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TOTAL TECHNOLOGY PhD THESIS

CHRISTOPHER J ROBINSON

End-Wall Flows and Blading Design For Axial Flow Compressors

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Supervisor: Dr A B McKenzie

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ABSTRACT

The flow in multistage axial flow compressors is particularly complex in nature because of the proximity of moving bladerows, the growth of end-wall boundary layers and the presence of tip and seal leakages and secondary flow. The problems associated with these phenomena are at their most acute in the latter, subsonic stages of the core compressor, where Reynolds numbers are modest and the blading has low aspect ratio. Indeed, much of the inefficiency of axial stages is believed to be associated with the interaction between blading and end-wall flows.

The fact that the end-wall flow phenomena result in conditions local to the blade which are quite different from those over the major part of the annulus was appreciated by many of the earliest workers in the axial turbomachinery field. However, experiments on blading designs aimed specifically at attacking the end-loss have been sparse.

This thesis includes results from tests of conventional and end-bent blading in a four-stage, low-speed, axial compressor, built specifically for the task, at a scale where high spatial measurement resolution could be readily achieved within the flowpath. Two basic design styles are considered: a zero $\alpha 0$ stage with DCA aerofoils and a low-reaction controlled-diffusion design with cantilevered stators.

The data gives insight into the flow phenomena present in 'buried' stages and has resulted in a much clearer understanding of the behaviour of end-bent blading. A 3D Navier-Stokes solver was calibrated on the two low-reaction stators and was found to give good agreement with most aspects of the experimental results. An improved design procedure is suggested based on the incorporation of end-bends into the throughflow and iterative use of the 3D Navier-Stokes solver.

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5 BLADING DETAILED DESIGN

6 LOW-SPEED COMPRESSOR RIG AND INSTRUMENTATION

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NOMENCLATURE

AE	Advanced Engineering
AR	Aspect Ratio
CD	Controlled Diffusion
Cp	Specific Internal Enthalpy
CPA	Critical Path Analysis
DCA	Double Circular Arc
DOC	Direct Operating Cost
EIC	Engineer In Charge
FACE	Fan And Compressor Engineering CAD Suite
GE	General Electric Aircraft Engine Group
H	Enthalpy
Hx	Boundary Layer Shape Factor
HP	High Pressure
h/t	Hub/Tip ratio
IEB	Inviscid End-Bends
IGV	Inlet Guide Vane
IVP	Inverted Velocity Profile
L	Length Scale
L-R	Low-Reaction
$\dot{M}$	Mass Flowrate
MEFP	Moore Elliptic Flow Program
MIT	Massachusetts Institute of Technology
MTOW	Maximum Take-Off Weight
N	Rotational speed
NOB	Number of Blades
OGV	Outlet Guide Vane
P, p	Total, static pressure
Δp	Bellmouth Inlet Depression
ΔP	Mean Stage Pressure Rise
PV	Private Venture
P&W	United Technologies Pratt and Whitney

r	Radius
RAE	Royal Aerospace Establishment
RM	Research Manager
SFC	Specific Fuel Consumption
s/c	Space/Chord ratio
T, t	Total, static tmperature
tc/h	Tip Clearance/blade height
U	Blade speed
$\underline{u}$	Velocity vector
Va, Vx	Axial Velocity
vx	Axial Velocity in the Boundary Layer
$V\Theta$	Whirl Velocity
$v\theta$	Whirl Velocity in the Boundary Layer
VAP	Variable Axial Profile
VWD	Variable Work Done
α	Gas Angle
α'	Blade Angle, Gas Angle in the Boundary Layer
ε	Gas Deflection
ρ	Density
μ	Viscosity
η	Efficiency
τ	Torque
Θ	Camber
Θx	Axial Momentum Thickness
ζ	Vorticity, Stagger

1. INTRODUCTION

1.1 General Background

The cost of fuel burned by the engines typically accounts for 20% of aircraft Direct Operating Costs in the medium and long range market segments. Competitive fuel burn is an essential ingredient for successful marketing campaigns, leading ultimately to profitable engine programmes. The thermodynamic cycles for large civil aero engines, such as the Rolls-Royce RB211 series and the International Aero Engine consortium V2500 (fig 1.1) are optimised to minimise fuel consumption with achievable levels of component efficiency, compressor pressure ratio and turbine entry temperature. As turbomachinery researchers find ways of reducing losses, it becomes worthwhile for engine designers to aim for higher cycle pressure ratios, as illustrated by fig 1.2. This presents compressor design engineers with the dual requirements for high shaft pressure ratios and high compression efficiency, while maintaining adequate surge margin for safe, transient operation.

As the cycle peak pressure and temperature are raised, the optimum bypass ratio increases, so the core engine size for a given thrust reduces and deleterious viscous flow phenomena make the achievement of high efficiency more difficult, particularly in the latter stages of high pressure core compressors.

Over the 50 year history of the aero gas turbine, the most significant progress in compressor technology has been in achieving well matched designs, more nearly 'right first time', and in reducing the levels of aerofoil profile loss, such that today's industry standard machines have polytropic efficiencies of around 90%. Much of the outstanding inefficiency that is potentially reducible is believed to be associated with interaction between blades and near end-wall flow phenomena.

Though this has long been recognised as a problem area, there has been relatively little published work to guide designers, and that which exists is not systematic and often contradictory.

For example, Rolls-Royce has demonstrated improvements in efficiency of the order of 1% (so a 10% reduction in losses) together with increased surge margin in tests of one of its core compressors by improving the radial matching of rotor and stator blade geometries to annulus wall boundary layer velocities. This involves near-wall camber and stagger modifications known as 'end-bends'. (Freeman and Dawson, 1983). However, experience has revealed that these modifications can not be relied upon to improve performance, highlighting a need for further research.

Compressor performance levels are normally demonstrated using high-speed, multistage rigs which are expensive to manufacture and test and not well suited to detailed surveys of the internal flow because of their small physical size. Cost effective research, which is aimed at establishing and understanding the important fundamental flow phenomena, can be carried out on simpler vehicles, such as the large-scale, low speed axial flow 4-stage compressor rig (LSRC) which is installed at the Cranfield Institute of Technology.

The work reported in this thesis is an element of an ongoing programme on this facility, which includes tests of end-bent blading compared directly with conventionally designed stages. This is aimed at increasing understanding of the flow/geometry interaction with the overall objectives of maximising end-bend benefits and improving design techniques.

1.2 Outline of the Thesis

Chapter 2 represents a cost-benefit analysis for the LSRC programme, firstly explaining the relevance of advances in compressor technology to the performance and marketing of aero-engines. The concept of managed risk, 'evolution of technology' is discussed, whereby advances in compressor technology are progressed through fundamental research on simple rigs, then verification in high-speed machines which are more representative of the engine environment. Finally, performance and mechanical integrity are demonstrated in spool or engine tests prior to ultimately specifying hardware for live, commercial engine projects. The role of the LSRC in this process is described and the technical and financial pros and cons of research on this facility are compared with other means available to Advanced Engineering Research Management.

This has been a Total Technology Programme and the author's learning experience from involvement in the management process is documented in Chapter 3.

Chapter 4 reviews the literature which has most direct relevance to blading design in the end-wall region. Firstly, a brief description is given of the important 3D flow phenomena occurring in embedded stages of axial compressors. Most of the published material relevant to compressor technology concentrates either on describing conditions through conventionally designed machines, or on the achievement of well matched designs. Not much has been published on blading designed to reduce near-wall loss and the material reviewed in section 4.4 is believed to be comprehensive and not previously collated.

The blading design for the LSRC described in Chapter 5 covers two styles; the first is representative of a moderately loaded mid-stage of the high pressure (HP) core compressor from the three shaft, RB211 series of engines and uses DCA aerofoils with a conventional 'correlation' approach to the selection of profiles.

The second is based on a highly loaded mid-stage of an early version of the V2500 HP core compressor, featuring controlled diffusion aerofoils. The low-speed blading was tailored in such a way that the diffusion experienced by the aerofoil boundary layers would be equivalent to that experienced by the high-speed counterparts. Though this design philosophy is known to be used by other researchers in the USA (Wisler, 1984) its detail has not been published and the development of this non-standard approach within the Rolls-Royce computer aided design system was the work of the author.

Chapter 6 describes details of the LSRC and its instrumentation. The basic rig and facility were procured by CIT prior to the start of the work reported here. However, the author was closely involved in the aerodynamic commissioning of the rig, taking a leading role in the development of a suitable pneumatic traverse instrument. The second version of the LSRC, to suit the low-reaction V2500 designs, was completely specified by the author in terms of scantlings and instrumentation, benefiting from the initial experience of running the RB211 style standard.

Most of the aerodynamic data was recorded by staff at CIT under the direction of the author, who was responsible for all data analysis downstream of the basic raw data state. This included the successful application of an existing axisymmetric throughflow program to the reduction of area traverse data.

These data, presented in Chapter 7, are certainly unique in their description of severe end-bend geometries and provide a comprehensive description of the flow through embedded core compressor stages, which is not available from any other facility in the UK.

Chapter 8 reports the application of a 3D Navier-Stokes (N-S) solver to the conventional and end-bent stators from the two low-reaction builds, with the objective of assessing the code from the point of view of potential 'effectiveness' in the compressor aerodynamic design process. This was the first application of a new curvilinear mesh generation program to a compressor example and also the first use of this N-S solver in the core compressor environment at Rolls-Royce. Calculation results of exit whirl angles, losses and surface static pressures are compared with experiment and then the MEFP results are post-processed to reveal further details of the flow not normally visible to the experimentor.

This programme of work has resulted in a much clearer understanding of the behaviour of end-bent blading in the buried stage environment and calculations with a throughflow code in Chapter 7, and MEFP in Chapter 8, have paved the way towards improved design techniques.

Each chapter is concluded with its own discussion. The most important conclusions are drawn together in Chapter 9, while Chapter 10 presents suggestions for the continuation of this research.

2. COST - BENEFIT ANALYSIS

2.1 Introduction

A formal cost-benefit analysis is not explicitly performed for compressor research projects such as the LSRC programme. A Brochure contains estimates of the costs of the proposed package of work and sets down the Technical Objectives and Technical Benefits, with the limited objective of assisting well-informed managers within the Company and at RAE to make comparative, subjective judgements.

While the costs are relatively easy to quantify in principle, the benefits are rather less tangible. There already exists a level of compressor technology within Rolls-Royce which is, at this point in time, competitive with that offered by other engine manufacturers. Improvements to this level of technology will be accrued from the LSRC programme: for these to be worthwhile, they must ultimately contribute to increased profits for the Company, either by reducing the costs associated with engine project development or by supporting the continuation and expansion of Rolls-Royce's share in the growing world market for aero engines.

The first half of this chapter explains the relevance of compressor technology to the aero engine business, highlighting its importance in the engine layout and thermodynamic cycle and its influence on the costs of operating an aircraft. This discussion suggests that advances in compressor technology are worth pursuing, despite the fact that it is a maturing science.

The Company must not be tempted to overreach technology, as project cost and time overruns are then almost inevitable.

It has to evolve research strategies which keep important technical milestones within reach so that eventual implementation on project engines is low risk. The second part of this chapter describes the means of advancing compressor technology, focusing on the role of the LSRC.

It will become clear that a cost-benefit analysis for the present project is somewhat subjective. It simplifies to:

i) Advances in compressor technology will continue to play an important role in the marketing of aero-engines.

ii) The LSRC programme is a cost-effective means of advancing compressor technology as part of a structured technology acquisition and demonstration process. Some of the data and understanding which it can provide are not available to Rolls-Royce from any other source.

iii) Taking a realistic view, there is a strong probability that lessons learnt directly from the LSRC will occasionally save builds in a high speed compressor development programme, as well as contributing to a shorter development path in the future as part of the Core Research Strategy. A saving of <u>one</u> high-speed, multistage compressor re-blade and test would pay for the entire LSRC programme to date.

2.2 World Aerospace Market 1988 to 2010

The total annual aerospace market in the western world is expected to be worth $2080 Billion (Yates, 1987) of which civil and military aircraft account for approximately 75%. While the military market is static, growing at less than 1% per annum, civil aviation is expected to grow by 5% per annum, doubling by the year 2000.

This expansion is driven by increasing wealth and willingness to travel throughout the world. Fig 2.1 shows a rough breakdown into market segments: the long range and twin engined (medium) range segments are expected to be worth $550 billion. Engines in these segments account for roughly 30% of the cost of the aeroplane, so the large thrust aero engine market should be worth around $180 billion. Rolls-Royce currently holds 13% of the market for the larger thrust aero engines, behind Pratt and Whitney (18.7%) and General Electric/SNECMA (68.3%). This is similar in proportion to Britain's share in the world aerospace market (15%).

The 50 year history of commercial air transport has shown a strong trend of falling numbers of major companies taking part and in the number of new aircraft programmes beginning. There are only three major manufacturers producing large transport aeroplanes since the withdrawal of Lockheed with the L-1011 Tristar: Airbus Industrie (the European Consortium), Boeing and McDonnel-Douglas (MDC).

The number of engine manufacturers has similarly declined such that the large civil business has only three major manufacturers. There are, however, a number of smaller companies participating on the periphery such as SNECMA (who have a significant share of the market in collaboration with GE on the CFM56 and CF6 engines) Turbomeca (France), Allison and Garrett (United States) MTU (Germany) and JAEC (the Japanese Aero Engine Consortium). These companies are not currently independent competitors in the large, civil engine business but are available as collaborators or 'risk sharers' on future projects.

2.3 Relevance of compressor technology to the Aero Engine Business

The principal business of the Company is to profit from the 'design, development, manufacture, sale and in-service support of gas turbine engines and ancillary equipment' (Company accounts, 1987). Minimising the cost of these activities allows the Company to offer engines at competitive price while maintaining profitability. In view of the fact that compressor technology is perceived to be mature, it is relevant to consider the importance of further advances, likely to be expensive to attain, in the future marketing of Rolls-Royce engines.

2.3.1 Customer Needs

In the marketing of aero-engines, as with any other product, it is important to understand the needs of the customers. The airline executives are the actual buyers of engines, but the concept of the 'customer' extends to the aircraft manufacturer, the fare paying passengers, the flight and maintenance crews and the people who share the environment around airports.

The air-travel business has matured and the airline executives are not primarily interested in high technology for its own sake: they have focused on the need to operate their fleets of aircraft at a profit. Simply, the earning power of a commercial aircraft is judged by its Direct Operating Costs (DOC) which comprise its price, the maintenance costs, the cost of the fuel which it burns and the cost of providing the crew. The civil aviation market is highly competitive and any advance in technology that can reduce operating costs will be taken seriously.

The aircraft manufacturers are sophisticated arbitrators of the claims made by the engine companies for advances in performance and maintainability of their products. They need to be convinced that the engines are technically viable and that their delivery will be compatible with their own programmes for airworthiness certification and aircraft delivery.

The market for aircraft can be categorised into long, medium and short haul aeroplanes. In the long and medium range segments, the aircraft manufacturer generally offers at least two choices of engine from the three major manufacturers promoting competition, while in the short range segment an aeroplane is often sold with one engine model only. Clearly it is important to have a good relationship with the aircraft manufacturer in the short range segment and if possible establish a 'niche' market. In the other segments, it is highly desirable to be the 'launch engine', the one with which the aircraft is initially certified, as many subsequent airline buyers tend to follow the lead of the launch customer. The second engine onto an airframe may require a new installation which must be certified as airworthy with an expensive flight test programme, at cost to the engine manufacturer.

The needs of the fare paying passengers are closely aligned with those of the airlines, being keen to have low cost transport (low ticket costs derived from low operating costs) but with special emphasis on safety, reliability and comfort. Aircraft accidents, though rare, are often major tragedies involving large loss of life and 'public concern' is quickly generated if a particular product seems to be accident prone. Reliability, as opposed to mechanical integrity, is also relevant to the passengers: there is little redundancy in the fleets of airlines and unscheduled maintenance tends to result in delays and inconvenience to passengers.

2.3.2 Structure of Airline Costs

It is normal practice for airlines to divide their accounts into operating and non-operating items, with the aim of separating both costs and revenues which are not directly associated with the airlines own air services. The non-operating items would include such things as gains and losses from retirement of property or equipment (for example, if an aeroplane is sold off for other than its depreciated book value) business of affiliated or subsidiary companies, foreign exchange transactions and government subsidies. On the operating side, the accounts are further subdivided into direct and indirect operating costs. The aforementioned direct operating costs include all the costs that are associated with and dependent on the type of aircraft being used. This includes all flying expenses, maintenance and overhaul costs and aircraft depreciation charges. Indirect operating costs are those which are independent of the aircraft type and these tend to be passenger related (such as passenger service costs and ticketing and sales costs) and general airline administration charges. The split between direct and indirect operating costs is typically 60% to 40% respectively.

2.3.3 Direct Operating Costs

The engines influence all three elements of the direct operating costs. Fig 2.2 shows a breakdown of DOC for the IAE V2500 engine (in which Rolls-Royce has a 30% share) in the Airbus A320 and the RB211-524 engine in the Boeing 747. These two engine/airframe combinations operate in different segments, the A320 being medium range while the B747 operates over a much longer stage length. The major difference between the two segments is the split between the cost of operations and the cost of fuel which are both strong functions of stage length.

The B747 spends much of its typical 400 minutes 'block' (flight) time at cruise and consequently spends more on fuel than landing and ground handling fees. The converse is true for the A320 which has a typical flight time of only 100 minutes.

An obvious impact of compressor technology is through its contribution towards fuel consumption. The impact of SFC in terms of money saved through burning less fuel is actually modest as the price of fuel today is relatively low: in real terms it is at a similar level to that of the mid-1970s, though fig 2.3 shows a temporary doubling of price in the interim years. For the V2500 and RB211 examples shown in fig 2.2, a change in compressor efficiency of 1% would translate to 1% and 0.4% of SFC respectively, which is worth $12000 and $33500 per aircraft per annum. Across a whole fleet of aeroplanes this can become a substantial contribution towards profit, or alternatively a cost to Rolls-Royce in fuel consumption guarantees, should the efficiency be less than offered.

2.3.4 Potential for Future Fuel Consumption Improvement

The modest savings in operating cost per annum estimated above are 'marginal' in the sense that they are perturbations about an existing engine and cycle. The benefits can be much greater if an increase in compressor efficiency can be accounted at the design stage. If the engine burns less fuel, then the mass of the aeroplane (including fuel for the mission) will be lower for a given payload, so the thrust requirement will reduce and the engines can be smaller, hence lighter, and so on. In addition, the optimum cycle pressure ratio may be raised as a direct result of higher available compressor efficiency.

The Specific Fuel Consumption (SFC) is an industry standard measure of the overall efficiency of an engine at a given flight speed (SFC is inversely proportional to overall efficiency). It is defined as the fuel flow rate divided by the net thrust and is often quoted at a cruise condition of 0.8 Mach number at an altitude of 35,000 ft on a 'standard' day (where ambient pressure and temperature at sea level are 14.7 psia and 288K respectively).

At cruise, the net thrust of the engines must balance the drag of the aeroplane for steady flight at constant speed. The drag force multiplied by aircraft speed represents an expenditure of energy which can be considered as the ultimate in SFC. At cruise, this is typically 0.2 lb/lb/hr.

The overall efficiency of the engine is the product of two components, the thermal efficiency, which is the rate at which kinetic energy is added to the air passing through the engine as a proportion of the rate of energy supply from the fuel, and the propulsive efficiency, which expresses the extent to which this added kinetic energy appears as useful work to propel the aircraft.

As pressure ratio increases, compressor delivery air temperature increases and there is a practical limit imposed by the capacity of compressor and turbine rotor materials to withstand high temperatures. This maximum pressure ratio limit of about 80:1 raises the irreducible SFC to 0.27 lb/lb/hr. To achieve SFCs approaching this value the engine designer has to influence both thermal and propulsive efficiencies.

There has been considerable improvement in component efficiencies since the jet engines built by the early pioneers first flew during World War II. The early engines must have had component efficiencies of around 70% to be self-sustaining and capable of producing thrust.

Typical compressor polytropic efficiencies today are around 88 to 90%, so their losses have been reduced by some 60% over the past 40 years. Fig 2.4 shows the historical development in SFC through improvements in the thermal and propulsive efficiencies. Whittle's W1 engine in the E28/39 prototype fighter aircraft had an overall efficiency of only 10%, comprising 20% thermal efficiency and 50% propulsive efficiency. By the time the jet propulsion concept had been developed into the Ghost engines for the de Havilland DH106 Comet 1, the thermal efficiency had improved to 35% while the propulsive efficiency was unchanged resulting in an overall efficiency of 17%, and SFC of 1.15 lb/lb/hr at cruise. The last of the simple subsonic turbojets produced by Rolls-Royce, the Avon for the Comet 4, had an incremental rise in thermal efficiency as compressor designers advanced the pressure ratio and improved materials allowed higher turbine entry temperatures. The trend of small increments in thermal efficiency with time has been maintained over the years, and step changes in performance have come through advances in propulsive efficiency with 'bypass' engines.

The thrust produced by an engine depends on the difference between the forward speed of the aircraft and the jet exhaust velocity, multiplied by the mass of air passing through the engine; high propulsive efficiency requires the differential of these velocities to be minimised as the kinetic energy expended is proportional to the square of jet velocity.

Bypass engines move a large mass of air through the engine with a small exhaust velocity. The first generation of bypass engines, typified by the Spey, passed about half of the ingested air through the core compressor, combustor and turbine, exhausting with a moderate jet velocity from the LP turbine and the remaining air passed around the core, leaving with a low jet velocity resulting in a mean exhaust velocity lower than that of a pure jet.

The Spey increased the propulsive efficiency to 60%, reducing SFC to 0.82 lb/lb/hr. The latest generation turbofans, such as the RB211 and V2500 have bypass ratios just above 5:1 and 75% propulsive efficiencies. The thermal efficiencies of these two engines are similar, with the RB211 slightly higher reflecting a cycle optimised to give minimum SFC at cruise of 0.61 lb/lb/hr. This figure is about half way towards the ultimate value of 0.28 lb/lb/hr from the baseline of the Avon in 1958.

Further advances in propulsive efficiency are achievable: propellors have very high propulsive efficiency at low aircraft speed. In fact, it has taken 30 years of expensive turbofan and airframe development to achieve the same standard of fuel economy as the Dart turboprop powered Vickers Viscount on 500 NM stages. The contemporary jet powered aircraft burnt double the amount of fuel, which was the price paid for speed and comfort through reduced cabin noise and vibration levels.

The propulsion concepts offered in the medium and long range market segments may well be different in the future. In the medium range segment, the use of advanced propellors offers the prospect of a step change in fuel consumption without a fundamental need for improvements in the thermal efficiency. The thrust requirement for these aircraft (typically seating 150 people) permits the use of two engines which can be mounted in the tail, easing the cabin noise problem inherent with propellors. The propfan appears to be a good concept for the future, but it will only be commercially attractive once the price of fuel rises such that its improved fuel economy offsets the expense of buying the new aircraft and engines.

In the long range sector where fuel burn is more important, the trend is to have larger aeroplanes, which have proportionally less drag, to maximise the payload per unit weight of airframe and to offset the penalties of aircrew costs and traffic density.

Typical thrusts required for these large four-engined aircraft are around 60,000 lbs, and a high cruise Mach number is required in order to reduce journey times making the propfan uncompetitive.

SFC can be improved for the turbofan by increasing bypass ratio, but the exchange rate is very small when the additional drag and weight of the larger fan cowl is taken into account. Also, by definition the core becomes physically small making it more difficult to maintain high component efficiencies, so thermal efficiency would suffer in the limit and there may be no nett benefit. A much better prospect for the turbofan is an improvement in thermal efficiency brought about by further component loss reduction at more moderate levels of bypass ratio.

Fig 2.5 shows the trade-off between thermal efficiency at cruise and component efficiency at varying levels of turbine entry temperature. At a turbine entry temperature of 1400K, a 4% component efficiency improvement (which is a 30% reduction in losses from today's in-service levels) would be worth 15% of SFC. The technology advances in component design would read across to the smaller cores needed for propfans but the potential for efficiency improvement is likely to be diminished by small scale effects. It should be noted that the improvements must be across all turbomachinery components to achieve this level of improvement.

Furthermore, the jet engine is a complex product which has components carefully matched to achieve high performance. Failure of individual components in meeting their design objectives has a high gearing effect on overall engine performance.

2.3.5 Factors Influencing Choice of Engines

The direct operating cost is the principal, tangible driver affecting the choice of a particular engine/airframe combination. The important, external influences on DOC are the price of fuel, the interest rate on borrowed capital and the general rate of inflation. Airline operators buy new aircraft both to expand their fleets to take up an increased demand for air travel, and to replace older equipment which has become too expensive, either in terms of fuel burn or maintenance.

The history of fuel price, shown in fig 2.3, shows the 1988 price to be around 65 cents per gallon. At this level there is little incentive to buy new. Fig 2.6 shows a breakdown between direct operating costs of a 1960's technology Boeing 727-200 with P&W JT8D low bypass ratio turbofans (similar to the Spey) and a projected 1990's technology Boeing 7J7 with GE UDF propfans. The B7J7 is only slightly cheaper on flight crew, despite having only two compared to the three on the B727, as they would command higher salaries for flying the high technology aeroplane. The big differences appear between cost of ownership and the price of fuel burnt. The B7J7 would be a brand new aircraft requiring large loans with considerable interest payments and heavy depreciation, while the B727 could be bought cheaply second hand having been written-down over a number of years. The fuel burn is substantially reduced by 62% for the high-tech B737, 50% coming from the more fuel efficient engines and the remaining 12% by reducing the drag of the airframe. However, the nett result is that the B7J7 is 11% more expensive to operate with fuel at 65c per gallon and interest at 7.5%. With such high interest rates, the fuel price would have to exceed about 120c per gallon to make the high technology B7J7 more attractive.

While DOC is probably the principal influence on choice of engines, there are other newer issues which are taken into account. Government legislation on noise and emissions precludes the use of some older aircraft at certain airports and could be the driver for high technology engines in the face of low fuel prices.

Larger airlines may have their own maintenance and overhaul facilities and the choice of engine may well be influenced by a desire for fleet commonality. Smaller airlines use other agencies to maintain their aircraft and this will be much less of an issue.

Finally, airlines are keen on 'track record' achievements of reliability and performance retention. In addition, they will seek reassurance on purchasing a particular manufacturer's engines that the Company has the financial backing and technical capability of providing in-service performance improvement packages to help keep operating expenses competitive. Advanced Engineering research has an important role in demonstrating the Company's commitment towards keeping its technology base competitive.

2.3.6 The Consequences of Technology Shortfall

An established product with a proven track record of reliability can continue to sell with a quite conventional level of technology. Financial adjustments in the sales contract can overcome obsolescent or deficient technology.

The relationship between technology and sales can be illustrated by a simple 'S' curve in fig 2.7.

As time passes, the market expects a higher level of technology and the curve moves to the right and a manufacturer failing to reach the plateau can expect to lost market share.

This phenomena is analogous to a failure to innovate, described by Foster (1987). An appropriate example concerned the Rolls-Royce Dart turboprop which, for many years, was the sole engine offered in the short-range Fokker F27. The engine had only minor development since its launch in the 1940's and it was displaced from its position in the F27 by a high technology competitor from Pratt and Whitney. The PW120 used 30% less fuel, through effective application of technology developed on other engines, resulting in a nett 5% reduction in DOC.

Having optimised an engine cycle at the design stage, a minor shortfall in component performance is not too serious - in fact most engines enter service below specification and are retrospectively refined over the first year or so of operation by parts replacement. If the compressor is below its target efficiency, the turbine may have to run hotter to achieve the specified thrust. This reduces the life of the turbine, a cost to be borne by the engine manufacturer, but it is tolerable. It will, however, affect the productivity of the aeroplane as more fuel will have to be carried and burnt, again financed by the engine manufacturer under fuel-burn guarantees within the purchase contract. The aircraft's range performance is illustrated in payload-range diagrams, shown schematically in fig 2.8. Over short range, the payload capacity is effectively limited by the volume of the aircraft, the number of seats and the space available for cargo. As range increases, the aeroplane has to carry progressively more fuel until the total maximum weight of aircraft structure, fuel and payload reaches that which cannot be exceeded for safety, known as the Maximum Take-Off Weight (MTOW). To fly beyond the range at MTOW, the aircraft must sacrifice payload for fuel and the range is 'weight limited'.

The second cut-off occurs when the fuel tanks are full and is the effective maximum range of the aircraft, although the plane can fly further by reducing payload since a lighter aircraft consumes fuel at a lower rate. The secondary lines on fig 2.8 show the effect of a 1% worsening of fuel consumption which would be caused by a 1% shortfall in compressor efficiency.

If the compressor was substantially short of efficiency (or pressure ratio) the whole engine concept can be undermined if it is no longer practicable to run the engine hotter to achieve the required thrust while the compressor is developed. The balance of the engine matching is upset and necessary changes made to the engine cycle require expensive changes to structural components of the engine, such as those which would be caused by an increase in core compressor discharge by reducing pressure ratio.

Core problems are not unusual in engine development, but Rolls-Royce has endured particularly bad press in two cases: the RB211-22B for the Lockheed L-1011 Tristar in 1971, and the V2500 for the A320 in 1987.

The early V2500 High Pressure compressor was some 2-3% short of efficiency which again caused problems in terms of turbine life to achieve the specified thrust. The core flow was increased by 8% necessitating a change from a two stage to a three stage booster behind the fan. The HP compressor annulus was not dramatically changed, as the compressor was re-matched on a lower working line (lower pressure ratio, higher mass flow) but the entire HP compressor was replaced eventually by a re-designed machine towards the end of the 18 tests in the development programme.

The V2500 did suffer mechanical problems too during its development phase, which were also widely reported in the press, but the major culprit was deemed to be the HP compressor (Aviation Week and Space Technology 14 Dec 1987 Vol 127 No. 4 pp 32-34). The hostile media can have an important influence on marketing. The V2500 example above was extensively reported in the US-based press fuelled, presumably, by information leaked from within the IAE consortium followed by the loss of the launch customer, Lufthansa.

Both the RB211 and V2500 problems were ones of overreaching demonstrated technology in a fiercely competitive and demanding market.

2.4 The Fundamental Problem

The basic problem, faced by all engine manufacturers, is that it may take longer to develop a new aircraft engine than a new aeroplane. Furthermore, a reduction in overall fuel consumption of the order of 25% is necessary to make a new aeroplane/engine configuration sufficiently attractive for it to be selected ahead of current, in-service products. This target generally implies a level of component performance which is certainly ahead of in-service standards and possibly even in advance of demonstrated technology. The motivation to take a commercial risk on the level of attainable technology is driven by the intense competition between manufacturers, who are striving to have the 'launch engine' on a new aircraft.

Just as competition between manufacturers ensures that the level of technology in the engine is high, the unit cost is kept low, minimising ownership and operating costs to the benefit of the airline customers and ultimately, to the fare paying passengers. The combination of short, challenging development programmes leads to high expenditure through the early years of a project's life-cycle followed by a slow recovery to break-even which is typical of the aerospace business. Fig 2.9 shows a typical life-cycle curve. The large, negative cash flow over the two or three years following the project launch are the result of expensive development as the engines are refined to achieve their promised levels of performance. Typical engine projects take between 10 and 15 years to break even, with a small profit margin on each unit sold. Much of the Company's profit is actually made on the sale of spares which are necessary to maintain performance and mechanical integrity over an engine's useful life.

2.5 Extent of Spend on Research and Development

In the Company accounts, spend on product development is bracketed with Research. In 1987, a total of £294m was spent on R&D which was approximately 14% of turnover. R&D as a proportion of turnover has remained at about this level since 1984, and the figure is believed to be comparable with that of the competitors. Spends greater than 10% of turnover are considered large even for high technology industries.

The bracketing of R&D can, however, hide swings in the balance between the two elements: the Company noted that an increase in gross R&D spend between 1986 and 1987 was the result of investment in the -524G and V2500 engine projects, hinting that spend on Research had not significantly changed over the two years.

The split between the Research and Development elements believed to be 25% to 75% on a trend of increasing the proportion spent on research from somewhat less than 10% in the mid 1970's to a target of approximately one-third by the early 1990's.

Taking the 1987 figures as representative for an order of magnitude calculation, engine development absorbed around £220m and research approximately £74m. These figures are not the total expenditure on research within the Company, however, as some of the work is contract-funded by external sources, such as the Department of Trade and Industry (DTI) and the Ministry of Defence (MoD). The DTI may also contribute some launch aid for new engine projects. The nett spend on R&D in 1987 was £187m which is equivalent to 9% of turnover or 54% of operating profit.

The £74m or so spent on research in Advanced Engineering represents the investment by the Company across technology which is necessary to keep products competitive and to reduce the high spend on development by improving the accuracy with which the product is designed. The challenge is to minimise the area beneath the product life cycle curve by well targeted research in advance of project launch, as shown schematically in fig 2.10. Ruffles (1986) estimated that 20% of development costs are irreducible, which represents the notional spend necessary to demonstrate that the product has achieved its design objectives for performance and that it is airworthy.

Apart from the obvious benefit to Rolls-Royce, there is a benefit to the customer of a smoother transition into service. Of the 75% of R&D spend on Development, only 40% is spent curing problems before entry into airline service: the balance is incurred while the engines are actually in use.

2.6 The Problem of Overreaching Technology

Analysis of engine development cost and timescale overruns in the 1970's showed that for projects which were launched on a well established technology base, the cost and (usually) the timescales were predictable, as shown in fig 2.11. However, where the projects overstepped previously demonstrated technology, both cost and timescale overruns were, apparently, inevitable (Ruffles, 1986). The crucial point is that the concept of the project must be 'right first time': configurational changes during the engine development process are particularly expensive to implement. Trying to recover from a non-competitive position of technology during this phase not only prolongs the project break even date but it can also jeopardise the liquidity of the Company as it did with the RB211 in 1971.

Demonstration of the necessary aerothermal and mechanical technologies in advance of project commitment could well be considered commercially mandatory: it is arguably too difficult to advance these technologies, necessary to achieve (and retain) performance, weight and time targets, while ensuring that a new design has the necessary mechanical integrity and durability.

2.7 Core Compressor Research Strategy

To avoid overreaching its technology, the Company requires both a competitive data base and a means of applying this technology accurately and consistently across its product range. Important milestones for future technology demonstration must then be identified. These are directly related to the Company's perception of future market needs and its product strategy.

A Compressor Research Strategy must then be evolved to keep these milestones within reach, which involves embarking on a continuum of technology starting with fundamental research on simplified rigs, such as the LSRC, moving into exploratory development on high-speed multistage rigs followed, for radically new mechanical configurations, by spool performance demonstration (i.e. compressor, combustor and turbine) prior to full engine demonstrator testing. Each stage in this process involves progressively greater expense and complexity.

The LSRC is an integral part of the Core Compressor Research Strategy, which has two parallel objectives:

i) to demonstrate levels of component performance which are appropriate for current and future project needs

ii) to incorporate existing compressor technology in to an accurate computer aided design system.

The first point is purposely generalised: compressors are selected to represent the best compromise between several conflicting requirements for efficiency, weight, length and cost. The criterion used to select compressor design specifications will be an overall engine/aircraft system figure of merit, such as direct operating cost for the civil market or thrust to weight ratio for the military, rather than a figure which relates to the compressor alone. The global challenges are to improve performance (pressure ratio per stage and efficiency, for example) and ruggedness simultaneously and to maintain or reduce weight and cost.

