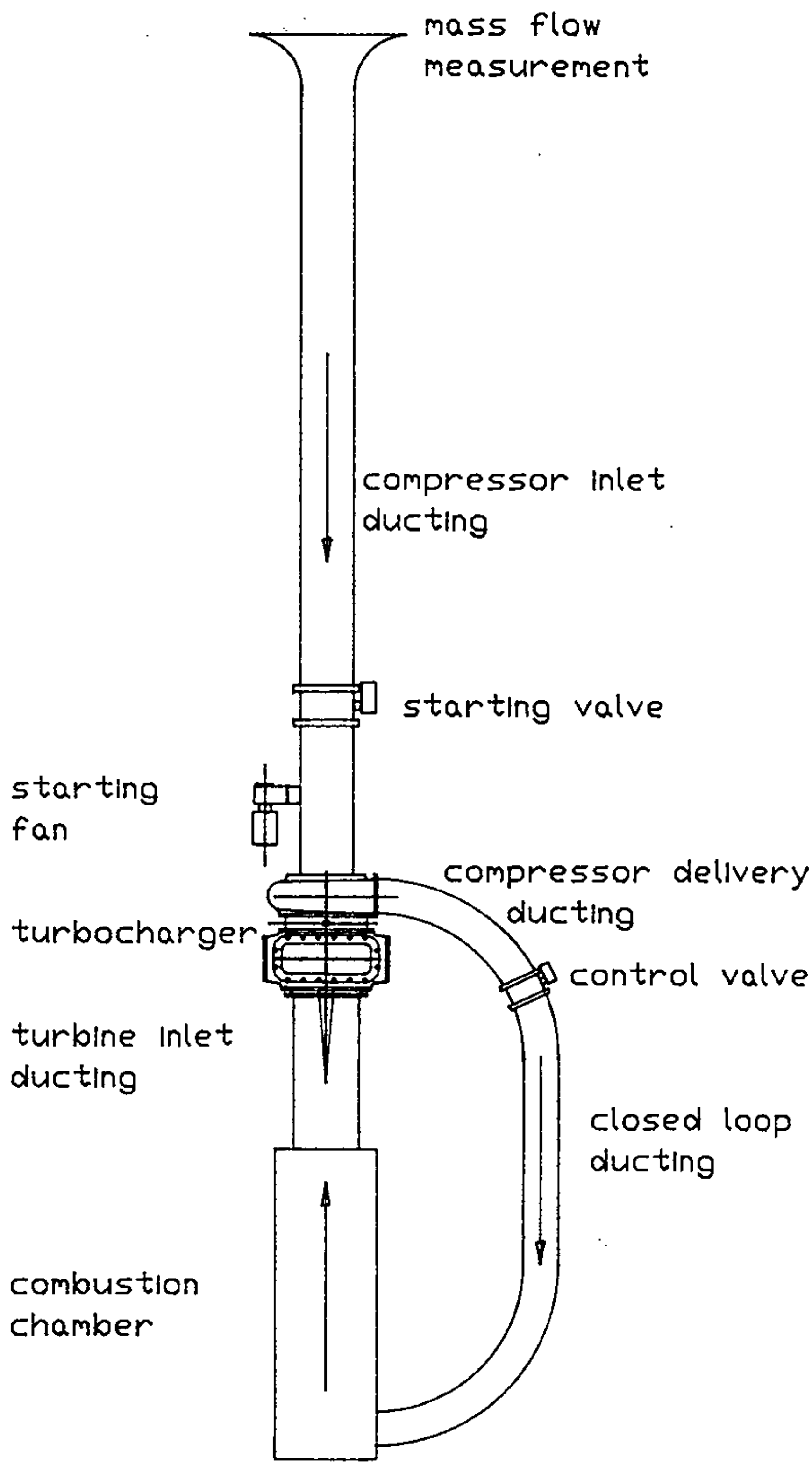


Figure 9.5: NA355 test rig schematic

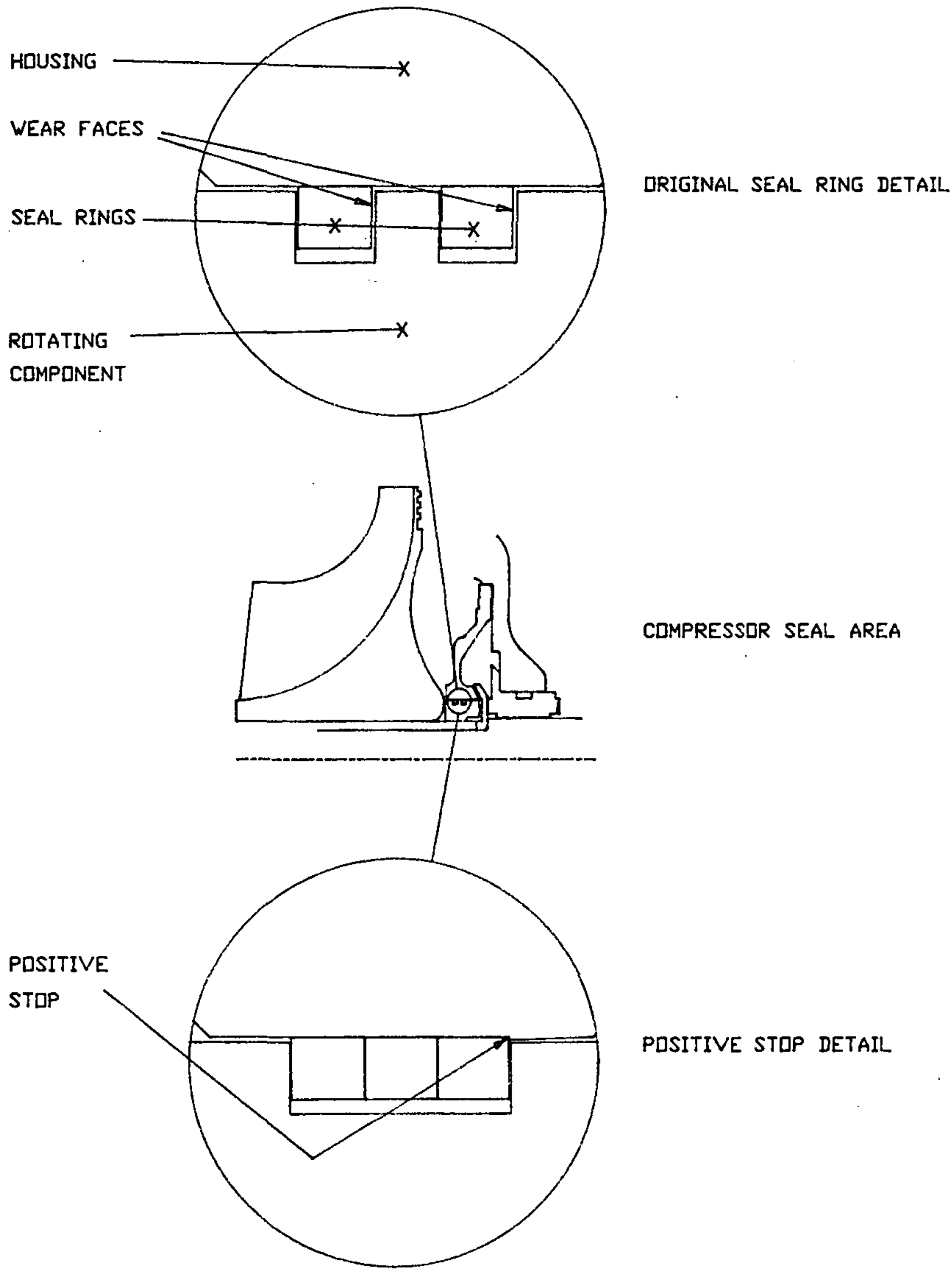


Figure 9.6: Positive stop arrangement

CHAPTER 10

CONCLUDING REMARKS

10.1 The market

The way that Napier chose to gather and analyse the market information was typical of the type of company and of the type of product. A method of operating was chosen and depended on particular individuals with their special attributes. For the General Manager to actively guide the marketing policy would not work, possibly, in larger companies or where consumer goods are involved.

A company can use the spur of competition to good effect. Less competition within the turbocharger business would have meant that the designs would have stagnated. To develop a product to remain competitive, and if necessary to adapt and adopt ideas so that the product will have a vital edge is all important.

10.2 The existing product

The problem with a reasonably successful design is to determine when its successor should be introduced. By the time the sales figures for a product begin to fall the starting date for its successor may well be passed. Also for a reasonably successful design an amount of inertia to change, or improvement, is often felt. Sometimes this is the result of slightly mis-placed loyalty and pride in the product. It was necessary to be objective and clear in the assessment of the existing product.

10.3 Proposals

The gestation time for the NA355, between concept and sufficient sales to make its introduction worthwhile was about 4 years, although the first sale was within 2 years. This could only have been achieved by adopting the philosophy of incorporating as many components from the previous turbocharger as possible. The approach could have been simply to improve the existing product, possibly with the proviso that nothing could be changed. However, the approach was to decide what was achievable by a new product, and then to try to adapt as many of the existing product components to achieve the new turbocharger.

10.4 Company targets

Introduction of a new product could have been considered as the activity of the engineering department. This would have ensured failure of any new product. The company must be solidly behind any new project, so that jointly agreed targets are achieved or changed to suit the situation. Napier chose to spread its resources during the period of this project, thereby delaying the NA355. This is not necessarily undesirable, and in fact some advantages can be seen. It would have been undesirable if this delay had just happened, but by planning to change priorities it became acceptable. Moving a project at an acceptable pace can improve the amount of "thinking" time by the design function, and can remove unnecessary pressures. Introducing an element of science into the project planning was beneficial.

