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**GAS TURBINE TRANSIENT PERFORMANCE:
HEAT SOAKAGE MODELLING**

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ABSTRACT

The ability to accurately extract steady state performance information from transient engine test data has the potential to significantly reduce test bed time and cost, depending upon the accuracy with which engine phenomena such as heat soakage and transient tip clearance changes can be modelled within the context of a whole engine performance code.

This thesis describes improvements to the GTTrans generic whole engine performance code thermomechanical heat soakage and tip clearance methods. This thesis describes the 'streams method' in which a collection of finite volumes are used to represent engine component hardware and perform transient heat transfer calculations accounting for both conduction and convection to and from non-trivial engine component geometries. Transient metal and gas temperature profiles are then used to estimate transient blade tip clearances. The streams method has been incorporated into a Rolls-Royce Trent 500 engine performance code. Accounting for heat soakage, tip clearances, and bleed temperature variations within the engine code affected the predicted steady state operation as well as making spool accelerations more sluggish and raising compressor working lines during transient manoeuvres. Although all three effects were significant, heat soakage appeared to have the largest effect on transient performance, particularly during the hot reslam manoeuvre.

Rolls-Royce PLC sponsored this EngD project laying down 5 project requirements that enhanced heat soakage and tip clearance methods should meet, i.e. handle both heat soakage and tip clearances, be physics based, run at or near to real time, meet given accuracy targets, and be capable of being calibrated. The streams method met the first three of these requirements, although with further development it might be possible to meet all five.

This thesis also contains analysis and recommendations regarding the process of technology transfer from the Cranfield University Technology Centre in Gas Turbine Performance to within Rolls-Royce.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A	Area
A	Intermediate variable
A	Current altitude
a	Intermediate variable
Ah	Reference altitude (high)
Alpha	Thermal expansivity at the appropriate metal temperature
B	Change in tip clearance due to presence of bleed
C	Correction factor
C	Instantaneous closure
c	Intermediate variable
Cp	Specific heat
Cs	Strip clearance
D	Instantaneous clearance
D	Total growth
dt	Length of time interval
E	Young's Modulus
F	Intermediate variable
F	Ratio of heat into fluid to work out of fluid
G	Intermediate variable
h	Heat transfer coefficient
K	Thermal conductivity
K1,..., K5	Empirically determined constants
KAQX	Conductance term between a volume and one of its neighbours
L	Length
M	Mass of the metal
M	Intermediate variable
m1, m2	Intermediate variables
Mass	Mass of the metal
MG	Mechanical growth
N	Rotational speed
n	Number of blades on the disc
P	Pressure
P	Pressure difference between interior and exterior surfaces
q	Quantity of heat
R	Radius
r	Radius
t	Time
t	Effective mechanical thickness
T	Temperature
T	Time constant
Tau	Thermal time constant
tau	Thermal time constant
Tm	Metal temperature
Tg	Gas temperature
TG	Thermal growth
U	Conductance term
Utotal	Total conductance term

U1,U2,U3	Individual conductance terms
W	Mass flow rate
X	Polytropic index of compression or expansion
X	Intermediate variable
X	X factor (fraction of component temperature rise)
X	Linear distance
Y	Intermediate variable
Z	Intermediate variable

SUFFIXES

0	start of time interval
1	end of time interval
1	start of process
1	point 1 in the solid
2	point 2 in the solid
2	end of process
26	compressor inlet
30	compressor outlet
b	blade
c	casing
d	datum
f	fluid
fluid	fluid
h	hoop
i	idle
in	inlet
m	max power
m	subject volume
n	neighbour
out	outlet
r	ring
r	radial
s	solid
solid	solid
ss	steady state
t	time

GREEK

α	Thermal expansivity
ρ	Density
ω	Angular velocity
ν	Poisson's ratio of the disc material
δ	Dovetail slippage
θ	Dovetail angle (from a tangent to the disc) in degrees
σ	Stress

Chapter 1: INTRODUCTION

ENGINEERING DOCTORATE

The work outlined in this thesis was performed in partial fulfilment of the requirements of an Engineering Doctorate (EngD) degree programme.

In the early 1990s, the then Science and Engineering Research Council (SERC, now EPSRC) set up a committee, chaired by Dr John Parnaby of Lucas Industries, in response to a belief held in industry, and supported by Government, that the traditional PhD research degree did not adequately prepare researchers for careers in industry. In 1992 following committee recommendations, SERC instituted the Engineering Doctorate (EngD) programme, and created three EngD centres at Warwick, UMIST and Swansea/Cardiff. In 1993 a further two centres were established at Cranfield and Brunel/Surrey. The EngD programme has now become EPSRC's flagship degree programme, and the number of EngD centres throughout the UK has grown to sixteen (EPSRC 2004).

The Engineering Doctorate is a 4 year research degree, awarded for industrially sponsored research and supported by a programme of professional development courses. It provides at least the intellectual challenge of a PhD in a framework of experience that helps prepare Engineering Doctors for industrial careers.

At Cranfield, the taught courses associated with the programme include a series of EngD modules (to broaden engineering awareness), specialist technical modules (usually taken from Masters degree programmes), and the compulsory core modules of the executive MBA at the Cranfield School of Management. Some EngD candidates are based at the sponsoring company whereas others are based at the Cranfield campus. The EngD project for each candidate is overseen by an EngD steering group, which consists of an academic supervisor from the relevant university department, a management supervisor from the school of management, and an industrial supervisor from the sponsoring company. The steering group meets with the EngD candidate approximately four times a year throughout the programme to receive reports from the candidate and to provide guidance on future work. One of the requirements of the EngD programme is that up to twenty percent of the thesis should deal with the business and commercial aspects of the research.

UNIVERSITY TECHNOLOGY CENTRE

The work outlined in this thesis was sponsored by Rolls-Royce PLC at the University Technology Centre (UTC) in Gas Turbine Performance Engineering at Cranfield University.

Rolls-Royce established the concept of the UTC in 1991 following a critical review of existing arrangements with British universities. The emphasis of these UTCs was placed not just on the quality of the individual research undertaken, but also on the level of technology transfer and uptake of the different technologies within the

company (Smith et al 1998). Rolls-Royce currently has over 20 UTCs at universities all around the globe. The UTC arrangement allows Rolls-Royce to reduce cost in its medium and long term research programmes, to access the academic capability base, and to provide itself with a pool of well qualified potential recruits. The UTC arrangement gives universities enhanced ties with industry, access to additional research funding, and projects of real industrial interest to work on.

At each UTC, Rolls-Royce sponsors research associated with a particular theme. In 1998 the UTC in Gas Turbine Performance Engineering was set up at Cranfield University, to conduct research into fields such as novel engine cycles, engine diagnostics, test bed simulation, windmilling, altitude relight, and engine/airframe integration.

TYPE OF CONTRIBUTION

Cranfield University's General Handbook for all Students at the Cranfield Campus gives guidance on the academic standards required in theses for the degree of EngD. According to the handbook an EngD thesis should:

"...demonstrate a significant contribution to learning; for example, through the discovery of new knowledge or the application of existing knowledge to new situations, the connection of previously unrelated facts, the development of a new theory or design, or the significant revision of older views or the improvement of an existing design."

The work outlined in this thesis falls into the category of 'the improvement of an existing design', more specifically the improvement of the GTTrans thermomechanical heat soakage and tip clearance methods, as described in the following chapters.

CHAPTER OUTLINE

Chapter 2 of this thesis describes the business case that led to sponsorship of this project. It gives a review of the relevant open literature, i.e. whole engine performance modelling; heat soakage phenomena; and existing heat soakage phenomena models. The chapter lists the project requirements laid down by the sponsor, identifies the research gap to be filled by this piece of research, and defines project scope. The purpose of the chapter is to justify the project scope and objectives.

Chapter 3 of this thesis describes the approach taken by the author to model convection, conduction, radiation, and thermomechanical engine geometry changes that occur transiently within a gas turbine engine. Sample calculations are compared against studies using the finite element analysis package ANSYS®. The chapter also describes how the author integrated these new methods into the whole engine transient performance code provided by the sponsor. The purpose of the chapter is to present the heat soakage and tip clearance models developed by the author.

Chapter 4 of this thesis describes the results of a series of studies making use of the methods developed by the author. The first study involved use of the new heat soakage and tip clearance methods to construct a multistage model of a Rolls-Royce Trent 500 high-pressure compressor. The results it produced were compared to those produced by a more authoritative method. The second study involved constructing a simplified single stage model of the same compressor and comparing the results it produced to those in the first study. The third study involved the new methods being used to construct multistage models of almost all of the turbomachinery in the Trent 500 engine. Comparative transient test runs were made between the original engine code, the code updated with the new models and methods, and a set of test data provided by the sponsor. The chapter also presents a discussion of the possibility of extracting steady state engine performance information from transient test data. The main purpose of the chapter is to establish the extent to which the heat soakage and tip clearance methods developed by the author meet the project requirements laid down by the sponsor.