Reliable performance prediction methods are required to carry out the design optimisation process and an accurate design system is necessary to ensure that the development phase is minimised.

The advances in Computational Fluid Dynamics (CFD), which have been made possible by increased computing power and capacity per unit cost, make it possible now to solve the equations governing viscous fluid flow in 3-D for some cases of turbomachinery interest, but not for the multistage machine. The flow in core compressors is both viscous and unsteady in time, with complex secondary flows, such as overtip leakage. The equations may be exact for laminar flow, but empiricism is necessary to model the turbulent regime which is more prevalent in practice. The use of such viscous, 3D software for multistage turbomachinery design is some years away. Faced with the prospect of designing these machines today, the engineer uses a range of part empirical, part theoretical techniques. Although the basic principles are common throughout the Company (and the industry) the methods and their rules of application have evolved differently through time at different sites. The Core Strategy aims to incorporate the best of the methodology into a unified design system. This has the benefit of enhancing product assurance (quality) and permits fast implementation of new ideas. Also, it pools the vast experience available across the Company and avoids duplication of effort.

2.8 Advances in Compressor Technology

There are two basic ways of advancing technology

i) by a parametric test approach
ii) by improved understanding of the flow physics leading to development of better design tools

The first can be successful and it has been used extensively historically. However, it can be expensive and it has the potentially more serious drawback that read across to other applications may not be successful if there is a lack of understanding of the physical mechanism which lies behind an apparent improvement.

The second fits comfortably with the concept of evolution of technology. The flow phenomena are studied in a simplified environment, such as the LSRC. The design system is then critically assessed against experimental data and the improved understanding allows computational techniques and empiricism to be refined, enhancing the quality of the system. This is a much firmer basis for carrying a level of technology across to other designs.

The concept of targeting research effectively and reducing the number of tests which take place in the future, aiming for a high yield of information from each, is entirely compatible with the drive for value for money from R&D spend. However, compressor aerodynamics is a relatively mature technology. The 'S' curve, relating attainment in technology to cumulative resource applied, shown in fig 2.12, suggests that further increases will be progressively more difficult to attain.

Relating this to compressor technology, advances attributable to single compressor tests, or even projects, are few. The understanding which is necessary to enable the advance in technology emerges often from the accumulation of experience over a number of experiments. Historically these would be the development path of a particular project compressor. There is a need to replace this expensive, rather unpredictable, source of 'secondary' research data with testing on cheaper vehicles, and to expect to continue with a high level of investment to obtain progressively smaller returns. It is not necessarily realistic or desirable to reduce the number of tests per se: focus will shift onto more complex issues.

2.9 Experimental Compressor Research

There are several relevant approaches to compressor experimental research, each with technical pros and cons as well as financial arguments. It is difficult to establish the absolute costs of the various options due to differences in accounting conventions between sites within Rolls-Royce and between external agencies. However, Compressor Research Management have an annual budget allocated by Advanced Engineering, so a pragmatic comparison for the purposes of the present report is to compare the costs of the various options as charged to this budget. It is recognised that this does, to a degree, force a short-term horizon as there is a temptation to move work outside the Company to take advantage of lower overheads. So long as the internal resources are fully utilised on other projects, this is a reasonable approach. It does require careful monitoring by the Company to ensure that any resulting streamlining of internal experimental facilities is in the direction of improving the efficiency of the service, rather than losing the capability of doing such experimentation. A base level of turbomachinery research is certainly necessary to keep personnel and equipment fit to tackle tests without falling back down a learning curve and incurring time and cost wasting facility debugging.

The range of experimental facilities open to the compressor engineer offers a balance between faithful reproduction of real, multistage compressors and their limitations and usefulness as a means of demonstrating fundamental flows of interest in an environment where they can be recorded and understood more readily. Fig 2.13 illustrates this point qualitatively and makes an order or magnitude cost comparison between the four basic categories based on recent, Rolls-Royce experience.

Multistage machines are the most expensive of vehicles and tend to be geometry specific, targeted at a particular engine development. At the other extreme, cascades are a much simplified arrangement and tend to be relevant across many compressor designs, though they can be configured to be project specific. The LSRC, which combines the size and some of the simplifying techniques of cascades with the complex flow structure of a multistage machine, and HP9, a single or two stage 'slice' of a multistage compressor, are someway between these two extremes.

The three curves shown in fig 2.13 relate to: the 'first cost' of purchasing a new rig to run on an existing facility; the cost of manufacture, fitting and testing of subsequent new blading standards; the cost per annum of supporting a programme on the cheaper facilities. In general, it is meaningful to consider one or two tests per annum on an expensive, in-house facility, which does not fully occupy the resource all year. For the cascade and LSRC, however, the personnel tend to be dedicated to that facility and it is more meaningful to consider occupying them for a fixed period of time to do a series of tests. The LSRC figures correspond to four builds per annum.

2.9.1 Multistage, High Speed Rigs

Testing on high-speed, multistage research rigs is an obvious avenue for exploring the behaviour of engine compressors. Although multistage research rigs are occasionally built for this role, it is a rare occurrence in Rolls-Royce, mainly because of the expense involved. The cost of a new rig, including the first set of blading, would be approximately £4000K. Subsequent blading manufacture costs between £50K and £75K per stage, depending on the manufacturing techniques, and fitting and testing add in the region of £250K. These figures represent averages obtained from current (1988) research and engine development projects and are inflation adjusted.

Such rigs are normally used only for performance demonstration towards the end of the cycle of evolving technology. A parametric study would be prohibitively expensive, and multistage machines are not particularly suitable vehicles for research into the physics of turbomachinery flows. The blades are physically small with heights of typically 0.75 to 1.5 inches. Intrusive instrumentation, such a pneumatic traverse probe, has a resolution limit of approximately 0.040" so it is difficult to resolve flow phenomena which are on a scale of a fraction of annulus height. Furthermore, the probes are large compared with other characteristic dimensions, such as blade thickness and spacing, and their presence can influence the very phenomena which they seek to record.

The flow in the real turbomachine can be far from axisymmetric, due to basic limitations of the facility, such as bends and valving in the flowpath both up and downstream of the compressor and clearance variations as well as the result of having differing numbers of blades per row, so it is difficult to ensure that limited surveys over a fraction of the circumference are truly representative of the average flow conditions. While the problem of intrusive instrumentation can potentially be overcome by the laser anemometer, the flow asymmetry, which compromises interpretation of the data, is a more fundamental problem. A full scale multistage compressor absorbs a great deal of power (18000 HP for the V2500) which limits its operation to specially built facilities, not normally found outside the Company. Their running costs are high and they are usually fully utilised in project development so there is pressure to turn over tests in a short time, which is not conducive to basic research.

2.9.2 Single Stage, High Speed Rigs

The high costs associated with multistage, high-speed facilities are crudely proportional to the number of stages, and the cost of experiments are an order of magnitude smaller for a single stage machine (fig 2.13). The technical weakness of this approach is that it loses the detail of the structure of the flow entering a buried stage which reflects the history of its passage through earlier bladerows in the real, multistage machine. This can be partially overcome by the use of passive devices, such as inlet guide vanes and spoilers, for thickening the annulus wall boundary layers. In addition, the single stage high speed rig suffers from the same small-scale limitation on resolution of phenomena as the full multistage. However, faithful reproduction of flow Mach and Reynolds numbers, the factors governing the extent to which viscous and compressibility effects are important, make these rigs particularly suitable for research into aerofoil profile shape.

2.9.3 Low Speed Turbomachines

The flow in a turbomachine is governed by a set of non-dimensional aerodynamic and geometric design parameters, and its performance can also be expressed in terms of a set of non-dimensional parameters. It is axiomatic in fluid mechanics that machines which are scaled such that the aerodynamic and geometric non-dimensionals are preserved will have similar non-dimensional performance. Reynolds number and Mach number have already been mentioned above. The remaining compressor aerodynamic performance non-dimensionals, $\Delta H/U^2$, the stage loading, Va/U, the flow coefficient and the degree of reaction (the proportion of the stage static pressure rise which occurs in the rotor) define the primary flow through the stage.

The primary flow, together with the annulus wall boundary layers and geometric non-dimensionals, such as radius ratio, aspect ratio, space to chord ratio and tip clearance to blade height, govern the secondary flow phenomena. Depending on the focus of the experiment, the researcher may choose some compromise to non-dimensionals regarded as having second order influence, in order to simplify and cheapen the experiment.

It is on this premise that the LSRC programme is founded. Reynolds number (the ratio of inertia to viscous forces on the fluid) is regarded as particularly important in the behaviour of vortex flow phenomena, such as annulus wall boundary layers, passage secondary flows and parasitic phenomena such as overtip leakage. These are all significant factors in the performance of buried core compressor stages. A basic limitation noted against high-speed machines is their small size: large scale is desirable from the objective of accurate resolution of flow phenomena. Reynolds number is proportional to the product of length scale and velocity, so a large size implies low velocities, if Reynolds number is to be preserved, and hence low Mach numbers.

By careful profile design, all the non-dimensional aerodynamic parameters of a high-speed machine can be reproduced in the low speed model, with the exception of Mach number. Also the low stress level present in this class of rig hardware permits much simpler mechanical design.

The density and blockage changes are roughly in balance, so the annulus walls can be parallel and identical blading can be used in each stage without significant axial mismatching. One benefit of the low stress levels it that die-cast aluminium blading can be used rather than expensive 'machined from solid' or forged blading which is necessary in the high speed environment.

Four stages are used in the LSRC. The third is the one of primary interest; the first two set up the detailed flow structure likely to be present at entry to any buried stage through interaction between primary secondary and parasitic flows; the fourth is present to model the downstream static pressure field.

The power necessary to achieve the required Reynolds number is inversely proportional to the linear scale of the compressor. This helps keep down the running costs but more significantly, when combined with the low stress levels, makes it practicable to site the LSRC at CIT away from the Rolls-Royce in-house test facilities .

This offers a number of potential benefits for a programme aimed at careful research into fundamental phenomena. The pace of testing is not disrupted by pressure to release the facility and it is possible to take advantage of local instrumentation expertise. The overall cost of the programme to Advanced Engineering is lower through avoiding the overheads of in-house facilities.

The financial argument for low-speed testing is powerful when compared to the high-speed equivalent. The cost of a new rig is another order of magnitude smaller than a single stage, high-speed rig (so two orders of magnitude away from the multistage high speed machine) as shown in fig 2.13. The cost of Manufacture, Fit and Test (MFT) for new sets of blading is reduced by a factor of 3 or 4 from the high speed, single stage machine.

The size of the blading is such that they can be fitted with surface static pressure tappings on both rotors and stators cheaply and with little compromise to mechanical integrity. High resolution measurements can be obtained without the need for miniaturised probes or any worry about instrument blockage.

Setting aside the fact that the LSRC cannot reproduce shock-wave driven phenomena, the major technical limitation is that associated with accurate measurement of small levels of pressure relative to ambient conditions. Thus, the problem of spatial resolution in the high-speed, small machines is replaced by another difficulty which requires particularly careful experimental design and practice to overcome.

Some single-stage, low speed facilities exist, sharing the pros and cons of size and measurement resolution with the LSRC. By having fewer stages, the building of these rigs is mechanically more convenient than the LSRC. The lack of simulation of buried stage entry conditions simplifies the experimental scope, but does permit detailed research into the physics of turbomachinery related flows in an environment which may be computationally more convenient, providing data for CFD code validation. The single stage large scale, low speed rigs fall between the LSRC and linear cascades in the balance of applied research and fundamental understanding. They are particularly suitable for the academic environment.

2.9.4 Linear Cascades

The most basic form of experimental research which is relevant to compressor aerodynamics is conducted on idealised flows such as boundary layer development on plates with adverse pressure gradients or 2D flow through cascades of aerofoils. The experimental geometries are dictated by computational convenience and ease of manufacture and are generally found in academic establishments. The concept of testing linear cascades of compressor aerofoils was extensively utilised by the early workers in turbomachinery (summarised by Gostelow 1984) and their results still have limited application to compressor design.

Cascade testing is used today principally in two areas of compressor technology: for studies of a simplified representation, usually at low Mach number, of the secondary flows which are present in turbomachines, and for studies of advanced profile shapes - the logical extension of the original cascade work. Low speed cascade testing is particularly suitable for detailed measurement and flow visualisation of the simplified flows, leading to enhanced understanding of the basic flow physics. As the aerofoils are scaled up in size to achieve representative Reynolds numbers, it is possible to achieve very fine spatial resolution of the flow phenomena, and these data are used extensively in the validation of the 3D viscous flow calculation programs, albeit on simplified geometry and at low Mach number. The high Mach number tunnels tend to be smaller in scale, but still detailed measurements of the blade surface boundary layer development can be made. These results are used in the validation of quasi 3D, viscous/inviscid interaction blade-to-blade codes which are used extensively in compressor design as a replacement, ironically, for cascade data correlations of bladerow loss and turning.

Linear cascades are by far the lowest cost option (fig 2.13) and are now usually run by universities and industrial research establishments rather than by the engine manufacturers.

The disadvantage of cascades is that, fundamentally, they can never reproduce truly representative secondary flow phenomena as these are a function of radial gradients of the primary flow which cannot, by definition, be reproduced by a linear, 2D cascade. Similarly, the influences of Coriolis forces, due to rotation, cannot be present.

The inlet boundary layers entering cascades are naturally collateral, whereas in a turbomachine the effect of stepping between stationary and rotating frames of reference gives the boundary layers a different nature relative to the bladerows. The relative inlet velocities are fairly constant in magnitude through the boundary layers, but varying in direction. A collateral boundary layer has vectors which are reducing in magnitude, with constant direction. The 'skew' of real turbomachinery boundary layers can be simulated to some extent in cascades at a cost of a more complicated mechanical arrangement, and the same is true for the relative motion between blades with clearance and endwalls. However, since the most profitable area for progress is compressor aerodynamics is likely to be in the intelligent handling of endwall flows, 2D cascades have low potential for future gains.

2.10 Summary of Technical Cost-Benefits

Compressor technology has matured to a level where advances in performance are progressively more difficult to attain: the flattening of the yield against investment curve (fig 2.12) suggests that smaller increments in the future will be more expensive. However, combined with other advances such as increased cycle pressure ratio they are still important. In the present climate of maximising profit and competitiveness through minimising spend on R&D, management has to invest shrewdly across the range of relevant means of researching into compressor technology, to ensure a blend between insight into the fundamental physics and project-related performance demonstration.

The LSRC has an important technical role in the core compressor strategy:

i) it models the flow structure present in buried, core compressor stages with the unique benefits that the scale is sufficiently large to permit detailed spatial resolution of phenomena and the flow is axisymmetric. These factors much improve the prospect of successfully interpreting the detail of the results and rationalising traverse data with compressor overall performance.

ii) it is comparatively cheap, with a potentially rapid design, manufacture and test cycle, which permits a degree of parametric study and speculative testing of high risk designs which would not normally be undertaken on more expensive, high-speed facilities.

There are secondary spin-offs for CFD and instrumentation technology. Use of the data in code validation and through providing the insight into flow physics which underwrites technical development of CFD methods. The relatively 'clean' environment of the LSRC, both literally and aerodynamically, make it ideal for gaining understanding of flow phenomena/instrument interactions which can usefully read across to more complex high speed machines.

Assuming that the facility is at least capable of delivering these technical benefits, it offers value for money as a means of studying Mach number independent phenomena in the multistage environment. The cumulative cost of the programme to date is £700K, and individual builds are approximately £50K. The improved insight into compressor behaviour gleaned from the LSRC programme has only to save one complete reblade, build and test of a multistage machine to offset the complete cost of the programme to Rolls-Royce so far.

Alternatively, ten LSRC builds can be carried out for the price of one high-speed, multistage test, or three for the price of a single-stage high speed experiment. Had the LSRC programme been completed in advance of the design of the V2500 Batch 1 blading, it would certainly have resulted in a more appropriate design for the very first build of that multistage design, which would have resulted in a saving of at least one build out of the eighteen carried out during the development programme.

2.11 Summary of Marketing Implications

The projected, future market for large civil transport engines is substantial, not only for Rolls-Royce, to the benefit of shareholders, but also for the UK economy. The competitive nature of the business, evidenced by the falling numbers of companies involved in aerospace, is likely to intensify putting Rolls-Royce under significant commercial pressure from larger, US competition. There is an emerging trend for collaborative and risk-sharing projects, as the manufacturers perceive the cost of launching brand-new engines alone across all market segments to be too high. Rolls-Royce has to maintain and extend its technology base to be able to take advantage of the future marketing opportunities either alone, or as a leader in collaborative ventures, if it is not to be squeezed out of the profitable large civil transport sector of the market.

Compressor technology will continue to play an important, enabling role in improving engine thermal efficiency, particularly for the largest thrust engines for long range aeroplanes, where cruise fuel consumption has high gearing on direct operating cost. The current climate of relatively low fuel cost and high interest rates is delaying the next generation of engines technology, but compressor technology still plays an important role in ensuring accurate first design, leading to minimal development costs and ultimately low unit cost in evolution of todays turbofan engines.

This delay places particular emphasis on todays fundamental research and technology evolution programmes, such as the LSRC, which will allow Rolls-Royce to offer competitive engine concepts for the future without overreaching technology.

3. PROGRAMME MANAGEMENT AND FINANCIAL CONTROL

3.1 Introduction

The technical work presented in this thesis is drawn from the low-speed 4 stage, Research Compressor Programme (LSRC) which is in progress at the Cranfield Institute of Technology. It is an unusual project for Rolls-Royce, having a large proportion of effort subcontracted, while being of significant monetary value both in absolute terms and as a fraction of the Research budget available for Compressor Technology.

The range of experimental facilities available to the compressor engineer for research projects were discussed in section 2.9 and graded in terms of technical realism and cost (fig 2.13). The present programme was launched as 'low cost-low risk'. Though the facility has been technically successful by providing high quality data, which gives a clear insight into the behaviour of end-bent blading in the core compressor environment, the project has not been exemplary from the management point of view.

Some of the management problems encountered were the result of the Rolls-Royce Research Management procedures of the day and are consequently shared with other contemporary projects. Others are specific to this particular project and lessons learnt will be useful to others embarking on similar programmes.

This chapter summarises the 'management' aspects of the programme, highlighting things which could have been better, noting where procedures have been improved with the benefit of this experience. Aspects which went well are also mentioned.

3.2 Funding of Research at Rolls-Royce

Research is funded by Advanced Engineering (AE), within Corporate Engineering, as though it is a project. It has an annual budget to spend across all disciplines within engineering, to cover manpower and internal and external spend on manufacture, fitting and testing. Private Venture (PV) research is funded entirely by the company from profits or reserves. Alternatively, 50% or 85% of the cost of particular projects can be funded by the Department of Trade and Industry (DTI) or the Ministry of Defence (MoD) depending on the generality of the research for the former and the applicability to military projects for the latter. Occasionally, an engine project will fund specific applied research outside the AE budget. The LSRC programme has received support from all these sources over its duration. The funding and monitoring of projects is essentially the same in all cases.

A project, or package of work, is the subject of a 'Brochure', which amounts to a cost-benefit justification for the research, and a statement of technical objectives. It includes a milestone programme and an annual spend profile. Providing that the technical case is considered to be sound and the spend profile and internal manpower requirements dovetail into the annual budgets imposed on Advanced Engineering Research, the Brochure will be approved and the work initiated for PV funded projects. Where the funding is shared there is a final technical and financial audit by RAE Compressor specialists acting as agents for the DTI.

Throughout the part of the present project which is reported here, the practice was not to allocate internal manpower to specific brochures. A number of men were allocated to the Compressor Research Manager, leaving it to him to deploy them across the projects under his control without a formal mechanism to ensure that all ongoing research could be supported.

3.3 Management Control

Projects are reviewed at four levels by Management, using information supplied by the Engineer in Charge (EIC) and the Research Manager (RM).

i) Informal Review with the Research Manager

The EIC supplies the RM with a general update verbally on a weekly basis at a section or administration meeting, covering the technical status and programme progress. The RM is made aware of any likely future funding problems and the milestones of future tests are updated in the week prior to the next level review meeting.

ii) The Bi-Monthly Brochure Review

These meetings are largely administrative, with a strong financial interest. The progress of all current Compressor Technology brochures is reviewed in terms of achievement of milestones against spend to date, with the help of summary charts which are designed to make slippage through time visible. The need for issue of brochure addenda is discussed and the DTI representatives report back on the status of brochures which have been submitted, but are awaiting approval.

iii) The Quarterly Management Meeting

This meeting is the forum where technical issues are discussed in much greater detail, and brochure programmes are again reviewed. The future programmes are assessed against the background of a presentation of engine project status by the Corporate Head of Compressor Technology. The sum of compressor brochures amounts to a Research Strategy, and this strategy is reviewed against the background of project current, and anticipated, requirements.

iv) Annual Review

The annual review covers similar ground to the quarterly meeting, but it collates all compressor related activity throughout the company and reviews this against market and project needs, and competitors activities. A review document is produced which reaches a wider circulation of Senior Company Management, the RAE, DTI and the Armed Services.

3.4 The Role of the Author

The author, as designated Engineer in Charge (EIC) for most of the programme, had firstly to sell the technical objectives to both Rolls-Royce Research Management and the representatives of the DTI in the form of the 'Brochure'. Then, as the programme progressed, spend and technical milestones had to be monitored against this document, providing Management with information for the review process. The EIC has to account for cost and programme overruns in re-issues of the brochures (addenda) which re-sell the remainder of the programme to the sponsors.

This has been a collaborative project with the aerodynamic design and the analysis of data being done by the author in Derby, while the manufacture, fit and test activities have mostly taken place at Cranfield. The author had the responsibility of providing the interface between the two sites. The split-site nature of the programme, and the fact that the DTI has been involved in the funding, placed a strong emphasis on communication.

3.5 The Brochures

The work included in this thesis was funded by two shared brochures and one private venture. B1C2/110D included all the DCA zero $\alpha 0$ testing and effectively paid for the commissioning of the facility.

When the project driven move to the low reaction compressor came in 1985, it was initially funded by the V2500 project directly. In early 1986, Management decided that it would be funded by Advanced Engineering under a new shared brochure. The by then retrospective element (hardware changes to the rig and blading which had already been purchased) was not acceptable to DTI and MoD, so that was funded entirely Private Venture (PV). The subsequent testing of the low reaction datum and end-bent blading was funded under B1C2/118D.

3.6 Initial Phase

Discussions commenced between CIT and Rolls-Royce early in 1981 which resulted in an eventual, successful submission to SERC, who provided the funding to establish the basic facility. These initial negotiations are particularly relevant as they set out the likely rate of testing which would be achievable and the levels of manpower necessary on-site at CIT for servicing the rig and analysing the data. These estimates turned out to be hopelessly optimistic, possibly the result of an inadequate specification from Rolls-Royce.

The initial Brochure submission was prepared with a proposal from CIT (Hetherington, 1981)) which estimated that the datum RB211-style build (DCA, zero $\alpha 0$) and six sets of end-bend blading could be tested absorbing half of the available facility time over the first two years of operation. Allowing for a two month commissioning period, it was accepted by CIT and Rolls-Royce that the build-test-strip cycle could be achieved within four weeks. With this information, the first network for the project was drawn, fig 3.1, and the blading manufacture of the inviscid end-bends was identified as being on the critical path.

In fact, once the rig was delivered in December 1982 and installed in the facility, it was beset by a series of mechanical and technical problems. In late 1984, towards the end of the timescale covered by the original Brochure, much of the budget had been spent without satisfactorily completing any of the planned tests, although the datum blading had been run for the equivalent of three or four builds accumulating data for debugging the instrumentation and experience in operating the rig.

The fact that this programme went wrong was really due to inadequate pre-project, non-technical analysis and planning. Firstly, the project was seen to be low-tech, which perhaps it was relative to the high-speed multistage experiment more usually experienced within the Company, but for a campus environment the project was actually technically challenging. The potential for engineering, commissioning and measurement problems was ignored by all parties in the early proposals. In fact, the rig was faced with teething troubles experienced by most new facilities. The endemic, high speed rig problems, such as probe blockage and spatial resolution are solved in the transition to a low speed, large scale, environment, but they are replaced with a new set of difficulties associated with calibration and accurate data recording of small pressure levels.

An internal memo by Smith (1981) suggested that the 4-stage programme could be accomplished even more cheaply than the eventual CIT proposal. These views were based on experience of the large-scale single stage Oxford (later Deverson) rig at the Whittle Laboratory, which is actually a much simpler facility mechanically and is significantly less ambitious in the scope of its measurements.

The fact that some of the technical issues were not well thought through and the estimates which went into the proposals, and hence brochures, were unsubstantiated immediately moved the project from 'low-cost, low-risk' into the high-risk category. Indeed, had the project not been allowed to overrun, then very few of the technical objectives would have been met by the end of the planned time and financial outlay.

The lack of relevant data or experience should not have been an excuse for lack of planning. For a relatively mature technology such as turbomachinery aerodynamics there is <u>always</u> some data of relevance.

In this case, the in-house experience with HP9, a single stage high-speed rig, should have been considered. It was commissioned in 1979 and over the first two years of operation, the programme was supported in Compressor Technology by a section leader and two engineers (all BSc or equivalent) backed up by the Rolls-Royce Test Facilities group which provides, a pool of technicians who build and test the rig. In addition, the engineers were supported by draughtsmen and 'spec. writers' who handle the routine elements of specifying rig parts and instrumentation required for each test. HP9 and the LSRC are actually very similar in terms of measurement objectives.

More information could easily have been gleaned from the in-house operation of the General Electric, low speed, 4-stage research compressor. GE utilise two full-time engineers plus a half share in an instrumentation specialist (all PhD or MSc). Two millwrights are available for rig assembly and there are two additional technicians, one specialising in instrumentation, the other in computing (data reduction and presentation).

At this level of resource, GE complete an annual programme of six to eight builds per annum, which is similar to that foreseen for the LSRC at its beginning with one research officer (PhD) and half a technician on site, supplemented by the EIC (part-time) in Derby.

Over the twenty months which elapsed between the delivery of the rig and the second submission of the Brochure (Addendum 1) most of the time-absorbing problems were aerodynamic in nature and their solution was fundamental to the value of the project. The instrumentation was all new to CIT and the data recording and automatic control hardware and software were relatively ambitious in scope and largely untried. Some delays were the result of mechanical problems and though their nature would have been unpredictable at the outset of the project, some allowance could reasonably have been included beyond the optimistic, two month commissioning period. Some of the mechanical problems were attributable to the relative inexperience of the rig detail designer in rotating turbomachinery: this has to be offset against the lower cost of this subcontract option.

Over this period the monitoring procedures were operative so Management was aware of the state of the programme. Their procedures were geared towards passive tolerance of this situation: the delays were caused by a series of technical problems which had to be overcome sequentially, so the programme was allowed to overrun. Only the marginal costs accumulated proportionally with time at CIT were considered, set against the benefits of the eventual usefulness of the facility. This was rather too simplistic a view, ignoring the costs of tying up internal manpower (not clearly attributed to the brochure) which perhaps could be more profitably deployed elsewhere, and the costs of not having the information which would have been available through successful achievement of the programme's technical objectives.

Had these costs been considered, a much stronger case could have been made for an increase in effort both on-site at CIT and in-house to accelerate the programme.

Rolls-Royce has changed its policy recently so that brochures now do contain specific internal manpower allocation.

3.7 Maturing Programme

3.7.1 Monitoring Cash Flow

This task was relatively straightforward for the present project, since most of the spend has been channelled through CIT. The EIC is required to acknowledge the receipt of the 'service' referred to in incoming invoices, so an informal system was initiated whereby the Research Officer at CIT copied a complete breakdown of each invoice to Compressor Technology as it was submitted to the Cranfield Accounts Department. As each invoice arrived through Rolls-Royce via the official route, showing only the final amount with a one-line description of the work done, the author could readily reconcile the invoice with the appropriate job.

Fig 3.2 shows the historical cash flow for each of the brochures which have funded the LSRC programme. The invoices or cash flows have been summed over each quarter. The brochures are shown separately and the upper line represents cumulative spend. Over 1983-1984, when only the original brochure was involved, the cash flow was approximately linear reflecting the fact that most of the spend funds personnel at CIT. The linearity reflects the maintenance of a capability, rather than a piecemeal project with a clearly defined start and finish, which might have been expected to follow an 'S' curve relationship. The scatter about the straight line is generally the result of ad-hoc invoicing, with occasional high spend on hardware.

3.7.2 Programme Management

Critical Path Analysis (CPA) is probably the most widely used programme management tool. Although applied at higher level for planning and monitoring commercial engine projects, its use was not particularly widespread within Rolls-Royce for 'small' research projects. It was used in this case, however.

There are two major advantages to using CPA over more simplistic planning techniques: it introduces a logical discipline into the planning process and it schedules the most effective use of resources, minimising timescales and costs. It is at its most powerful when a project has a large number of tasks and resources.

Much of the promise of CPA has been lost on the LSRC programme, as it has been so manpower limited and, by definition, all major test milestones involved the single resource of the facility itself. The logic of the programme generally reduced to a simple build-test-strip cycle although, on occasions, parallel design and manufacture activities were appropriate, involving resource at Rolls-Royce, TANDEE and RD Castings (the rig and blading subcontractors).

The early networks were hand drawn and were found to be adequate for the purpose of planning and monitoring. As part of the learning process, SuperProject Plus, an IBM-PC XT based software package was used latterly. This program was found to be particularly user-friendly and had ample scope for a project the size of the LSRC. It was able to produce hardcopy plots with similar layout to a hand-drawn version, making them easy to follow, although they became untidy with many activities.

CPA has largely been used as a monitoring tool, rather than for planning over most of the programme. It did prove useful both in highlighting point at which decisions had to be taken in order to avoid hazard to milestones, and as a means of drawing slippage to the attention of Management.

CPA was used to define blading design release dates to ensure that its supply never fell onto the critical path. The low reaction design was particularly difficult technically and could easily have delayed the test date by a late finish had the implications of slippage not been identified in advance.

The CPA plans were also used to help sell the need for increased manpower to the Company Management. Several essential items, but not on the critical path, were completed by graduate trainees under the authors supervision. An example was the data processing software for transferring raw aerodynamic data from magnetic tapes from CIT to the Company's SPECTRE database (programs VU03 and VU04).

The major criticism of the management of this programme is that the activities were consistently timed optimistically, without planning for unforeseen delays, even though the weight of experience over the years suggested that problems were statistically likely. This was defended by CIT (and condoned by Rolls-Royce) on the grounds that the problems did not involve any particular aspect of the facility but were genuinely random and unpredictable. This was probably a counterproductive step: a truly realistic programme, with the low testing rate which that would imply, might have actually helped to raise priority for additional resource.

CPA has now become much more widespread within the Company. For many years, there has been a mainframe based database which stores details and schedules tasks in progress or planned across all departments and sites (known as TASK-Q). The input/output processes were not at all user friendly in the past, but recently a PC based front-end programme has become available which can interface with the mainframe database (POWERPROJECT). In fact, such planning is now mandatory for all in-house, aerothermal rig programmes.

3.7.3 Dealing with Subcontractors

The principal subcontractor for the rig design and manufacture was a small company not under direct control of either CIT or Rolls-Royce (Tooling AND Equipment Engineers, TANDEE). They were found not to have any real scheduling or planning systems, which was a potential problem since they often were close to the critical paths. In collaboration with the Research Officer at CIT, the author introduced a simple system of progress reporting sheets which were updated weekly. Examples are shown in fig 3.3. Eventually, they were refined into loading plans for specific machines and were successful as a means of generating data for updating the CPA plans and identifying potential delays in advance.

The procedures in use in the Rolls-Royce Purchase Office were not well suited to this type of small programme, which typically required small value orders but with short lead times where it was also important to have continuity of supplier. Occasionally, it was found necessary to circumvent this organisation by ordering CIT to deal with all other suppliers, but eventually there was a change in procedure to allow the EIC to specify the source of bought-out items.

3.8 Funding

There is a very fine line between keeping the project shielded from commercial pressure on resources, and it losing relevance to the needs of the business. The early /110D and /118D programmes were certainly far too long and rigid. This had the effect of making it difficult to react to the findings of the research as it unfolds, which is an essential element of this hands-on, fundamental work. This resulted in an untidy state of brochure affairs. A situation was evolving where brochures with relatively large value were being opened, but not closed, so a more flexible approach to the funding problems was sought. In fairness, the programmes would probably have been broken into shorter components had the extent of more realistic timescales been understood at the outset.

However, there is the need to provide visibility to the levels of future commitment. The solution which has emerged is the 'Master Brochure', which estimates the level of funding required to support the LSRC for a three year period, without technically specifying the build content. The technical details of a small number of builds, covering a year's work at most, together with more precise costing, are then submitted as addenda to the master brochure. This allows Management to retain overall control on the detail of the programme, but it allows a much more effective response to changes in emphasis resulting either from commercial pressures or interesting findings.

This funding philosophy has recently been extended to other sites by forming University Technology Centres (UTCs) which are a more formal implementation of the master brochure.

3.9 Communication and Motivation

In the original proposals for work on the LSRC, CIT had responsibility for all data analysis to a specification laid down by Rolls-Royce. However, this rather pure and simplistic specification proved non-viable and the author took a lead role in the analysis using Rolls-Royce in-house software, which has become the standard analysis route. So following basic data reduction at CIT, responsibility for data analysis was transferred to Rolls-Royce in Derby. While this was expedient while the facility was absorbing a large amount of engineering effort, it has led to a longer term problem.

The research officers have overcome the engineering problems and have accumulated sufficient expertise in the hardware and software of the rig control and data recording systems that much of the technical challenge has disappeared. The building and operation of the rig still demands a high level of concentration, however, in order to maintain quality of data. The motivation for this attention to detail has been difficult to find, particularly for 'second generation' rig owners.

Initially, it was necessary to provide a means for CIT to access Rolls-Royce specialist knowledge in the solution of technical problems. As the representative of Rolls-Royce at CIT, it was important for the author to convey the Company's interest in the difficulties being faced on-site and to demonstrate that the data produced from the LSRC was being constructively used in ongoing analysis at Derby and was having some positive influence within Compressor Technology.

The telephone was used extensively in routine communication (often daily) and visits took place frequently (typically 2 or 3 days per month).

The author also became involved in meetings with the subcontractors who supplied the rig and blading, essentially from a position of demonstrating interest and willingness to be involved as a member of the team, rather than as a 'customer' or protector of the commercial interests of Rolls-Royce.

Meetings involving more senior managers were arranged in the early part of the programme. These were less positive, and may even have been destructive of morale. They tended to focus on activities at a rather low level and resulted in long lists of 'actions' which must have been demoralising for CIT. Also, these meetings took place too regularly and had the effect of rescinding responsibility from CIT, as they became a venue for presenting problems rather than solutions.

More recently, communication has suffered as control of the programme has been delegated from the original 'owners' to 'second generation' personnel. Two essential ingredients were lost at this time: the regular, informal interaction between the research officer and the EIC and their 'emotional' involvement in the success of the programme.