10.5 Detail

From a position some years after the introduction of the NA355 and with the evolution of other turbochargers of a similar family it easy to see how the turbocharger may be improved. However, these changes are relatively minor. Therefore, it must be concluded that in general the NA355 has stood the test of time. The techniques used in the design of the NA355 have been proved to be adequate for the purpose, and for that time. Some of the techniques currently being used are improvements upon those used during this project. The way in which the details were incorporated into a scheme and then the geometry had to be re-created by hand has changed dramatically with the introduction of computer aided design (CAD). This has changed the way that the design is considered, releasing some mental constraints whilst introducing others.

10.6 **Validation**

Proving that the turbocharger finally achieved was what was intended and required was an on-going situation for a considerable period of time. For those involved in the design function the process has not ended. Every new application introduces sometimes slight, sometimes considerable changes in the requirements. In the case of the NA355 a conscious decision was made to develop the turbocharger in the field. This was considered where the problems would be encountered, and where the solutions would have to be found. The amount of rig testing to determine the performance or to understand a mechanical feature has been considerable. However, if the philosophy of the NA355 had been different then the life of this turbocharger would have been far more traumatic.

10.6 **Conspectus**

The stated aims of the thesis are to report upon the strategic review process, the technical choices and the detailed design, testing and early field experience of a new large turbocharger.

It is considered that these aims have been achieved. Additionally, given the advantage of hindsight this retrospective reviewing process has allowed valuable analysis of the various aspects that comprise the introduction of a new product. This has allowed identification of both the less well executed aspects and also acknowledgement and recognition of the well planned and successful operations.