Chapter 5 addresses the management topic associated with this EngD project, i.e. technology transfer from university to sponsor. It gives a brief review of open literature relevant to the topic. It then describes the Cranfield UTC arrangement as it currently stands. The chapter then goes on to describe the technology transfer issues faced by the UTC. It then describes the author's proposal for a possible addition to the Cranfield UTC quality plan that would help address these issues. The chapter also discusses technology transfer issues relating to the heat soakage and tip clearance methods developed by the author, and ends with a list of recommendations.

Chapter 6 of this thesis is the conclusion. It describes the author's contribution. It also lists the author's recommendations for future work in the field of heat soakage and tip clearance modelling in a whole engine performance context.

Appendices at the end of this thesis support the material presented within each of the individual chapters.

Chapter 2: BACKGROUND

2.1 CHAPTER INTRODUCTION

The purpose of this chapter is to present and justify the scope and objectives for the project outlined in this thesis.

First the underlying business case for project sponsorship is discussed. Next, a survey is given of the open literature in the areas of engine performance models, heat soakage phenomena, and heat soakage modelling. After that, the research gap associated with this project is given. And finally the project scope and objectives are presented.

2.2 BUSINESS CASE

THE INDUSTRY

In October of 1932 Frank Whittle was granted a patent for the first turbojet engine. Working independently, Hans Von Ohain's gas turbine brought about the world's first jet powered flight in 1939. In the same year the Swiss company Brown Boveri completed development of the first modern land-based gas turbine. Since that time the world gas turbine industry has enjoyed continued growth. In 1999 the value of production of gas turbines was US\$34bn (Langston 2001). A significant part of the gas turbine industry is the large civil turbofan sector. Over the last 25 years the airline industry has grown at 6.6% per annum (Singh 2001). In spite of recent world events the large civil turbofan sector continues on its trend of long term growth. Figure 2.2.1 shows Rolls-Royce figures for engine delivery values for the past 20 years, and their expectation for the next 20 years (Miller 2004).

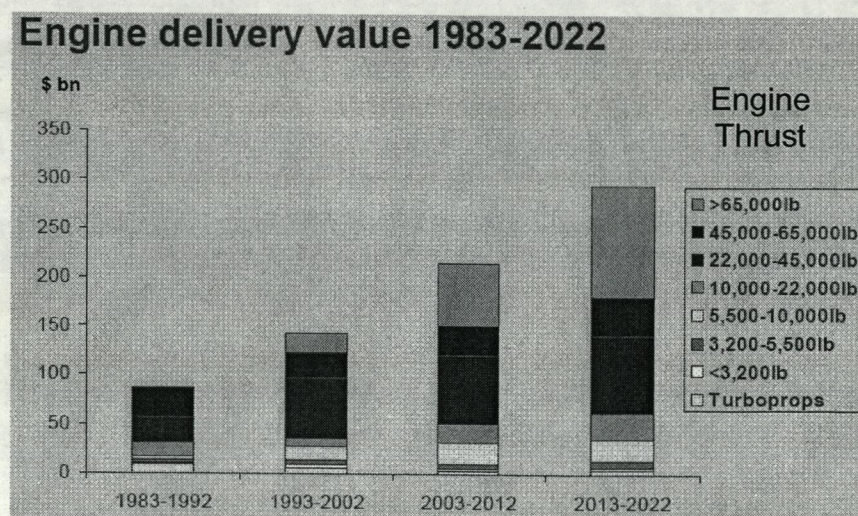


Figure 2.2.1 Aero engine delivery values (Miller 2004)

The large civil turbofan sector has just 3 main players: General Electric, Pratt and Whitney, and Rolls-Royce. General Electric (GE) of the US is a large conglomerate engaged in businesses ranging from jet engines to medical instruments to news and information. In 2003 GE's revenues totalled \$134bn, and sales for 'GE Transportation' (who are responsible for jet engines both military and civil, rail, etc.) totalled \$13.5bn (GE 2003 annual report). Pratt & Whitney (PW) also of the US, are a division of the United Technologies Corporation (UTC), who manufacture a host of goods from air conditioners to elevators and escalators. In 2003, UTC's revenue was \$31bn (UTC 2004). PW specialises in aerospace propulsion (civil, military, rockets), and in 2003 its revenues totalled \$7.5bn. Finally there is Rolls-Royce of the UK, who is not part of a larger conglomerate. Rolls-Royce produces gas turbine engines for power on land, sea, and air. About half of Rolls-Royce sales are in the civil aerospace sector. In 2003, Rolls-Royce revenues were £5.6bn (Rolls-Royce 2003 annual report), which is equivalent to \$10.2bn at August 2004 exchange rates. GE leads the overall market for turbofan engines (Opdyke & Lindstrom 2000). At the time of writing, the Rolls-Royce Trent family of engines had a 50% share of the large civil turbofan business, i.e. engines for twin aisle aircraft (Rolls-Royce 2004a).

This small oligopoly of large suppliers is typical of a maturing market. There is intense rivalry between the three. Large civil turbofans are very high technology products and specialist knowledge with any competitive value is kept securely in-house within these big three companies. Any new supplier wishing to enter the market would be unlikely to be able to make gas turbines as cheaply or to as high a standard as the incumbents, and would need to make massive investments in both R&D and manufacturing equipment. In addition, positive return on investment for new engines can take decades to materialise. This sector clearly has very large barriers to entry.

In recent years significant changes have taken place within the industry. Engine manufacturers are becoming more and more involved with aftermarket activities. The pressure to reduce turbofan list prices is forcing the engine manufacturers to turn increasingly to aftermarket activities to recover costs. The proportion of the Rolls-Royce civil engine fleet covered by TotalCare™ agreements now stands at 30%, and this is rising. Such agreements provide the operator with a fixed engine maintenance cost over an extended period of time. Operators are assured of an accurate cost projection and avoid the costs associated with unscheduled maintenance. In the first half of 2004, Rolls-Royce aftermarket activities represented 58% of civil engine sales (Rolls-Royce 2004 interim results).

Intense competition, market sensitivity to world security issues, slow return on investment, and increasingly strict noise and emissions regulations make for an extremely harsh business environment. In order to reduce business risk and reduce the typically huge costs of developing new engines companies often get together to form multi-company alliances. Current examples are the Engine Alliance GP7000 engine project (General Electric Pratt & Whitney), and the International Aero Engines V2500 engine project (Rolls-Royce, MTU, Pratt & Whitney, JAEC).

NEW ENGINE DEVELOPMENT

In 2000, Phil Ruffles (Director of Engineering and Technology, Rolls-Royce) speaking of the future of aircraft propulsion said that the aero-engine industry is fiercely competitive and best practice methods are being employed throughout Rolls-Royce to minimise the non-recurring cost of development (Ruffles, 2000). He stated that the acquisition costs of aero-engines are directly related to the efficiency at which a manufacturer can manage its product development.

Rolls-Royce is characteristically proactive in its attitude towards reducing the cost of ever better engines. New engine development programmes typically take years to complete and cost hundreds of millions of pounds. A considerable proportion of that time and cost is associated with engine development testing. Accurate engine simulation can help minimise costly trial and error development testing.

During development testing, following assembly of a new build of engine, the engine is placed on a test bed and its performance is measured under a variety of operating conditions. One such test involves running the engine at low power until the engine carcass has thermally stabilised, or 'soaked', and has a steady state metal temperature profile. At this point a set of engine measurements are made. The fuel flow is then stepped up a little, and test engineers again leave the engine to thermally stabilise (for a nominal 3 to 5 minutes) before taking a further set of engine measurements. This procedure continues until measurements at all of the engine settings of interest have been taken. The result is a profile of the steady state performance of the engine. In total this test can take two hours or more to complete (Martin 2001). Most of this time is spent waiting for the engine to thermally stabilise before taking measurements.

THE BUSINESS CASE FOR SPONSORSHIP OF THIS PROJECT

Prior to the start of this project, the Rolls-Royce engineers in charge of the Cranfield UTC together came up with the idea that if the thermal stabilisation phenomena could be accurately understood and quantified (i.e. modelled), then during development testing instead of increasing fuel flow in a series of steps each time waiting for thermal stabilisation to occur, perhaps a single transient manoeuvre could be performed in which transient engine measurements would be taken continuously. This would take only a fraction of the time to complete (c.f. figure 2.2.2). Using accurate thermal models it might then be possible to post process these transient engine measurements to estimate the steady state performance of the engine. If this could be done successfully, then it might be possible to achieve development testing cost reductions of up to an order of magnitude (Curnock 2001). The business case for developing accurate heat soakage phenomena models is clearly quite compelling.