There have been three major initiatives to promote liaison and improve morale. Firstly, in an effort to improve inter-site communication, regular informal meetings have been initiated, taking ½ day per month.

In the short term, this provides a venue where current concerns and progress can be reported and CIT can be given feedback of topical in-house activity. In the longer term, these meetings are intended to provide a basis for building a working relationship between the research officers and Rolls-Royce engineers which will promote working level communication.

CIT have also been given a more involved role in the running of the programme. In particular, they have been included in the negotiations with RAE on brochure proposals and have taken the initiative to generate their own work for the LSRC, compatible with Compressor Research strategy goals, submitting their own addenda to the Master Brochure. It is hoped that owning complete packages of work will improve integration of the team and result in continued attention to detail in the experimental programme.

Finally, Rolls-Royce is in the process of funding the development of the on-site data analysis software. The objective is to provide a package which is capable of mimicking the Derby analysis using a streamline curvature programme to help in the reduction of data. This will restore CIT to the status envisaged at the start of the LSRC programme and will provide the means for the gatherers of data to be more directly involved in the interpretation of results, the 'interesting' part of the job.

3.10 Major Lessons Learnt

This section is, essentially, a reminder for the author of what has been learnt by experience in the management aspects of this project. It may also be of use for others planning similar jobs.

Firstly it is necessary to define the project scope and 'deliverables' and to organise and agree the framework within which the project is going to be managed. For example, who is in overall charge of the programme, who is to be responsible for the elemental tasks and how will the monitoring be carried out and coordinated?

Then there is a need for careful planning of the programme with substantiated data, identifying the high risk activities as well as those falling on the critical path. If the data contributing to the plan is felt to be unsubstantiated or not as relevant as it might be, then time must be built in to allow for contingencies.

Having identified the tasks and their logical relationships, a CPA plan should be constructed and regularly updated. This should be further broken down into detailed workplans for each sub-resource which can be monitored to provide data for the main CPA plan. It is essential that potential slippage is identified as early as possible so that corrective action can be taken or at least Senior Management can be made aware of likely delays.

It is important to properly account for all the costs of the project, including manpower and other items which one might be tempted to bracket as overheads. If the project looks likely to overrun then Management can reassess the cost-benefit analysis.

The retention of some programme flexibility is desirable so that the experimental plans can be amended to take account of findings as they emerge. However, there is also a need for visibility of stable funding so that staff at the research agencies can be given contracts of reasonable length. This was not compatible with the operation of rochures over most of this programme, where rigid long term plans existed. The solution of the Master Brochure seems much more appropriate.

As well as the need for well motivated staff at research officer level, it is important to have enough high quality technician effort available. The present project was certainly undermanned at both levels. To generalise, the campus environment is not particularly suitable for 'screening' types of applied research, where there are many builds of short duration. This places far too heavy a burden on technician effort. It is probably better to concentrate on few builds measured thoroughly. This is what has actually been achieved in the present project.

The success of small scale research programmes is heavily dependent on the commitment and achievement of individuals. It has been understandably difficult to maintain morale of staff at CIT while they have not had responsibility for the complete task (recording but not analysing data) and they have gone through periods of feeling remote from the team in Derby. Split site projects place a particular emphasis on communication and it is, therefore, important to establish regular and structured exchange of information, both through face to face meetings and over the telephone.

4. REVIEW OF PREVIOUS WORK

4.1 Introduction

The drivers for advancing compressor technology were discussed with a civil engine bias in Chapter 2. The quest for military air superiority has also provided strong motivation for turbomachinery development. These market segments have significant overlap in turbomachinery technology and together they have inspired and underwritten enormous expenditure of both resources and effort in compressor-related research, which has resulted in a vast amount of published information.

Text books can be a useful introduction for those new to the subject, but they are sparse in compressor aerodynamics and almost all present information which had emerged by the early 1960's. 'Axial Flow Compressors' by Horlock (1958) is a useful basic text, while NASA SP-36 (1965) contains detailed descriptions of the then current design techniques (also providing the empirical information necessary to apply the methods) as well as qualitative descriptions of flow phenomena which are just as relevant today. 'Cascade Aerodynamics' by Gostelow (1984) collates much of the information published from such facilities which is most relevant to compressor turbomachinery, but by far the most useful book is 'Compressor Aerodynamics' by Cumpsty (1989). As well as being a comprehensive and critical reference guide, the text presents and explains much of the industry's folklore, often in print for the first time. It does not duplicate much of the basic applied maths which has become a part of most mechanical engineering undergraduate courses and is adequately covered by Horlock. The chapter dedicated to viscous effects is most relevant to the present thesis.

A complete review of the literature is beyond the scope of this chapter, so attention is focused on the work of particular relevance to blading design in the end-wall region, for compressors operating close to their design or peak efficiency points.

Most of the literature published on axial compressors over the past 50 years either relates to the understanding of the flow phenomena present in these machines or aims to help the designer achieve a well matched 'conventional' design. There have been very few published attempts aimed at perturbing these designs to specially account for the flow near the end-walls.

This is somewhat surprising, since as long ago as Howell (1942), this region has been blamed for a substantial proportion of compressor inefficiency. Cumpsty (1989) suggests that the lack of publications in this field may be the result of selective reporting, but he concedes that the interactions are complex and the hardest problems are naturally left until last.

4.2 Description of the 3D Flow Phenomena

Serovy (1985) presents one of the most recent literature reviews of secondary flow phenomena. The paper is without critical discussion, but it is useful as a comprehensive list of relevant references including earlier comprehensive review papers such as Lakshminarayana and Horlock (1963, 1973). Also, this paper collates some of the better schematic diagrams of secondary flow phenomena, from which that due to Inoue and Kuroumarou (1984) has been selected and reproduced here, with minor modifications, as fig 4.1. This particular description was produced following detailed measurements downstream of an isolated rotor with a hot wire anemometer and shows the features in a frame of reference moving with the rotor. It can equally well be used to describe the flow in cantilevered stator passages, though clearly the leakage flow would be at the inner end. Serovy underlines the point that the phenomena are not independent, but that they overlap and interact, so not all these discrete features will be discernible in every set of experimental data.

Passage Vortices - Classical Secondary Flow

Conceptually, the simplest form of secondary flow is the passage vortex which arises from the turning of a flow with non-zero vorticity in the streamwise direction (due to a radial gradient of total pressure towards a cascade end-wall, for example) through a finite angle. The turning of the main stream flow sets up a cross-passage pressure gradient which is largely maintained near the walls by the requirement for the flow to be in radial equilibrium. Considering the normal equilibrium equation, acting along the passage,

$$\frac{\partial p}{\partial r} = \rho \frac{V_\theta^2}{r}$$

the slower moving fluid near the end-walls must follow a tighter radius of curvature and it is swept towards the suction surface, in the direction of the pressure gradient, rolling up into a vortex. This phenomena was visualised by Herzig and Hansen (1955) by injecting smoke into the end-wall boundary layer ahead of a linear cascade. Two cascades were tested with deflections of 45 and 60 degrees and the strength or 'tightness' of the passage vortex was found to increase with turning. Herzig et al went on to make the positive recommendation that it would be worth spending time and effort to design blades with 'favourable' velocity profiles (avoiding regions of excessive deceleration) which would result in less severe local flow disturbances and smaller deviations from design gas angles.

Skew

The flow entering the bladerows is not collateral in general, but is 'skewed' such that the flow close to the walls is at high incidence compared to the mainstream flow. The main cause of skewing is the relative motion between successive bladerows. Low momentum fluid in the boundary layer of one passage acquires a large tangential component of velocity as it changes frame of reference between rotating and stationary components.

The action of viscosity is to destroy any cross flow in the boundary layer such that given a large enough axial distance, a collateral flow will result. Also, the no slip condition ensures that the large velocity magnitudes close to the wall are greatly reduced in a small axial distance. The blade force in compressors is also in the direction to destroy the skew.

Moore and Richardson (1957) investigated this phenomenon using a linear cascade, setting up the inlet skew with a jet of air blowing normal to the main throughflow velocity close to the wall just ahead of the bladerow. They found that there was much less overturning at exit from the cascade when the inlet boundary layer was skewed compared to the collateral flow case.

Bettner and Elrod (1982) established readacross of this phenomena to real turbomachinery flows with measurements of the casing wall boundary layer development through a stator of a single stage axial compressor.

Stators in axial compressors exhibit flow features which are most similar to those found in cascades. Generally, the stator experiences positive incidence near the walls, which is in the opposite direction to the secondary flow that is likely to be produced within the stator itself. The nature of the flow leaving the stator depends on the difference between these two effects. High stagger blades usually have small camber, so inlet skew is likely to predominate, leading to underturning in the stator. Conversely, for low stagger, high camber blading (common near hubs) a larger contribution from within the passage is likely, hence overturning.

By far the most significant collection of measurements of end-wall flows in multistage compressors was given by Smith (1970). Results of axial displacement thickness and tangential force defect are presented for many configurations. The understanding of repeating stages and correlations given for loss and blockage are still the best available in the open literature today. Between May 1959 and February 1967 41 builds of the rig were completed with a hub/tip ratio of 0.7 at a non-dimensional design point of $Va/U=0.6$ and $\Delta H/U^2=0.36$ with a type of controlled vortex flow.

This information formed the basis for endwall loss and blockage correlations in Koch and Smith (1976), who provided a semi-empirical method of estimating the efficiency of axial compressor stages.

Annulus Boundary Layer Calculations

These theoretical approaches usually treat the flow as uniform in the circumferential direction, using normal boundary layer methods to compute the axial development, although the tidy division into end-wall and free-stream is not well supported by observation.

Many of the more successful methods have used some formulation of the momentum integral equations. All can be described as control volume analyses with several unknowns; blade force defects, skin friction, velocity profiles and other parameters, such as entrainment. The calculations are applied through the bladerow taking the free-stream flow field to be prescribed, for example: Mellor and Wood (1971), de Ruyck and Hirsch (1985), Lindsay et al (1987) and Brochet and Falchetti (1987).

Stratford (1967) presents a simple analysis for the flow along the endwalls, ignoring skin friction and assuming incompressible flow. Results from this method have been found to be useful where the tip clearances are significantly smaller than boundary layer displacement thickness (so small force defects) and there is little or no change in momentum thickness across bladerows. Designers need displacement thicknesses for blockage and this requires further empiricism. This method was used in many Rolls-Royce axial compressor designs of the 1970's.

Lindsay et al (1987) suggested that the more ambitious boundary layer methods were unlikely to succeed even for compressors of modest loading compared to those found in aero-engine applications. Brochet and Falchetti (1987) did manage to obtain reasonable agreement by adjusting one of the parameters (the direction of the force defect) but this may not be generally applicable.

Boundary layer methods are still used by compressor designers as a consistent and systematic method of including many of the geometric and flow variables in a correlation scheme. The method in use at Rolls-Royce is that developed by Wright (1984) which was described in the open literature by Freeman (1985). The method, based on de Ruyck and Hirsch, was used in conjunction with an optimiser which found the best fit to streamwise and normal force defects from tests of many compressors.

Although the method gives good predictions of variables, it does not 'describe the flow', and is of most value in designing machines within the bounds of its empiricism. Few organisations design new machines completely from scratch these days, most have substantial in-house databases to guide designs. An approach which relies on data from earlier machines is at its most reliable when the new machine bears a strong resemblance to one of its predecessors.

Tip Clearance Flows

Dean (1954) is one of a number of excellent descriptive reports to emerge from MIT during the 1950's. A large amount of empirical data is presented which reveals the mechanism and influence of tip leakage flow in a rectilinear cascade. The data includes spanwise pressure distributions as well as pneumatic traverse measurements, but most impressive is the descriptive model of the tip leakage and secondary flow which is built from these data. Although the measurements were from a cascade, it is clear that the author understood the readacross of this information to the real turbomachine.

The description starts from an idealised inviscid flow pattern, then Dean incorporates the features of the real situation one at a time, starting with viscosity, which allows lift to be retained at the tip of the blade. Also, a viscous fluid cannot remain attached around the end of the aerofoil as an inviscid fluid can, but separates and follows the wall, as shown in fig 4.2. The leakage flow in the viscous case entrains fluid from away from the wall and the flow is <u>towards</u> the tip not away from it as an inviscid argument would predict.

The exposition continues through the case with incoming boundary layers, relative motion between the blade tip and wall and the effects of skew. Some results hinted at the fact that there might be a minimum clearance below which losses would start to rise, a phenomena which has been the subject of considerable discussion over the years, much of it summarised by Peacock (1982,1983).

Rains (1954) studied a three-stage axial water pump. For a rotor, the clearance vortex seemed to appear from the leading edge, whereas for the stator the clearance flow sprang from the quarter chord point. Rains suggested that the mechanism of loss generation was that the kinetic energy normal to the chord was lost.

Lakshminarayana and Horlock (1967) took measurements in a linear cascade with a clearance at mid-span. At very small clearances, the pressure distribution near the end was very similar to the ideal 2D case, but at larger gaps (4% chord) the influence of the clearance flow was marked up to 8% of chord along the blade, with a large drop in pressure on the suction surface just where the vortex might be assumed to affect it.

There are many published measurements from turbomachines, particularly from low-speed facilities and most interest has been in rotor tip clearances. Laksminarayana et al (1986) plotted the development of the clearance flow through the passage near peak efficiency and found collection of loss close to the pressure surface.

Inoue et al (1985) present measurements from an isolated rotor at several clearance levels. The region of high loss associated with the leakage fluid moved progressively towards the pressure surface as clearance was increased up to about 2% of chord, then it reversed its direction of migration.

Hunter and Cumpsty (1982) noted the collection of high loss fluid close to the pressure surface, particularly near to stall. This may be because the inlet skew was greater in this case, or because the clearance flow was rather stronger. Again, at clearances of greater than 5% of chord, the regular pattern disappeared.

Tip clearance has an important effect on both the end-wall and blade-to-blade flows in its vicinity. It is a complex phenomena; generally increasing clearance leads to reduced efficiency, loss in capacity and deterioration in the stall line. However, there is some evidence that there is an optimum value, below which the loss starts to increase.

Spanwise Mixing

Regions of high loss production experience an outflow of gas with raised stagnation temperature and reduced axial momentum. This is necessary if there is not to be a very high build-up of entropy near the walls of the multistage machine. The evidence suggests that equilibrium is reached in practice after two or three stages. Gallimore and Cumpsty (1986) found that there was no clear edge to the viscous region. Wisler et al (1987) found that there was, although there was high turbulence right across the annulus.

Adkins and Smith (1981) suggested that secondary flows are primarily responsible for both the spanwise mixing of flow properties and the deviation of bladerow turnings from 2D cascade theory. Their analysis was based on inviscid, small-perturbation, secondary flow theory, including non-free vortex main stream flow, end-wall boundary layers, tip clearances, stator shroud leakage and blade profile boundary layer and wake migration under centrifugal forces.

They model mixing as a diffusion process with a mixing coefficient found from calculated radial velocities. They also calculate secondary cross-passage velocities for use in estimates of under and over turning.

Gallimore and Cumpsty (1986) claimed that the above model was not correct and that spanwise mixing was dominated by a more random, turbulent type of diffusion. Their mixing coefficient was based on stage geometry, loss and flow coefficient.

Having incorporated their mixing coefficients into axisymmetric throughflow codes, both camps were able to demonstrate good agreement with different test cases. Wisler, Bauer and Okiishi (1987) diplomatically concluded that both models had important facets and consequently both are needed in throughflow calculations for multistage machines.

4.3 Compressor Design Systems

Behlke (1985) gave an interesting description of an "Integrated Research strategy" with "evolution of technology" which was seen as desirable in Chapter 2. Pratt and Whitney fund a 3-stage LSRC at the United Technology Research Centre and Behlke describes the role of this rig as a source of fundamental data primarily, then a first level validation for concepts prior to the commitment to a high speed rig test.

The paper gives a qualitative description of the extension of Controlled Diffusion Airofoil (CDA) philosophy to blading design near the endwalls. Semi-empirical corrections are applied to the radial distributions of loss and turning and integrated into the throughflow. During the blade-to-blade section design iterations, loss and turning are fed back into the throughflow. A similar approach is followed at Rolls-Royce.

As with all methods based around inviscid programs, some approximations are needed to account for multistage effects. Between rows, wakes decay, vorticity dissipates, spanwise profiles are mixed and a change in reference frame occurs. In calibration, this throughflow method appeared to work well on both P&W and GE LSRC cases.

4.4 Blading Design in the End-Wall Region

The earliest axial compressors were designed assuming that axial velocity was either constant across the span, or that it followed a straight line between hub and casing, using the change in angular momentum to determine the required temperature rise. Experimentally, it was found that stages were absorbing somewhat less work, and Howell (1942) introduced the 'work-done' factor, defined as the ratio of the likely actual work absorbing capacity to its ideal value. The explanation for this phenomenon is simply that the axial velocity is not constant or linear across the span, but becomes increasingly peaky as the flow proceeds, settling down to a fairly stable profile after 3 or 4 stages (Andrews, Jeffs and Hartley, 1951).

Although the blades were not specifically adjusted in the end-wall regions, this represents the first design technique to account for end-wall effects.

Fortunately, there is a natural tendency for conventional blading to add more work to the lower momentum boundary layer fluid, offsetting the deterioration in the total pressure profile to some degree. Considering a representative velocity triangle (fig 4.3) the work can be expressed in terms of rotor and stator relative exit gas angles and blade speed,

$$\frac{\Delta H}{U^2} = 1 - \frac{V_a}{U}\left(\tan\alpha_2 + \tan\alpha_4\right)$$

It is reasonable to assume as a first approximation that the blade deviations do not change with incidence, so as axial velocity increases, the work reduces and vice versa.

On a mass meaned basis, the overall work from a given stage reduces as the axial velocity profile becomes more fully developed. Howell's work-done factor was assumed to vary through the multistage compressor reflecting the initial rapid growth of the annulus wall boundary layers and acknowledging the existence of an eventual, repeating profile.

The fact that so-called 'conventional' blades operate at incidences well away from ideal towards the blade ends has been recognised since the beginning of axial compressor development. Attempts at designing stages which either take account of or modify the flow conditions around the ends of the blades date back to the early 1950's at NGTE.

Andrews et al (1951) suggested two strategies which were aimed at producing a design which has reasonable gas angles over more of the blade height, hoping for improved performance by reduction in secondary losses: to either accept the deteriorated profile or prevent the deterioration from occurring.

The first philosophy hypothesised that the poor axial velocity profile was caused by high losses at the ends resulting from high incidences. Variable Axial Profile (VAP) aimed to minimise losses by increasing blade inlet and exit angles at hub and tip of both rotor and stator. The second philosophy accepted that the losses were not incidence generated but were unavoidable. Variable Work Done (VWD) designed more work in towards the walls at a conventional level of axial velocity in order to achieve a flatter velocity profile. The blade geometry of VAP, VWD and the datum is illustrated in fig 4.4.

Overall performance was measured on six-stage builds of the C106 compressor at 0.75 hub/tip ratio and a Reynolds Number of 6.2×10^4 with small axial gaps (67% of chord) while detailed traversing took place on four-stage builds at increased axial gap (1½ chords). VWD had six stages of identical blading, while VAP comprised three sets of blades: standard in the first two stages, then two sets designed for the progressive velocity profile deterioration. Both standards of blading were designed to produce a common mean ΔT per stage at a flow.

VWD lost 2% peak efficiency relative to the datum, although it did achieve a much flatter velocity profile. The excess work designed in over the mean favoured the outer diameter (1.14 to 1.06 at the hub) and resulted in high velocities at around 80% height. This might have put these sections into negative incidence stall, thus contributing towards the poor performance. Andrews suggested that an excess of 1.1 (over mid) would be a more appropriate guide for future designs resulting in a more uniform axial velocity.

This test did demonstrate that varying the work along the span can enable the axial velocity to be kept near design so that only a small portion of the blade needs to suffer high incidences.

VAP failed in its design objective. The velocity profile degenerated continually through the machine and ended up even more peaky than the datum and it lost a further 1% of efficiency, so was 3 points down on the datum. The report lacks radial traverse information for VAP, but presumably, incidences were just as bad (or worse) than the datum case.

In 1959 Rolls-Royce did their own version of VAP (Payne, 1959). It was designed for the same air exit angles but lower axial velocities at ends and hence increased inlet angles. The design work was slightly higher. It also had reduced s/c ratio because of the increased camber. On test at high speed in a compressor designated 'H5', it lost of 1% of peak efficiency and 1½% of flow relative to the datum (H2).

The following year, Rolls-Royce tested a 4-stage 0.8 hub/tip ratio low-speed compressor with VAP and Inverted Velocity Profile (IVP) (McKenzie, 1960). VAP was similar in design to H5, except that the space chord ratios were maintained at the datum levels. IVP was a logical extension to VWD with a further excess of work designed into the hub and tip to pull yet more flow through the walls and unload the blade ends. Inlet angles were not increased, however, to keep down camber. All achieved the same $\Delta H/U^2$, but relative to the datum efficiency, VAP was +1% and IVP +2%. IVP achieved the flattest velocity profile.

The last test in this series in 1961 was reblade of rear 6-stages of the 9-stage Conway HP compressor. Both rotor and stator inlet and exit angles were increased, but with no change to the s/c ratios. This high speed test was very successful with 11% greater surge margin and a 1% efficiency improvement at design speed, more at part speed (Williamson, 1962).

Scrivener (1971) summarised this early work by concluding that the velocity profile only improved when the design work was increased at hub and tip via increased deflection and that there is a loss generating mechanism at the blade ends which involves secondary flows and the end wall boundary layer, rather than a blade element loss caused by incidences. If the profile can be prevented from deteriorating then the conventional inlet angles will be acceptable. On the basis of VAP he advised that changing the incidences to account for lower axial velocities made matters worse, a conclusion which is challenged below.

The IVP philosophy resurfaced as a candidate for the RB211-535E4 HP compressor, when it was rig tested as part of the development programme. It was found to pass too much flow at reduced efficiency with a worse surge line when compared to the datum compressor (Malley, 1986).

Secondary Flow

Some elegant mathematical treatments of classical secondary flow were produced in the 1950s, the simplest due to Squire and Winter (1951). For small perturbations about a free-vortex, inviscid flow, the secondary vorticity at exit is given by

$$\xi_2 = 2\left(\alpha_2 - \alpha_1\right) \cdot \frac{dv_1}{dz_1}$$

The Poisson equation

$$\nabla^2 \psi = \frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = \xi_2$$

can then be solved for the secondary stream function in a plane normal to the main stream flow downstream of the blade row. The secondary velocities can be found from the definition of the stream function, which then can be turned into angle perturbations.

The ability to calculate secondary flows led quickly into some attempts at modifying blade designs to minimise them. Ehrich (1955) used a conceptual model similar to that of Squire and Winter but a more general analysis incorporating not only the non-uniform inlet velocity profile, but also the radial variation of blade angle.

He made classical assumptions of incompressible and inviscid flow and small turning, hence small cross passage pressure gradient and little distortion of Bernoulli surfaces, so this work is limited to small perturbations.

Ehrich derived a general equation of the secondary flow stream function

$$\nabla^2\psi = \underbrace{2\epsilon\left(\frac{\partial u}{\partial y}\right)}_{\mathrm{I}} + \underbrace{u\left(\frac{\partial \epsilon}{\partial y}\right)}_{\mathrm{II}} + \underbrace{\left(\delta_1 - \delta_2\right)}_{\mathrm{III}}\left(\frac{\partial u}{\partial y}\right)$$

I secondary flow introduced by development of vorticity in the direction of flow as fluid is turned in the cascade

II secondary flows induced by the variation in turning angle

III contribution from deviations of the inlet and exit angles from the mean levels. These were defined to be small, so this term was ignored.

Complete elimination of secondary flow required

$$\frac{1}{u}\frac{\partial}{\partial y}\left(\epsilon u^2\right) = 0$$

or εU^2 = constant, so as velocity decreases, turning should increase.

Three cases were investigated experimentally, building up the most general

a) cascade with non-uniform turning and constant U to study II

b) cascade with uniform turning and non-uniform velocity profile to study I

c) superposition attempting to hold εU^2 = constant

This strategy appeared to work reasonably well with good agreement between experiment and theoretical calculation. The most notable departures were seen towards the endwalls; although the boundary layers had been bled ahead of the cascade there was some growth within the passage.

The camber was a little low (19.5°) for direct read across to compressor applications and there was no measurement of losses.

Taylor, Stevenson and Dean (1954) is another contemporary contribution from the MIT Gas Turbine Laboratory. The paper starts with a discussion of the desired annulus boundary layer flow, wisely noting that only tests of rotating machinery would ultimately answer questions associated with the details of secondary flow in real compressors. They defined an objective of keeping all boundary layer fluid against walls and preventing or at least delaying separation.

The simplest control means suggested was to energise the wall boundary layer by adding more work near the end-walls than in the free stream, which is essentially VWD as described above. This would result in more nearly axisymmetric flow and may lead to the elimination of corner stall in conservatively designed stages.

Another idea was to re-energise the low momentum fluid through mixing. The twisting of stream surfaces could allow this by bringing boundary layer fluid to one side of passage with corresponding main stream flow moving towards wall. In other words, skewing of Bernoulli surfaces may be sometimes desirable.

They recognised the fact that the boundary layer may have constant velocity but varying direction (skew) noting this to be very different from the standard cascade situation (collateral flow) and suggested the possibility of an equilibrium state for the boundary layers, the repeating stage situation later found experimentally by Smith (1970).

Taylor concluded that elimination of spanwise velocities was possible with mild upstream vorticity, but thought this to be impracticable where strong vorticity exists (ie in the boundary layers) where air would not follow the surface and would separate.

Experimentally, Taylor investigated "forward leaning" the ends of the cascade (the so-called Zika tip) aiming to allow the secondary flow velocities to proceed past the trailing edge without stalling the suction surface. This was successful in removing the corner stall, but there was vigorous mixing downstream of the cascade with the result that the total mixed out loss was unchanged despite an initial improvement within the passage. There was a reduction in tangential blade force for the forward leaning tip, a result of the strong cross-flow behind the blade, leading to underturning.

Finally, they mentioned the possibility of energising the boundary layer by using tip slots, thus allowing some high energy fluid from the pressure surface to pass through the blade and that bleed holes could be advantageously placed in a blade row which had badly behaved boundary layers.

Having noted Ehrich's suggestion of increasing camber towards the walls, Martin (1959) proposed the opposite!

Her objective was to vary the camber towards the blade tips aiming for greater uniformity for the downstream blade rows and hoping to achieve considerable improvement there by reducing spanwise variations in outlet angles. It was hoped to reduce corner stall in the subject blade row also.

At low incidence, some control over the spanwise variation in the cascade outlet angle was achieved through a rather crude geometry change, though reducing the camber seemed to have little influence on losses at low incidence. At increased incidence, the angle changes are not shown, but the report suggests a significant reduction in mass meaned loss. Mass meaning may be obscuring the results; a loss coefficient (based on inlet dynamic head) would have been more appropriate. As far as read across to turbomachinery is concerned, the blades had unusually low camber and quite a severe velocity profile was chosen for convenient theoretical analysis.

Glynn and Marsh (1980) produced an elegant secondary flow analysis for an annular cascade with the basic assumption that there is no variation in the secondary velocity components in the direction of primary flow at exit from the cascade. The secondary flow stream function was defined for a plane which contains the blade trailing edges.

If a uniform flow is turned in an annular cascade which is a non-free vortex design then there is a variation of lift along the span. If there is no inlet shear, there can be no distributed secondary vorticity (which is Kelvin's theorem). When both effects are present, a non free vortex design and inlet shear reinforce each other at one end of the blade and oppose each other at the other end.

Their resulting expression of the Poisson equation in polar coordinates was

$$\frac{\partial^2\psi}{\partial r^2} + \frac{1}{r}\frac{\partial\psi}{\partial r} + \frac{1}{r^2\cos^2\alpha_2}\frac{\partial^2\psi}{\partial\theta^2} = \xi_{sec}\sec\alpha_2 + \frac{u}{r}\frac{\partial}{\partial r}\left(r\tan\alpha_2\right)$$

where

$$\xi_{sec} = \frac{\xi_n}{\cos\alpha_1\cos\alpha_2}\left[\frac{1}{2}\left(\sin 2\alpha_2 - \sin 2\alpha_1\right) + \left(\alpha_2 - \alpha_1\right)\right]$$

The second term quantifies the effect of a non-free vortex design.

James (1982) applied a Rolls-Royce secondary flow program (L009) using the above vorticity model to the geometry of a single stage, low reaction high speed compressor (HP9). For the IGV, the inviscid code predicted the exit angle quite well, though the boundary layer thickness at inlet was chosen somewhat arbitrarily to achieve the best match to measured angle. The prediction was improved if the program was re-run with primary angles used from the first pass. The calculated difference in $\Delta\alpha$ reduces with subsequent iterations until eventually it tends to zero.

At rotor exit, the angles predicted by the inviscid code and those deduced from a throughflow analysis of test data began to disagree. While the test data suggested over turning at the walls, the program predicted increasing exit angle towards both ends of the blade.

James concluded that either viscosity has a large effect on the flow within the stator, or the dominant effect is the unskewing of the inlet boundary layer. If the latter were true then one could still get reasonable results by modelling the degree of unskewing. The best 'prediction' or agreement was found when it was assumed that rotor/stator interaction completely destroyed the boundary layer at rotor exit.

James suggested a design mode for the secondary flow program which can be considered a development of that proposed by Ehrich earlier. Considering Marsh's stream function equation, the second term is the non-free vortex contribution to the secondary flow. The object of end-bends can be thought of as creating enough 'non-free vorticity' to cancel the secondary streamwise vorticity at exit to a blade row.

Birch (1984) performed a theoretical study on skew with a 3D viscous flow solver, comparing calculated results with measurements from the same single stage rig test used by James. The prediction of both IGV and rotor exit whirl angles was qualitatively good with this viscous program. In the relative frame, the rotor had high whirl at inlet near both walls and a fairly flat exit angle profile resulted, with an indication of some skewing in the opposite sense at both walls (the rotor had almost zero tip clearance). A similar inviscid calculation resulted in increasing $\alpha 2$ at the walls, just as predicted by James' simpler secondary flow method above.

Birch suggested that the destruction of skew in the axisymmetric turbulent boundary layer by viscous forces was not particularly well described by the simple isotropic Prandtl mixing length model of turbulence. However, no matter how imperfect this model, his work showed that viscous terms are definitely required when skew is present.

Dang (1985) was the last to attempt application of secondary flow techniques to blading design. The proposal was similar in principle to Ehrich and James as reported earlier, but involved a power law relationship to prescribe the swirl distribution. It did not appear to result in a practicable blading design technique.

Calvert and Came (1978) estimated the ultimate polytropic efficiency of contemporary core compressors to be about 95%, starting from the achieved levels of 89%. The irreducible 5% of inefficiency was associated with skin friction on the blades and walls and for operation at reasonable levels of stage loading. The theoretically reducible 6% was attributed to secondary losses, tip clearances and blade-row interaction. They suggested that perhaps one third of this 6% could be recovered in practice and their report provided the technical justification and much of the motivation for the more recent studies on end-bends at Rolls-Royce, of which the present work is one.

The modern history of end-bends at Derby stems from Freeman (1978). Freeman proposed a small perturbation to a conventionally designed stage. The axial velocity profiles are estimated on both hub and casing and the blade is recambered at the leading edge, to achieve a sensibly constant incidence across the span, and at the trailing edge, to match the gas angles which would result assuming a constant tangential blade force through the boundary layer. This technique has become known as Inviscid End-Bends (IEB) but it is inviscid only in the sense that the approximate calculation of blade force does not include viscous terms.

It was successfully tested on both zero αo and 50% reaction blading in the HP9 single stage, high speed rig. Generally, end-bends have given at least a 1% improvement in peak efficiency with an increased surge margin over a range of space to chord ratios with conventional profiles in this configuration (Harris, 1985). A 2-stage version of HP9 has also been tested with end-bends, but the experience was not so clear cut. Conclusions on the performance of the end-bends tended to be obscured by difficulties in read across of build standard and instrumentation between tests.

There has also been rather mixed results on high speed, multistage compressors with end-bends. They were successfully applied to a zero $\alpha 0$, HP compressor, which is in airline service (Freeman, 1985, fig 4.5). HP10 was a six stage research compressor with 50% reaction stages designed within the RB211-524 HP annulus line of the day. HP11 was a reblade of HP10 with inviscid end-bends. The test results showed a significant fall in capacity for the end-bent version, with a drop in peak efficiency of between 0.5 and 1.5% at the high speed end of the characteristic. Freeman (1982) suggested that the rotors may have already been choked in the original HP10 'parent' design. Further reducing the throat areas with the end-bends could then have worsened the stage matching.

Two other methods have emerged from within Rolls-Royce. One, the so-called 'Controlled Vortex', is of the type where the vortex flow is altered by changing the stator exit gas angles and is a close relative of VWD. This has featured in many of the military core compressors and, latterly, has also been successfully applied to civil machines.

The second was based on a 3-zone throughflow method where the end-walls and core flows were calculated separately then matched and the near wall gas angle perturbations found using an inviscid secondary flow method (Garwood, 1974). In view of the difficulty in predicting secondary flow angle changes through more than one bladerow with inviscid methods, this method has not been pursued.

There have been few tests of end-bent blading reported in the open literature since the early NGTE work; most have appeared recently while the present work has been in progress.

Behlke (1985) reports that the first generation of Pratt and Whitney CDA aerofoils had stalled profiles near endwalls, despite leading and trailing edge over cambers. Modifications were carried out in three steps. The most influential was to align inlet metal and gas angles, which led to reduced loss and increased turning. Secondly, the exit metal angle was increased, reducing turning which led to reduced loss. Finally, the aerofoil geometry was adjusted to be more aft loaded, which reduced loss a little and contributed to increased turning. Modifications were carried out to both rotors and stators; the extent of the modifications depends on spanwise location and axial position in compressor.

The improvements were demonstrated at high speed on a full sized model of the middle three stages of the PW2037 HPC. 'CDA-II' apparently produced 1.5% increase in working line efficiency and 8% increased surge margin.

However, most of the quoted benefit appears to be through an improvement in matching of the locus of peak efficiency with the working line. The three stages would probably not operate on a common working line since they have different flow speed relationships, so for a comparison, it is fairer to look at peak efficiencies. The actual change in the peak value at 100% design speed was less than 0.20%. There were improvements in characteristic shape, particularly a steeper pressure rise towards stall and broader efficiency curve. Surge margin was also increased through raising the surging pressure ratio.

Cai et al (1985) applied end-bends to rig and engine compressors. The description of the modifications is somewhat confusing in the text, but two stators of a three stage machine were apparently involved with a de-camber over the rear 50% of chord. The radial extent of the stator changes was not reported and the rotors were apparently not changed. Although not specifically mentioned, the modifications appear to have been inspired by the publications from GE (Wisler, 1984). The stated objective was to improve conditions onto the following rotors.

This moderately loaded machine (2.5:1 in 3 stages) gave a dramatic improvement of 3.2%, but from a relatively modest level with a small drop in capacity. It was also backed up with an engine test which showed reducing SFC. Though the surge margin improvement looks genuine there may be a rematching effect in the efficiency gain and the compressor did not actually reach peak efficieny at all speeds.

Brochet and Falchetti (1987) present the SNECMA strategy, which builds on their background in secondary flow computation. Secondary flows are considered as two 3D boundary layers coupled to a throughflow calculation. The boundary layers are an extension of that published by Mellor and Wood, extended to work within the bladerows. Local values of secondary effects are calculated, averaged across the pitch.

The paper goes on to describe a design application to stators. The resulting blades show a substantial increase in stagger towards the walls as the sections are adjusted to match stator leading edge incidence. This reduces the section loading and reduces secondary flow phenomena and better matches the flow onto the rotors. The paper does not quantify the performance improvements.

NASA sponsored a large amount of work on the GE 4-stage LSRC, studying multistage compressor exit stage phenomena. The final contract report by Wisler (1981) provides a useful summary of the important results. Wisler (1984) gives wider publicity to the same information. This study had similar objectives to the present work, aiming to reduce the losses associated with endwall phenomena by a target 15% through modifications to traditional (baseline) vector triangle geometries and by tailoring aerofoil shapes.

The baseline design had rotors with circular arc thickness distributions and NACA-65 stators with maximum thickness and camberlines chosen such that the aerofoil surface velocity distributions modelled those of a stage from a high-speed machine. The compressor had 0.85 hub/tip ratio, aspect ratio of 1.2 and blade chordal Reynolds number of 360,000. Rotor tip clearances were 1.36% of annulus height and stators were shrouded. Rotor 1 was fitted with a grooved casing treatment to make sure that the first stage would not be the stall limiting blading.

Tests of the baseline blading had revealed high stator exit swirl angles near both endwalls and regions of separated flow near the hub. The most significant improvement in performance was obtained by increasing the stator stagger towards both hub and casing annulus walls, reducing the axial velocities in these regions relative to the baseline standard. Stagger was increased by 8°-10° near both walls and reduced by around 2° at mid-height.

The end-bent stator was stacked at 30% chord rather than 50% to minimise the acute angles where the blade and casing intersect. The design point and average stator exit tangential momentum were maintained.

The end-bends led to improved non-dimensional characteristics. Peak efficiency was increased by 0.4%, implying a 10% reduction in endwall loss. The characteristic steepened towards stall, though the stage stalled at a higher Va/U.

At peak efficiency, the traverse results showed slightly lower losses for end-bent stator at the hub, but towards stall, the re-staggered stator was much improved over the datum. Vane surface statics showed that the re-staggered blade had indeed lower leading edge loading and apparently less suction surface separation. This standard of blading was incorporated into the core compressor for the NASA/GE Energy Efficient Engine programme.

A rotor with increased hub work gave a small additional improvement in efficiency when run with the restaggered stator. The tests including local aerofoil profile modifications (rotor with aft loaded tip sections and stator with forward loading at the hub) showed no particular improvement.

An additional objective of this study was to determine whether single stage experiments can prove useful in predicting the performance of multistage machines. Significant differences were found, particularly in the pressure rise characteristics, due to the fact that an inlet guide vane does not perform like a compressor stator at off-design conditions and the downstream stages in the multistage configuration exert a stabilising influence on the first stage. The radial profiles of loss coefficient were also notably different to those deduced from the buried stages.

All the designs discussed so far have been stacked on radial lines which is generally convenient from the stressing point of view. However, inclining blades introduces a radial component of blade force which might be expected to be beneficial if the flow has to move radially inwards. The most thorough analytical treatment of this was by Smith and Yeh (1963) but there was an experimental study by Breugelmans Carels and Demuth (1984) using a rectilinear cascade. A NACA-65 blade with 30 degrees camber and 28.9 degrees stagger was inclined at a number of angles by rotating about the stagger line. This introduced both dihedral and sweep, in the aerodynamic sense, so the sweep was removed by rotating about an axis normal to the plane formed by the leading edge and the chord.

The results showed a large increase in loss near the acute suction surface endwall corner and a reduction in the obtuse corner. Breuglemans et al went on to test a blade which was stacked such that the suction surface at both ends intersected with the endwall at an obtuse angle, and this gave the disappointing result that although their was a small reduction in loss in the corners, the mid-span region of the blade was substantially worse.

4.5 The Role of CFD

Over the past decade, advances in Computational Fluid Dynamics (CFD) which have taken advantage of faster, cheaper and larger computers, have opened up the possibility that some of the fundamental compressor research could be performed numerically. Dunham (1986) reviewed the history and state of the art of the application of CFD in improving turbomachinery.

Reliable 3D Euler solvers have been available for a number of years and these have had a significant impact on turbomachinery design. Morgan (1984), for example, reports a 1% efficiency increase in the RB211-524 turbine efficiency which was achieved by changing the stacking of the nozzle guide vane, guided by an inviscid time-marching code which was based on Denton (1982).

However, Euler solvers do not give information on loss production and, by definition, cannot model phenomena such as the destruction of inlet skew by viscous shear, which has already been mentioned as a significant action in core compressor stages. Attention has moved towards solvers of the Reynolds averaged Navier Stokes equations, such as the pressure correction Moore Elliptic Flow Code (MEFP) (Moore, 1985) and the time-marching Dawes code (Dawes, 1986).

4.6 Discussion

The flow in a core compressor is particularly complex and the evolution of viscous and 3D effects through the multistage machine cannot yet be calculated by today's most sophisticated software running on the fastest computers in a time scale which is practicable for compressor design. However, the combination of experiment backed up by calculations of increasing complexity have resulted in a comprehensive 'engineering' description of the 3D flow phenomena, identifying blade/endwall interaction as a major source of loss.

Dihedral and sweep do not seem to have been widely applied, possibly as they may complicate mechanical design.

One common tack which has emerged is aimed at minimising secondary flows by arranging the primary flow angles across the span in such a way that the secondary streamwise vorticity is cancelled. However, the problem with application of techniques based on inviscid secondary flow theory is that no one has managed to evolve a reliable method capable of accurately tracing secondary flow through more than one or two bladerows (perhaps with the exception of Brochet and Falchetti, 1987). A method capable of routine application to a multistage core compressor does not appear to have emerged in the published literature.

The second popular strategy has been to develop spanwise whirl laws which are aimed at making the best of the near wall flow conditions, with some minor adjustment in the free-stream. VWD was the first of these, followed by IVP and indeed many current compressors at Rolls-Royce have a stator exit whirl law which is related to these. However, designing more work into the ends pulls more flow through these regions and though this may unload the blading, there may be a mass meaning disadvantage on efficiency.

GE and SNECMA have used a similar throughflow design technique, but have opted for a whirl law which reduces flow through the ends and may result in more favourable incidences onto a following rotor. A successful implementation appears to be worth about 0.5% of efficiency.

The Rolls-Royce IEB technique has shown the largest published performance increment on an already competitive machine (1.0%) but experience has been mixed particularly with low-reaction multistage compressors. The Pratt and Whitney CDA-II appears to result in blading which looks similar, particularly for low reaction stages.

These results have demonstrated that the end-loss can be reduced but there has been a notable lack of success on occasions with what appear on the surface to be very similar strategies. This suggests that great care must be taken in their implementation and further work is justified to add to the fundamental understanding of their behaviour.

5. DETAILED BLADING DESIGN

5.1 Introduction

Tests of four sets of blading are described in this thesis: two free-vortex (datum and end-bent) and a corresponding low reaction pair. There was a time-lag of about eighteen months between the design of the two sets, with the free-vortex being the earlier. Both designs were carried out within the available corporate design system and were influenced by the then prevalent design philosophies.

The free-vortex series were based on an aerodynamic non-dimensional design point which is typical of mid-stages of the HP compressor found in the RB211 series of three shaft engines. The blading has conventional Double Circular Arc (DCA) thickness distributions on circular arc camber lines. It was a straightforward design drawing on correlations of cascade data.

The low reaction blading, though complementary to the free-vortex series, was more closely tied to an engine project. The design point of the low-speed stage was chosen to be that of stage 7 of the Batch 1 V2500 HP core compressor, and the aerodynamic and geometric non-dimensionals were copied, as closely as was practicable. Furthermore, the blade profiles were manipulated such that they mimicked the non-dimensional surface velocity distributions predicted for the high-speed blading.

Both end-bent stages were defined using the procedure proposed by Freeman (1978) which is summarised in Appendix 5.1.

Briefly, this involves taking a conventional set of blading, making an estimate of the axial velocity profiles likely to be present in practice throughout the stage, then modifying the camber and stagger of the blades within these boundary layers to design for constant tangential blade force, avoiding spanwise variations in incidence. The objective is to minimise radial pressure gradients which could drive spanwise flows and result in streamsheet twisting.

The theory is 'viscous' only in the sense that it recognises the axial velocity profiles, which are the result of viscous effects. The calculations of blade force are completely inviscid, so the theory has become known as Inviscid End-Bends (IEB).

Other important assumptions are that: the method was conceived as a 'small perturbation' theory, which was not, therefore, fed-back into the axisymmetric throughflow; it takes account of clearance flows only indirectly, through the separate boundary layer calculations. It does not differentiate between wall boundary conditions such as rotating or non-rotating surfaces.

5.2 Corporate Computer Aided Design System

The corporate design system FACE (Fan and Compressor Engineering) covers the entire suite of software from preliminary design calculations, through to the detailed aerodynamic design and geometrical specification of blade sections. Fig 5.1 shows a block diagram of the relevant part of the system, highlighting programs involved in this design.

5.3 DCA Zero α0 Series

5.3.1 Non-Dimensional Parameters

Both the straight and end-bent free-vortex blading have the following parameters in common.

Tip Diameter	48.0"	
Hub/tip ratio	0.85	
Aspect Ratio	1.5	
$\Delta H/U^2$	0.34	
Va/U	0.55	
s/c rotor	0.988	mid-height
s/c stator	0.956	
Rotor t_c/h	0.01	
Rotational Speed	1000 rpm	
No. rotors	59	
No. stators	61	

5.3.2 Annulus

A plot of the design annulus is shown in fig 5.2. The final flowpath ahead of the compressor was not finalised at the time that the blading was designed, so fig 5.2 is not an accurate model of the intake.

The aerodynamic details for the datum blade were computed using a blockage annulus calculated by Green's method, which predicted a blockage of approximately 5% at stage 3. The extent of end-bend on a blade is directly proportional to the local boundary layer thickness, so even a repeating stage design would require slightly different blades for each stage as the boundary layer grows through the machine.

It was assumed that the blockage would remain sensibly constant through the compressor so that only one set of blading would have to be manufactured for use in all four stages. Provision was made for the addition of spoilers upstream of the compressor so that the inlet boundary layers could be artificially thickened up to the predicted asymptotic level if necessary.

5.3.3 Conventional Blading

The aerodynamic design details have been derived assuming axisymmetric flow, including the effects of streamline curvature, in the Q263 throughflow program, using 9 radial calculation stations. The basic level of loss was predicted by preliminary design software (AC162) but crudely modified to acknowledge the existence of the boundary layer by raising the loss at the blade-ends and lowering it at mid-height. The blades were designed to produce a pressure ratio of 1.013 per stage, which corresponds to an overall temperature rise of 4.7K at the estimated adiabatic efficiency of 90.8%. The high space chord ratios (typical of research compressors at the time) led to fairly high hub diffusion factors, particularly on the rotors (0.43). Although the machine runs at low speed, the vector triangle geometry and aerodynamic loadings are representative of most RB211 HP compressors.

Both the rotors and the stators were bladed to a 'nominal' four degrees of negative incidence, although there was some radial variation in incidence on both blades. The large negative incidence results in a higher camber than is strictly necessary for this duty so that the blades could be restaggered should the compressor be short of work when tested. This was a reasonable precaution in view of the uncertainty over the exhaust system losses of the new facility.

The blades were stacked to produce 9 equi-spaced, plane sections. The design details of the rotor and stator are shown in table 5.1.

5.3.4 Inviscid End-Bends (IEB)

The blades were designed on 13 streamlines, with a concentration towards the blade-ends.The desired alterations to the air angles were calculated following the procedure described in the Appendix, but automated by using 'exec-files' which operated on the output of the throughflow in a database retrieval program (G653). The blockage was assumed to be a constant 5% throughout the machine, which corresponds to displacement thicknesses of 0.090 ins on either wall. With the assumption of a 5th power law velocity profile, the modifications covered 0.54 ins at both root and tip of the 3.6 inch long blades.

The blade geometry was derived by interpolating between the existing sections of the datum rotor and stator then applying the calculated gas angle changes directly to the metal angles. The finished blades have 5 mm fillet radii round the bosses and the rotor tip clearance is .036 ins. There is clearly no point in insisting on the calculated angle changes in these regions of the span. The convention adopted was to use the calculated angles up to the start of the fillets, and then follow a straight line, tangential to a curve of delta angle against radius up to the ends of the blades, thus enabling the milling tool to follow a smooth path as it cuts the master aerofoils.

The manufacturing constraint of 0.006 in thickness for leading and trailing edges forced a thickening of stator leading edge, root and tip, and rotor leading and trailing edges at the tip.

Rather than allowing the thickness to decay to the minimum allowable, the edge radii were thickened such that the normal thicknesses for the blades with end-bends are the same as for the datum blades. The procedure used was firstly to rotate the plane sections to look along the blade edges (in G367, the stacking program). The lean of the blade-ends, relative to the free-stream, (θ) can then be measured (fig 5.3) and the leading edge radii increased by:

$$\text{New radius} = \text{Old radius}/\cos\theta$$

Both the radial distribution of aerofoil maximum thickness and chord were unchanged from the datum designs and the sections were again stacked on their centroids. Each had 49 plane sections with 5 mm separation over the free stream, and 1 mm separation at the ends to give fine definition of the end-bends.

Plots of the blade angle distributions are given in figs 5.4 and 5.5 with the datum blade angles for comparison. The geometric changes can be summarised as increases in stagger and camber at the blade ends of both rotor and stator. Full details of the end-bent rotor and stator are given in tables 5.2 and 5.3.

5.4 Controlled Diffusion Low Reaction Series

5.4.1 Preliminary Design

The low reaction blading was based on stage 7 of the V2500 compression system reported by Dunster(1985). In detail the design of each stage of this machine is subtly different.

However, stage 7 is representative of the basic philosophy and is the most highly loaded. Also, the hub to tip radius ratio is similar to the LSRC at this point. The high speed stage contracts from 0.853 at rotor inlet to 0.876 at stator outlet, compared with the constant 0.85 for the LSRC.

The mid-height non-dimensional geometric parameters, which were interpolated from Dunster's plots, and the aerodynamic loading parameters taken from Waymont (1985) can be summarised as:-

	Rotor	Stator
Aspect Ratio	1.58	1.55
s/c	0.772	0.825
$\Delta H/U^2$	0.45	
Va/U	0.70	
$\alpha 0$	19.2°	
tc/h	0.015	0.018

The free vortex blading described above has lower non-dimensional loading than the proposed low reaction design. For a given annulus area, the power required to run the rig is proportional to

$$Va/U \times \Delta H/U^2 \times U^3.$$

Running the free vortex blading for long periods of the design speed of 1000 rpm had already caused trouble with the DC motor controller resulting in a significant loss of rig running time. Subsequent testing had been done at 900 rpm. The low reaction design speed was fixed at 875 rpm, so the power required was then only about 20% greater than that absorbed by the datum blading.

The preliminary compressor design program (AC162) was again used to estimate the blade losses and establish the approximate design point and scantlings to give the required non-dimensional duty. As a first pass, the annulus wall boundary layer blockage was assumed constant across all 4 stages at 9%.

The increase in blockage over that assumed for the free vortex design was mainly the result of increasing rotor tip clearances from 1% to 1.4% and introducing a clearance of 1.8% at the shroudless stator hub. The blockage factors were about the same as those assumed in the V2500 stage 7 design. Since the hub/tip ratio is also similar, the end-bends covered a similar proportion of annulus height.

The radial profile of loss predicted by AC162 was then adjusted to give the same radial distribution as had been assumed for the V2500 blade (Waymont, 1985). The mass-meaned loss of the profile was set at the level of the unmodified, AC162 losses.

AC162 was finally re-run with the new loss profile supplied as input to give a final estimate of gas angles and the likely deviation angles. A first pass at the blading geometry was obtained by adopting an incidence of -4° and a circular arc camber line.

The annulus height of the LSRC is constant at 3.6 ins, so the constraint of the V2500 aspect ratios fixed the chords at 2.28 ins and 2.32 ins for rotor and stator respectively. To match the V2500 space/chord ratios, the rotor and stator blade numbers should have been 78 and 73 respectively. In fact, 72 stators were chosen to offer the possibility of changing the number of inlet struts to 18 (an exact divisor of 72). In the event, the front bearing support structure, which contains the 20 struts, was not changed to keep down the costs of the modification.

The blade numbers differ slightly from the V2500 values in mainly because of the differences in hub/tip ratio, since

$$NOB = \frac{\pi AR}{s/c} \frac{1 + (h/t)}{1 - (h/t)}$$

The number of rotors and stators in each stage of the LSRC is constant to preserve axisymmetry. The number of IGVs was chosen to be the same as the number of stators. The aspect ratio of the IGV was chosen to be smaller than the stator value (chord increased) resulting in a s/c of 0.67 which gives the vanes adequate overlap. Provision was made for altering the IGV stagger by $\pm 5^\circ$ without the need to dismantle the rig.

5.4.2 Throughflow

Annulus

The estimated axial projections of the blading were developed into a preliminary annulus diagram, shown in fig 5.6. The annulus includes the area contraction across the struts between stations 4 and 5. The RAE Lag Entrainment boundary layer within the throughflow program (Green, Weeks and Brooman, 1973) was used to estimate the growth of the boundary layer up to rotor 1 entry (plane 8). The stations upstream of the contraction did not model the intake shape, as the low velocities in the approach to the airmeter throat were known to create problems for the boundary layer code.

The boundary layer development through the compressor was estimated using a semi-empirical, mean-line method (VN08) which has correlations to take account of blade loading, geometry and clearances. The new method was calibrated against experimental data from the free-vortex version of the rig.

VN08 predicted the development of thick boundary layers for the low reaction compressor, which were close to separation, particularly at stator root. The average blockage through the study stage calculated by VN08 was about 10%, slightly greater than the preliminary assumptions. The predicted blockages were smoothed slightly in the axial direction, but essentially became the blockage annulus which was used as input to the throughflow program.

Inlet Conditions

Mass flowrate was set so as to achieve the required Va/U at stage 3. The compressor is designed to pass a flow of 28.6 lb/s on a standard day at 875 rpm.

Blade Loss

The radial profiles of loss coefficient deduced from the shape of the V2500 design values and the AC162 levels were used for the repeating rotors and stators.

Data on IGV losses is relatively sparse. The radial profile of loss measured downstream of the HP9 IGV, which achieves 21° of turning with a space chord ratio of 1.2, was used directly. This was perhaps an underestimate of the losses, as the space chord ratio is higher, but the low deviation, supercritical style profiles planned for the LSRC should have generally lower loss than the HP9 C5 aerofoils.

Blade Turning

Blade turning was specified by rotor pressure ratio and stator exit gas angle ($\alpha 4$). The rotor pressure ratio was adjusted slightly from the AC162 value to give the desired $\Delta H/U^2$ at stage 3.

The V2500 stator exit angles had a slightly parabolic radial profile, from about 18.5° at the wall to 19.2° at mid height. This appeared to be a function of the curvature of the streamlines in the high speed case, which had a non-uniform axial velocity profile. The LSRC blading was designed for a constant $\alpha 4$ of 19.2°.

5.4.3 Aerofoil Modelling Principles

The objective of this exercise was to model the non-dimensional flow conditions through stage 7 of the V2500 compressor in terms of the relevant non-dimensional compressor loading parameters and by paying particular attention to the predicted behaviour of the blade suction surface boundary layers.

A simple geometric scale of the high speed aerofoils would not have produced the desired result. Fig 5.7 shows a comparison between the calculated suction surface velocity distributions, at low and high Mach number, for a DCA stator. The low speed case has significantly less diffusion and a lower acceleration rate at the leading edge. The general geometry modification in the scale from high to low speed can be deduced from this plot. The profile must be thickened, to raise the peak velocity, and the chordwise camber distribution altered to reduce the effective aerodynamic incidence (ie reduced leading edge loading).

This type of transformation is well known in classical aerodynamics. The combination of Gothert's rule with the Incompressible Similarity law for isolated aerofoils has the result:

> "The dimensionless pressure distribution, lift coefficient and moment coefficient will be the same for compressible and incompressible flow if the profiles are affinely related in such a way that the compressible profile is smaller in camber ratio, thickness ratio and angle of attack by the factor $\sqrt{1-M_\infty^2}$ "
>
> Shapiro (1953)

Although the turbomachinery case is mathematically more complicated, one might expect the same general result: to produce similar behaviour in an incompressible flow, the profile should be thicker and have more camber than in a compressible flow.

5.4.4 Aerofoil Detailed Design

The practical approach to the profile design, in the absence of a 'direct' design tool, was to apply the factor $\sqrt{\frac{1-M_{LS}^2}{1-M_{HS}^2}}$ to the high speed maximum thickness to chord ratios, maintain the s/c and aspect ratios and then use a Prescribed Velocity Distribution (PVD) like option within the blade-to-blade analysis program (VP33, FINSUP) to converge on the desired suction surface velocity distribution. The starting profiles were generated by a generic supercritical aerofoil generator (VP99).

The 'trial and error' nature of the design was particularly time consuming, involving nested iteration: initially to produce a section which would achieve the required turning; secondly to alter the suction surface velocity distribution to achieve the design intent.

A major difficulty lay in defining a starting aerofoil which was suitable for small modifications with FINSUP. VP99 is a correlation of high speed profile parameters, based on several 'pure' supercritical designs. With experience, it was possible to modify the default camber and thickness distributions using options within the correlation program to generate reasonable starting profiles. Initially, the root and tip sections for the IGV and stator, and root, mean and tip sections for the rotor, were defined by this method. Intermediate streamline sections were then generated by interpolation using G367, the stacking program. These were rotated to give the correct exit angle, and then adjusted in camber and thickness distribution, using the design mode of FINSUP, to give the required velocity distribution. Finally, the boundary layer behaviour and section deflections were checked using FINSUP.

IGV

The geometrical properties, defined in the preliminary design, were not altered during the detailing phase. Small modifications to the VP99 - generated profiles were necessary to produce the smooth velocity distribution shown in fig 5.8. The suction surface has a controlled diffusion, similar to a decelerating bladerow, while the pressure surface is designed to have a gently accelerating flow. The loading is concentrated over the first half of the vane, while the last 15% of chord is designed for zero lift and an accelerating flow, intended to keep the deviation angles small. The sections were evaluated over $\pm5°$ incidence to ensure that the vane could be restaggered should the expected nearly zero deviations not materialise.

The vane was specified to manufacture on 20 plane sections, centroid stacked without leans.

The OGV is downstream of all the measurement planes and is only required to straighten the flow before the exhaust and throttle. In order to keep costs down, the IGV aerofoil was used upside down in the annulus (ie the boss was attached to the opposite end) rather than producing another bladerow. It should perform the duty as both IGV and compressor exit profiles are designed to be a constant 19.2°.

Rotor and Stator

The V2500 HP7, mid-height maximum thickness to chord ratios input to the profile generation program were multiplied by the factors defined above; 1.8 for the rotor and 1.56 for the stator. The stator s/c ratio was maintained at the V2500 levels but the rotor s/c was slightly reduced due to the late addition of an extra blade (to 79 off) for the convenience of rig mechanical design.

The 'converged' velocity distributions are shown in figs 5.9 and 5.10 and the suction surface non-dimensional distributions compared with the V2500 basis in figs 5.11 and 5.12. Both low speed sections reproduce the suction surface overall diffusion rates, but the low speed rotor has initially greater diffusion just downstream of the peak velocity location, while the turbulent boundary layer is thin. The FINSUP integral boundary layer calculation predicted the rotor boundary layer, with this modification, to be in slightly better condition than the stator. The peak shape factors are just below 2.0 in the turbulent boundary layer towards the trailing edge, at mid-height, for both blades. The suction surface boundary layers on both high-speed blades appeared to be close to separation.

Both rotor and stator were specified for manufacture on 20 equispaced plane sections, stacked on their centroids without leans. Full details of the IGV, rotor and stator are given in table 5.4.

5.4.5 Application of End-Bends

Inviscid end-bends were applied to the rotor and stator as for the DCA series. The VN08 run, which predicted the blockage annulus shape, was used to provide the displacement thicknesses. At the time of this design, the exec files has been consolidated onto the departmental library as 'EBEND', also invoked from the database retrieval program, G653. A 1/5 power law velocity profile was assumed, as for the high-speed V2500 design.

The datum blade plane sections were rotated into their flattest attitude (to avoid re-entrant sections after end-bending) then the air angle deltas were applied to the inlet and outlet metal angles. The procedure used was to rotate the section about the centre of gravity through the exit angle delta, recamber the leading edge to achieve the inlet angle change, blending in at 70% of chord, then redatum the section to the centre of gravity.

The new plane sections were then transferred to the NGS (Numerical Geometry System) for viewing along the leading and trailing edges at their likely die attitudes. The final trailing edge end-bends had to be moderated for manufacture on both rotor and stator. The end-bend deltas were reduced to the limit that the trailing edge line at the blade ends made a maximum angle of 50° to the die horizontal, then the remaining sections altered to give smooth changes through the end-bend regions. Intermediate sections were added to both rotor and stator to control the inevitable abrupt changes in surface shape.

Full details of the end-bend blades are given in table 5.5 and the inlet and outlet blade angles are compared with the datum version in figs 5.13 and 5.14.

5.5 Discussion

The calculation routines for the end-bend angle changes in sections 5.3.4 and 5.4.5 produced deltas of 40° to 45° at the annulus walls. Had the blades been forged, as high-speed blades are, the maximum allowable delta would have been 15° which would have been a poor approximation to the desired angles. Also, some of the end-bend effect is lost in the fillet radii, which are necessary on all blades for mechanical integrity.

It was explained in section 5.3.4 that the blade leading and trailing edges had to be thickened because of the end-bend lean. The only way of doing this was to grossly thicken the edge radii on the streamline sections. Since this is the edge that the air actually 'sees', the performance may be compromised.

With hindsight, a better approach to the design of the repeating low-reaction rotor and stator would have been to code a simple, incompressible, PVD design tool. However, the time allocated to the design was based on experience in the routine use of existing methods. Also, the ground rules for the modelling process were not well defined initially, but evolved during the design process.

The design process was slow due to the time required to change between programs VP99 and FINSUP during multiple iterations to define a starting profile with the required turning. This could, perhaps, be short-cut in the future by using the profiles designed in this exercise to produce a revised correlation which would improve the first guess. Also, the runs of FINSUP with the boundary layer switched on were extremely slow to converge.

Although the velocity triangles are similar, there is an appreciable rise in static temperature across the high speed rotor which causes a corresponding rise in the local speed of sound. At mid-height, this results in a compressible (V2500) rotor inlet to stator inlet Mach number ratio of 1.08 compared with the incompressible (LSRC) value of 1.04. This implies that the transformations to be applied to the rotors and stators will differ, even for this low reaction example (or indeed the 50% reaction case). This means that the changes in the tangible properties of the blading, such as incidence and camber, will not be the same for the rotor and stator. The changes in the geometrical properties in the high to low speed transformation are consistent with those reported by Wisler (1984) for the CF6 LSRC model.

The aerofoil scaling procedure (sections 5.4.3 and 5.4.4) is strictly only relevant at the design point. Therefore, the readacross of conclusions concerning changes in stall margin from low to high speed must be treated with some caution. It should be noted that the blade cambers and incidences were not considered to be primary features of the design, consistent with current Compressor Office design practice. The values given in the tables are deduced from the final profile shapes.

6. LOW SPEED COMPRESSOR RIG AND INSTRUMENTATION

6.1 Introduction

The emphasis in the LSRC programme is to produce aerodynamic data which is representative of the flow in today's high pressure compressors. The performance of any turbomachinery stage is influenced strongly by others around it. In order to obtain meaningful results from a model test which will read across to the engine environment, it is necessary to have a sophisticated simulation of the likely stage entry conditions for a single stage model, or to test a complete multistage compressor.

Four stages were chosen for the present study: the first two are used to set up repeating flow conditions of velocities and whirl angle, the third is the study stage and the fourth provides the appropriate downstream static pressure field.

The rig has been designed with mechanical flexibility in mind so that blade staggers, tip clearances and the location of traverse instrumentation can easily be varied. Also, the arrangement of stationary blading, struts and the inlet and exhaust systems is such that the flow is likely to be periodic, that is repeating in a pitchwise sense.

The target for radial resolution of flow properties was 1% of blade height, while the circumferential sweeps should cover at least one and a half stator pitches. The area traverse mechanism permits measurement of several axial planes simultaneously.

This chapter starts with an explanation of how the rig was sized followed by a description of the mechanical design. The fixed and traversable instrumentation is then described and with an outline of the logging system including the reduction of data. Some features of the internal flow are discussed and finally the experimental uncertainty is assessed.

6.2 Sizing

The rig was initially conceived as a model of a mid-stage of a free-vortex, zero $\alpha 0$, HP compressor, typical of the RB211 family of engines. The real machine has 5 or 6 stages, depending on the version, varying in hub to tip radius ratio from front to back. A mean level of 0.85 was chosen for the low speed model.

The major technical motivation for a large scale rig is the potential ease of making detailed measurements. A useful side effect is that the power necessary to achieve a given chordal Reynolds number (Re) is inversely proportional to tip diameter, so the low speed rig has a small power requirement.

It is generally accepted that the limit of resolution of miniature, pneumatic instruments is about 1mm, so a target of 1% resolution of annulus height implied that the blade height should be around 100mm. In fact, a tip diameter of 48 inches was chosen giving a blade height of 3.6 inches (91.44mm).

6.3 Mechanical Arrangement

Fig 6.1 shows a photograph of the rig and fig 6.2 shows a simplified cross-section through the compressor and drive. Figs 6.3 and 6.4 show more detail of the zero $\alpha 0$ and low-reaction compressor working sections.

The intake to the compressor is a logarithmic spiral airmeter, which is cast in fibreglass. When the rig is running, it is shielded by a large debris guard which is covered by glass-fibre filter cloth.

The supports which hold the main bearings are machined from solid steel plate. The front support has 20 evenly spaced struts which are uncambered aerofoils with C4 thickness distributions. These are followed by a 5% area contraction which helps to thin the wall boundary layers and mix out the strut wakes ahead of the compressor.

Spoiler rings can be added to the inner and outer annuli should the thickness or profile of the inlet boundary layers need to be altered. These have not been used to date.

The outer casing comprises a series of cast, aluminium rings which are located by dowels and held to the end plates by a series of axial tie bolts. The rings holding the stators are radially bored and counterbored for the blades, while the casing segments over the rotors have static pressure tappings and mountings for the traverse gear. The rings over rotors 3 and 4 (and the IGV and OGV, where fitted) are supported on grease lubricated, rolling contact bearings and can be driven circumferentially by a computer controlled stepper motor. This provides the tangential movement of the traverse gear relative to the fixed stator vanes which is necessary for an area traverse. These features are illustrated in fig 6.5.

The rings holding the rotor blades are machined from forged and rolled aluminium rings, while the spacers, which form the seal faces under the stator shrouds, are made from aluminium castings. Adjacent rings are located by dowels and the drum is backed by steel end plates. The rotating assembly is held together by axial tie bolts, similar to those of the stationary casing.

The rear support is machined from solid steel and has 16 uncambered struts. On builds with non-zero $\alpha 0$, an outlet guide vane (OGV) is fitted to straighten the flow before it passes through the exhaust system. This is not a particularly crucial bladerow as all measurements are taken ahead of it. It is merely there to avoid high incidences onto the exit struts which could raise the total loss of the outlet system and preclude operation at the high flow end of the compressor characteristic. The OGV was made by casting a set of IGV's with the boss on the opposite end of the aerofoil, which avoided the cost of a completely new die.

The double, angular contact, ball bearing at the rear provides the axial location, while the front bearing is a simple roller. Both bearings are grease lubricated.

Downstream of the rear support struts, the air diffuses along an axisymmetric duct. The compressor back pressure is controlled by a motorised, conical throttle (not shown in fig 6.1). There is also an electrically operated flap valve which can be actuated to drop the compressor quickly out of stall.

The rig is driven from the rear by a 160 HP (120 KW) DC, compound wound electric motor, which drives through a 5:1 epicyclic reduction gearbox and a strain gauge torquemeter. The torquemeter is protected from a seizure of the drive shaft by shear pins. The drive system is mounted within the inner annulus line at the rear of the compressor. Services to the system are passed through faired tubular supports towards the rear of the exhaust ducting, where the gas velocities are at their lowest, thereby minimising intrusion into the flowpath and spoiling axisymmetry.

6.4 Blading Manufacture

The rotor and stator blades are low-pressure die cast in LM24 aluminium alloy. The critical part in the process is the manufacture of the tooling. Producing the bulk castings is a straight forward operation, although the scrap rate is particularly high due to the tight aerofoil profile tolerances.

The two portions of the the die which form the aerofoil surfaces are milled either directly from 'negative' surface co-ordinates or copy milled from master aerofoils. Both methods have been used successfully, but direct machining of the dies is preferable as it eliminates surface form errors which would be introduced in the copying process. Machining cusps are removed from both the mating and aerofoil surfaces by hand, then the die is hardened.

The use of circular bosses, primarily dictated by an initial requirement to adjust blade staggers, is actually quite convenient for manufacture as it allows blades to be cast in their flattest attitudes (optimum for casting) without having to worry about the orientation of a ring platform segment.

The first set of blades was designed to allow 'easy' variation of clearances. These blades had undersize bosses and a gap between the end of the aerofoil and the annulus line. They sat on plastic shims in the recessed holes, the idea being that the shim thicknesses could be changed to alter tip clearances. In practice, it was necessary to fill the gaps around the aerofoil bases and form fillet radii by hand with 'elastic plastic padding' to achieve a satisfactory aerodynamic finish. This was extremely time consuming and subsequent sets of blades had bosses cast to size with fillet radii around the aerofoil bases. An example of the two standards is shown in fig 6.6. Future clearance changes will require traditional cropping. To return to tight clearances it will, of course, be necessary to buy extra sets of blades. In view of the time element in smoothing the annulus and applying fillets with the original blades, the cost of this option is likely to be reasonable in comparison.

The finished blades are checked by a technique of 'eyelashing' several plane sections (typically 5) to produce 10x life size plots which are compared against computer generated drawings from the blade file. The accuracy of the aerofoil surfaces produced by this process has been found to be very high. Most of the problems encountered have been in producing acceptable leading edges. In order to achieve small edge radii, it is necessary for the porous metal to exit from the die along these edges. This leaves a rough finish which must be tidied by hand. It is clearly difficult to ensure consistency with so many blades in the four-stage set. In spite of this difficulty, it has been possible to maintain 0.006 inch edge radii on the DCA sections and acceptable profiles on the thicker, low reaction blading.

The blades were tested to determine their ultimate strength in the 'radial' direction. They were found to break at a tensile stress of 10 tons/sq.in which is roughly 25x the nominal stress due to rotation. In practice the aluminium blades have proved to be durable and tough. There has not been any incidence of chipping or flaking, even on the extremes of the end bends.

6.5 Instrumentation

Fig 6.7 shows a schematic of the measured parameters and their locations and fig 6.8 shows a block diagram of the logging system.