In addition to the potential cost savings associated with a reduction in development test bed time, superior modelling capability of heat soakage phenomena in whole engine performance codes could assist the understanding of the transient turbine entry temperature overshoot experienced by engines following acceleration manoeuvres. Engine life is particularly sensitive to temperature overshoots, and time between overhauls is a key economic driver for gas turbine users.

In addition, the ability to more accurately model how engine stability margin vary throughout transient manoeuvres could potentially improve our ability to produce safer engines (Fiola 2002).

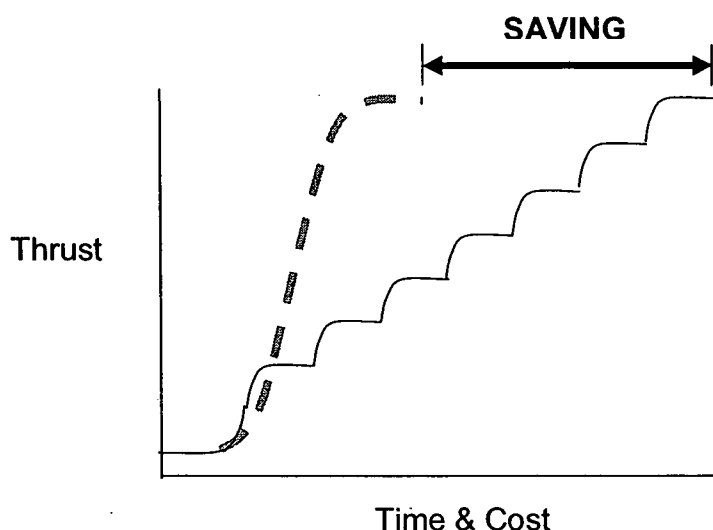


Figure 2.2.2 Potential development testing time and cost saving to be made with the appropriate use of accurate heat soakage models.

2.3 LITERATURE REVIEW

2.3.1 SECTION INTRODUCTION

The purpose of this literature review is not to introduce the basics of gas turbine theory; rather it is to outline the information available in the open literature that is relevant to this project.

The first part of this section very briefly introduces gas turbine performance modelling, and then goes on to discuss a selection of whole engine performance codes that can be found in the open literature. The second part of this section goes on to describe the types of physical phenomena that are associated with gas turbine transient manoeuvres and heat soakage. The final part of this section introduces a selection of methods reported to have been used to model the various heat soakage phenomena.

This section serves as a useful introduction to chapters 3 and 4 of this thesis.

2.3.2 WHOLE ENGINE PERFORMANCE MODELLING

PERFORMANCE MODELLING

Detailed descriptions of the fundamental concepts behind design point, steady state, and transient whole engine performance modelling are readily available in the open literature (e.g. Walsh & Fletcher 1998, Cohen et al. 1996, and Fawke & Saravanamuttoo 1971).

Design point performance calculations define a new engine's cycle parameters, configuration, component performance levels, and component sizes required to meet a particular market need. Only when design point calculations are complete can the performance of the engine at other operating conditions be attempted.

In order to perform steady state off design performance calculations, generally a whole engine performance code must be constructed. Within this code, the steady state performance of each component under a variety of operating conditions is represented on a set of maps. When performing steady state calculations, for any given handle (e.g. fuel flow, thrust, etc.) the whole engine performance code iteratively selects operating points on each of the engine component maps until the matching constraints of rotational speed, mass flow, and powers associated with each component are compatible with all of the other components in the engine.

When performing transient performance calculations two different approaches are available. There is the continuity of mass flow (CMF) method, which is particularly suited to slow transients. And there is the inter-component volumes (ICV) method, which is particularly suited to fast transient. The CMF method operates in a very similar way to that of the steady state matching routine in that it is iterative, but instead of matching speeds, mass flows, and powers, it only matches rotational speeds and mass flows. The residual shaft power imbalances are used along with shaft inertias and a time interval to calculate shaft accelerations and thus shaft speeds for the next time interval. The ICV method is straight through rather than iterative, but to prevent instability the method uses very small time intervals between calculations. At each time interval the difference in component mass inflows and outflows are used to calculate component inlet and outlet temperatures and pressures. Residual shaft power imbalances are used to calculate accelerations and rotational speeds for the next time interval. The ICV method is better for fast transients because it accounts for mass packing in the primary gas path volumes associated with each component. The CMF method is better for slow transients where the assumption of zero mass flow accumulation within components is more justifiable.

WHOLE ENGINE PERFORMANCE CODES

There are a great many steady state and transient gas turbine performance codes reported in the literature. Some are only able to perform steady state calculations, whereas others can also perform transient calculations, and many of them are related, i.e. based on the same original source code. Sanghi et al (2000) gives quite a comprehensive overview of engine simulation during the past 50 years. The paper concludes by saying that recent developments in whole engine performance modelling

are mostly to do with computational platforms, implementation techniques to speed up execution times, and strong graphic user interfaces. The basic ideas underlying the methods are quite mature. In recent years there have been no radical changes to mathematical modelling and component representation (Sanghi et al 2000).

Early analytical and experimental investigations into gas turbine dynamic behaviour began in the 1950's. Otto and Taylor (1950) were probably the first to show that a whole single spool engine could be approximated by a first order system to determine the rotor speed response to a step change in fuel flow (Sanghi et al, 2000). As years went by, linear models were soon replaced by the far more versatile engine component based analogue and digital computing approaches, such as those described by Fawke and Saravanamuttoo (1971). The digital engine component based approach has now become very widespread and has been adopted to develop essentially all of the performance models around today.

At Cranfield University, Palmer (1968) developed TURBOCODE, a component based steady state performance prediction programme from which another steady state code was developed, i.e. TURBOMATCH (Palmer 1981). TURBOTRANS was then developed (Palmer & Yan 1982) which could perform transient calculations. TURBOTEST and TURBOFLEXI (Palmer & Gu 1983) then followed which were developed to simulate engine tests. Not long after TURBOCAL (Douglas 1986) was developed in an attempt to combine the functions of the above programmes into one single programme, and to add the capability to produce performance graphs. Since that time Cranfield's gas turbine performance efforts have largely focussed upon development of its steady state modelling capability, although more recently there has been a revival of interest in transient performance modelling (i.e. the author's project).

Authors from all over the world have written papers about gas turbine transient performance codes, for example Torella (1989) Italy, Duponchel et al (1992) France, Zhu et al (1996) China, Akhmedzianov (1997) Russia, Kong et al (1999) South Korea. Some of them include heat transfer between working fluid and engine metal, whereas others do not (notably the land based gas turbine engine codes). Many authors have written their source codes in computer languages such as FORTRAN or C++, whereas some have constructed transient engine models using generic digital simulation environment packages such as SIMULINK (Bettocchi et al 1996, Crosa et al 1998, Kim 1999), ACSL (Dale 1992), and TURVIS (Blotenberg 1993). Pilidis (1983) at Glasgow University developed a code named GTTrans, which was unique in that it included a built in thermomechanical model of the engine, albeit simplified, to calculate mechanical and thermal tip clearances and seal geometry changes and their effects on performance. The code was developed to include pioneering heat soakage work carried out by Maccallum (1970, 1973, 1977, 1979). It included the effects of heat soakage on compressor boundary layers, bulk fluid heating/cooling, and their effect on component characteristics and surge margins.

Available free on the internet is a demo version of GSP (NLR 2001). Development of the programme was initiated in 1986 at the Aerospace Department of the Delft Technical University in the Netherlands. It was based on NASA's DYNGEN programme (see below) but improved on numerical stability and extended the range of engine configuration simulations. It computes design, off design, and transient

performance calculations for all kinds of gas turbine engines. Bulk heat transfer and mass packing are the only transient phenomena included.

NASA has been very active in gas turbine performance simulation from the beginning. Their transient performance efforts have produced such codes such as DYNGEN (Seller & Daniele, 1975), DIGTEM (Daniele et al, 1983), DEAN (Sadler and Melcher, 1985), HYTESS 2 (Merrill, 1986), and others. With time they have grown in sophistication. GETRAN (Schobeiri et al, 1994), has been developed under contract with the NASA Lewis Research Centre. It was based on a code called COTRAN developed by Schobeiri whilst at Brown Boveri Company, Switzerland, in the mid 80's. GETRAN is capable of simulating the nonlinear dynamic behaviour of arbitrarily configured engines. Row-by-row diabatic and adiabatic calculation procedures are implemented and flow angles are found from row velocity diagrams. Off design efficiencies are calculated from total loss correlations. The code is capable of simulating aircraft engines having up to 5 spools with variable geometry and blade cooling. Heat transfer coefficients are obtained from Nusselt number correlations. The combustion chamber temperature calculations include convection and radiation. Transients including rotating stall and surge in individual compressor stages can be simulated. Different numerical solvers are used depending on the strength of the transient. Computing time varies from faster than real time to 10 times real time depending on the engine configuration and nature of the transient.