All the pressures are recorded through a series of five computer controlled, oil-free scanivalves, each equipped with a 0 to 0.5 psi Druck transducer. Initially oil-lubricated scanivalves were chosen, but these were abandoned after oil in the pressure lines was found to cause scatter in the data.

The fixed instrument data is recorded automatically by the test-bed PDP 11/34 computer via a 16-bit analogue to digital converter. The data is corrected on-line for start of test zero offsets and converted into engineering units. Transducer drifts are logged throughout testing and are stored along with the rest of the data on floppy discs for subsequent off-line analysis.

The settling and scanning time combined is roughly 25 seconds per data point, resulting in a time of typically 35 minutes for a tangential traverse sweep. At each traverse point, the compressor speed, torque, inlet depression and moving wall statics are logged along with the yaw probe pressures.

6.5.1 Steady State Performance

The compressor overall performance is expressed in terms of a 'mean stage' characteristic which is deduced from measurements of the total (for the zero $\alpha 0$ builds) or static (low-reaction tests) pressure rise across the four stages, mass flowrate, torque and rotational speed.

The total pressure rise is measured across inlet and exit rakes. For non-zero $\alpha 0$ builds, the rakes are downstream of the IGV's and upstream of the OGV's and they are set at the mean, nominal flow angle (19.2 degrees). There are 4 rakes on both inlet and exit planes approximately 90 degrees apart, but circumferentially verniered across IGV or strut wakes.

Each rake has 12 limbs with a concentration towards the annulus walls. The tubes have Kiel shrouds to make them reasonably insensitive to the incident flow direction. Their angular characteristics were checked in a wind tunnel.

The overall static pressure rise is measured across the same planes as the total with sets of wall tappings. Both inlet and exit have 5 tappings distributed around the circumference, again verniered across a stator pitch.

The rakes are verniered in an attempt to account for the pressure wakes from the upstream bladerows. The wall statics need to be distributed across a pitch also to give a reasonable mean of the static pressure field which arises from the potential flow around the aerofoils.

Compressor shaft speed is measured using an optically actuated, digital tachometer and the torque by a strain gauge torquemeter. Input power is then calculated from the product of torque and speed.

The measurement of mass flowrate uses a pair of standard NPL pitot-static probes situated at mid annulus height in the bellmouth. The intake has been traversed by the NPL probes to determine a discharge coefficient.

There are arrays of verniered wall statics between each bladerow. These are recorded to give a monitor of axial matching.

All pressures are logged relative to the ambient static pressure, Bar, which is obtained regularly throughout the day from a mercury column barometer. The inlet total temperature is recorded by 5 thermocouples which are distributed around the debris guard.

6.5.2 Pneumatic Area Traversing

Each of the three planes around the study stage can be circumferentially traversed on both port and starboard sides of the rig simultaneously. Six computer controlled traverse units can be positioned on these rings at four locations on each (two upstream and two downstream of the rotor). The fixed casings over the first two rotors allow radial traversing at five circumferential locations which vernier across a stator pitch. Each traverse gear carries a pneumatic wedge probe which can be moved radially into and out of the casing and rotated about their own axes. The drive is provided by two stepper motors mounted on the gear. Fig 6.9 shows a photograph of the traverse gear.

The brackets which locate the traverse gears are fitted with static pressure tappings which are displaced by a small amount tangentially from the traverse probes. These provide measurements of the wall static pressure field for comparison with values measured in the flow by the traverse probes.

Movement between the area traverse data points is controlled by the test bed computer. The probes are moved in 0.004 inch steps with automatic backlash compensation. They can be positioned radially to ± 25 μm (0.02% height) circumferentially to ±0.1 mm (0.3% pitch) and to ± 2 min rotationally about their own axes.

The initial design of pneumatic probe was based on the NASA 60 degree combination wedge described by Glawe, Krause and Dudzinski (1968). The CIT head does not have a thermocouple, but an additional 'heel' static pressure tapping was included in the hope that it would prove to be insensitive to yaw angle. The probes were calibrated for pitch and yaw in Rolls-Royce wind tunnels.

The heel static was found to be heavily depressed and required a large correction factor in comparison with the side statics, so it has not been used.

A more serious problem emerged when the probes were traversed radially across the intake to gain confidence in their suitability for use in the more complex compressor internal flows. An interaction between the end-wall boundary layer and the probe stem disturbed the indicated static pressure over an unacceptable proportion of the annulus. A new probe was, therefore, developed with a smaller stem blockage and wedge angle (23 degrees) based on a standard Rolls-Royce design. The new probe was carefully checked in the known tunnel flows and was found to give a reliable indication of static pressure much closer to the walls. The traverse results from the two LSRC probes (and the Rolls-Royce 'parent') are compared in fig 6.10. Fig 6.11 shows photos of the two wedge probe heads together with a cobra (not used in the present experiments). Figs 6.12 and 6.13 are engineering drawings of the two wedges.

Although the computer control system was designed with auto-yaw in mind, this feature has not been used in practice due to the relatively long settling times resulting from low pressures and long lines between probe head and transducers.

The probes are set at a fixed yaw angle for each sweep during traversing, chosen to be aligned with the expected absolute flow angle within a 10 degree tolerance, and 'indicated' total and static pressures are recorded. These are then corrected using tunnel-measured calibration curves as described in Appendix 6.1.

All traversing is carried out with fixed throttle setting and at constant corrected speed.

The radial location of the probe's bottom total pressure tapping is the nominal immersion of the traverse. The side statics, and hence the deduced yaw angles, are radially displaced by 2.5% of annulus height from the nominal.

The only manual intervention with the test data at this stage was to check that the probe calibration software had behaved reasonably through the stator wakes. The calibration curves are not really valid for the highly sheared wake flow and the deduced angle tends to overshoot on one side of the wake and undershoot on the other. Each sweep was checked to see that the plus and minus overshoots cancelled. If they did not, the yaw angles were re-averaged with a linear variation assumed between the measurements at either side of the wakes.

To date, calibration tests have been carried out on No 9 airflow rig on Sinfin A Site. This comprises a closed tunnel with a 6 inch diameter working section. The flow condition is set by throttling the tunnel to a total to static pressure ratio corresponding to the desired nominal Mach number. The probe is rotated through ± 30° relative to the nominal zero in 5° increments. The five pressures are recorded manually on water manometers. The nominal zero is set by balancing the probe side statics in a flow with high dynamic head, then angles relative to the zero are measured on a fixed protractor. All the CIT probes have been calibrated at 0.1 Mn only, which is characteristic of the flow in the LSRC. Calibrations for the 3 probes used in the present study are given in fig 6.14.

6.5.3 Buried Stage Flow Phenomena

The four-stage compressor was designed to set up a repeating flow, to be representative of the environment in which axial stages are obliged to operate in multistage machines.

Fig 6.15 shows a comparison between total pressures measured downstream of stators 2 and 3 on the DCA build with end-bends. The local pressures at each radial and circumferential location have been normalised by the average pressure in each plane.

Very close agreement is obtained, suggesting that the flow features have not changed significantly across the the third stage, giving confidence that the measurements around stage 3 are a good representation of those in a buried stage.

Figs 6.16 and 6.17 show data from a circumferential sweep of the pneumatic probes downstream of the rotor and stator of the study stage from the DCA datum build. There is a large pitchwise variation in the static pressure, as the probe detects the potential influence of the stator row behind it, such that the dynamic head varies by about 40% from the mean. These plots underline the need for careful verniering of several casing static pressure measurements across a stator pitch if they are to be used to make deductions on stage performance.

6.5.4 Circumferentially Meaned Blade Element Performance

The average total pressures and yaw angles were derived from circumferential traverses in the absolute frame of reference for each radius traversed. The total pressures and yaw angles obtained were then entered into an axisymmetric throughflow program (Q263) together with the total mass flow derived from the airmeter and compressor rotational speed. The program then calculated the static pressures, air angles and velocities throughout the stage, both in the stationary and rotating frames of reference, satisfying continuity and radial equilibrium.

Analysing the results in this way at face value gave an unreasonable sharing of the stage loss between the rotor and stator: approximately zero rotor loss and a corresponding high stator loss. Other investigators, such as Cyrus (1986), have experienced this problem, which lies in the interpretation of total pressure and yaw angle data taken at rotor exit with stationary pneumatic traverse instruments.

The flow field behind the rotor in the relative frame has been measured by both pneumatic traverse instruments rotating with the rotor (Wisler, 1984) and by stationary, fast-response instruments (Hirsch and Kool, 1976, for example, from which fig 6.18 is reproduced). The rotor exit absolute total pressure is reasonably constant in the pitchwise direction. The rotor wake relative total pressure deficit appears as a large perturbation to the absolute flow angle, repeating at rotor passing frequency. This fluctuating flow is time-meaned by the stationary traverse probe. The amplitude of the angle variation is greater than half of the included angle of the probe, so it is likely to be stalled for at least part of the cycle and, therefore, into the non-linear part of its characteristic. The main problem is, however, attenuation in the relatively long lines connecting the probe head to the transducer in the scanivalve box. The procedure adopted to adjust the rotor exit conditions was as follows: the total pressure profile was altered in level such that the static pressure derived from the total mass flow within the throughflow agreed with the measured wall statics, then the angle was adjusted to give the average level of temperature rise deduced from the overall characteristics, less a 1% deduction for windage and bearing friction. This procedure places most confidence in the measured wall static pressure: it was felt that this quantity would suffer the least unsteady, pitchwise variation out of the available, measured parameters.

6.6 Experimental Uncertainty Analysis

6.6.1 Single Sample Uncertainty Theory

In most cases considered here, results are calculated from product strings such as

$$R = x_1^{\,a} \cdot x_2^{\,b} \cdot x_3^{\,c} \ldots$$

Where $x_i = \hat{x}_i \pm \delta x_i$

$\hat{x}_i$ - is the best estimate

δx_i - is the likely uncertainty interval

If the uncertainty in each variable is described by the same odds (which it is if the interval is expressed as $\pm\ 2\sigma$) then the following formulae apply to the uncertainty in the result (Taylor, 1982)

likely error

$$\delta R = \left[\left(\frac{\partial R}{\partial x_1} \delta x_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} \delta x_2 \right)^2 + \ldots \right]^{1/2} \qquad 6.1$$

relative uncertainty

$$\frac{\delta R}{R} = \left[\left(\frac{a\,\delta x_1}{x_1} \right)^2 + \left(\frac{b\,\delta x_2}{x_2} \right)^2 + \ldots \right]^{1/2} \qquad 6.2$$

Equation 6.1 is the general expression, and 6.2 refers to the product strings. 6.1 was shown by Kline and McClintock (1953) to assess the uncertainty in most engineering applications with good accuracy.

The conditions to be satisfied for the method to be rigorously applied are: each variable must be independent and conform to a Gaussian distribution and the odds of each uncertainty must be the same (ie 95% for $\pm\ 2\sigma$).

6.6.2 Overall Performance

The following expressions can be written down, providing that the pressure rise is a small fraction of the inlet pressure:

$$\dot{M} \propto \left(\frac{P\Delta p}{T}\right)^{1/2} \qquad 6.3$$

$$\frac{\Delta H}{U^2} \propto \frac{\tau}{\dot{M} N} \qquad 6.4$$

$$\frac{V_a}{U} \propto \frac{\dot{M} T}{N P} \qquad 6.5$$

$$\eta \propto \frac{\Delta P \dot{M} T}{\tau N P} \qquad 6.6$$

$$\frac{\Delta P}{\rho U^2} \propto \frac{\Delta P T}{N^2 P} \qquad 6.7$$

Equation 6.2 can be applied to all the expressions 6.3 to 6.7 to provide equations for estimating the fractional uncertainty in each result. For example:

$$\frac{\delta\eta}{\eta} = \left[\left(\frac{\delta\Delta P}{\Delta P}\right)^2 + \left(\frac{\delta\dot{M}}{\dot{M}}\right)^2 + \left(\frac{\delta T}{T}\right)^2 + \left(\frac{\delta\tau}{\tau}\right)^2 + \left(\frac{\delta N}{N}\right)^2 + \left(\frac{\delta P}{P}\right)^2\right]^{1/2}$$

The uncertainty interval for each result has been established from a series of fixed instrumentation scans from two low reaction builds of the LSRC near the peak efficiency point.

$P = 29.856 \pm 0.005$ in Hg $\qquad \delta P/P = 0.00017$

$T = 282 \pm 0.4$ K $\qquad \delta T/T = 0.00142$

$\Delta p = 51.5 \pm 0.6$ mm H_2O $\qquad \delta\Delta p/\Delta p = 0.0117$

$\tau = 529.2 \pm 1.0$ ft lb $\qquad \delta\tau/\tau = 0.00189$

$N = 841.2 \pm 0.2$ rpm $\qquad \delta N/N = 0.00024$

$\Delta P = 502.8 \pm 1.5$ mm H_2O $\qquad \delta\Delta P/\Delta P = 0.00298$

for total pressure rise

Inserting these values of fractional uncertainties into the equations deduced by applying 6.2 to 6.3 to 6.7, the fractional uncertainty of each result has been found, for example, to be:

$$\frac{\delta\eta}{\eta} = \pm\, 0.007$$

0.2 to 0.3 %

6.6.3 Traverse Data

The bias error on probe calibration is minimised by setting up at high dynamic head. However, the coefficients are inaccurate to some degree because of manometer reading errors during the testing.

The three coefficients are defined as

$$C_S = \left(\frac{P_T}{p_t} - 1\right) \Big/ \left(\frac{P_i}{p_i} - 1\right)$$

$$C_T = (P_i - P_T) \,/\, (P_T - p_T)$$

$$C_{yaw} = (S_2 - S_3) \,/\, (P_T - p_T)$$

Where $\quad p_i = (S_2 + S_3)/2$

The above expressions can be partially differentiated with respect to the component measurements to give influence coefficients such as

$$\frac{\partial C_S}{\partial p_i} = f\,(P_T,\, p_T,\, P_i,\, p_i)$$

It is generally accepted that water manometers can be read to ± 1 mm H_2O. With this estimate of the uncertainty in each individual measurement, equation 6.1 can be used to estimate the likely errors in C_S, C_T and Cyaw which are introduced by the limitation on reading the manometers.

For example:-

$$\delta C_S = \left[\left(\frac{\partial C_S}{\partial p_i}\,\delta p_i\right)^2 + \left(\frac{\partial C_S}{\partial P_i}\,\delta P_i\right)^2 + \left(\frac{\partial C_S}{\partial p_T}\,\delta p_T\right)^2 + \left(\frac{\partial C_S}{\partial P_T}\,\delta P_T\right)^2 \right]^{1/2}$$

Care must be taken to keep consistent units in the calculation of influence coefficients, which have the same units as $(\text{pressure})^{-1}$. Table 6.1 summarises the influence coefficients and sample results from Cranfield probe E003 at 0.1 Mn, with units in $(\text{mmH}_2\text{O})^{-1}$. The table shows that the C_S and C_T coefficients are weakly dependent on yaw angle, as ds, although small compared to S_2, is present in the influence coefficients.

The size of the uncertainty was a surprise initially. It is, however, magnified by the low Mach number level, as one would expect, since as Mach number increases, the fixed uncertainty in the gauge pressure (± 1 mm) becomes a smaller proportion of each reading.

The next step is to find the knock-on effect of the calibration curve uncertainties in the probe data correction algorithm.

Assuming that the iteration described in the Appendix is simply a one pass correction routine, the influence coefficient can be deduced by partial differentiation of the equations

$$p_{T\,New} = P_T \Big/ \left[C_S\,(1 + C_T)\left(\frac{P_i}{p_i} - 1\right) + 1 \right]$$

$$P_{T\,New} = p_{T\,New}\left[C_S\left(\frac{P_i}{p_i} - 1\right) + 1 \right]$$

The one pass assumption is reasonable for a probe that is within ± 10° of the actual flow direction. Full convergence typically takes three passes, but the bulk of the correction is on the first pass. The resulting influence coefficients are given in Table 6.2 along with typical data taken from a sample point in a stator three exit area traverse using probe EØØ3. Also shown in the table are values of influence coefficient found by systematically varying the input to the full, iterative data reduction algorithm. The agreement is good enough to justify the use of the 'one pass' assumption.

Taking typical values for the Eϕϕ3 coefficients from Table 6.1 as

$$C_T = -0.02 \pm 0.02$$

$$C_S = 0.83 \pm 0.02$$

and again assuming ± 1 mm H_2O as the precision of the basic P_i, S_2 and S_3 measurements, the likely uncertainty in the true corrected total and static pressures is ± 2 mm H_2O and ± 3 mm H_2O respectively, using equation 6.1. The details of the calculation are shown in Table 6.2.

The deduced yaw angle influence coefficients are also given in Table 6.2. Although the nominal correction is small, the uncertainty in the value turns out to be large in comparison at ± 0.5°. Again, if the calibration uncertainty could be removed, the accuracy would be improved to ± 0.3°.

The accuracy of the basic measurements (P_1, S_2 and S_3) is actually limited by the calibrations of the individual transducers, which use water manometers, rather than the logging process.

6.7 Discussion

Bellmouth dynamic head is the primary component in the mass flow calculation and, as seen in section 6.6.2, it has an important impact on all the derived parameters. It is vital that this measurement is controlled between successive tests as misinterpretation of the flow can cause misleading results.

The dynamic head in the flow measurement plane is smaller than it might have been. The rig has an area contraction across the inlet struts, designed to mix out the wakes and thin the annulus wall boundary layers ahead of the compressor. If the complete flowpath had parallel walls, the dynamic head would be about 36% greater and the uncertainty in flow would reduce to ± 0.36% with corresponding improvements in the efficiency and work accuracy.

There may be some scope in future blading designs to increase dynamic head by running the compressor at higher power. This would also increase chordal Reynolds numbers which would help blading design. However, there may be practical difficulties in increasing the torque and speed demanded of the motor with the thyrister controller and in brush wear.

The most widely published low speed, four-stage rig is run by General Electric in Cincinatti. They have accumulated at least twenty-five years of experience with their facility and it is worth comparing their stated accuracy (Wisler, 1984) with that achieved at CIT.

	GE	CIT
N	0.25%	0.02%
τ	0.07%	0.19%
Va/U	0.15%	0.6%
η	0.15pts	0.6pts

For GE to achieve their stated efficiency accuracy, they must know the instantaneous speed to much greater precision than 0.25% of design speed. Assuming that their speed measurement has similar precision to that at CIT, the major difference is caused by the estimated error in the inlet dynamic head, which affects the mass flow. GE claim an accuracy of ± 0.025% of a full scale pressure reading of 50 inches H_2O, which is roughly 0.3 mm H_2O: half that estimated for the Cranfield case. The dynamic head in the measurement plane is about 25% greater than at CIT, so one would expect a fractional uncertainly in flow about 40% of that estimated for the CIT facility, which largely accounts for the difference in the flow coefficients and efficiencies. Although the GE torque measurement is claimed to be more precise, the mass flow uncertainty is dominant in both cases. GE use a steam turbine as prime mover which may produce a smoother drive than the CIT electric motor with its pulsed controller, hence a smaller torque uncertainty.

This chapter has demonstrated the difficulty of making high precision measurements of blade element performance in a low speed, small dynamic head environment, with simple, pneumatic instrumentation. The problems of spatial resolution and probe blockage in the small scale, high speed case are traded for difficulties associated with small absolute and difference measurements. The LSRC has achieved basic flow symmetry both locally (pitch to pitch) and circumferentially (port to starboard), a spatially accurate and reliable traversing system which enables the flowfield to be defined with a manageable amount of data in reasonable elapsed time.

The traverse probes can, unusually, be traversed simultaneously across the three planes comprising the study stage, so there are three instruments and transducers to calibrate separately. Errors in the calibrations are of first order importance in the results from each plane rather than being a second order bias across all results.

7. AERODYNAMIC RESULTS

7.1 Introduction

Results are included from four builds of the four-stage compressor featuring the blading described in Chapter 5. Two builds were of a zero $\alpha 0$ stage with DCA profiles, one conventional and one with end-bends, both with shrouded stators. There are two builds of the low-reaction compressor, one conventional and one with end-bends, but with Controlled-Diffusion (CD) profiles and cantilevered stators.

The tests were carried out over several years and were not conducted in the logical sequence presented here. During this period, the instrumentation was extended in scope to include aerofoil surface static pressure tappings, so coverage is not complete for all builds. Where fitted, hub, mid and tip correspond to 9, 50 and 91% height on the DCA builds and 4, 50 and 88% on the CD examples.

The layout of this chapter can be summarised as follows. Firstly, the results from each build are presented in turn, in the order: overall characteristics, traverse data and, where available, aerofoil surface statics. The overall performance data is based on gross torque, uncorrected for bearing friction and windage, and flow coefficient is calculated using the metal annulus area. Only sufficient plots to describe the main features of the traverse data are presented, but full output for the study stage (stage 3) from both peak efficiency and near-stall operating conditions is available in Appendix 7. All data has been scaled to ambient conditions of 14.7 psia and 288K.

The blade surface static pressures were measured in the second stage and are presented relative to ambient inlet pressure and normalised by a dynamic head based on rotor tip speed.

Finally, the experimental findings are drawn together and discussed in section 7.6.

7.2 DCA Datum

7.2.1 Overall Performance

Overall characteristics are presented in fig 7.1 at a corrected speed of 950 rpm. It was measured with a rotor tip clearance of 1.0% of annulus height and had shrouded stators. The blading exceeded its design work but appears to be well matched with peak efficiency occurring near the design flow coefficient.

A logical definition of 'surge' margin is the amount by which a stage can be throttled before it stalls whilst running at a constant speed. For a high speed machine, this would be expressed by the percentage change in the exit flow function ($M\sqrt{T2}/AP2$) between the working line and stall. In this low speed case, with small stagnation temperature and pressure rise, this can be approximated by the change in non-dimensional flow coefficient. Defining the working line to intersect the characteristic at a flow coefficient of 0.55, the compressor had a stall margin of 26%.

For a repeating stage of 'ideal' blading, where the deviations of both rotor and stator and the axial blockages are not a function of the operating point, the relationship between non-dimensional work and flow would be a straight line passing through $\Delta H/U^2 = 1.0$ at zero flow, with gradient $-(\tan\alpha_2 + \tan\alpha o)$. So, for this zero αo stage, the gradient would be $-\tan\alpha_2$. The mean stage vector triangle geometry implies that the rotor relative exit gas angle should be about 50.2°, so the gradient of the work coefficient with flow would be about -1.2, which it is in fig 7.1 for values of Va/U above 0.65. As the flow coefficient reduces, the work characteristic gradually becomes less steep as both blockage and deviation increase, until just below the peak efficiency flow where the gradient begins to rise.

7.2.2 Area Traversing

Area Traversing took place at three flow coefficients: 0.455, 0.550 and 0.637 corresponding to near stall, peak efficiency and a high flow condition towards choke. The circumferential sweeps consisted of 82 data points across one and a half blade pitches, with concentration of points through the wakes. 14 radial heights were traversed. Detailed analysis of this data has concentrated on the two lowest flow coefficients.

Fig 7.2 shows contour plots of total pressure at stator exit for the three operating conditions, giving an appreciation of the data coverage. At peak efficiency, the wakes are generally thin and 'two dimensional', thickening slightly towards the annulus walls adjacent to the suction surface. The near stall point has significant thickening of the wakes on the suction side of the aerofoil all the way up the span, with the hub showing most distress. Also visible are wakes from upstream stators, situated on the pressure surface side of the free-stream. These are most obvious in the near stall traverse. The end-wall boundary layers seem to be thin at all operating conditions supporting the idea of a profile rather than a wall-stall.

The peak efficiency and near stall operating point data have been reduced by circumferentially averaging total pressures and yaw angle on the three traverse planes (stage inlet, rotor 3 exit, stage 3 exit) and using these to define loss and turning for an analysis run of the axisymmetric streamline curvature program (Q263). Appendix 7.1 contains full output from the throughflow analysis at both operating conditions.

Fig 7.3 shows the rotor relative gas angles. The blade is exhibiting 2D behaviour between 30 and 90% of annulus height at peak efficiency. Near stall behaviour is similar although the 2D nature is confined to 50 to 90% of height. The incidences over this region at peak efficiency are close to the nominal design values of -3 to -4 degrees. Incidence at peak efficiency increases sharply between 0 and 5% and 90 and 100% of height by 6 to 8 degrees.

The radial profile of deviation at peak efficiency is exactly equal to the design assumption between 30 and 50% span and close to design right out to 90% height. At each end of the rotor there are significant departures from design caused by three-dimensional effects. Towards the casing, there is a region of large underturning caused by the tip leakage vortex, whilst at the hub there is evidence of a classical secondary flow passage vortex. The tip underturning is similar at both flow coefficients, while the deviation close to the hub increases significantly towards stall. So much of the annulus exhibits behaviour which is quite clearly not 2D that the term 'secondary flow' does not seem particularly appropriate, though the flow phenomena are similar to those observed in cascade experiments with finite incoming boundary layers.

The radial profiles of loss coefficient are shown in fig 7.4. At peak efficiency the rotor has a fairly constant loss across the free-stream region, with minimum value of 3 to 4% of inlet dynamic head. The loss has a minimum between 85 and 90% of height before increasing rapidly towards the casing due to the loss associated with the tip clearance.

Near the hub, the similarity with deterministic secondary flow observed in cascades is again present: the location of peak loss is 10% of height away from the wall and there is little loss indicated very close to the hub. At near-stall, the radial profile is similar to that at peak efficiency, but is approximately double in level. Also, the location of peak loss is further from the wall at 20% of height and the indicated loss drops below zero in the region of overturning.

The stator incidences across the free stream at peak efficiency shown in fig 7.5 are also lower than the nominal values by a degree or so, but the increase towards both walls is greater than that experienced by the rotor. Over the inner 30% of annulus height the incidence increases progressively to 20 degrees above the nominal, while the increase at the casing is about 15 degrees over the same radial distance. The stator experiences a much larger swing in incidence than the rotor as it moves towards stall: incidence is increased by 10 to 15 degrees across most of the stator span, a result of the high reaction design.

The radial profiles of stator deviation exhibit the now familiar under and overturning near the casing. The blade is close to achieving the design deviation of just under 15 degrees across most of the annulus at peak efficiency and deviations are everywhere 2 or 3 degrees higher near-stall. Near the hub, however, there is an apparent anomaly where the deviation continues to rise towards the wall. With shrouded stators, one might have expected to see similar behaviour to that at the casing. Instead, the stator behaves either as if it has substantial overtip leakage (the flow pattern looks very similar to that seen in the relative frame at the rotor tip) or that the sections are stalled. The loss (fig 7.6) contains the same anomaly, even down to the minima at 10 to 15% of height at peak efficiency. The mid-span, peak efficiency loss of about 5% of inlet dynamic head is a little higher than that deduced for the rotor and the loss core near the casing is substantially larger than its equivalent at the rotor hub. At near-stall, the stator loss coefficient increases substantially across the whole span, most significantly over the inner half of the annulus.

Fig 7.6 suggests that there are very high losses in the stator hub region, approaching 50% of inlet dynamic head near stall.

The apparent lack of secondary flow near the hub may be the result of a lack of streamwise vorticity in the stator inlet conditions. The measured total pressure in this plane shows little evidence of boundary layer deficits. In fact, the total pressure rises towards the wall at the near-stall operating point. It should also be noted that the stator platforms are contained within the stator axial chords and the probe always approaches a running inner wall.

Fig 7.7 shows the stage exit total pressure profiles. Wisler (1980) published results from tests of a similar shroud box geometry on the GE low-speed, 4-stage compressor.

7.3 Inviscid End-Bends (IEB)

7.3.1 Overall Characteristics

The non-dimensional, mean stage characteristics for the IEB blading are compared to the datum build in fig 7.8. Though there is more scatter in the end-bend data, there appears to be no change in the value of peak efficiency. The IEB stage seems to have matched at a slightly lower flow, such that peak efficiency occurs at a flow coefficient of 0.53. The value of work at a flow is reduced, also giving the appearance of a stage with somewhat lower flow capacity than the datum.

In judging the 'surge' margin, it seems fair to assume that the flow capacity of the end-bent stage would have to be restored to that of the datum, in practice by a small re-stagger. Adjusting the characteristics of the end-bent stage such that it would achieve a pressure rise coefficient of 0.31 at a Va/U of 0.55, the stall margin would be 31%, a modest improvement of 5% on the datum.

The behaviour of the work coefficient with flow is very similar to that of the datum blading, as is the rate of reduction in stage efficiency with both increasing and decreasing flow away from the peak efficiency operating point.

7.3.2 Area Traversing

This build had similar traverse coverage to the DCA datum above and was measured at nominally the same flow coefficients. Fig 7.9 shows corresponding contour plots of total pressure at stage 3 exit for the two lower flow coefficients, and these can be compared to those from the conventional blading in fig 7.2. The plots are very similar to the earlier ones at peak efficiency in terms of wake thickness. They are distinguishable as being from end-bent blades by the tangential displacement of the wakes over the last 10% of span near both inner and outer walls, indicating that the trailing edges of these stators do not lie on a radial line. Near stall, there is again thickening across the entire span, much as for the datum (fig 7.2) with a suggestion that the hub of the end-bent blade is slightly further away from stall than the datum.

The peak efficiency and near stall operating point data have again been reduced by throughflow analysis and full output is given in Appendix 7.2. The general character of the stage behaviour is much as for the datum case so this is not reviewed in detail in isolation. Of more interest is the comparison between the conventional and end-bent blading.

The IEB rotor inlet metal angle proves to be a good match to the relative air inlet angle near the tip, though too much of the blade has been modified at the hub, as shown in fig 7.10. The radial profile of inlet gas angle is similar for both rotors.

The exit conditions are notably different, however, at both the hub and casing. At the hub of the end-bent stage, the gas follows the trend of the 2D deviation until closer to the wall, though there is still overturning in the last 5% or so of annulus height. The rotor outlet angle is two to three degrees lower in spite of the small increase in the exit metal angle. The reduced deviation could be due, in part, to a favourable axial velocity ratio occurring across the end-bent rotor hub, which is not present in the datum case, though this is only significant in the innermost 5% of annulus height. At the tip, the end-bend results in an increase to the exit metal angle, so the blade has increased stagger but similar camber to the datum. The clearance flow results in a very similar profile of deviation over the outer half of the blade, so there is an increase in the exit gas angle which goes in sympathy with the change to the metal.

The radial profile of rotor loss coefficient in fig 7.11 is interesting when viewed alongside the observed differences between the deviation of the conventional and end-bent rotors. The bulge in loss noted at 10% of annulus height for the conventional blade is not present in the end-bent data, where the blade has reduced deviation. The profile loss is reasonably constant across most of the inner half of the annulus until it falls to be negative very close to the hub. At the tip, the end-bent blade has the higher loss coefficient. It also has smaller inlet dynamic head, though this alone does not account for the increased loss.

Turning to the stator gas angles at peak efficiency, in fig 7.12, the end-bent inlet metal angle is particularly well matched to the incident gas in both hub and casing boundary layers. The inlet air angles are around 10 degrees higher near the tip for the end-bent stage and 2 to 3 degrees lower near the hub. Across mid-span, the inlet air angle is similar in both cases.

The exit gas angle is almost identical for the conventional and end-bent stages, as is the deviation, since the zero $\alpha 0$ stage had no change to the exit metal angle. There seems to be a small radial redistribution of loss coefficient, shown in fig 7.13, with the end-bent blade having slightly more loss between 10 and 20% of height, but slightly less between 70 and 95% of height.

Comparing the stage exit profiles of total pressure in fig 7.14, the end-bent stage appears to be more hub biased than the datum, which is reasonably symmetrical about mid-span.

The blade element performance comparison near stall suggests that the behaviour has been little changed by the application of end-bends. The rotor inlet gas angles are very similar for the two standards of blading, so the end-bend benefits from reduced incidence in the hub. The end-bent stator also has reduced hub incidences, and this extends much further along the span than the region of end-bending, as shown in fig 7.15. Fig 7.16 shows the pressure loss coefficients for the two stators near stall. The reduced near-wall incidences of the end-bent blade near the hub seems to have resulted in lower loss over this region such that its loss is more uniformly bucket shaped than the datum, which has high hub loss. This is consistent with the contour plots discussed earlier (figs 7.2 and 7.9).

7.3.3 Aerofoil Surface Static Pressures

The aerofoil surface static pressure distributions are shown for the end-bent blade in figs 7.17 and 7.18. They are presented relative to the compressor inlet pressure and are normalised by a dynamic head based on rotor tip speed. The rotor boundary layers seem well attached at hub, mid and tip at both flow coefficients.

Peak suction near the tip is closer to the trailing edge than at other heights, an influence of the tip leakage vortex. The measurements show substantial retained lift at the last chordwise tapping. This partly reflects the fact that tappings could not be placed particularly close the trailing edge with such thin aerofoil sections, but is basically a fundamental feature of DCA profiles on circular arc camber lines.

The rotor shows increased leading edge loading at the near-stall operating conditions, but there is still monotonically increasing static pressure on the suction surface suggesting that if the flow separates at this condition, then this occurs downstream of the last tapping. One feature of the rotor pressure distribution which does look odd is the region of apparently constant static pressure on the pressure surface close to the trailing edge near the tip.

The stator in the zero αo case has less lift than the rotor which is very clear in fig 7.18, although it does have a much lower inlet dynamic head from which to produce its pressure rise. At peak efficiency, the mid-height tappings suggest that the flow is well-attached, while those near the ends are showing a tendency to be separated even at this near-design flow coefficient. Near stall, all sections suffer increased leading edge loading and even the mid-height tappings now show almost constant pressure along the suction surface towards the trailing edge. The behaviour of these surface static measurements corresponds well with that seen in the stator exit contour plots in fig 7.9 and the blade element performance deduced from the traverse data.

7.4 Low Reaction CD Datum

7.4.1 Overall Performance

Fig 7.19 shows the mean-stage, overall characteristics of the low-reaction CD datum blading. It was measured at a corrected speed of 850 rpm and had rotor tip and stator hub clearances of 1.4% and 1.8% of annulus height respectively. This stage has exceeded its design point work; the excess could be accounted for by the contribution of friction and windage together with an overestimate of blockage. The relationship between work coefficient and flow is almost linear, with gradient close to the ideal calculated from the mean-stage values of $-(\tan\alpha 2 + \tan\alpha 0)$.

The pressure rise coefficient is also different in character to the DCA, zero $\alpha 0$ curves. The CD datum exhibits little of the 'roll-over' seen in fig 7.1 before terminating in a rather abrupt stall. There is consequently only 2% reduction in efficiency between the peak and that at the stall point. The stage has a smaller stall margin (16%) compared to that of the zero $\alpha 0$, DCA datum, estimated by arbitrarily assuming that the working line would intersect the characteristic at a flow coefficient of 0.7.

A graphic illustration of the reduced surge margin was that the CD blading required a different start-up procedure to the DCA. The practice with the DCA builds was to leave the throttle fixed at a partially closed condition, corresponding to a traverse operating point, and to allow the rig to run up in speed from rest at this throttle setting. The CD blading, however, started up in rotating stall when the throttle was set for the peak efficiency operating point. This problem was circumvented by using the exhaust dump valve to increase the effective throttle area and to move the compressor working line away from stall.