NASA has taken the next step in terms of added detail and is now pursuing its Numerical Propulsion System Simulation (NPSS) project. NPSS focuses on the integration of many different disciplines such as aerodynamics, structures, and heat transfer along with the concept of numerical zooming between 0- dimensional to 1-, 2-, and 3-dimensional component codes. 0-dimensional analysis is equivalent to using the standard steady state component map for the calculation. 1-dimensional analysis is where gas properties are calculated along the length of a particular component. 2 and 3-dimensional analysis extend to the use of Finite Element Analysis and Computational Fluid Dynamics software (see Figure 2.3.2.1).

Convergence of the system is arrived at in the following way. First the 0-D NPSS model comes to a thermodynamic balance, i.e. the engine code converges. The 1-D model of say the HPC is then executed at the same operating conditions as the 0-D HPC. The higher fidelity 1-D HPC will have a slightly different operating point, which is allowed to 'float' a little. The new 1-D operating point is then imposed upon the 0-D component map and the whole engine simulation is again brought into thermodynamic balance. In like manner higher order analyses can be incorporated. Directly replacing the 0-dimensional code with its 1-dimensional counterpart was not attempted as this technique would have destroyed the plug and play capability of the NPSS and would have precluded use of a consistent methodology for even higher order analysis, e.g. 3D CFD (Follen & auBuchon 1999).

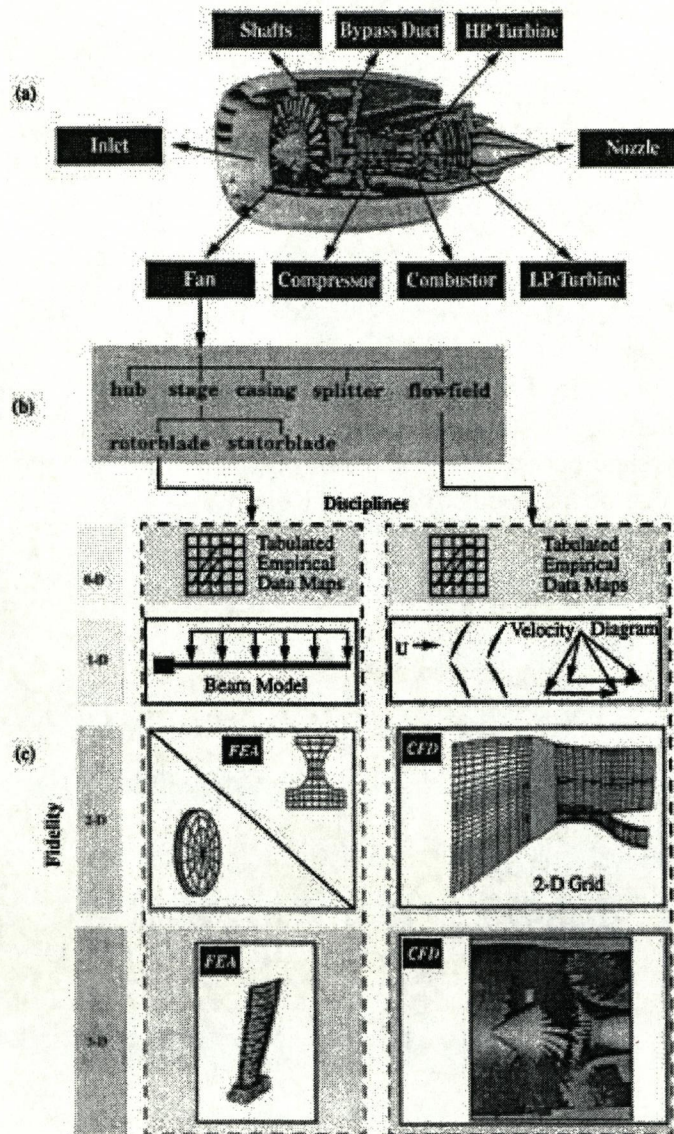


Figure 2.3.2.1 The 'zooming' principle (Reed & Afjeh 2000).

One potential problem with the zooming approach is when a low order component model (say 0-D IPC) is used upstream of a high order component model (say 3-D HPC). The problem is that the accuracy of output from the higher order model can never be any greater than the accuracy of data fed to it by the low order model. However, this situation also occurs when running a 3D CFD analysis as a standalone simulation - one must assume specific inlet and exit conditions (typically a velocity or pressure profile). Zooming is still quite new, and it is extremely difficult to go from 0-D to 3-D, especially without human intervention. However, going from 0-D to 1-D in compressor models can be done successfully (Reed 2001).

Not fully available in the open literature are descriptions of the engine manufacturers' proprietary performance codes. An example of which is the Rolls-Royce Aero-engine Performance code (RRAP). RRAP can perform steady state and transient performance calculations (ICV method) and has been used to develop many of the world's best engines.

2.3.3 HEAT SOAKAGE PHENOMENA

This subsection starts with a brief discussion of engine heat flows, and follows with a description of how these heat flows observably affect transient engine performance. The remainder of this subsection is divided into a discussion of how heat soakage affects the geometry, aerodynamics, and thermodynamics of a gas turbine engine.

TRANSIENT HEAT FLOWS

Figure 2.3.3.1 shows the difference in temperature profile throughout a large triple spool civil turbofan engine between idle and full power. The figure shows that the temperature difference between idle and full power in all of the engine's components (except the fan), can be hundreds of degrees Kelvin.

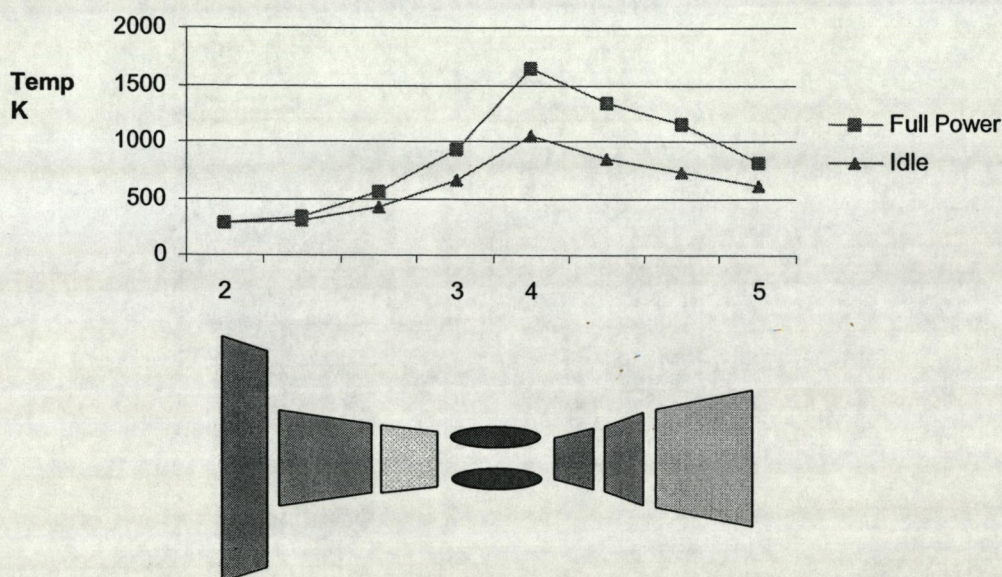


Figure 2.3.3.1 Typical Large Turbofan Engine Station Temperature Profiles at Idle and Full Power

The large temperature differences between the engine metal and primary gas path give rise to large heat flows, which can significantly affect engine performance both during and after a transient manoeuvre.

EFFECTS OF HEAT SOAKAGE ON ENGINE PERFORMANCE

A number of early researchers made investigations into how gas turbine engines behaved during and following engine transient manoeuvres. Bauerfeind (1968) and Thomson (1975) reported that the time taken for an engine to complete a speed

transient could be 20 to 30 percent longer than predicted by adiabatic transient performance models. Maccallum (1970) reported that a 1.5% to 2% thrust shortfall below the stabilised value for a typical single spool engine at the end of an acceleration speed transient often occurred, and that this shortfall could persist for some minutes. Maccallum & Qi (1989) reported that when an engine is rapidly re-accelerated following a slam deceleration from a fully soaked high power setting (i.e. the hot reslam or Bodie manoeuvre), then surge is more likely to occur than if the same engine is accelerated from cold (i.e. a cold slam acceleration). These effects occur as a consequence of heat soakage.

The effect of heat soakage on an engine's performance depends very much on the size and type of engine. Over the years engines have tended to become lighter and more powerful, and have developed sophisticated cooling and thermal barrier coating systems. It is unlikely therefore that heat soakage will affect a newer lighter engine to the same extent as an older heavier engine. The speed with which the transient manoeuvre is carried out also has a large effect on the severity with which heat soakage effects are experienced by the engine. The more slowly a transient manoeuvre is performed then the more closely the engine approaches steady state behaviour and the effect of heat soakage will become less observable. Conversely, in some engines during fast transients as much as 30% of the over-fuelling used to accelerate an engine can be absorbed by the engine carcass via heat soakage (Walsh and Fletcher 1998). Furthermore, the severity with which heat soakage effects are experienced also depends upon where the engine is in its operational envelope. Pilidis (1983) showed that the effects of heat soakage are more noticeable at altitude than at sea level, largely due to the fact that at altitude mass flows and heat transfer coefficients are much lower, resulting in a slower engine thermal response.