The dump valve was closed once the compressor was part way towards the nominal operating speed, enabling the remaining run-up to be carried out along the peak efficiency working line.

7.4.2 Area Traversing

The conditions downstream of the Inlet Guide Vane (IGV) were checked with a 'skeleton' traverse, consisting of a mid-pitch radial traverse and hub, mid and tip circumferential sweeps at the peak efficiency flow coefficient. The radial traverse results are shown in fig 7.20. The yaw angle is close to the design intent of 19.2 degrees over most of the annulus height, so the blade has achieved its near-zero deviation, the exception being close to the hub where the fluid overturns. Very close to the wall the gas picks up a tangential component of momentum from the rotating drum, which is in the same sense as any passage secondary flow in this accelerating bladerow.

The total pressure is plotted relative to ambient in fig 7.20. The small, negative free-stream value represents the loss incurred across the intake debris filter, and is approximately 4% of dynamic head. The total pressure profile was converted to velocity using the casing statics and assuming radial equilibrium, and the axial component was integrated to determine the compressor inlet blockage for comparison with that assumed in the design. Hub and casing displacement thicknesses were 0.0385 and 0.037 inches respectively, corresponding to 1.0% and 1.1% of annulus blockage. The total blockage was two thirds of that calculated by the lag-entrainment boundary layer, which is a reasonable approximation, considering that the method makes no attempt to model the influence of the IGV, other than the drop in static pressure across it.

Area traversing took place at three flow coefficients: 0.617, 0.686 and 0.833. The low-reaction compressor has increased axial gaps between bladerows compared to the zero $\alpha 0$ version, so the traverse probes are situated further downstream of the blade trailing edges. Some mixing out of the wakes occurs before the measurement plane, so the pitchwise total pressure gradients are less severe. Circumferential sweeps of 35 points covering 1.2 stator pitches were found adequate to define the wakes and again, 14 heights were traversed at rotor 3 inlet, exit and stator 3 exit planes.

Fig 7.21 shows the stator exit total pressure contours for three operating conditions: near-stall, peak efficiency and a high flow condition. The peak efficiency plot shows thin wakes over much of the span, between 20 and 70% of height, where the behaviour could be described as 'two-dimensional'. Towards the casing, the wake thickens on the suction surface where any incoming boundary layer fluid accumulates, having been swept across the passage by classical secondary flow. At the inner end of the stator, the picture is dominated by the large core of low momentum fluid, which is the hub leakage flow, situated at roughly mid-pitch. The suction surface of the stator is 'clean' near the hub, and the wakes are locally no thicker than at mid-height. The peak total pressure contour does not extend right the way across the passage, even at mid-span. The wake from an upstream stator, which is almost, but not quite, mixed out, is visible next to the pressure surface at the peak efficiency flow coefficient.

The character of the flow does not change dramatically as the compressor is throttled, but the flow near the end-walls seems to be deteriorating faster than at mid-span as stall is approached. The casing corner stall penetrates farther across the pitch near-stall than at peak efficiency, though it does not seem to have progressed in the spanwise direction. The leakage loss core grows visibly as the flow is reduced, but it does not appear to change its pitchwise location. The suction surface inside the 20% of height dominated by the clearance flow remains clean at all operating conditions.

As for the DCA builds, the peak efficiency and near-stall data has been reduced by throughflow analysis and full output is given in Appendix 7.3.

The rotor relative gas angles shown in fig 7.22 suggest that the 'free-stream' region covers 30% to 70% of annulus height. Rotor incidence around mid-span is around one degree below the nominal design value at peak efficiency, but it increases by 8 to 10 degrees towards both walls as the axial component of velocity falls through the boundary layers. The rate of increase of incidence is somewhat greater at the hub than the casing. The radial profile of incidence is similar at the near-stall operating condition, with a general increase in level of 3 degrees across the free-stream. The rate of increase is less steep towards the walls than at peak efficiency.

The small rise in incidence at 20% annulus height in the peak efficiency data is due to the shape of the absolute total pressure measured at stator 2 exit which repeats downstream of stator 3 (fig 7.23). Fig 7.3 shows a similar effect on the incidence of the DCA datum rotor.

The radial profile of rotor relative exit angle changes little between the two operating points. There is a similar portion of 2D behaviour where the rotor overturns by about half a degree from the design deviation of 2.4 degrees. Towards the tip, over the outer 15% of height, there is a large increase in deviation due to the tip clearance influence such that the tip deflection is reduced to about 20 degrees in spite of the increased incidence. At the near-stall condition, the rising deviation starts slightly inboard of the peak efficiency location and rises at a reduced rate. The radial extent of the clearance contamination is greater than that seen in the DCA results, but similar as a proportion of the gap dimension (of the order of 10x the span).

At the rotor hub, both operating conditions have a classical secondary flow pattern, with a region of small underturning, peaking at 15% of annulus height, followed by overturning close to the wall. The secondary flow deviations are quite small in magnitude, 3 degrees at most, presumably as a result of the inlet skew which has already been seen in the profiles of relative inlet angle. The near-wall overturning is slightly greater at the near stall operating condition where there was a stronger profile of relative total pressure. The combination of high inlet angle and overturning leads to gas deflections which are locally 15 degrees greater than the design values.

The radial profiles of rotor loss coefficient, shown in fig 7.24, are also similar for the two operating points, mirroring the profiles of deviation. The minimum profile loss is approximately 6% of relative inlet dynamic head and there is a shallow bucket across the free-stream region. The deleterious influence of the clearance flow can again be seen in fig 7.24, where the loss coefficient rises rapidly towards the casing to a value of 20% of inlet dynamic head.

At the hub, the secondary flow pattern is apparent as the loss core is located at the point of peak underturning, some 15% of annulus height away from the wall. Very close to the wall, within the innermost 5% of height, there is a region of indicated 'negative' loss.

Despite the indication of excessive 2D loading parameters, the rotor appears to operate stably at the near-stall condition and is well behaved at peak efficiency.

The symmetries in the rotor and stator behaviour are even more striking for the CD datum than they were for the zero $\alpha 0$ DCA stage.

The stator is subjected to increased incidences towards both annulus walls (fig 7.25), 8 degrees at the hub and 6 at the tip, with the mid-height values about one degree below the nominal value. The bulk increase in level towards stall is fairly constant across the span at 4 degrees. The clearance effect on the stator hub flow in the absolute frame is very similar to the behaviour at the rotor tip in the relative frame of reference, with large deviations rising to about 20 degrees close to the wall. At the casing, there is again evidence of a rather weak passage vortex giving rise to around 3 degrees extra deviation some distance away from the wall. The maximum underturning occurs at 90% of height at peak efficiency, but about 10% further inboard near-stall. The underturning also seems to percolate almost to mid-span at the lower flow coefficient.

The stator mid-span loss coefficient in fig 7.26 is high at 7% of inlet dynamic head (as was the rotor). The radial profile shows the same features as the rotor, with a large loss associated with the hub clearance flow and a core of low momentum fluid sitting a finite distance inboard of the casing, the result of secondary flow at the fixed end of the stator. There is a particularly low value of loss at 15% of height at the peak efficiency condition. This seems most likely to be the result of a radial redistribution of some higher energy fluid from further towards mid-span at stator inlet. This feature is not as noticeable at the near-stall condition, but there is a minimum in loss at 20% height. There is a more obvious increase in stator loss at the lower flow coefficient than there was for the rotor, particularly in the clearance dominated, near-hub region and between 60 and 85% of span.

At the peak efficiency operating point, the velocity profiles from the throughflow have been integrated to give boundary layer parameters in both axial and tangential and streamwise and normal coordinate systems. Definition of the boundary layer edge is somewhat subjective. It was found in this case by superimposing design, inviscid axial velocity profiles and the actual, measured curves and the edge was defined to be the height at which the two lines diverged.

The results are summarised in table 7.1 together with VN08 predictions, run at the test levels of clearance. The program appears to have overpredicted blockage by about 2% which has contributed towards the excess of work noted in the overall characteristics.

7.4.3 Aerofoil Surface Static Pressures

This build had partial coverage of surface static pressures tappings. The rotor had both suction and pressure surface tappings, but only at mid-height. The stator had suction and pressure surface at mid-height, and suction surface only at hub and tip. The results are shown in figs 7.27 and 7.28 for both peak efficiency and near stall operating points, again relative to compressor inlet total pressure and normalised by a dynamic head based on rotor tip speed. The chordwise coverage is much improved relative to the DCA blading: the aerofoils were much thicker.

The rotor at mid-height shows apparently well attached flow at mid-height throughout the severe diffusion along the suction surface. At near-stall, there is a modest increase in loading at the leading edge consistent with the increased incidence observed in fig 7.22. Immediately following peak suction at both peak efficiency and near-stall there is a hint of a laminar separation bubble at approximately 22mm aft of the leading edge. The mid-height pressure distribution measured on the stator at peak efficiency shows it to be rather more highly loaded near the leading edge than the rotor. As incidence increases at the lower flow coefficient, a leading edge 'overspeed' occurs and while this does not appear to have caused the boundary layer to separate at the condition at which the measurements were taken, it does suggest that the stator is set closer to stall than the rotor, as the change in incidence with flow which the two blades experience is similar. This may indicate a degree of mismatching which could be the cause of the abrupt stall.

The stator hub and tip measurements imply increased leading edge loading relative to the mid-height conditions, and the overspeed worsens towards stall. The mid and tip pressure profiles are very similar in character, as one would intuitively expect for this almost constant section blade, while the hub shows peak suction to be moved axially rearwards. The remainder of the suction surface then experiences a much steeper diffusion, but it does not appear to cause separation. The movement of peak suction is similar to that seen at the rotor tip of the DCA, end-bent blades in fig 7.17. The hub of the DCA stator did not show any axial movement of peak suction, but rather a high leading edge loading, so it seems likely that it is suffering a corner-stall rather than being influenced unduly by any leakage under the part of the hub not covered by the shroud ring.

7.5 Low-Reaction Inviscid End-Bends

7.5.1 Overall Performance

The application of end-bends to the CD stage, shown in fig 7.29, results in a similar change to the mean-stage characteristics as that observed for the DCA datum in fig 7.8, though the effect is more pronounced, reflecting the larger changes to both rotor and stator geometry. The end-bent test was run at the same speed and with the same clearances as the CD datum.

The end-bent CD stage has a lower capacity than its datum. Peak efficiency is unchanged in level but it occurs at a lower flow coefficient. The end-bent blading stalls at the same flow as the datum, indicating an effective reduction in stall margin.

The end-bent results are obscured to a degree by axial mis-matching. The four stages with end-bends applied to both rotor and stator were tested behind the conventional inlet guide-vane, which had a constant exit whirl angle of 19.2 degrees, so the first stage had substantially less pre-swirl over two-thirds of the annulus height than did subsequent stages. The compressor never achieved the repeating flow condition, which is clearly illustrated by the axial distribution of reaction (defined as the proportion of stage static pressure rise occurring in the rotor) shown in fig 7.30 for the peak efficiency operating condition. The reaction of all four stages with the datum blading is reasonably constant and close to the design value of 58%, while the end-bend build exhibits an axially falling reaction. The first stage reaction is similar to that of the datum build, while the latter stages seem to be tending towards about 45%.

7.5.2 Area Traversing

Fig 7.31 compares the stator exit contour plots at the peak efficiency and near-stall points; fig 7.21 helps in their interpretation. The end-bent stator wake is displaced tangentially towards the casing as it follows the blade profile shape, but it does not thicken on the suction side of the passage as it did for the conventional stator in fig 7.21. This trend is also apparent at the near-stall point. At the hub, the large core of low momentum fluid in the leakage vortex is again visible in fig 7.31, but it is tucked under the wake on the pressure surface side of the passage. The situation barely changes towards stall.

Full output from the throughflow reduction is given in Appendix 7.4.

The rotor deviations deduced from fig 7.32 show surprisingly little difference between the conventional and end-bent blades, considering the substantial changes made to the geometry. They are almost identical between 30% height and the tip, but they do differ near the hub. While the conventional blade has under and over-turning, the deviation of the end-bent blade rises steadily towards the wall.

The rotor incidences for the conventional blade at peak efficiency are fairly uniform across the mid-span region, with about 5 degrees steady increase through the wall boundary layers. While the rotor inlet angles of the end-bent stage were increased to try and match the expected rise in the gas angles, the relative inlet air angle <u>falls</u> between 30% height and the hub and 70% and the casing. With the applied increase in inlet metal angle, this results in some large negative incidences, up to -20 degrees, near the annulus walls.

The end-bent rotor loss coefficient shown in fig 7.33 is nearly uniform between 15 and 70% height, though there is a small minimum at 50% height. Near the hub, the loss again becomes negative as it has for all the previous rotors. The region of high loss between 10 and 15% on the conventional blade is not present in the end-bent results. Towards the tip, there is a minimum at 80% height before the loss rises steeply towards the casing at a faster rate and to a higher level than that of the conventional blade.

The stator deviations deduced from fig 7.34 have a similar pattern to that of the rotor across the entire annulus, but the end-bent vane has a region of increased deviation between 10 and 20% height.

The rate of increase of the end-bent stator inlet metal angles is steeper than the measured rise in the conventional stator gas angles in fig 7.34. However, the end-bent stator experiences a much steeper rising gas angle so the match between metal and gas is coincidentally good for the end-bent blade.

The modification to the stator loss at the casing for the application of end-bends (fig 7.35) is similar to that already seen at the rotor hub. The peak seen at around 85% height in the results from the datum stator is smaller and closer to the casing in the end-bent build.

The end-bent rotor is unloaded across the mid portion of the span and it experiences higher loading at the ends, both in terms of diffusion factors and the static pressure rise coefficient. The increase in the loading parameters is the result of the substantial loss in relative inlet dynamic head experienced by the end-bent blade.

While the end-bent rotor loses inlet dynamic head, the converse is true for the stator and the blade ends are unloaded relative to the conventional stator.

A striking feature of the end-bent results shown in fig 7.36 is the change in axial velocity which occurs well away from the walls. The velocity increases by up to 6% at mid-height and falls towards both walls, by over 20% at the casing. The datum rotor has an axial velocity ratio close to one across most of the span. Although fig 7.30 showed that the end-bent compressor did not achieve a fully repeating flow condition, the stator axial velocity ratio in fig 7.36b is approximately the inverse of that of the rotor. This rising and falling of axial velocity is likely to persist however many stages are present. It is inherent in this implementation of end-bends, as discussed below.

The comparison of stage exit conditions in fig 7.37 shows that the concept of boundary layers and blockage must be treated carefully when such large end-bends are present. The exit total pressure profile has less of a defect in the end-bent case than the datum (particularly at the casing), the result of additional work done at the rotor ends which boosts the stator inlet total pressure over the inner and outer 20% of annulus height. However, the fluid has such a high swirl angle at stator exit in these near-wall regions that the axial component of velocity is much lower than it was for the datum stage, giving the appearance of thick boundary layers.

The traverse data for the end-bent low-reaction stage at the near-stall operating condition proved to be the most troublesome dataset to analyse. Though there is less confidence in the absolute levels of blade element performance for this case than for other builds, the trends indicated may still be useful. The reason for the analysis difficulty lies partly in the general unsteadiness of the flowfield, as this data was taken with the compressor closest to its eventual stall point, but also because the probes were set further off nominal angles throughout much of the traversing.

The changes in incidence and loading are as expected for the reduced flow coefficient with similar changes apparent for the rotor and stator in this low reaction stage. The stator inlet angle shown in fig 7.38 rises near-stall such that the flow is close to the tangential direction. Both the stator inlet and exit angles show reducing gradient as the casing is approached, and while deviations have risen between 5 and 20% and 80 and 95% height, there is little change across mid-span. The stator loss-coefficients in fig 7.39 are rising towards both walls at the near-stall point, more steeply near the hub.

7.5.3 Aerofoil Surface Static Pressures

The end-bent, low-reaction blading was fitted with surface static tappings at hub, mid and tip of both rotor and stator: the results are shown in figs 7.40 and 7.41 for both peak efficiency and near-stall operating points scaled and normalised as for previous builds.

Both rotor and stator pressure distributions at mid-height are similar to those measured on the datum stage (figs 7.27 and 7.28). At the tip, the rotor has peak suction further towards the trailing edge, as did the DCA end-bent rotor in fig 7.17, though this effect is most striking at the stator hub, where peak suction is almost at mid-chord. The rotor hub seems to have somewhat less lift than at other sections at both flow coefficients and both hub and tip sections show particularly low leading edge loadings consistent with operation at large negative incidences seen in fig 7.32.

The stator pressure distributions tie in well with the contour plots in fig 7.31, where the mid and near-tip wakes looked very similar in character. The pressure distributions are almost identical at these two heights at both operating conditions. Near the hub, the stator suction surface experiences a diffusion which is even more severe than that seen on the datum blade. The end-bent stator does not suffer the spikes seen at the leading edge of the conventional blade towards stall. Neither bladerow shows any obvious sign of separated flow at any height measured.

7.6 Discussion of Experimental Findings

7.6.1 Comparison of Datum Stage Overall Characteristics

The design point non-dimensional parameters of the zero $\alpha0$, DCA datum give it a characteristic which is steeper, thus inherently more stable, than that of this low-reaction stage. There are, however, other important differences which are likely to play a part in the reasons for the apparent lack of stall margin of the low-reaction compressor. The CD profiles were designed to have velocity distributions which result in the turbulent boundary layers being close to separation over much of the blade suction surfaces, with peak form-factors at about 60% of chord. The circular arc camberline aerofoils have peak form factors at their trailing edges. If stall is the result of turbulent separation, then the location of the peak form factors at 60% chord of the low reaction blading might imply sudden detachment over 40% of chord rather than a progressive creeping forward of the separation point from the trailing edge in the case of the DCA's.

With hindsight, looking back at the design of the low-reaction blading, the stators were arranged to have increased leading edge loading (figs 5.9 and 5.10) which is also apparent in the surface static pressure measurements (figs 7.27 and 7.28). This may be the root cause of the premature stall of the stator.

Another possible explanation is that the laminar separation bubble, which may occur at these low Reynolds numbers, bursts and fails to reattach as it encounters the strong deceleration following the peak suction surface velocity.

The controlled diffusion blades have achieved their predicted low deviations. The IGV had almost no deviation, the result of the long flat aft region with generally accelerating flow. One of the potential advantages of the more modern, CD profiles is that the design deviations are small compared to those of the circular arc camberline DCAs and there is consequently less scope for error in their derivation. As the controlled, suction surface diffusion should eliminate, or at least reduce, any separated flow, the calculation of the contribution of the viscous effects to loss and deviation using the integral boundary layer within the blade-to-blade code should be that much more accurate. It should be noted that the estimation of both blockage and deviation go hand-in-hand towards achieving the design point.

As is often the case with research which is strongly linked to the fast-moving engine project scene, there are rather too many differences between the DCA and CD tests to be confident of attributing differences in observed behaviour to either profile shape, choice of reaction or stator hub clearance. However, the fact that the measured peak efficiencies of the two datum stages are equivalent reflects well on the low reaction blading, as it had 0.4% greater rotor tip clearance and a 1.8% gap at the stator hub, and has substantially thicker profiles than the zero $\alpha 0$ DCAs. The low-reaction stage is also the more highly loaded.

The rather low level of absolute polytropic efficiency can be attributed to:-

a) low Reynolds number
b) lack of windage and friction accounting
c) large clearances for the low-reaction build

7.6.2 Compressor Reynolds Number

Both the DCA and CD series of tests were conducted at non-dimensional speeds somewhat 'off-design' for reasons related to the facility rather than the aerodynamics. The compressor was originally designed to run at 1000 rpm (a tip speed of 209 ft/s) which implied a chordal Reynolds number of approximately 300000. However, when the DCA datum blading was tested during the rig 'shakedown', problems were encountered with the DC motor's thyristor controller and speed had to be limited to 900 rpm.

As the present rig was considered marginal from the low Reynolds number point of view, characteristics were measured at corrected speeds of 800 and 1000 rpm during the shakedown with the DCA datum blading. The results are shown in fig 7.42; within the accuracy at which the data was recorded at the time, the non-dimensional, mean-stage characteristics were deemed to be unique giving confidence that the chosen speed (900 rpm) was sufficiently high.

The low-reaction design point represents a higher work and flow at a tip-speed than the zero $\alpha 0$ series. Rather than running the risk of incurring further problems with the electrical control system or motor, the design speed was set at 875 rpm which kept the power demand close to demonstrated experience. However, when tested, the rig exhibited substantial acoustic distress at the nominal design speed. The traversing schedule called for long-term operation at each throttle setting, so in view of the fact that the tolerance to fatigue of the alloy blading was unknown, the speed was adjusted to 850 rpm, which gave the blading the same chordal Reynolds Number as the zero $\alpha 0$ when running at 900 rpm.

This decision to run at low Reynolds number was not taken lightly. The preliminary design choice of 300000 was chosen carefully to be the smallest for direct readacross to engine compressors (typically 500000 to 1000000).

It could, of course, have been made larger by increasing the diameter of the rig or by running it faster. To have increased the diameter would have resulted in an extremely unwieldy rig, noting the 'square-cube' law. The existing hardware is about at the limit of what can be handled by one or two fitters operating in a campus environment. Running the rig faster was not particularly appealing from the aerodynamic point of view. As tip speed is increased, compressibility effects become more significant and there would have been an appreciable density change across the four stages, which could have caused some axial mismatching, given that the same blading was to be used in each stage to promote the repeating flow conditions.

The influence of Reynolds number on compressor performance has been reported by a number of investigators, such as Schaffer (1980) and Lawson (1953). To summarise the consensus of their results, loss and deviation change little with increasing Reynolds number above 500000 and they change rapidly as it reduces below 100000. The abrupt discontinuity which occurs somewhere between 100000 and 200000 and is felt to be associated with separation of the laminar boundary layer, which is likely to depend on surface roughness and incident turbulence levels.

Fig 7.42 at the time vindicated the use of low rotational speed. With hindsight it seems likely that the deviations measured will be representative of those at higher Reynolds numbers, while the level of overall performance may be slightly degraded.

7.6.3 Features in the Traverse Data

Many of the observed features seen in the traverse data support the existence of deterministic secondary flows, which are present in spite of the high gas angles near the walls at entry to bladerows:

under and over turning near fixed ends

blade pressure distributions modified by the action of the tip clearance vortex

the clearance influence on underturning.

This generally supports the Adkins and Smith model of spanwise mixing, suggesting that the Gallimore and Cumpsty pure turbulent diffusion approach needs development or at least ad hoc modification near the end-walls to deal with tip clearance and secondary flow.

The phenomena associated with tip clearance seem to read across from cascade experiments. There is significant underturning which affects the blades for a distance of up to 10x the gap along the span and the blade pressure distributions are altered by the leakage vortex just as Rains (1955) and Lakshminarayana and Horlock (1962) observed for blades in cascade.

The use of boundary layer codes to calculate the flow along the annulus walls in 1D preliminary design software must be approached with some caution. The contour plots of total pressure downstream of the conventional stages (figs 7.2 and 7.21) show apparently thin wall boundary layers, and much of what is accounted as blockage in the circumferentially meaned sense looks to be wake fluid, particularly in the case of the DCA datum.

Also, for the low-reaction end-bent blading, the choice of whirl distribution and radial equilibrium have resulted in what seems to be thick boundary layers if judged in terms of the profiles of axial velocity, but the total pressure stays high towards the walls.

7.6.4 Behaviour of End-Bent Blades

The overall performance of the end-bend blades reported here is disappointing in the sense that they have not resulted in a positive demonstration of improved performance which can immediately read across to project designs. The most encouraging behaviour observed in the traverse data was seen at the fixed ends of the blades, where the flow does seen to be more two-dimensional in nature.

In its current implementation, the IEB technique has some shortcomings:

- it was conceived as a small perturbation theory, yet there is nothing in the design procedure to stop large changes in geometry being made,

- there is a lack of symmetry in the treatment of rotors and stators, whereas the experimental evidence suggests strong similarity between the two,

- it does not differentiate between blades with and without clearance.

- it can result in a loss in stage flow capacity.

The observed change in reaction (fig 7.30) might have been expected from the implied change to the vector triangle geometries. The design of the end-bends ignored the radial equilibrium implications of arbitrarily designing for a radial pressure gradient which is inconsistent with the assumed whirl velocities.

This flaw in the IEB design procedure was most clearly revealed by the low reaction test. It revealed the potential for the chosen whirl angles to adjust the radial static pressure gradients and axial velocities. The shape of the design vector diagram for the end-bent blading implies a significantly lower reaction near the blade ends than in the free-stream and consequently a lower rotor static pressure rise at the hub and tip with a corresponding larger static pressure increase at the ends of the stators. Since the radial gradient of static pressure implied by the design velocity triangles is not consistent with radial equilibrium, it follows that some departure from the design triangles must take place to satisfy equilibrium.

This is achieved by variation of the axial velocity ratio, resulting in a rise in axial velocity across the mid-span of the rotor and a corresponding fall across the end-bent stator. Changes of opposite sign might be expected across the ends and this is broadly supported by fig 7.36.

The average reaction of the end-bent stage would be expected to be somewhat lower than the conventional stage, based simply on the average of the stator outlet angle. As the conventional IGV was retained for the end-bent test, the first stage has higher reaction than subsequent stages. Had the IGV been changed for one with an exit whirl distribution similar to that of the end-bent stators, then it is likely that a more constant axial distribution of reaction would have resulted. There would still be a continuing change in the axial velocity profile across successive rotors and stators, which would persist no matter how many stages were present.

The end-bent inlet metal angles were increased near the walls in the hope of maintaining incidence at 2D levels across most of the blade span. In fact, the axial velocity profiles at rotor inlet resulted in reduced relative inlet angles near both walls, so incidences in these regions were far more negative than intended, up to -20 degrees in places. This may not have resulted in drastic losses in this low speed model, but it would have been far more serious for a high Mach number cases. The fact that the end-bent stator inlet metal angles matched the gas so well was good fortune. The measured angles from the datum test are much less severe than those chosen for the end-bent design.

A potentially serious consequence of the IEB design is the loss in relative inlet dynamic head at both ends of the rotor, which results in very high aerodynamic loadings for this high cambered bladerow.

Having seen the side-effects of IEB in the low-reaction tests, it is interesting to look back at the zero $\alpha 0$ series to see whether they demonstrated similar behaviour. The coverage of fixed casing static pressure tappings was less extensive, so the traversing wall tappings around stages 3 and 4 have been used. Comparing area means of these at peak efficiency, the end-bent stage 3 had a reaction of 85% compared with 87% for the datum. This small change is the result of keeping the stator exit angles fixed (at zero) for these high reaction stages.

The positive features of IEB are that

they generally deliver a fuller total pressure profile

there is evidence of a reduction in secondary flow, probably the result of a fuller total pressure profile,

they have not cost in stage efficiency despite the obvious flaws in the design: very large negative incidences on the rotor and reduced rotor inlet dynamic head.

However, they have not helped with the free end, in fact there is some evidence to suggest that higher stagger blades may be detrimental to the flow here.

An important point is that the phenomena observed for the low-reaction blading would not have been revealed in a single stage build.

7.6.5 Synthesising the Behaviour of the End-Bend Blading

That the analysis software could cope well enough with the severe profiles encountered was encouraging, as the same program (Q263) is at the heart of the design procedure. As a first pass at simulating the behaviour of the low-reaction end-bent stage using the throughflow program, the 'design' calculation was re-run, specifying blade turning with the assumed relative exit gas angles, and retaining the original profiles of loss coefficients, mass flow and rotational speed.

Two calculations were run; the first with constant inlet whirl angle of 19.2 degrees, to simulate the first stage, and the second with the inlet whirl profile set equal to the stator outlet angle, simulating a buried stage.

Though this calculation was not particularly rigorous, it managed to reproduce two of the most important features; the low reaction of the 'buried' stage and the unusual axial velocity ratios.

Fig 7.43 shows the axial distribution of reaction, replotted from fig 7.30, showing the values deduced from the two 'design' throughflows. Fig 7.44 shows the axial velocity ratios from the buried stage throughflow compared with the test analysis version.

Had these throughflows been run as part of the end-bend design process (and believed) they could have resulted in a much more appropriate choice of rotor inlet blade angles. It is fair to say that this simplistic modelling would be even less exact for a high speed stage where high temperatures would be generated near the walls in conventional inviscid throughflows. However, it should certainly be handled by the emerging viscous throughflows.

There is no evidence to suggest that maintaining reaction with radius or holding the mean stage value constant is of particular importance, so long as radial equilibrium considerations are accounted for in the design process.

7.6.6 Alternative End-Bend Strategies

The success of the crude 'design' throughflows suggested that it might be beneficial to look at the vortex flow implications of some of the alternative end-bend strategies.

The calculations were based on the design point of the low-reaction stage. The free-stream stator exit angle was maintained throughout at 19.2 degrees and allowed to vary by an arbitrary amount of ± 7 degrees at the walls, over about 20% of annulus height.

The rotor end-wall angles were allowed to vary by ± 3 degrees relative to the datum levels, and the bulk level adjusted to maintain the stage pressure rise of the datum. All runs had common flow and rotational speed, and loss coefficient profiles for the subject stage (3) were assumed fixed throughout.

The procedure for running the throughflow was first to guess the stage 3 inlet total pressure profile, which was input to define stator 2 loss for the first iteration. For subsequent iterations, the stator 2 exit total pressure profile was fed back from the stator 3 exit profile, less the stage pressure rise. If necessary, the rotor exit angles were adjusted bodily to maintain the desired level of stage pressure rise. Typically 5 iterations were necessary to converge the solution.

This exercise was not aimed at rigorous performance prediction. For that, the throughflow would have needed both a loading based loss model and incorporation of spanwise mixing terms to dissipate the high temperatures which would build up near the walls. The objective was to examine trends in loading parameters for moderate perturbations in gas angles.

i) Datum Case

The datum case assumed an inlet total pressure profile which was a sanitised version of those measured earlier. The loss coefficients were based on the test measurements, but were assumed to follow a conventional bucket shape and be equal for both rotor and stator at mid-height.

The datum stage had rather low inlet dynamic head at both rotor and stator hubs with consequently high static pressure loading in these regions relative to mid-span.

ii) Datum Rotor, Reduced Stagger Stator

Reducing the stator exit angles near the ends leads to increased work and hence a fuller total pressure profile and more uniform axial velocity. The rotor inlet dynamic head is increased at both hub and tip while the mid-span level is little changed, and the stator inlet dynamic head becomes more radially uniform, relatively higher at the ends but reduced across the free stream. The extra dynamic head near the end-walls reduces the extreme values of the loading parameters for both rotor and stator.

However, the redistribution of flow has the potentially damaging effect of reducing the mass-meaned efficiency: relatively more flow passes through the less efficient parts of the stage. For the stage to be more efficient, therefore, the unloading of the blade ends must result in substantially reduced loss firstly to overcome the loss due to mass meaning and then to bring about any net improvement over the datum stage.

The stator inlet angles fall in sympathy to the outlet angles, so the blade would not need more camber but would need to be set at somewhat lower stagger than the datum near the end-walls. This is effectively the Controlled Vortex approach.

iii) Increased Camber Rotor, Reduced Stagger Stator

Applying a similar sense of modification to the rotor (ie reducing exit angles near the ends) boosts the stator inlet dynamic head at both ends and improved the loading parameters slightly for both bladerows. Yet more flow is pulled through the less efficient end-wall regions, however, such that at a level of loss coefficient which is common to the datum vortex flow, mass meaning would lose 1% of efficiency. This is effectively the VWD case tested by Andrews et al (1951).

iv) Datum Rotor, Increased Stagger Stator

Closing the ends of the stators has an opposite effect and blade loadings increase as the flow passing through the near wall regions reduces, which looks attractive from the mass meaning point of view. This is the approach apparently used by General Electric (Wisler, 1984).

v) Increased Stagger Rotor, Increased Stagger Stator

Allowing the rotor stagger to increase at the endwalls further increases the loadings near the end-walls and results in the most extreme total pressure and axial velocity profiles. This is VAP as tested by Andrews et al, 1951).

There is enough physics incorporated into the standard throughflow program for it to be a useful aid to the thinking process when planning a whirl law strategy. It appears that there is not a definitive answer to the question 'what is the optimum whirl law ?' but that the choice depends on the specific case under consideration. The fact that there apparently exists contradictory evidence in the open literature, where particular strategies worked well on some occasions but gave opposite results on others, supports this view.

A consistent approach is suggested:

for highly loaded stages, there seems merit in unloading the ends of the blades using reduced stator staggers towards the walls and accepting the mass meaning disadvantage. This might be expected to lead to improved surge margin, since the ends are likely to stall first.

for lightly loaded stages, it might be possible to tolerate the increased near-wall loading implied by reduced stator staggers and gain efficiency through the mass-meaning argument. Choosing a datum rotor, or even adopting a small near-wall camber increase, seems a sensible option to ensure strong near wall total pressure profiles.

The implication is that for moderately loaded stages, the datum whirl law may well be a sensible compromise, moving in one direction or the other depending on other design constraints, benefiting from the 'natural' increase in work which occurs as axial velocity reduces through the boundary layers with constant deviations.

7.6.7 Readacross to High Speed Compressors

The development path of the V2500 high-speed compressor through to an interim specification used in development engines ('Plan B') shows a substantial departure from the original Batch 1 standard modelled here by the low-reaction, end-bent blading. The end-bend angle perturbations were much smaller and the rear stages became higher reaction. Both changes minimise the 'reaction' problems of the IEB technique.

Fig 7.45 illustrates a readacross between low and high-speed data: it shows the axial distributions of reaction for configurations tested early and late in the development program. In all stages featuring severe end-bends, of the type tested in the LSRC, the test reactions (static pressure definition) were found to be substantially lower than those assumed in the design.

These project multistage compressors were rather poorly instrumented by research standards and were without traverse information. The single-stage HP9 compressor is equipped with traverse gear at stator exit, and improvements in performance have been demonstrated with end-bends for both 50% and zero $\alpha 0$ stages.

The major problem with the single stage examination of end-bends is the lack of a correct simulation of the conditions found at entry to a buried stage. In HP9, the inlet gas angles were the result of a spoilt total pressure profile and were uniquely known in advance. The HP9 results are still encouraging as they do suggest that end-bends can give a performance improvement when they are well matched to the local incident flow conditions.

HP9 unfortunately ran with unrealistically low tip clearances, not representative of those achievable in the engine environment. Also, the single stage test would not have revealed the reaction problem, and there now exists evidence (Wisler, 1981) that the results of a given stage geometry are quite different when tested in isolation compared to a multistage test with identical blading.

The hypothesis applied in the setting up of the HP9 model was that the action of viscosity and the blade forces eliminate any tendency for the boundary layers to leave bladerows with other than collateral flow. This is not supported by the angle measurements taken within the LSRC, nor those from C147, a high speed multistage compressor as shown in fig 7.46.

7.6.8 Loss and Deviation Correlations

The symmetries in the rotor and stator behaviour give some hope that a simple correlation, such as that proposed by Roberts and Serovy (1988), might be adequate as a first approximation to bladerow loss and deviation in the presence of secondary flow. Fig 7.47 summarises the deviations from the low-reaction tests at the peak efficiency operating condition. With the exception of the rotor hub, there exists strong symmetry between conditions at fixed and free ends of these bladerows.

8. 3D NAVIER-STOKES ANALYSIS OF THE L-R STATORS

8.1 Introduction

It was suggested in Chapter 4 that advances in Computational Fluid Dynamics (CFD) have opened up the possibility that some of the more fundamental compressor research and design optimisation could be performed numerically. There are a number of 3D Navier-Stokes (N-S) solvers reported in the literature, employing a range of algorithms and turbulence models. There has been a steady stream of publications describing impressive results from the application of these programs to cascade and compressor examples. These were often the work of the program authors and can be considered part of the essential code validation phase. There is not, however, any evidence known to the author of such codes breaking into the design environment.

The objective of the work reported in this chapter was to calibrate one of the N-S solvers by calculating the flow through some of the bladerows already tested, in readiness for its use as an aid to the design of the next generation of end-bent blading.

This chapter explains the factors influencing the choice of solver and the problems encountered in the preparation of meshes prior to actually running the program. Results of calculations for both the conventional and end-bent, low-reaction stators are then compared with test data from their respective peak efficiency operating points. Having established the basic validity of the models, the power of the 'theoretical' solution is then exploited as some features of the flow which are beyond the reach of the experimenter are explored. Finally, the experience relevant to the use of such programs in the design environment is discussed.

8.2 Choice of Solver

8.2.1 Quality and Acceptance

Miranda (1982) gives a particularly clear perspective of the factors influencing CFD program 'effectiveness' in the design environment. He defines effectiveness as the product of 'quality' and 'acceptance', emphasising that the impact of a given code in the design process depends not only on how good the program itself is, but how widely used or accepted it is. In the present context, 'quality' refers to the accuracy and realism of the solution, while 'acceptance' has to do with the user interface and cost. Factors influencing quality are:

i) **Adequate simulation of physics.** The mathematical model should clearly be capable of modelling the flow features which are significant to the current problem.

ii) **Complex geometry.** In this case, the highly three-dimensional geometry of the end-bent blade and the detail of the tip-gap.

Perhaps the overwhelming factor influencing the choice of program in practice is that of availability. At the time the present exercise was started (in early 1988) Rolls-Royce had two potentially suitable programs: MEFP and 3D VICTA. The former has already been briefly mentioned. VICTA is based on a time-marching algorithm and has its roots in the technology emerging from the Whittle Laboratory which has spawned the programs by Denton (1982) and Dawes (1986).

The current work hypothesises that both the available programs are capable of satisfying the 'quality' criteria. While it was recognised that the empiricism within the 3D flow codes awaits a thorough evaluation for some of the more detailed aspects of the flow in turbomachines, the judgement was that the level of technology would be adequate for the programs to be applied along with engineering judgement, much in the same way as experimental results are used.

Factors influencing program acceptance are:

iii) **User friendliness.** The input and output should be automated, clear and logical.
iv) **Robustness.** The program should not require a high level of specialised expertise on the part of the user, nor should it require excessive adjustment of input control parameters to achieve a successful run.

v) **Reasonable computer demand.** Both cost and adequate turn over of jobs are important for the design activity.

It was not clear that the peripheral software supporting the N-S solvers would allow the system as a whole to satisfy these acceptance criteria.

8.2.2 In-House Status

As far as the end-user is concerned, MEFP and VICTA solve an equivalent set of basic conservation equations and employ similar semi-empirical turbulence models, based on the Prandtl Mixing Length concept. The pressure correction algorithm is better suited to the lowest Mach number flows than is time-marching which, in its usual formulation, relies on appreciable changes in density to achieve reasonable rates of convergence.

Both programs were available both on the in-house, IBM mainframe computer and on a bureau Cray XMP vector processor and both provided equivalent output for storage on the Blade-File database and could, therefore, make use of the same post-processing software.

They differ in their choice of grid systems. MEFP uses a fully curvilinear grid with mesh lines passing through the solid bodies (such as blades) in the calculation domain, as shown in fig 8.1. VICTA, on the other hand, uses a 'sheared-H' grid which has been the standard option for the time-marching solvers with roots at the Whittle Laboratory. The sheared-H grid system is most appealing in its simplicity and is certainly the most acceptable to users, in terms of time taken to produce a mesh from scratch. However, this system is at its weakest with profiles featuring the combination of thick leading edges and high stagger; both features were present in the end-bent stators under investigation. At high stagger, the slope of the suction surface is such that, even at very small axial spacing, the blade-to-blade lines at constant x do not give a very accurate definition of the aerofoil surface.

Compressor examples are particularly sensitive in this respect as the highest Mach numbers are generated in this region and grid inaccuracies may generate spurious losses which will be convected throughout the solution domain. The modelling of geometry in the vicinity of the blade ends with tip clearance is also weak with the sheared-H grid system. The blade must be tapered, thus removing the sharp corner from which the leakage flow is likely to separate in practice as shown in fig 8.2. The curvilinear grid system again gives a much sharper definition of the true geometry.

VICTA was, at the time, a relatively new program and was not extensively validated on compressor cases, though it had been assessed on some turbine examples. MEFP had a more appropriate pedigree, having been extensively validated by the program authors on both turbomachinery and standard viscous flow test cases in adverse pressure gradients.

8.2.3 Choice - MEFP

MEFP was chosen after a preliminary evaluation using VICTA, which was primarily motivated by the appealing ease with which grids could be generated. The leading edge grid definition afforded by he sheared-H system was fairly poor as expected but, more significantly, the program struggled to converge even when the level of Mach number was increased artificially to around 0.3, and the solution appeared unreasonably dominated by very large viscous effects. At design incidence at the mid-span of the stator, the calculated velocity vectors in fig 8.3 show the flow to be hopelessly stalled, whereas indications from surface static pressure measurements and traverse results (Chapter 7) together with design calculations with blade-to-blade codes, suggested normally unstalled flow and low loss.

This was to be the first application of the new MEFP mesh generation software to a turbomachinery example in polar coordinates, and the first example with such strongly three-dimensional geometry.

8.3 Mesh Generation

8.3.1 Basic Input Data

The blading analysis software suite has been designed as a 'forward' route, which means that the 3D analysis is assumed to follow basic section design, which results in a set of 2D streamline sections which define the blade geometry (fig 5.1). In this case, while streamline sections existed for the conventional blade, the end-bends were applied later to the plane sections, and this blade only existed in the latter form. The first step, therefore, was to interpolate the plane section definition back onto stream surfaces, known as 'reverse-stacking'.
A special set of streamline definitions was established for this task with the blade metal angles used as blade turning input to the throughflow.

One significant problem which was encountered was that the end-bends had been applied with the plane sections set more open than their true aerodynamic stagger, simply to avoid the possibility of the suction surface becoming non-monotonic in x. When the end-bent planes were set at their correct rig stagger for the reverse stacking operation, the suction surfaces were no longer single valued at x locations near the leading edge at both hub and casing. This was overcome by fitting a very small 'leading edge' circle, not at the true leading edge, but at the position along the suction surface at minimum x, as shown in fig 8.4. The 'pressure surface' was then redefined to incorporate the lower part of the old suction surface and the leading edge region in addition to the original pressure surface.

The annulus line, being parallel, presented no problems. It is passed into the mesh generator in the form of the analysis throughflow solution, which can also be used to provide the aerodynamic boundary conditions for the 3D calculation.

8.3.2 Master Geometry Definition

There are two parts to the mesh generation software for MEFP. Firstly, a skeleton mesh is produced which contains a full description of the blade geometry but only the bounds of the remainder of the domain (program JA37). This mesh is similar in appearance to a sheared-H grid, except that it is continuous through the solid blade.

All CFD programs work best with grids which tend towards a high degree of orthogonality and where the cell aspect ratios and spacing between adjacent grid lines is of order 1. The concentration of grid lines required to give the fine definition of the leading and trailing edges in the JA37 skeleton mesh makes this difficult to achieve within a reasonable total number of grid points, so the grid must be flared. This task currently requires a second program, VN97.

The programs work with the range of sections defining the whole blade simultaneously. The user concentrates on one section only, say the mid-height, and alterations such as the addition of mesh lines and flaring operations are copied to other sections automatically.

The mesh generation software is command driven and it runs in an operating environment which allows commands to be stored, then re-executed. Precisely the same instructions were applied to generate the end-bent stator mesh as had been used on the straight blade, to ensure that the mesh quality was comparable between the two. Indeed, across the middle portion of the blades, where the geometry is identical, the meshes are also identical.

At the trailing edges, the exit mesh angles are aligned with the blade outlet angles, giving a concentration of points through the wakes.

The original intention was similarly to align the mesh inlet angle with the blade inlet angles, but this posed problems with the end-bent blade. It has very high inlet angles, tending towards 90 degrees on the walls, and had the blade and grid inlet angles been equal there would have been severe interference between the meridional lines passing through the leading edge and the blade-to-blade lines passing around the nose of the aerofoil, resulting in a high degree of non-orthogonality. The angle between conjugate grid lines should not become small, since flow gradients approximately normal to the grid lines cannot be resolved with the same accuracy as gradients along grid lines, which leads to difficulties in the discretisation of diffusion terms. The code may still have converged with a highly sheared grid, but the solution would have been likely to contain large truncation errors.

The convention adopted was to choose a constant mesh inlet angle for all sections, to be roughly the average of the end-bent metal inlet angles. The mesh corner point was allowed to drift rearwards along the suction surface towards the ends of the end-bent stator, as illustrated in fig 8.5.

VN97, the mesh manipulator, was originally intended to work only on the definition of blade geometry provided by JA37, which had the link to the master geometry on the Blade File. However, this was found to be inadequate during the present work and a VN97-Blade File link had to be established to ensure that the calculation mesh accurately represented the true geometry.

In general, there is little point in over manipulating a grid to achieve an aesthetically pleasing result. The MEFP solver itself is the best judge of mesh quality and there is certainly merit in proceeding to an early first run of the program. Scrutiny of the results should reveal where any further manipulation might pay dividends.

The final calculation meshes had 75 axial points, 35 hub-to-casing and 39 blade-to-blade. 45 of the axial points were within the blade. 11 of the hub-to-tip points were in the clearance gap, with the twelfth on the end of the blade, 1.8% of annulus height above the rotating inner wall. The pitchwise points within the clearance gap were also arranged in a geometric progression, such that the first point was 1% of gap height away from the corner. This ensures that there is accurate resolution of the pressure gradient causing the separation of the leakage fluid from the tip of the blade (Moore, 1988).

8.4 Moore Elliptic Flow Program (MEFP)

8.4.1 Governing Flow Equations

The N-S equations express the time rate of change of mass, momentum (three components) and energy for a control volume of fluid. These five equations have ten unknowns, so additional equations are required to couple the thermodynamics to the properties of the gas. For turbomachinery cases, closure is achieved by means of the equation of state and the assumption of a perfect gas, specifying the Prandtl number, assuming a relationship between the viscosity coefficients and by using Sutherland's Law for the laminar viscosity. The system of equations could, in principle, be used to simulate large-scale turbulent phenomena, but computer storage is not yet large enough to accommodate the number of grid points which would be required to achieve adequate resolution for simulating such details for the complex geometry of the turbomachine. Currently, turbulent flows are computed by first expressing the N-S equations in Reynolds-averaged form, which means replacing the three velocities by the sum of mean and fluctuating components, then time-averaging.

Comparing the resulting momentum equations with the originals, for laminar flow, extra terms arise, known as the Reynolds stresses. These are approximated using the Boussinesq hypothesis, which allows the viscosity to be replaced by an 'effective' value. In the equations written below, μ can stand for the molecular viscosity (for laminar flow) or the effective viscosity (for turbulent) flow.

In addition, MEFP assumes that:

i) the flow is steady in the relative frame of reference, so time derivatives are zero.

ii) the second coefficient of viscosity is zero.

iii) the body force term is represented by centrifugal and coriolis terms. The equations can be written for a stator:

<u>Mass</u> $$\nabla.\rho\underline{u} = 0$$

<u>Momentum</u> $$\rho(\underline{u}.\nabla)\ \underline{u} - \nabla.(\mu\nabla\underline{u}) = \nabla.\overline{(\mu\nabla\underline{u})}^{T} - \nabla p$$

<u>Energy</u> $$\rho\underline{u}.\nabla H - \nabla.\mu\nabla H = 0$$

where $H = Cpt + \frac{1}{2}(\underline{u}.\underline{u})$

<u>State</u> $$p = \rho Rt$$

The viscous terms in the momentum conservation equations are split between the right and left hand sides, primarily to reduce the number of iterations necessary to reach the converged solution. The viscous term on the left hand side also helps to keep the finite difference equations stable.

8.4.2 Turbulence Model

Depending on the balance between deterministic secondary flows and turbulent diffusion in the evolution of the secondary flow phenomena in turbomachinery bladerows, McNally and Sockol (1985) suggest that good predictions of bulk flow should be possible with a relatively simple turbulence model, provided that the numerical techniques used to solve the governing equations are sufficiently accurate.

MEFP, in common with most other solvers for turbomachinery flows, uses the simple Prandtl mixing-length model to calculate the turbulent viscosity

$$\mu_T = \rho L^2 \left(\frac{du}{dy}\right)$$

where $\left(\frac{du}{dy}\right)$ is a three dimensional velocity gradient.

The laminar viscosity is given by Sutherland's Law:

$$\mu_L = 0.0000004.t^{0.68}$$

The program was run in fully turbulent mode with effective viscosity given by the sum of laminar and turbulent components. Close to the walls, this is modified to be:

$$\mu = [\mu_L (\mu_L + \mu_T)]^{\frac{1}{2}}$$

This model has been validated by the Moores on a number of Stanford Conference test cases (Moore and Moore, 1981).

The mixing length is set to be the larger of 8% of the width of a boundary (or shear) layer and a free-stream value, and then the minimum of this and 41% of the distance to the nearest wall. Very close to the wall, the program uses the Van Driest correction:

$$L = 0.41y\,[1 - \exp(-0.038\; y\sqrt{\rho\tau}/\mu_L)]$$

The effects of free-stream turbulence are included in the calculation of free-stream mixing length, which is correlated as:

$$L_{FS} = 0.387.tu.S$$

Boundary or shear layers are found using a test function based on a 3D pressure gradient.

The turbulence model lacks a laminar to turbulent transition calculation (although zones of the calculation region can be constrained to have laminar viscosity) and the second order effect of surface curvature.
The test function does not account for the different length scales which might be relevant to different regions of the flow.

8.4.3 Boundary Conditions

The experimental data consisted of area traverses of total pressure and yaw angle, both upstream and downstream of the bladerows, together with compressor mass flowrate and speed. These were reduced to radial profiles of flow properties as described in Chapter 7 and are illustrated schematically in fig 8.6.

The following parameters were specified at entry to the calculation domain:-

i) Radial profiles of static pressure, axial and tangential components of velocity. The radial component of velocity, which was not measured and deduced to be small by the throughflow analysis, was ignored.

ii) Stagnation temperature was assumed constant at the inlet plane, set at a level deduced from the compressor overall performance.

The total pressure (from $p + 0.5\rho(Vx^2+V\theta^2)$) and whirl angles (from $\arctan(V\theta/Vx)$) were then fixed on the inlet plane. The axial rate of change of pressure correction is uniform across the downstream boundary, and the level is set to achieve the 'specified' mass flowrate (from integrating $\rho Vx\,dr$ at inlet).

The domain repeats along the first and last blade-to-blade planes. The whole of the inner wall is rotating with a tangential velocity 35% greater than the mean axial velocity. The flow velocity is set to zero relative to all solid boundaries (the no-slip condition).

8.4.4 Solution Algorithm

MEFP uses a 'pressure correction' algorithm. In simple terms, this means that the momentum and continuity equations are combined to yield an equation for updating the pressure field, which ensures that continuity is satisfied in the converged solution. The corrected pressure gradient is then used in the momentum equations to update velocity, followed by a solution of the energy equation to find temperature. Finally, density is calculated from the equation of state.

This may be best understood when contrasted with the more straightforward time marching algorithms in which, typically, the continuity equation is first solved for density, then the momentum and energy equations are solved for velocities and temperatures. Finally, the equation of state yields pressure.

For most cases of interest in turbomachinery design, the two approaches will give equivalent results. However, time-marching is less appropriate for the present example. With Mach numbers of order 0.1, the flow is effectively incompressible, so there is little or no change to density calculated in the first step of solution procedure. The algorithm may converge but it is unlikely to satisfy the continuity equation.

The continuity and momentum conservation equations are discretised over finite volumes. The continuity control volumes have the grid points at their corners. Convection and pressure terms are discretised using linear variations of velocity and pressure between the grid points using a second order accurate Taylor series expansion. The momentum control volumes are upwinded so that, in order to follow the local flow direction, their shape changes as the calculation proceeds.

The advantage of the upwinded control volumes is that they remove the need to add 'numerical diffusion' or explicit smoothing to stabilise the solution process, and MEFP is unconditionally stable, independent of cell Reynolds number (Peclet number).

MEFP is described in much greater detail by Moore (1985). In standard form, it uses an iterative procedure which is similar to Patankar's SIMPLER algorithm (Patankar,1980). This approach has been found to converge quickly with good accuracy and be relatively insensitive to non-uniform grid spacing without relaxation, apart from the averaging of velocities and pressures from alternate passes. This approach worked well on earlier, coarse mesh, conventional stator examples. However, the calculations on finer grids (particularly for the end-bent blade) proved to be less stable and a conventional SIMPLER algorithm was adopted, with a relaxation factors of 0.3 applied to the pressure correction and 0.5 applied to velocities instead of alternate pass averaging of velocities and pressure. This adversely affected the rate of convergence, but ensured stability.

The calculations were started from a simple algebraic guess of the flow field. After an initial continuity pass, 70 pressure correction passes were found necessary to achieve convergence.

8.5 Calculation Results and Comparison with Experiment

8.5.1 Radial Profiles of Loss and Deviation

The results are firstly considered quantitatively, in terms of the radial distributions of loss and deviation, which are parameters of particular interest to compressor designers, especially in the regions where they depart from idealised, 2D cascade behaviour.

The circumferentially averaged whirl angles downstream of both the conventional and end-bent stators are compared to corresponding experimental data in fig 8.7.

The calculation results for both blades agree well with experiment close to the hub, where the flow is dominated by the clearance gap and the rotating wall. The position of overturning is correctly predicted to be between 10% and 15% of annulus height, but the magnitude is overestimated by about 2.5° in both cases.

The calculation results for the conventional blade follow the test data closely across the centre of the span. Towards the outer wall (which has no clearance) there is a classical, secondary flow pattern with initial underturning, followed by overturning very close to the casing. The magnitude of peak underturning at 85% annulus height is overpredicted by about 1.5°.

The gas angle agreement across the middle third of the span of the end-bent blade is less exact, being similar in shape but 1° greater in level. The calculation results and measurements converge towards the casing, agreeing on the amount and location of maximum underturning.

Although the blades have quite different radial distributions of exit metal angle, the radial profiles of deviation are surprisingly similar, as noted in Chapter 7. Both calculation results show strong overturning adjacent to the casing which is localised and not detected by the traverse instrumentation. Although the uncertainty in measured yaw angle is ±0.3° in the free-stream, it may rise towards the wall where there is a large gradient of total pressure as the side statics are radially displaced from the total pressure tappings on the wedge probes.

Fig 8.8a compares the measured profiles of loss coefficient for the two blades, defined as total pressure loss as a fraction of local inlet dynamic head, and fig 8.8b shows the corresponding calculation results. As expected, much of the measured loss is the consequence of the hub leakage flow, with a peak total pressure loss of 25% of inlet dynamic head at 2% of annulus height.

The test data (fig 8.8a) suggests that the conventional blade has lower loss in the region 5% to 20% of annulus height, while the end-bent blade appears to have lower loss between 75% and 90% of height. The conventional blade has a peak in loss at 85% of height, perhaps indicating a re-distribution of low energy fluid driven by classical secondary flow. The end-bent stator shows a smaller peak loss towards the outer wall, and this is located much closer to the casing.

The calculated profile of loss coefficient (fig 8.8b) mirrors the measured trend between 5% and 20% of height, but does less well near the tip, detecting a marginally smaller loss for the end-bent blade between 80% and 90% of height, but no spanwise migration of low energy fluid. MEFP has generally overpredicted the losses relative to the test data, with the discrepancy being greatest towards both annulus walls.

The reason for the apparent overestimate of free-stream profile loss is 'spurious' entropy generation in the grid just ahead of the leading edge of the blade across the entire span. Plotting the axial distribution of total pressure for the mid-height region of the blade (fig 8.9) reveals that about half the loss occurs outside the bladerow. If the grid could have been further improved in this region, then it looks as if the calculated profile loss would have given very close agreement with experiment, at least across the free-stream region.

8.5.2 Total Pressure Contours

Fig 8.10 compares the predicted contours of total pressure, normalised by the mean inlet dynamic head, downstream of the conventional stator with measured values from area traversing. The shape and pitchwise location of the clearance loss core at the hub is accurately reproduced by the calculation. Also, the calculated results show the low loss region adjacent to the suction surface near the hub.

The action of the rotating hub is responsible for pulling the loss core so far across the passage, reinforcing the clearance flow itself. The results of earlier calculations performed with the MEFP with hub clearance, but without inner wall rotation in fig 8.11 show this loss core to be integral with a suction surface corner stall, similar to that visualised by Dong et al (1986).

At the tip, the calculation and measured data agree on the existence of a suction surface corner stall and its spanwise extent. Across the remainder of the span, the calculated wake is rather thicker and shallower than is suggested by the experimental results. This is partly due to the contamination by entropy generated ahead of the blade, but also indicating that mixing is occurring too quickly just downstream of the trailing edges.

Apart from the wake shape factors, the only significant, qualitative disagreement between calculation and experiment in fig 8.10 is the shape of the peak pressure contour, which extends all the way across the passage from the suction surface to the edge of the pressure surface boundary layers in the calculation, but only 2/3 of pitch in the experiment.

This region of low total pressure adjacent to the pressure surface found in the experiment is actually a wake from the upstream stator row (stator 2) which is progressively mixing out as it passes through the third stage. Fig 8.12 shows the measured conditions at entry to the conventional stator, where there are three distinct regions of low pressure fluid: the remains of the upstream stator wake and clearance loss core, and an apparently thick annulus wall boundary layer on the casing. The latter is the time averaged interpretation from the stationary, pneumatic probe of the unsteady, rotor tip leakage flow. The data shown in fig 8.12 was circumferentially meaned for definition of the upstream boundary conditions for the calculation, denying the program the chance to accurately track the passage of the stator 2 wake through the third stator.

The contour plots of total pressure calculated and measured for the end-bent stator are shown in fig 8.13. Fig 8.13b shows a shallow, broad wake, similar in character to the conventional blade result in fig 8.10b. Again, the clearance loss core has moved across the passage to rest adjacent to the pressure surface in good agreement with the test data in fig 8.13a.

8.5.3 Surface Pressure Distributions

The calculated surface static pressure distributions are shown in fig 8.14 for the conventional stator and fig 8.15 for the end-bent: these can be compared to measured values (from stage 2) in figures 7.28 and 7.41.

Only the suction surface was instrumented at all heights for the conventional stator. There is good agreement on the relative magnitudes and axial location of minimum pressure. In particular, at the near-hub plane, MEFP captures the influence of the leakage flow in moving peak suction axially rearwards. The distribution along the remainder of the suction surface agrees well at hub and mid sections, and around mid-height on the pressure surface.

At the tip, the MEFP results show a reduced rate of diffusion towards the trailing edge relative to the other sections, which is present in the experimental data, but to a much lesser degree. This is broadly consistent with MEFP predicting more secondary flow and a larger extent of suction surface corner stall than probably occurred in practice.

A standard, blade-to-blade calculation of the flow near the end-walls does not, of course, include the influence of leakage and does not give a particularly useful indication of the ability of the blade to cope with the prevalent flow conditions. Fig 8.16 shows the 2D calculated pressure distribution for the near-hub section from FINSUP running at the measured inlet Mach number and incidence. Peak loading occurs right at the leading edge and the excessive diffusion along the suction surface causes the boundary layer to separate (a converged, 'viscous' solution was not possible). While the plot of the velocity vectors near the hub from the MEFP solution in fig 8.17 is consistent with the section operating at incidence (the stagnation point is towards the pressure surface rather than on the leading edge) the pressure distribution near the hub (fig 8.18) is similar to that at mid-height, but modified by the leakage.

The mid-height velocity vectors in fig 8.17 show the stagnation point to be right at the leading edge, so the blade is well aligned with the flow.

Towards the casing, the vectors again show a degree of positive incidence, which is reflected in the measured and MEFP pressure distributions. The 2D code again overestimates the extent of the leading-edge loading for the given inlet conditions, as shown in fig 8.19.

Good agreement is also obtained between calculation and experiment for the end-bent stator. The calculated, mid and tip pressure distributions are almost identical (fig 8.15) as they were in the experiment, giving further support to the perceived success of the end-bend at the fixed end of the stators. Neither test nor calculation show any evidence of corner stall for the end-bent blade near the casing which was also concluded from the contour plots of total pressure in fig 8.13.

At the hub, the influence of the clearance flow is less pronounced in the MEFP results, though the trends are reproduced correctly. Peak suction is at lower pressure at the hub than for other heights (in contrast to the conventional stator) and it is moved axially rearwards.

The hub seems to be experiencing less negative incidence in the calculation than in the experiment which might be a consequence of a lack of a repeating flow (the calculation has stage 3 stator inlet conditions while the measurements of pressure were on stage 2). The lack of agreement on the distortion of the hub pressure distribution could infer that too little leakage flow has been predicted.

One final disagreement which is worthy of note is that the calculation predicts too high a pressure towards the trailing edge of the pressure surface at the hub in the calculation for the end-bent blade (fig 8.15). The prediction for the conventional blade shows a similar effect, but there is no experimental evidence to contradict it.

Examination of the stagnation points calculated by MEFP, shown in fig 8.20, suggests that the end-bend modifications to the inlet angles were too large. At both ends of this stator, but particularly near the casing, the stagnation points have drifted around onto the suction surface. At mid-height, the blade is still well aligned with the flow.

8.5.4 Calculated Flow Streaklines

The calculation results discussed so far have all had experimental data available for comparison. The numerical approach has the benefit that data can be post-processed to study details of the flow within bladerows which are not so readily accessible to the experimenter.

Fig 8.21 shows calculated surface streaklines from the conventional blade. The suction surface corner stall can be seen near the casing and its influence in deflecting the flow towards the hub is apparent well into the mid-span region. Near the hub, there is a strong radial flow on the suction surface as fluid is entrained by the clearance jet as it emerges from beneath the tip of the blade, It is this radial flow which convects higher pressure fluid from the 'free-stream' flow towards the hub, resulting in the region of low loss around 15% height, as noted from fig 8.8.

On the pressure surface, the streaklines near the hub show a division between fluid migrating radially up the pressure surface and fluid passing beneath the end of the blade. The radial migration appears to be the result of a 'scraping vortex', where the hub boundary layer is stripped off the wall as it hits the stationary blade. This scraping vortex was observed to be the dominant secondary flow phenomenon at the tip of an isolated rotor blade by Phillips and Head (1980). In the present example, the relative motion between the stationary blade and the moving wall is equivalent to the moving blade, stationary casing condition at a rotor tip.

With the exception of the pressure surface migration, the remainder of the phenomena apparent in the calculation results support the clear description of the clearance flow, published by Dean (1954), which was based on observations from experiments on a linear cascade both with, and without, a moving end wall.

Fig 8.22 focuses on the detail of the secondary flow beneath the blade at about 60% of chord. There is a stagnation point (a separation line) about 1% of annulus height from the blade tip. The flow which passes through the gap separates from the pressure surface corner. The separation bubble is rather smaller than might have been expected by analogy with long orifice flows. It is possible that the bubble is being suppressed by the action of the moving wall, which has a similar tangential velocity to the mean clearance flow, easing the passage of the leakage flow beneath the blade.

More likely, the lack of a significant vena contracta is a consequence of the over prediction of viscosity in the gap by the simple, mixing-length turbulence model, which has been set to pick out boundary and shear layers on the scale of the passage, rather than the tip gap. A lower viscosity in the vicinity of the gap would result in a larger separation zone beneath the blade, but the tendency for the clearance flowrate to reduce in proportion would be offset by the reduced, viscous resistance to this leakage flow.

The corresponding, meridional streaklines for the end-bent blade are shown in fig 8.23. On the suction surface, there are large radial flows towards the hub near the trailing edge, which are very similar in character to those observed on the conventional blade in fig 8.21.
At the casing, the end-bend has successfully alleviated the suction surface corner stall and, in general, there is noticeably less stream-sheet twisting over the tip portion of this blade compared to its conventional counterpart.

Near the hub however, the end-bend results in fig 8.23 show a noticeable radial component of velocity near the leading edge, as the throughflow is deflected over the leakage flow which, for the high stagger, end-bent hub, has a larger forward component of velocity.

8.6 Discussion

8.6.1 Implications for End-Bend Design

The absolute accuracy of the MEFP computations is discussed below. The modelling is clearly imperfect to some degree, but can such programs contribute to the design of unconventional blading?

Given a reasonably realistic throughflow computation, such as the methods incorporating spanwise mixing by Gallimore (1986) or Adkins and Smith (1982), to define the inlet conditions to each bladerow, the 3D programs could be used to refine the semi-empirical deviation assumptions and to help correctly align the blades with the flow. The program should be capable of warning of significant corner stalling.

Whilst the absolute accuracy of the loss predictions is perhaps the most disappointing aspect of the MEFP results, the program correctly attributed highest loss in the presence of a tip gap to the end-bent stator. This is also the region of the blade which is least well handled by the standard blade-to-blade codes. It is quite likely that optimum aerofoils from the leakage point of view are quite unconventional in comparison with those best for the mid-span region, in terms of stagger, camber and thickness distributions. The 3D code could be used to evaluate these non-standard profiles and to give guidance on how far they must extend away from the walls if they are to influence conditions near to the ends.

8.6.2 Quality

The agreement between the MEFP calculation results and the test data from the two stators is encouraging. The program gives a good description of the 3D deviations of both bladerows and it correctly attributes lower loss to the low stagger hub of the conventional stator when compared to the end-bent version.

Earlier runs of the program, with coarser grids, suggested that the radial profiles of 3D deviation are not critically dependent on accurate modelling of the flow through the tip gap, provided that the rotating inner wall boundary condition is implemented. On the other hand, the correct pitchwise positioning of the clearance loss core and the detail of the blade spanwise flows near the hub were only detected with fine gridding in the gap.

The simple, mixing length turbulence model generally performed well as the calculation results appeared to capture the essential physics of viscous, blade-end flows, giving confidence that the code can be used to guide end-bend stator developments. Although the magnitude of losses was overpredicted much of the discrepancy in comparison with experiment was seen to be the fault of poor gridding.

The model is probably weakest in the wake immediately downstream of the stator trailing edges, and in the tip gap. The flow remains attached to the blade right around the trailing edge and the rate of mixing out of the wakes is much greater than that suggested by the experimental data. Also, the flow in the tip gap lacks a substantial vena contracta. These effects imply that the flow in these regions is too viscous. On the other hand, the prediction of outlet angle near the casing looks as if the calculated flow lacks viscosity. The phenomena demand their own, differing length-scales: the MEFP model has only one which is optimised to resolve features on the scale of the passage width.

These observations indicate that this simplest of turbulence models may have reached the limits of its usefulness for compressor applications. While the calculation of the basic flowfield was limited by the combination of grid resolution and solver accuracy, there was little point in employing undue sophistication in the turbulence model. However, now that these aspects have improved and computer hardware has developed both in size and speed, users would probably be prepared to invest in the additional mesh resolution and computer run time required for a more complex turbulence model.

The extent of the predicted, spanwise flows on the blade surfaces raises questions on the validity of the use of 2D (or quasi 3D) programs for assessing blade element performance for low aspect ratio blading, particularly towards the annulus walls. The 2D codes also give an incorrect assessment of aerofoil loading in the leading-edge region for near-wall profiles operating at incidence. The 3D solution captures the spanwise interaction between near-wall and free-stream regions which the 2D, by definition, cannot.

8.6.3 Acceptance

There are two issues in mesh generation: will the algorithm converge successfully and how accurate will the results be? In designing the algorithm, the programmer has some freedom to balance accuracy and robustness through the choice of differencing scheme and the implementation of explicit smoothing.

MEFP has been designed to be an accurate algorithm. It uses a second order accurate discretisation of the convection terms in the equations of motion which does not introduce implicit smoothing associated with less accurate schemes, nor does it use explicit smoothing, such as the second and fourth order schemes used in the time-marching algorithms (such as VICTA).

The use of second order accurate discretisation and interpolation over 1/8th control volumes ensures that the MEFP solutions should be less grid dependent than other contemporary algorithms. However, the convergence behaviour is more dependent on mesh quality.

Producing the first mesh for the conventional stator did take an unreasonable amount of time, due mainly to the fact that the software was new, containing some 'bugs', which had to be fixed, in addition to the requirement for some further development. The process of producing the second mesh, for the end-bent blade, by re-executing the stored commands, took around three hours which is still longer than one would like for a design application, but is of the right order of magnitude.

For a new application it is often worth exploring several routes through mesh manipulation, perhaps storing some of the intermediate steps, rather than trying to produce a definitive 'exec' file on the first run through.
Retracing steps is not particularly wasteful, since much of the time spent is in decision making and this is likely to be quicker on the second time round.

Many of the problems in the earlier phase of the grid generation were the result of the conventions adopted for geometries on the Blade File. These evolved at a time when 'correlation' profiles were prevalent and blades were defined with edge radii and curves which were monotonic in x. These concepts are much less meaningful for today's more arbitrary profiles, and are a particular hindrance for the combination of thick leading edges and high stagger.

9. CONCLUSIONS

There is a large potential market for aero gas turbines and Rolls-Royce must remain technically competitive to maintain or improve on its market share. Although compressor aerodynamics is a relatively mature science, advances in performance are still worth pursuing as they can have a significant impact on engine operating cost through fuel burn when combined with improvements in other components and re-optimised thermodynamic cycles.

Technical developments must be through managed risk, 'evolution of technology' and the LSRC has a cost-effective role to play in this process.

The research programme must be well defined in advance so that there can be a clear division of responsibilities (especially true for a collaborative project). The project should be carefully planned with substantiated data, or contingency must be allowed for unforeseen problems. Careful monitoring is required throughout the life of the project so that potential delays can be drawn to the attention of management and the cost-benefit case can be re-assessed.

A flexible approach to funding is appropriate so that the experimental programme can be adjusted in order to react to the findings of the research as they emerge or to respond to changes in the Company's priorities. There must also be visibility of stability of funding for projects with large involvement of university or research establishments so that staff can be given contracts of reasonable length.

With any collaborative project, but especially the LSRC programme, there is a strong emphasis on communication and it is important to promote the feeling of belonging to an 'integrated team'.

The literature survey revealed that there exists a reasonably clear picture of the 3D flow phenomena present in axial stages, but that attention has been focused on achieving well matched designs. No clear, unambiguous guidance has emerged for minimising end-wall losses, though some strategies have shown positive results in certain circumstances.

The modern blading design system was found not to be particularly suitable for low-speed, prescribed velocity distribution profiles. Particular problems encountered were associated with the unusually thick profiles in combination with high staggers. The boundary layer model in the blade-to-blade software was found to be difficult to converge at low chordal Reynolds numbers.