GEOMETRY

During transient manoeuvres the geometry of gas turbine components undergoes changes due to both thermal and mechanical influences. When solids are heated and cooled they expand and contract, and when solids are subject to centrifugal stresses or stresses due to differential pressures they deflect. In addition, the mechanical properties of engine materials all change with temperature. The thermal and mechanical behaviour of gas turbine components are interconnected.

Maccallum (1973) reported that when engine components are heated up and cooled down primary gas path flow areas can change. In addition, blade tip clearances and labyrinth seal clearances are also likely to be altered.

The rate of change of thermal expansion of any particular piece of metal is dependent upon its thermal capacity, the temperature difference between it and the fluid with which it is in contact, and convective heat transfer coefficient that exists at their interface. Figure 2.3.3.2 shows a schematic of a single stage of turbomachinery. In the primary gas path mass flows and thus heat transfer coefficients are high and blades and vanes are generally very thin. As a consequence, the thermal response times for blades and vanes are relatively short. Metal thickness in the casing is also quite thin; however bleed mass flows and thus heat transfer coefficients are generally quite small. This results in casing thermal response times that are generally longer

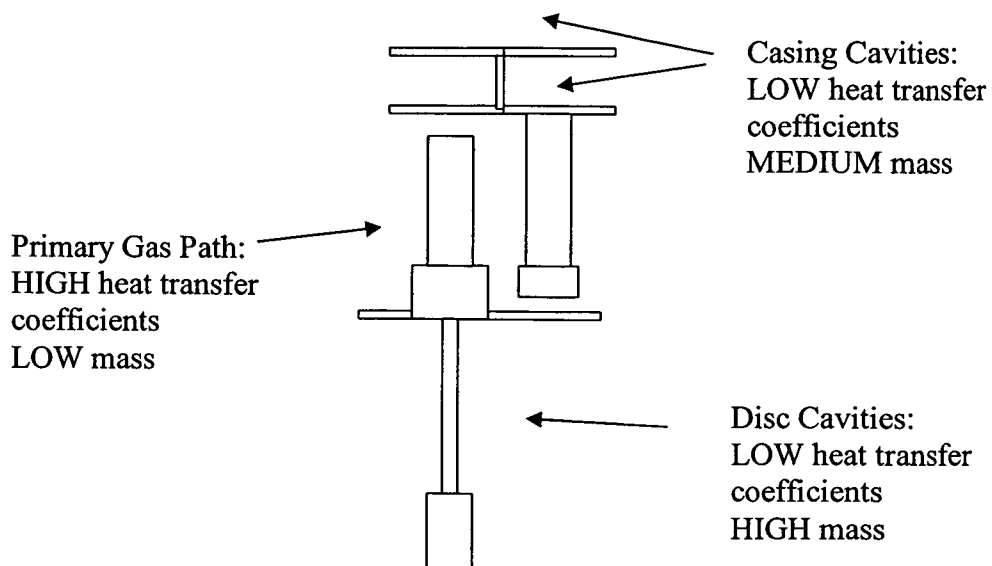


Figure 2.3.3.2 Transient convective heat transfer conditions in different parts of a single stage of turbomachinery

than for blades and vanes, especially when casings have thermal barrier coatings or are double skinned, in these cases conduction rather than convection can be the rate determining process. Mass flows in the disc cavities can be similar to those in the casing cavities, however, high rotational speeds can result in much higher heat transfer coefficients. Disc cobs and diaphragms are generally relatively thick and so thermal response times of the rotor drum can often be much longer than that of the casing. In addition to thermal expansions, when an engine changes speed then centrifugal stresses in the rotating parts also change leading to change in the radial dimensions of those parts. Figure 2.3.3.3 shows how these effects combine to alter radial blade tip clearances from the time the engine leaves the hanger until cruise.

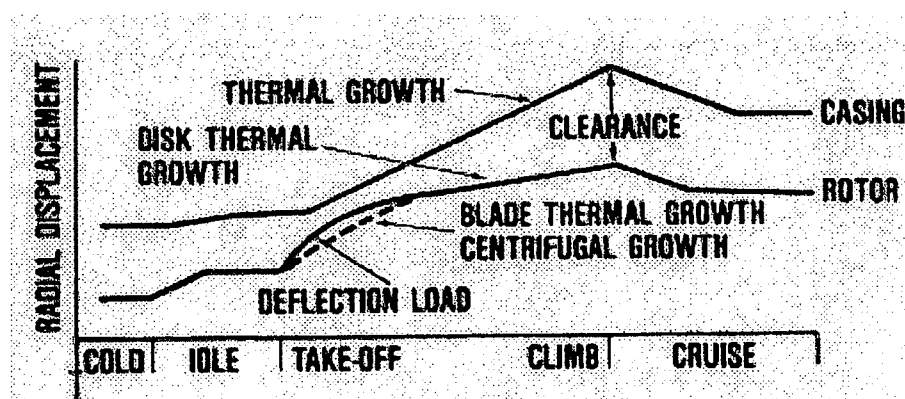


Figure 2.3.3.3 Transient radial blade tip gap changes from leaving the hanger until cruise (Booth 1985)

Figure 2.3.3.3 shows that transient tip clearance behaviour is not trivial. Differential rates of thermal growth between casings blades and discs can lead to transient tip clearances that are quite different from their steady state values at corresponding instantaneous operating conditions. Figure 2.3.3.4 shows how tip clearance might be expected to change during a slam acceleration manoeuvre.

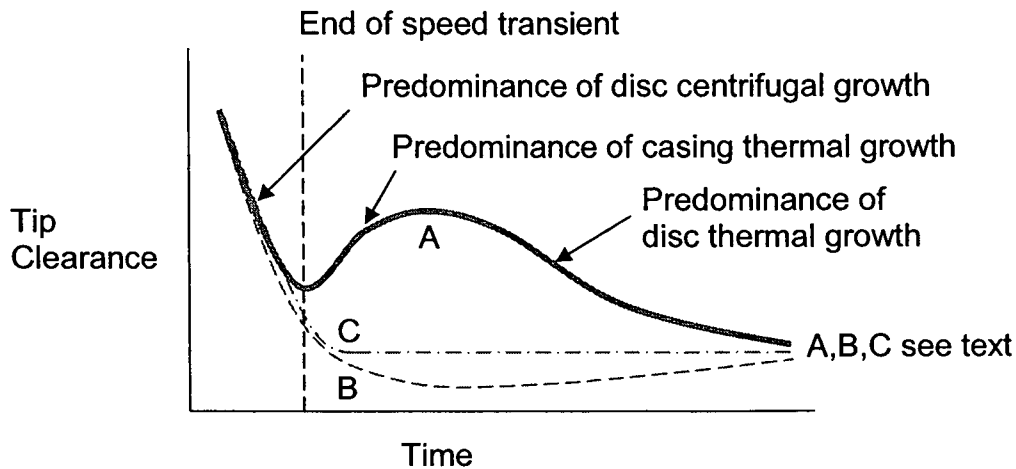


Figure 2.3.3.4 Transient radial blade tip gap changes during and following a slam acceleration (Naylor 2002a)

In case A of figure 2.3.3.4 the thermal response of the casing is assumed to be faster than that of the rotor, and so following the speed transient tip gaps open up slightly as the casing thermally grows faster than the rotor. Over time the thermal growth of the rotor catches up eventually bringing the blade tip clearance down to its steady state value at the higher power setting. Case B in figure 2.3.3.4 assumes that the thermal response of the casing is slower than that of the rotor, the result is a tip clearance which reaches a minimum as the rotor thermally stabilises but which opens up again until the casing reaches its steady state thermal growth. Case C shows the situation where thermal growth in the casing and the rotor takes places at approximately the same speed.

During transient manoeuvres, axial components of blade tip clearance can change as well as radial components. The pressure difference between the pressure and suction surfaces of rotor blades leads to blade bending. Furthermore, compressor pressures rise from front to rear, but in the turbines pressures fall from front to rear. This results in significant amounts of mechanical shaft stretch in addition to shaft elongation due to thermal expansion (Wilson 2003). This can be particularly problematic when rotors and casings are conical rather than cylindrical, i.e. have had angle. Figure 2.3.3.5 (Neal 1982) shows how tip clearance can change due to axial movements in components with had angle. Where casings have a non-zero had angle tip clearance calculations should include both axial and radial components. The axial component of tip clearance increases with increasing had angle, and gets worse the further away the blade row is from a fixed (as opposed to floating) shaft bearing.

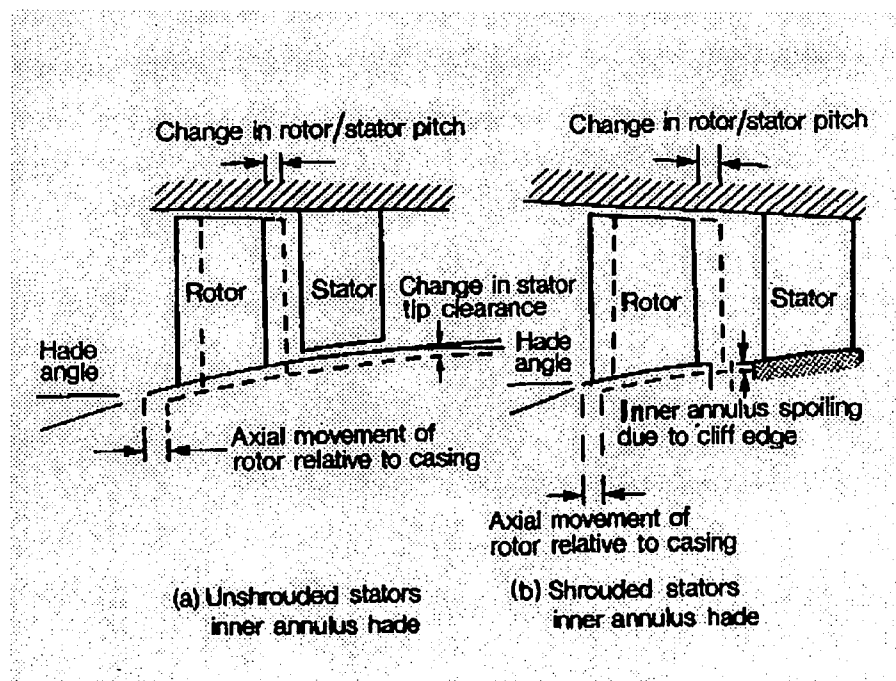


Figure 2.3.3.5 Axial movement of rotor relative to casing (Neal 1982).

Many researchers (e.g. MacCallum 1979, Neal 1982, and Pilidis 1983) report that during transient manoeuvres losses in performance can occur due to seal clearance changes. This is certainly true when labyrinth seals are used. More recently, brush seals have been developed which have much lower leakage rates than traditional labyrinth seals (Holle 1992). Their bristles continually touch their contact surfaces making flow rates less sensitive to geometrical changes. However, brush seals are not appropriate in every location, and strategically placed choked orifices are used to control sensitive flows in many modern engines (Rowe 2000).

AERODYNAMICS

Flow through turbomachinery blade passages is highly complex. Turbulence, boundary layers, vorticity, and shockwaves all contribute to total pressure loss and component inefficiency. For a compressor, Foley (1995) gives descriptions of the following flow structures:

Passage vortices – the slower moving fluid in the hub boundary layer turns through a greater angle in the blade passage than the bulk of the flow maintaining exit plane pressure profile, resulting in the passage vortex.

Blade boundary layers – the blade boundary layers grow along the length of the blade before leaving the trailing edge to form the wake. The thinner the boundary layers, the thinner the wake, and the lower the profile losses. Turbulence in the boundary layer also affects losses.

Annulus wall boundary layers – intense pressure gradients, tip leakage flows, and other steep variations in endwall flows make conventional boundary layer representation not even approximately valid in the endwall regions. However, for want of a better method, boundary layers are generally used to model the endwall flows.

Corner stalls – the casing boundary layer and the blade suction surface boundary layer interact causing the flow to separate and stall the blade in the suction corner.

Scraping vortices – as the blades move relative to the casing they scrape up the endwall boundary layer. This then rolls up into a vortex alongside the pressure surface of the blade.

Trailing vortices – these occur as in isolated airfoils, however, the presence of other blades, casing boundary layers, centripetal forces etc, make description of this flow much more complicated.

Horseshoe vortices – these are generated by incoming wall boundary layers meeting the blade leading edge at the tip and root of the blade. The boundary layer flow then rolls up into a vortex before progressing through the blade passage.

Tip leakage flow – believed to occur due to the pressure difference across the tip of the blade and of undeflected fluid in the annulus wall boundary layer. This is generally agreed to be the largest contributor to inefficiency and reduced stall margin. The roll up of the jet into a vortex and its part in loss generation is an area of continued research.

Shockwaves – these are becoming more important as turbomachinery is developed to rotate faster and faster. The study of shockwaves in blade passages is a large area of research.

Shock losses of approximately one third of the total pressure losses can occur in blade tip regions with high Mach numbers. The location and angle of the shock affect the magnitude of the losses (Schobeiri 1998).

Unsteady effects – such as wake chopping are very complicated and involve consideration of all of the secondary flows. Wake chopping causes engine vibration.

In the turbine flows are also very complicated; Figure 2.3.3.6 shows the review of flow phenomena occurring in a turbine blade passage from Yamamoto (1988).

To give an idea of the magnitude of the aerodynamic losses Carter (1949) reported that for compressors with 90% polytropic efficiency, the remaining 10% is made up of 4% blade profile loss, 4% secondary flow losses, and 2% annulus wall losses. More recently Storer & Cumpsty (1994) state that as much of 30% of the total loss in compressors is sometimes attributed to the flow through the tip clearance, but that they suggests 10% is a more realistic figure, and for well designed compressors this corresponds to only about 1% decrement in overall efficiency. They also state that in the past it is probably that some of the loss in the endwall region has been wrongly attributed to clearance flow.

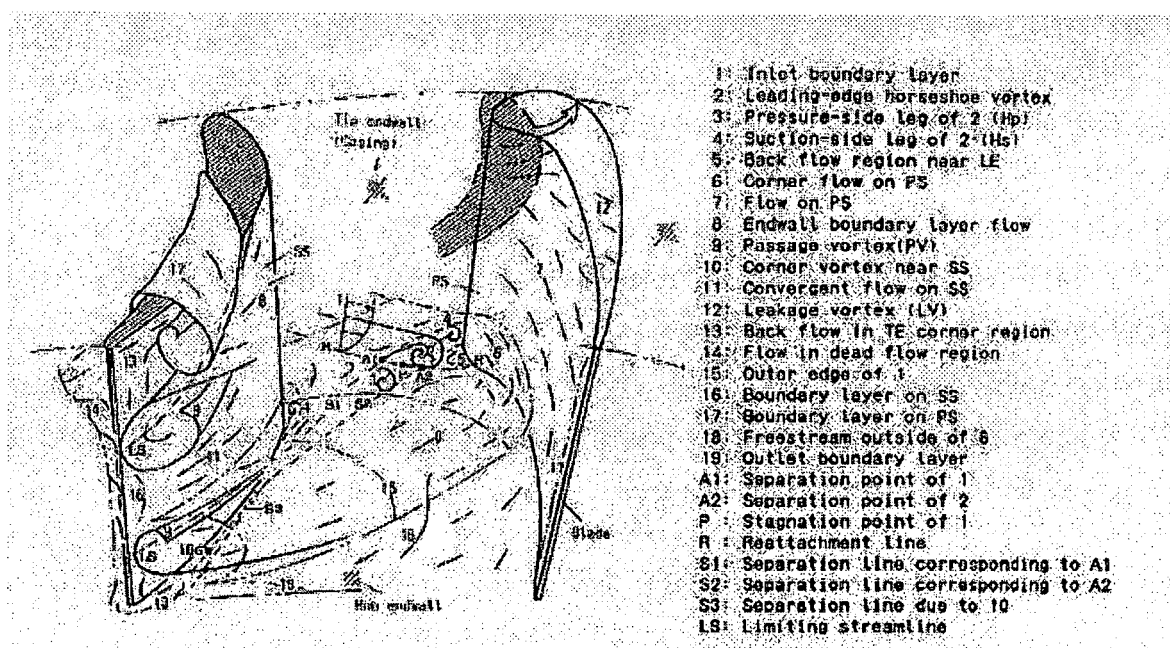


Figure 2.3.3.6

Review of flows in turbine passage drawn with the right blade a little opened in order to see the flows near the pressure surface. (Yamamoto 1988).

Quantification of the sizes of these various loss phenomena is clearly a difficult task. Moreover, the relative importance of any one of these flow structures is invariably dependent on the individual component geometry, its free-stream flow and the presence of other interacting secondary flows. What is more, these secondary flows are all inter-related, so much so that Chen & Tsai (1998) using computational fluid dynamics have advocated that for a compressor a non-zero tip clearance exists at which the passage vortex interacts with the tip leakage vortex in such a way that overall vorticity strength is reduced, resulting in an optimum non-zero tip clearance. This sheds doubt on the assumption that component efficiency is highest at zero tip clearance. Chen & Tsai (1998) show that secondary flow structure interaction cannot be ignored.