The adoption of increased thickness and camber to represent more accurately the suction surface diffusion found in high speed compressors resulted in a more appropriate model, but this transformation is only strictly valid at one operating point. Conclusions on the stalling behaviour may not directly read across to high speed, though the trends might be expected to be of the same sign.

The LSRC has become an established and technically viable facility and results have contributed towards improved understanding of the flow in multistage core compressors and the data has been used for validation of CFD codes. It has not demonstrated success so far as a vehicle for rapid-response screening tests, which are particularly demanding on technician effort.

The accuracy of data recorded is acceptable for detailed traverse analysis at this point, but some improvement may be necessary for the precise measurement of performance necessary for screening tests, if this use of the rig is to be pursued.

Accurate measurement of mass flow is perhaps the most difficult to achieve as the dynamic head in the measuring section is of the order of 50mm of water gauge, which places a demanding requirement on the calibration of the transducer used in its measurement.

The compressor chordal Reynolds number is bordering on the lowest acceptable value for meaningful readacross of data to the high speed environment. At the current value, the loss may be slightly compromised (higher) but deviations are likely to be representative.

The detail design of the traverse instruments must be carefully considered if they are to be used to measure static pressure and angle in close proximity to the outer casing. It appears that probe blockage is also an important design parameter as well as avoidance of stem leakage.

Many of the features observed in the traverse data could be classified as 'deterministic secondary flows'. The under and over turning seen near the fixed ends of both rotors and stators is a particularly striking example. The extent of the deviations from the primary angles is certainly suppressed by the presence of skew in the relative inlet flow to the blades, but classical secondary flow is clearly present too.

The symmetries in the rotor and stator behaviour give some hope that simple correlations might be adequate as a first approximation to blade row loss and deviation in the presence of secondary flow.

It seems unlikely that the pure turbulent diffusion model of spanwise mixing will be completely satisfactory for multistage core compressors without some extension to cover tip clearance and secondary flow, in the light of the aerodynamic results presented here.

The phenomena associated with tip clearance, at least at the stator inner ends, are just as found in cascade experiments. There is significant underturning which affects the blades for a height of up to 10x the gap and the blade pressure distributions are altered by the leakage vortex.

The overall performance of the end-bend blades reported here is disappointing in the sense that they have not resulted in a positive demonstration of improved performance which can immediately read across to project designs.

The positive features of IEB are that

they generally deliver a fuller total pressure profile and have not cost stage efficiency, despite some obvious flaws in the design technique,

there is evidence of a reduction in secondary flow at the fixed ends of both rotors and stators,

they resulted in improved stall margin for the zero $\alpha 0$ case.

Conditions did not improve at the free end; there is some evidence to suggest that higher stagger blades may be detrimental to the flow there. It is quite likely that optimum aerofoils from the leakage point of view are quite unconventional in comparison with those best for the mid-span region, in terms of stagger, camber and thickness distributions.

In it current implementation, the IEB technique has some shortcomings:

- it was conceived as a small perturbation theory, yet there is nothing to stop large changes in geometry being made,

- there is a lack of symmetry in the treatment of rotors and stators, whereas the experimental evidence suggests strong similarity between the two,

- it does not differentiate between blades with and without clearance,

- it can result in a loss in capacity.

The low reaction test with end-bends did not have a constant reaction through each stage. The IGV should have been changed to deliver the end-bent stator exit whirl angle.

The phenomena observed in the data from the low-reaction blading would not have been revealed in a single stage build.

The standard axisymmetric throughflow proved to be capable of reproducing the most important flow features seen in the low-reaction traverse data. It appears that this existing software can be a useful aid when reasoning a change to whirl philosophy.

The MEFP program has been successfully validated against the two low-reaction stators, one conventional and one with end-bends. The results suggest that the code can be used to guide the design of end-bent stators, with the suggestion that some development of the supporting software will much increase its appeal and acceptance for designers.

The calculated, 3D deviation for both bladerows agrees well with the experimental data, particularly in the near hub region, which is strongly influenced by the tip leakage flow.

Radial profiles of loss show reasonable agreement with experiment, although the magnitude of loss is overpredicted across the span. The discrepancy is greatest near both annulus walls.

The mixing length turbulence model has performed reasonably well in general, although its limitation to a single characteristic length scale has emerged as a weakness.

Application of end-bends at the fixed (casing) end of the vane successfully alleviates the suction surface corner stall which is present in the conventional stator results, which is in agreement with the measured data.

Significant spanwise flows exist along the blade surfaces near the free ends, which may compromise the effectiveness of conventional, axisymmetric, 2D design procedures.

The change in inlet metal angle necessary to match the leading edge of the aerofoil to the stagnation point is somewhat less than would be deduced from purely 2D arguments. The 3D software can provide more appropriate guidance.

A degree of readacross has been established in the observed flow phenomena and the behaviour of end-bent blading between low and high-speed machines.

10. SUGGESTIONS FOR FURTHER WORK

The most important lesson learnt in implementation of end-bends is that they should be correctly accounted in the design process at the throughflow stage. The first suggestion, therefore, is to re-design the low-reaction end-bent stage with essentially the same philosophy as IEB, but with the likely gas angles included in Q263 so that radial equilibrium effects can be allowed for.

The low-reaction end-bent rotor tested in the present work may have been excessively cambered at the hub, so a somewhat less severe end-bend is recommended for the re-design.

The calibration of MEFP reported in Chapter 8 showed enough promise for it to be useful for checking the design of the next generation of end-bends. The calibration/validation process of MEFP and other CFD tools now needs to be extended to the flow in rotors to see whether similar guidance in their design might be forthcoming with todays levels of turbulence modelling technology.

While the end-bends showed most positive results at the fixed ends of both rotors and stators, it was concluded that current designs may not be optimum where clearance gaps exist. A CFD-based design study of end-treatments for the clearance case is suggested, pending the outcome of the rotor flow calibration.

The finding that the radial profiles of deviation were actually very similar for conventional and end-bent blades offers the hope that these data can be developed into usable empiricism which will be required for use in more advanced throughflow codes.

If the CFD input/output processing software can be made more user friendly, then the time seems right to begin some numerical experimentation on, for example, non-radial stacking axes. It would also be interesting to see whether the programs can predict such things as optimum clearances, even with apparently imperfect turbulence models.

The blading design system is rather unwieldy when there is a need to move between programs whilst iterating towards a converged aerofoil surface velocity distribution and gas deflection. It may be cost effective to program a 'PVD' inverse design program and to incorporate some automation into the derivation of adequate camber.

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AERODYNAMIC AND BLADING DETAILS -

CRANFIELD LOW SPEED 4-STAGE : DATUM FREE VORTEX BLADE

BLADE ROW	PASSAGE RADII AT BLADE INLET	PASSAGE RADII AT BLADE OUTLET	$\alpha_{1(s)}$	$\alpha_{2(s)}$	U INLET	Va INLET	ΔT	No. OF BLADES	STREAMLINE CHORD C	$s/c_{(s)}$	$t/c_{(s)}$	$\alpha'_{1(s)}$	$\alpha'_{2(s)}$	θ	δ	i	$\frac{RADL}{T}$	$\frac{RADT}{T}$	TOTAL MACH NUMBER (INLET)	$\frac{\Delta p}{p}$ (COMP)	D_F	PLC	$\frac{P_{02}}{P_{01}}$
ROTOR	20.445	20.452	58.39	44.09	178.42	109.8	1.2	59	2.384	0.913	·090	62.26	28.37	33.89	45.32	-3.87	·06	·06	0.188	0.4256	0.4302	·0436	1.0135
	20.884	20.889	58.93	45.92	182.25	109.8	1.1		2.386	0.932	·082	62.80	32.13	30.67	47.47	-3.87	·06	·06	0.191	0.4176	0.4114	·0332	1.0134
	21.323	21.326	59.46	47.55	186.08	109.8	1.1		2.389	0.951	·075	63.32	35.62	27.70	49.47	-3.86	·06	·06	0.194	0.4090	0.3944	·0244	1.0133
	21.763	21.763	59.97	48.99	189.92	109.8	1.1		2.391	0.969	·067	63.83	38.80	25.03	51.32	-3.86	·06	·06	0.197	0.3999	0.3800	·0185	1.0132
	22.202	22.200	60.47	50.22	193.75	109.7	1.1		2.393	0.988	·060	64.33	41.59	22.74	52.96	-3.86	·06	·06	0.200	0.3909	0.3691	·0155	1.0132
	22.641	22.636	60.95	51.14	197.58	109.7	1.1		2.395	1.007	·052	64.81	43.84	20.97	54.33	-3.86	·06	·06	0.203	0.3821	0.3659	·0199	1.0132
	23.080	23.073	61.43	51.88	201.41	109.7	1.2		2.396	1.025	·045	65.28	45.80	19.48	55.54	-3.85	·06	·06	0.206	0.3733	0.3668	·0284	1.0133
	23.518	23.510	61.88	52.62	205.23	109.7	1.2		2.399	1.044	·037	65.74	46.82	18.92	56.28	-3.86	·06	·06	0.209	0.3648	0.3684	·0372	1.0134
	23.955	23.947	62.32	53.31	209.05	109.6	1.2		2.400	1.063	·030	66.18	45.66	20.52	55.92	-3.86	·06	·06	0.212	0.3565	0.3709	·0467	1.0134
STATOR	20.454	20.461	33.48	0.0	—	109.5	—	61	2.383	0.884	·06	39.31	-15.01	54.89	12.15	-5.83	·06	·06	0.118	0.2452	0.4062	·0531	0.9995
	20.890	20.896	32.25	0.0	—	109.6	—		2.385	0.902	·06	38.08	-14.82	52.90	11.63	-5.83	·06	·06	0.116	0.2395	0.3928	·0416	0.9996
	21.327	21.331	31.14	0.0	—	109.7	—		2.387	0.920	·06	36.97	-14.70	51.67	11.13	-5.83	·06	·06	0.115	0.2339	0.3806	·0313	0.9997
	21.763	21.765	30.23	0.0	—	109.7	—		2.389	0.938	·06	36.05	-14.69	50.74	10.68	-5.82	·06	·06	0.114	0.2286	0.3707	·0226	0.9998
	22.198	22.200	29.55	0.0	—	109.6	—		2.392	0.956	·06	35.36	-14.81	50.17	10.27	-5.81	·06	·06	0.113	0.2224	0.3645	·0183	0.9998
	22.634	22.634	29.42	0.0	—	109.5	—		2.394	0.974	·06	35.21	-15.26	50.47	9.97	-5.79	·06	·06	0.113	0.2152	0.3663	·0230	0.9998
	23.070	23.096	29.58	0.0	—	109.4	—		2.396	0.992	·06	35.35	-15.89	51.24	9.73	-5.77	·06	·06	0.113	0.2074	0.3721	·0310	0.9997
	23.507	23.504	29.81	0.0	—	109.1	—		2.398	1.010	·06	35.55	-16.83	52.38	9.36	-5.74	·06	·06	0.113	0.1995	0.3782	·0391	0.9996
	23.945	23.938	30.12	0.0	—	108.7	—		2.400	1.027	·06	35.83	-16.79	52.62	9.52	-5.71	·06	·06	0.113	0.1904	0.3846	·0474	0.9996
UNITS	ins.	ins.	deg.	deg.	ft/s	ft/s	K	—	ins.	—	—	deg.	deg.	deg.	deg.	deg.	—	—	—	—	—	—	—

Table 5.1

AERODYNAMIC AND BLADING DETAILS

CRANFIELD LOW SPEED FOUR STAGE : INVISCID END-BEND ROTOR

BLADE ROW	PASSAGE RADII AT BLADE INLET	PASSAGE RADII AT BLADE OUTLET	$\alpha_{1(s)}$	$\alpha_{2(s)}$	U INLET	Va INLET	ΔT	No. OF BLADES	STREAMLINE CHORD C	$s/c_{(s)}$	$t/c_{(s)}$	$\alpha'_{1(s)}$	$\alpha'_{2(s)}$	θ	δ	i	RADL/T	RADT/T	TOTAL MACH NUMBER (INLET)	P_{02}/P_{01}		PLC
ROTOR	20.399	20.399	81.30	13.20	177.98	27.23	1.2	59	2.385	0.9110	·091	72.22	29.16	43.06	50.69	9.08	·059	·059	0.162	1.0136		·0448
	20.426	20.426	70.79	36.81	178.22	62.10	1.2		2.385	0.9122	·090	71.31	29.40	41.91	50.36	-0.52	·059	·059	0.169	1.0135		·0442
	20.453	20.453	68.24	41.98	178.45	71.22	1.2		2.385	0.9133	·090	70.39	29.68	40.76	50.01	-2.15	·059	·059	0.172	1.0135		·0439
	20.507	20.507	65.45	44.38	178.93	81.75	1.2		2.385	0.9156	·089	69.98	30.11	39.87	50.05	-4.53	·060	·060	0.177	1.0135		·0425
	20.561	20.561	63.71	45.05	179.40	88.63	1.2		2.385	0.9179	·088	68.27	30.40	37.87	49.34	-4.56	·060	·060	0.179	1.0135		·0411
	20.759	20.759	60.15	45.48	181.12	103.95	1.2		2.386	0.9264	·085	64.69	31.75	32.94	48.22	-4.54	·060	·060	0.187	1.0134		·0363
	20.922	20.921	58.48	45.46	182.54	111.97	1.2		2.387	0.9333	·082	63.02	32.39	30.63	47.71	-4.54	·060	·060	0.192	1.0134		·0325
	20.994	20.993	58.40	45.67	183.17	112.70	1.2		2.387	0.9364	·081	62.94	33.00	29.94	47.97	-4.54	·060	·060	0.193	1.0133		·0308
	21.300	21.299	58.77	46.80	185.84	112.69	1.1		2.389	0.9495	·075	63.30	35.40	27.90	49.35	-4.53	·069	·060	0.195	1.0132		·0249
	21.751	21.749	59.30	48.31	189.77	112.67	1.1		2.391	0.9687	·068	63.82	38.70	25.12	51.26	-4.52	·060	·060	0.198	1.0132		·0185
	22.201	22.199	59.82	49.58	193.70	112.65	1.1		2.393	0.9880	·060	64.35	41.59	22.76	52.97	-4.53	·060	·060	0.201	1.0132		·0156
	22.652	22.649	60.32	50.51	197.64	112.63	1.1		2.395	1.0072	·052	64.82	43.83	20.99	54.33	-4.50	·060	·060	0.204	1.0133		·0203
	23.102	23.099	60.81	51.29	201.56	112.61	1.2		2.397	1.0265	·044	65.30	45.87	19.43	55.59	-4.49	·065	·065	0.207	1.0134		·0289
	23.479	23.477	61.38	52.02	204.86	111.77	1.2		2.398	1.0427	·038	65.88	46.92	18.96	56.40	-4.50	·073	·072	0.210	1.0134		·0366
	23.641	23.639	63.30	53.37	206.27	103.76	1.2		2.399	1.0496	·035	67.75	47.91	19.84	57.83	-4.45	·079	·077	0.207	1.0134		·0403
	23.820	23.819	66.50	55.37	207.83	90.35	1.2		2.400	1.0573	·032	70.97	49.18	21.79	60.08	-4.47	·090	·087	0.203	1.0134		·0440
	23.946	23.945	71.20	57.19	208.92	71.13	1.2		2.400	1.0627	·030	75.67	50.20	25.47	62.94	-4.47	·110	·100	0.198	1.0134		·0468
	23.973	23.972	73.47	57.05	209.16	62.08	1.2		2.401	1.0638	·029	77.96	50.80	27.16	64.38	-4.49	·122	·109	0.196	1.0134		·0474
	23.999	23.999	81.93	22.77	209.39	29.67	1.2		2.401	1.0650	·029	79.29	51.40	27.89	65.35	2.64	·133	·116	0.189	1.0135		·0479
UNITS	ins.	ins.	deg.	deg.	ft/s	ft/s	K	—	ins.	—	—	deg.	deg.	deg.	deg.	deg.	—	—	—	—	—	—

Table 5.2

AERODYNAMIC AND BLADING DETAILS

CRANFIELD LOW SPEED 4 STAGE RIG : INVISCID END BEND STATOR

BLADE ROW	PASSAGE RADII AT BLADE INLET	PASSAGE RADII AT BLADE OUTLET	$\alpha_{1(s)}$	$\alpha_{2(s)}$	U INLET	Va INLET	ΔT	No OF BLADES	STREAMLINE CHORD C	$s/c_{(s)}$	$t/c_{(s)}$	$\alpha'_{1(s)}$	$\alpha'_{2(s)}$	θ	δ	i	$\frac{RADL}{T}$	$\frac{RADT}{T}$	TOTAL MACH NUMBER (INLET)	$\frac{P_{02}}{P_{01}}$		PLC
STATOR	20.399	20.399	84.89	0.0	—	27.02	—	61	2.383	0.8820	.06	58.07	-15.05	73.12	21.51	26.82	.083	.059	0.272	0.9994		.0549
	20.426	20.426	64.98	0.0	—	61.64	—		2.383	0.8831	.06	56.49	-15.03	71.52	20.73	8.49	.081	.059	0.131	0.9995		.0540
	20.453	20.453	58.38	0.0	—	70.70	—		2.383	0.8842	.06	55.36	-15.02	70.38	20.17	3.02	.078	.059	0.121	0.9995		.0534
	20.508	20.507	50.80	0.0	—	81.16	—		2.383	0.8864	.06	53.02	-14.99	68.01	19.02	-2.22	.076	.059	0.115	0.9995		.0519
	20.562	20.561	46.04	0.0	—	88.00	—		2.383	0.8887	.06	50.21	-14.97	65.18	17.62	-4.17	.072	.059	0.114	0.9995		.0504
	20.760	20.759	36.42	0.0	—	103.24	—		2.385	0.8968	.06	42.77	-14.87	57.64	13.95	-6.35	.065	.059	0.115	0.9996		.0448
	20.921	20.921	32.00	0.0	—	111.24	—		2.385	0.9035	.06	38.35	-14.80	53.15	11.78	-6.35	.061	.059	0.117	0.9996		.0407
	20.993	20.993	31.47	0.0	—	111.98	—		2.386	0.9064	.06	37.81	-14.78	52.59	11.52	-6.34	.059	.059	0.118	0.9997		.0389
	21.299	21.299	30.71	0.0	—	112.02	—		2.387	0.9190	.06	37.06	-14.70	51.76	11.18	-6.35	.059	.059	0.117	0.9998		.0317
	21.749	21.749	29.76	0.0	—	112.00	—		2.389	0.9375	.06	36.06	-14.68	50.74	10.69	-6.30	.059	.059	0.116	0.9998		.0227
	22.198	22.199	29.07	0.0	—	111.97	—		2.392	0.9560	.06	35.39	-14.82	50.21	10.29	-6.32	.059	.059	0.115	0.9998		.0185
	22.647	22.649	28.94	0.0	—	111.87	—		2.394	0.9745	.06	35.22	-15.28	50.50	9.97	-6.28	.059	.059	0.114	0.9997		.0232
	23.097	23.099	29.12	0.0	—	111.67	—		2.396	0.9930	.06	35.36	-15.99	51.35	9.69	-6.24	.059	.059	0.114	0.9996		.0320
	23.475	23.477	29.59	0.0	—	110.83	—		2.398	1.0085	.06	35.86	-16.78	52.64	9.54	-6.27	.063	.059	0.114	0.9996		.0386
	23.638	23.639	33.48	0.0	—	102.83	—		2.399	1.0152	.06	39.73	-17.12	56.85	11.31	-6.25	.067	.059	0.110	0.9996		.0418
	23.818	23.819	41.24	0.0	—	89.42	—		2.400	1.0226	.06	47.47	-17.53	65.00	14.97	-6.23	.076	.059	0.106	0.9996		.0453
	23.945	23.945	54.82	0.0	—	70.35	—		2.400	1.0278	.06	53.76	-17.87	71.63	17.95	1.06	.102	.059	0.109	0.9996		.0477
	23.972	23.972	61.79	0.0	—	61.38	—		2.401	1.0289	.06	55.70	-17.95	73.65	18.88	6.09	.128	.059	0.116	0.9996		.0483
	23.999	23.999	81.05	0.0	—	29.26	—		2.401	1.0300	.06	58.90	-18.00	76.90	20.45	24.15	.148	.059	0.217	0.9995		.0487
UNITS	ins	ins.	deg.	deg.	—	ft/s	—	—	ins.	—	—	deg	deg.	deg.	deg.	deg.	—	—	—	—	—	—

Table 5.3

LSRC LOW REACTION DATUM DESIGN AERODYNAMICS AND BLADING DETAILS

BLADE ROW	PASSAGE RADII AT BLADE INLET	PASSAGE RADII AT BLADE OUTLET	$\alpha_{1(s)}$	$\alpha_{2(s)}$	U INLET	Va INLET		No. OF BLADES	STREAMLINE CHORD C	s/c	t/c	α_1'	α_2'	θ	δ	i	AVR	$\Delta H/U^2$	Va/U	D_F	$\frac{\Delta p}{p_1}$	$\overline{\omega}$
	20.440	20.451	0.0	19.2		110.5		72	2.896	0.615	.085	-8.2	19.2	-27.4	0.0	8.2						.0850
	20.881	20.888	0.0	19.2		110.4			2.896	0.628	.085	-8.2	19.2	-27.4	0.0	8.2						.0738
	21.324	21.325	0.0	19.2		110.3			2.896	0.642	.086	-7.8	19.2	-27.0	0.0	7.8						.0650
	21.767	21.763	0.0	19.2		110.2			2.896	0.655	.086	-7.0	19.2	-26.2	0.0	7.0						.0650
IGV	22.208	22.200	0.0	19.2		110.1			2.896	0.669	.087	-6.0	19.3	-25.3	0.1	6.0						.0650
	22.649	22.637	0.0	19.2		110.1			2.896	0.682	.087	-4.8	19.3	-24.1	0.1	4.8						.0650
	23.089	23.074	0.0	19.2		110.2			2.896	0.695	.088	-3.4	19.3	-22.7	0.1	3.4						.0650
	23.527	23.512	0.0	19.2		110.2			2.896	0.709	.089	-1.9	19.3	-22.1	0.1	1.9						.0738
	23.961	23.949	0.0	19.2		110.3			2.896	0.721	.089	-0.5	19.3	-19.8	0.1	0.5						.0850
	20.612	20.617	44.2	12.5	157.4	119.2		79	2.265	0.724	.128	55.7	10.4	45.3	2.1	-11.5	0.9984	.5679	.7576	0.4619	.4618	.0720
	21.015	21.018	44.9	15.8	160.5	119.5			2.259	0.740	.125	55.1	13.2	41.9	2.6	-10.2	1.0034	.5292	.7447	0.4475	.4533	.0540
	21.420	21.419	45.6	18.7	163.6	119.6			2.256	0.755	.122	54.8	16.3	38.5	2.4	-9.2	1.0065	.4960	.7312	0.4354	.4463	.0400
	21.825	21.820	46.4	21.3	166.7	119.2			2.253	0.770	.119	54.5	18.9	35.6	2.4	-8.1	1.0060	.4706	.7155	0.4280	.4454	.0340
ROTOR	22.230	22.221	47.2	23.4	169.7	118.9			2.251	0.785	.117	54.6	21.0	33.6	2.4	-7.4	1.0047	.4509	.7003	0.4235	.4465	.0320
	22.635	22.622	48.0	25.1	172.8	118.5			2.247	0.801	.114	55.0	22.6	32.4	2.5	-7.0	1.0020	.4392	.6854	0.4295	.4520	.0358
	23.037	23.023	48.8	26.5	175.9	118.0			2.243	0.817	.111	55.5	23.9	31.6	2.6	-6.7	0.9964	.4322	.6711	0.4372	.4619	.0450
	23.436	23.424	49.7	28.0	179.0	117.2			2.244	0.831	.108	56.5	25.2	31.3	2.8	-6.8	0.9849	.4281	.6551	0.4512	.4788	.0608
	23.830	23.825	50.5	29.6	182.0	116.4			2.247	0.843	.106	58.0	26.3	31.7	3.3	-7.5	0.9680	.4260	.6397	0.4664	.4999	.0800
	20.619	20.632	47.7	19.2		119.0		72	2.29	0.787	.095	50.7	15.1	35.6	4.1	-3.0	1.002			0.4846	.4902	.0770
	21.018	21.029	46.5	19.2		120.0			2.29	0.802	.095	49.6	15.7	33.9	3.5	-3.1	0.997			0.4683	.4722	.0652
	21.416	21.427	45.5	19.2		120.5			2.29	0.816	.094	48.6	16.0	32.6	3.2	-3.1	0.994			0.4543	.4562	.0550
	21.813	21.824	44.9	19.2		120.2			2.29	0.831	.094	48.0	16.2	31.8	3.0	-3.1	0.995			0.4450	.4442	.0476
STATOR	22.211	22.221	44.6	19.2		119.7			2.29	0.846	.094	47.8	16.2	31.6	2.8	-3.2	0.996			0.4416	.4351	.0440
	22.609	22.619	44.5	19.2		119.0			2.29	0.861	.094	48.0	16.2	31.8	3.0	-3.5	0.999			0.4404	.4313	.0481
	23.008	23.016	44.8	19.2		118.0			2.29	0.876	.094	49.0	15.9	33.1	3.3	-4.2	1.005			0.4447	.4294	.0550
	23.412	23.414	45.4	19.2		115.9			2.29	0.892	.094	50.4	15.3	35.1	3.9	-5.0	1.017			0.4505	.4278	.0605
	23.821	23.811	46.2	19.2		113.1			2.29	0.907	.095	52.0	14.7	37.3	4.5	-5.8	1.035			0.4556	.4251	.0660
UNITS	ins	ins	deg	deg	ft/s	ft/s			ins			deg	deg	deg	deg	deg						

Table 5.4

LSRC LOW REACTION END-BEND DESIGN AERODYNAMICS AND BLADING DETAILS

BLADE ROW	PASSAGE RADII AT BLADE INLET	PASSAGE RADII AT BLADE OUTLET	$\alpha_{1(s)}$	$\alpha_{2(s)}$	U INLET	Va INLET	No OF BLADES	s/c	t/c	α_1'	α_2'	θ	i	AVR	$\Delta H/U^2$	Va/U	D_F
ROTOR	20.612	20.617	49.6	-19.8	157.4	83.3	79	0.763	.134	60.5	-14.5	75.0	-10.9	0.9984	0.813	0.929	0.6917
	21.015	21.018	47.4	7.5	160.5	103.3		0.757	.127	57.2	6.2	51.0	-9.8	0.9996	0.616	0.644	0.5625
	21.420	21.419	46.4	16.8	163.6	114.4		0.764	.122	55.3	14.8	40.5	-8.9	1.0016	0.522	0.699	0.4755
	21.825	21.820	46.4	21.3	166.6	119.2		0.777	.120	54.5	18.9	35.6	-8.1	1.0060	0.471	0.715	0.4316
	22.230	22.221	47.2	23.4	169.7	118.9		0.793	.117	54.6	21.0	33.6	-7.4	1.0047	0.451	0.701	0.4300
	22.635	22.622	48.0	25.1	172.8	118.5		0.808	.114	55.0	22.6	32.4	-7.0	1.0020	0.439	0.686	0.4330
	23.037	23.023	49.0	26.2	175.9	116.7		0.823	.112	56.2	23.5	32.7	-7.2	0.9947	0.438	0.663	0.4511
	23.437	23.424	51.9	24.5	179.0	104.1		0.840	.109	59.2	22.0	37.2	-7.3	0.9846	0.482	0.582	0.5466
	23.830	23.825	56.7	9.5	182.0	81.4		0.870	.108	65.2	10.5	54.7	-8.5	0.9691	0.608	0.447	0.7854
STATOR	20.619	20.632	66.1	35.5		83.2	72	0.787	.095	69.0	30.0	39.0	-2.9	1.0021			0.7471
	21.018	21.029	55.0	25.6		103.2		0.802	.095	58.4	22.0	36.4	-3.4	0.9891			0.5904
	21.416	21.427	48.5	21.5		114.5		0.817	.094	51.0	18.0	33.0	-2.5	0.9848			0.4996
	21.813	21.824	44.9	19.2		120.2		0.831	.094	48.0	16.3	29.7	-3.1	0.9947			0.4453
	22.211	22.221	44.6	19.2		119.7		0.847	.094	47.8	16.2	29.9	-4.2	0.9964			0.4415
	22.609	22.619	44.5	19.2		119.0		0.861	.094	48.0	16.2	30.9	-3.5	0.9992			0.4605
	23.009	23.016	45.7	20.2		116.1		0.876	.094	50.0	16.5	33.5	-4.3	0.9936			0.4627
	23.412	23.414	52.2	24.3		102.7		0.892	.094	57.0	21.2	35.8	-4.8	1.0063			0.5515
	23.821	23.811	64.9	35.4		79.1		0.908	.095	69.3	20.0	29.3	-4.4	1.0358			0.7303
UNITS	ins	ins	deg	deg	ft/s	ft/s				deg	deg	deg	deg.				

Table 5.5

		$-10°$	$0°$	$10°$
$\partial C_s/\partial S_2$	$\dfrac{P_i(P_t/p_t - 1)}{(P_i - S_2 + dS/2)^2}$	0.0113	0.0101	0.0117
$\partial C_s/\partial(dS)$	$\dfrac{-P_i(P_t/p_t - 1)}{2(P_i - S_2 + dS/2)^2}$	−0.0056	−0.0051	−0.0059
$\partial C_s/\partial P_i$	$\dfrac{-(S_2 - dS/2)(P_t/p_t - 1)}{(P_i - S_2 + dS/2)^2}$	−0.0112	−0.0101	−0.0116
$\partial C_s/\partial P_t$	$\dfrac{(S_2 - dS/2)}{p_t(P_i - S_2 + dS/2)}$	0.0128	0.0121	0.0130
$\partial C_s/\partial p_t$	$\dfrac{-P_t(S_2 - dS/2)}{p_t^2(P_i - S_2 + dS/2)}$	0.0129	0.0122	0.0131
δC_s		±0.025	±0.023	±0.025
C_s		0.877	0.830	0.892
$\partial C_T/\partial S_2$	0	0	0	0
$\partial C_T/\partial(dS)$	0	0	0	0
$\partial C_T/\partial P_i$	$\dfrac{1}{P_t - p_t}$	0.0146	0.0146	0.0146
$\partial C_T/\partial P_t$	$\dfrac{-(P_i - p_t)}{(P_t - p_t)^2}$	−0.0140	−0.0143	−0.0138
$\partial C_T/\partial p_t$	$\dfrac{(P_i - P_t)}{(P_t - p_t)^2}$	−0.0005	−0.0003	−0.0008
δC_T		±0.020	±0.020	±0.020
C_T		−0.037	−0.020	−0.056
$\partial C_{yaw}/\partial S_2$	0	0	0	0
$\partial C_{yaw}/\partial(dS)$	$\dfrac{1}{P_t - p_t}$	0.0146	0.0146	0.0146
$\partial C_{yaw}/\partial P_i$	0	0	0	0
$\partial C_{yaw}/\partial P_t$	$\dfrac{-dS}{(P_t - p_t)^2}$	0.0057	0	−0.0046
$\partial C_{yaw}/\partial p_t$	$\dfrac{dS}{(P_t - p_t)^2}$	−0.0057	0	0.0046
δC_{yaw}		±0.017	±0.015	±0.016
C_{yaw}		−0.389	0	0.315

INFLUENCE COEFFICIENT UNITS : $(mmH_2O)^{-1}$

δS_2 , $\delta(DS)$, δP_T , δp_t , δP_i = $\pm 1\ mmH_2O$

Probe E003 Calibration Coefficient Uncertainties

Table 6.1

		CALCULATED	DEDUCED
$\frac{\partial p_t}{\partial P_i}$	$\frac{1 - C_S C_T}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	1.00	0.2
$\frac{\partial p_t}{\partial p_i}$	$\frac{(P_i/p_i)^2 C_S(1+C_T)}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	0.82	0.6
$\frac{\partial p_t}{\partial C_S}$	$\frac{-P_i(P_i/p_i - 1)(1+C_T)}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	−113.8	−112.1
$\frac{\partial p_t}{\partial C_T}$	$\frac{-P_i(P_i/p_i - 1)C_S}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	−96.3	−94.3
$\frac{\partial P_T}{\partial P_i}$	$\frac{C_S^2(1+C_T)(P_i/p_i - 1)^2 + 2C_S(P_i/p_i - 1) - C_S C_T + 1}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	1.01	1.0
$\frac{\partial P_T}{\partial p_i}$	$\frac{(P_i/p_i)^2 C_S C_T}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	− 0.02	−0.05
$\frac{\partial P_T}{\partial C_S}$	$\frac{-P_i C_T(P_i/p_i - 1)}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	1.79	1.72
$\frac{\partial P_T}{\partial C_T}$	$\frac{-P_i C_S(P_i/p_i - 1)\left[C_S(P_i/p_i - 1) + 1\right]}{\left[C_S(1+C_T)(P_i/p_i - 1) + 1\right]^2}$	−97.2	−94.3
$\frac{\partial \alpha}{\partial P_i}$	$-\frac{d\alpha}{dC_{yaw}} \cdot \frac{(s_2 - s_1)}{(P_i - p_i)^2}$	−0.001	−0.002
$\frac{\partial \alpha}{\partial s_2}$	$\frac{d\alpha}{dC_{yaw}} \frac{1}{(P_i - p_i)}$	0.20	0.27
$\frac{\partial \alpha}{\partial s_1}$	$-\frac{d\alpha}{dC_{yaw}} \frac{1}{(P_i - p_i)}$	−0.20	−0.27
$\frac{\partial \alpha}{\partial C_S}$			− 0.29
$\frac{\partial \alpha}{\partial C_T}$			~ 0
$\frac{d\alpha}{dC_{yaw}}$	SLOPE OF CALIBRATION CURVE	23.3	23.6

P_i = 387.0 mmH_2O gauge
p_i = 270.3 mmH_2O gauge

$\delta P_i = \pm 1.0$ mmH_2O $\delta p_i = \pm 1.4$ mmH_2O
δC_S, δC_T, δC_{yaw} FROM TABLE I.

P_t = 388.8 ± 2.0 mmH_2O
p_t = 291.7 ± 3.0 mmH_2O
α = 0.2 ± 0.5°

Table 6.2 Uncertainties in corrected pressures and angle

Low-Reaction Datum Blading at Peak Efficiency

Measured Values are in Brackets

Inner Wall

	%Blockage	Θx (ins)	Hx
Rotor in	5.4 (3.8)	0.116 (0.076)	1.83 (1.82)
Rotor out	3.7 (3.9)	0.096 (0.082)	1.52 (1.69)
Stator out	6.0 (3.8)	0.132 (0.074)	1.78 (1.84)

Outer Wall

	%Blockage	Θx (ins)	Hx
Rotor in	3.5 (3.3)	0.078 (0.088)	1.50 (1.35)
Rotor out	5.4 (3.8)	0.106 (0.095)	1.70 (1.46)
Stator out	3.9 (3.3)	0.089 (0.093)	1.48 (1.28)

Table 7.1 Summary of Axial Boundary Layer Parameters For VN08 Predictions and Measurement