Roy & Kumar (1999) outline a method for incorporating models for flow losses into a compressor performance prediction code. With it, they were able to estimate with some success the performance characteristic of many different multi-stage axial flow compressors. They used a mean line performance estimation method incorporating stagnation pressure loss coefficients to account for secondary losses, hub end-wall losses, tip clearance losses, flow angle deviations, and tip clearance related blockage. Their report focused only on the prediction of steady state performance characteristics. If we were in a position to model how each of these loss mechanisms varied with heating and cooling of the primary gas path metal, then we might be able to use these models to estimate changes to component characteristic maps arising from heat soakage during transient manoeuvres. Given the complexity of blade passage flow (see above) this is clearly a mammoth task, and without the benefit of experiment and/or computation fluid dynamics some may argue that it is impossible.

To their great credit MacCallum & Grant (1977) made just such an attempt. They developed a method to estimate the transient changes to compressor map that occur as a result of heat soakage by accounting for heating and cooling of the aerofoil boundary layers. Heating or cooling of the boundary layer affects both the density and viscosity of the fluid in the boundary layer. MacCallum & Grant (1977) report that on heated compressor blades boundary layer thickening takes place more on the suction surface than on the pressure surface. This asymmetry results in a change in angle of departure of the flow as it leaves the blade. These changes to flow angles (c.f. figure 2.3.3.7) effectively change the velocity triangles and thus compressor performance. The thickened wake also acts as a blockage to the flow.

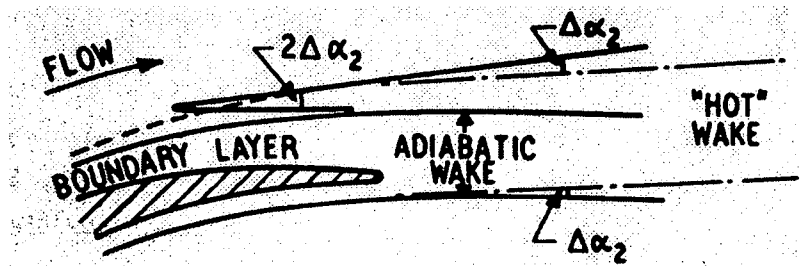


Figure 2.3.3.7 Compressor aerofoil wake and boundary layer changes due to boundary layer heating.
(Maccallum 1977)

A great deal of attention has been given to analysis of the losses occurring due to secondary flows in the blade tip region of turbomachinery. Even though tip clearances may not be the largest source of aerodynamic losses, the reason for all of this attention may well be due to the fact that other sources of loss are not as amenable to manipulation as the tip clearance losses (Ameri et al 1999). In addition, compressor surge margins are very sensitive to tip clearance variations. Blade tip clearance changes also affect a component's flow capacity. Figure 2.3.3.8, taken from Kurzke (1992) demonstrates how changes in tip clearance can affect the steady state performance map of a compressor.

One of the most undesirable aerodynamic events that can occur within a gas turbine is surge; it can seriously damage an engine. The compressor surges when a sufficient portion of its blades stall and thus fail to deliver pressure rise. When a jet engine compressor surges flames can come out of both ends of the engine, alarming pilots, passengers, and engine manufacturers alike. The blade passages in the compressor bring about pressure rise by diffusing the air. This process involves slowing the air down relative to the diffusing blade passage. An adverse pressure gradient exists along the blade passage. Air in boundary layers travels slower than it does in the bulk of the gas, and in the presence of the adverse pressure gradient the flow will separate from blade surfaces if it has insufficient momentum to overcome the adverse pressure gradient, leading to flow recirculation and blade stall (see Walsh & Fletcher 1998). Heat flowing from metal to gas not only thickens boundary layers to make flow breakdown more likely but also causes rematching of stages in multistage components. The result is that compressor surgelines can fall quite significantly during the re-acceleration part of hot reslam manoeuvres (Rowe 2000).

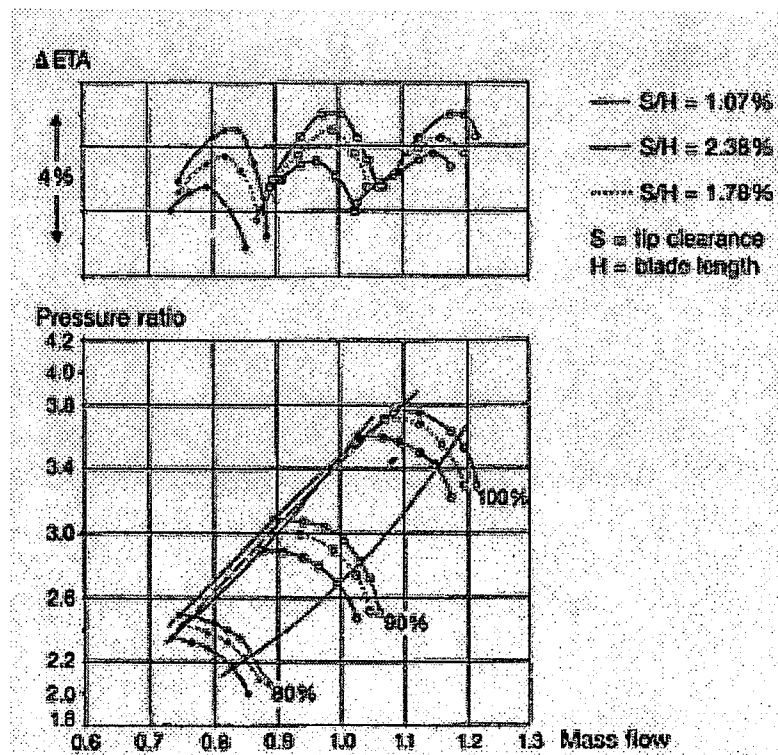


Figure 2.3.3.8 Effects of Tip clearance on Compressor Characteristics (Kurzke, 1992)

THERMODYNAMICS

Heating and cooling of the primary gas path fluid changes the thermodynamics of an engine. Indeed, intercooling between compressors and reheating between turbines is commonly done on land based gas turbines in order to reduce compressor power, increase turbine power, and thus increase the net available power from the engine. The difference between conventional intercooling and reheat and that which occurs due to primary gas path heat soakage during transient manoeuvres is that during transients the heat transfer takes place amongst the aerofoils, whereas conventional intercooling and reheat takes place in dedicated heat exchangers. Heating and cooling of the primary gas path fluid changes its density and therefore changes the relationship between axial and rotational flow velocities. This changes the component's velocity triangles and thus its performance (Maccallum & Qi, 1989). This phenomenon is more noticeable in the rearward stages of a component than in the forward stages, and this inevitably leads to a rematch of the stages within the component itself (internal rematching) and thus to a change in its overall performance (Walsh & Fletcher 1998).

Previously it was mentioned that an engine is more likely to surge during a hot reslam manoeuvre than during a cold slam acceleration. As part of an experimental investigation into hot reslam surge, Rowe (2000) carried out a series of calculations looking at the effect of internal rematching. His method was to split the compressor axially into individual stages, he then performed stage heat transfer calculations, and

finally restacked the stage maps to discover whether or not internal rematching was the cause of the hot reslam surge problem. The findings of the study were that internal rematching alone did not fully account for the drop in surge margin encountered during hot reslam manoeuvre tests. The conclusion was that the factors controlling the initiation of surge and therefore the drop in surge line were not observable from the station data measurements made in the experiments.

The normal assumption for steady state gas turbine calculations is that the processes of compression and expansion occur adiabatically. This may be a reasonable assumption under steady state conditions where real engine component heat flows are very small compared to engine component powers, however, under transient conditions, heat flows can no longer be ignored. The F parameter is defined as the heat flow into the fluid divided by the work done by the fluid. Maccallum & Qi (1989) reported calculations for F parameter values of up to 35% during transient manoeuvres. The effect of the F parameter on the adiabatic compression and expansion equation can be seen from equation 2.3.3.1 (adapted from Maccallum & Qi 1989). From the equation we can see that heat flow in or out of the working fluid during compression or expansion alters the pressure ratio associated with a particular temperature ratio. When this occurs inside of a gas turbine the result is a change in its thermodynamic cycle.

$$P2/P1 = (T2/T1)^{X / (1-F)} \quad (2.3.3.1)$$

P1, P2	Pressures at the start and end of the process
T1, T2	Temperatures at the start and end of the process
X	Polytropic index of compression or expansion
F	Ratio of heat into fluid to work out of fluid

An engine's thermodynamic cycle is not only altered due to primary gas path heat flow, but cycle changes can also occur due to heat flows in the secondary gas path, i.e. bleed passages. Geometry changes in labyrinth seal clearances can affect critical bleed mass flows (MacCallum 1979, Neal 1982, and Pilidis 1983), and when bleed temperature and mass flow change, then the mixed out temperature of the primary gas path also changes, thus changing the engine's thermodynamic cycle.

2.3.4 EXISTING HEAT SOAKAGE PHENOMENA MODELS

Gas turbine calculations vary from the very simple to the very sophisticated. Some calculations can be performed quickly to a reasonable accuracy with just a few pages of computer code, whereas highly accurate calculations can require the use of heavy weight numerical codes such as computational fluid dynamics (CFD) or finite element analysis (FEA) and can take many weeks to set up and converge upon a solution. When calculations start to become complicated and time consuming a common approach is to create maps that characterise the results of the heavy weight calculations. These maps can then be used quickly and conveniently. Examples of all three approaches can be found in the open literature. Clearly, whether a particular approach is good or bad depends entirely upon the nature of its application. The most extreme example is NASA's NPSS project (see Reed & Afjeh 2000). One of the aims

of NPSS is to use CFD and FEA to create a complete three-dimensional engine performance model.

The methods outlined below are far less resource hungry than NPSS and are more amenable for use in a project such as that outlined in this thesis. For convenience the review of how different authors have included heat soakage models into whole engine transient performance codes has been divided into categories. The categories are: heat soakage, tip and seal clearances, and transient component characteristics.

HEAT SOAKAGE

Elder (1974) outlines a method in which transient heat exchange is calculated between the air in the primary gas path of an axial flow compressor, and the metal of its blades and annulus casings. The method uses the calculated transient heat flows to update the delivery temperature of the compressor. The RRAP engine performance code uses the same approach in its heat soakage subroutine, BR081. BR081 is used to calculate primary gas path heat soakage in all of the major engine components (Grainger-Allen et al 2000). BR081 represents the hardware of a component in a very simple way. The annulus of the component is represented by a tube of equivalent hydraulic diameter, and for components with blades, the blades are represented by a flat plate (see figure 2.3.4.1).

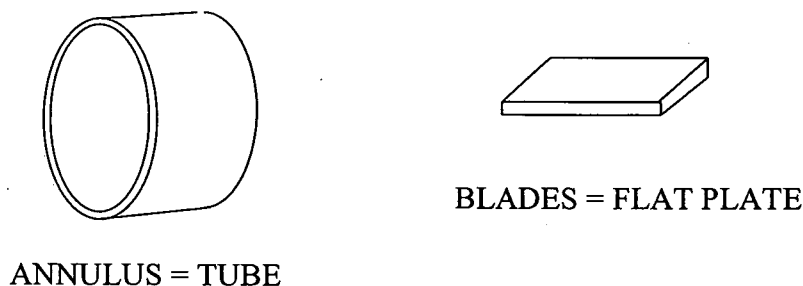


Figure 2.3.4.1 **BR081 representation of gas turbine component hardware**

Annulus and blade (i.e. tube and flat plate) thermal mass, wetted area, specific heat, etc. are set at values representative of the real component. The metal components are assumed to have infinite thermal conductivity, and no thermal communication takes place between blade and annulus metal. Heat transfer coefficients for the annulus are calculated using standard Nusselt number correlations for turbulent flow in a pipe, and for the blades they are calculated using standard Nusselt number correlations for turbulent flow over a flat plate. Initial gas and metal temperatures are assumed to be equal to the mean of the steady state inlet and outlet temperatures. BR081 transient heat transfer calculations use the lumped capacitance approach and are performed by calculating the thermal time constants for both the annulus and the blades (for the annulus only in components with no blades) according to equation 2.3.4.1

$$\tau = M.C_p / (h.A) \quad (2.3.4.1)$$

$$T_{m1} = T_{g0} - (T_{g0} - T_{m0})^{-dt / \tau} \quad (2.3.4.2)$$

$$q = (T_{m1} - T_{m0}) . M . C_p / dt \quad (2.3.4.3)$$

τ	Thermal time constant of the metal part
M	Mass of the metal
C_p	Specific heat of the metal
h	Heat transfer coefficient
A	Wetted area
T_{m1}	Metal temperature at end of time interval
T_{m0}	Metal temperature at start of time interval
T_{g0}	Gas temperature at start of time interval
dt	Length of time interval
q	Quantity of heat absorbed/released by the metal

Once time constants have been calculated and a new transient gas temperature has been established then they are used to calculate the new temperature of the metal according to equation 2.3.4.2. Once the new metal temperature has been calculated, the amount of energy absorbed or released by the metal during the time interval is calculated according to equation 2.3.4.3. Knowing the amount of heat absorbed or released by the metal we assume this is the same amount of heat absorbed or released by the gas. Equation 2.3.4.3 (exchanging metal temperatures for gas temperatures), can then be used to calculate the gas temperature change resulting from the heat transfer. The author of BR081 uses an iterative rather than straight through analytical approach to do this. Once the size of the quantity of heat has been calculated the effect of non-adiabatic compression and expansion can be estimated in turbines and compressors. The adiabatic component pressure and temperature ratios (i.e. those taken from the component map at the start of the time interval) are used to calculate the instantaneous polytropic index of expansion or compression from equation 2.3.3.1 (excluding the F parameter). Calculating the instantaneous value of the F parameter from the heat transferred and the instantaneous component power, equation 2.3.3.1 is then used to calculate the non-adiabatic component exit pressure.

Following the speed transient, the BR081 component metal temperatures take a while to catch up with the mean component gas temperature. BR081 continues to alter outlet temperature and pressure until the difference between mean gas temperature, annulus temperature, and blade temperature, eventually decays to zero. Factors are applied to heat transfer coefficients to change thermal time constants for each of the components where this is necessary to bring engine responses closer to those observed in the real engine.

The GTTrans engine performance code (Pilidis 1983), as part of more elaborate set of calculations (see later), uses a very similar approach to calculate primary gas path heat soakage. One of the main differences being the choice of heat transfer coefficient and Nusselt number correlations. Correlations for flow over aerofoils are used to calculate blade heat transfer coefficients, and those for flow through rotating concentric cylinders are used for annulus heat transfer coefficients. Like BR081,

GTTrans also assumes metal components do not thermally communicate and have infinite thermal conductivities. It too uses the lumped capacitance approach to calculate transient heat transfer. Component outlet temperatures and pressures are updated using the F parameter in a similar manner to that used by BR081.

Fiola (1993) presents results of a study in which the high-pressure turbine of the SNECMA LARZAC 04 engine was modelled using finite difference approach (see later). The components constituting the primary gas path were subdivided to a greater extent than in either BR081 or GTTrans. Both heat transfer and mechanical strain calculations were performed using the finite difference method, which was incorporated into a whole engine performance code.

Riegler (1999) dispenses with the idea of using any kind of hardware representation within a whole engine transient performance code and instead proposes a method whereby primary gas path heat soakage is calculated non-dimensionally using Biot and Fourier numbers. However, the workings of the method presented are still based on Nusselt numbers for flow through tubes, mean component inlet and outlet temperatures, and mean component temperatures. It is different to the BR081 and GTTrans methods in that it accounts for steady state heat flows (the Fiola (1993) finite difference algorithm includes conduction and so it too accounts for steady state heat flows). Riegler suggests that values for the parameters in the non-dimensional heat soakage calculations can be determined for each component by experimental investigations or by calculation using the results of thermal analysis performed by heat transfer specialists.

TIP & SEAL CLEARANCES

Khalid and Hearne (1980) present a method for transient tip clearance calculation in which each component is represented as three distinct parts: a casing, a blade and a disc. Changes in the relevant radii of each of these three parts are used to calculate changes in tip clearance. The equation used to calculate the change in radius of each of the three parts under steady state conditions is given as equation 2.3.4.4 below (adapted from Khalid and Hearne 1980). According to Khalid and Hearne (1980), the transient thermal growth of each of the three parts is proportional to its transient metal temperature, which is an exponential function of time, and so the transient thermal growth of each of the three parts can also be expressed as an exponential function of time using different time constants for each individual part.

$$\begin{aligned} X &= K3 \text{ for discs, } K4 \text{ for blades, } K5 \text{ for casings} \\ Y &= P_{out} \text{ for case, } N^2 \text{ for blades, } N^2 \text{ for discs} \end{aligned}$$

$$\Delta R = K1.T_{in} + K2.T_{out} + X.Y \quad (2.3.4.4)$$

X, Y	Intermediate variables
K1,..., K5	Empirically determined constants
P _{out}	Pressure out of the component
N	Rotational speed of the component
R	Radius
T _{in} , T _{out}	Temperature at inlet and outlet of the component.

In their paper Khalid and Hearne scale the blade, casing and disc time constants against the square root of component mass flow to take account of varying heat transfer coefficients. Transient tip clearances are found from equation 2.3.4.5 (adapted from Khalid and Hearne 1980).

$$\Delta R_{t1} = \Delta R_{ss1} - (\Delta R_{ss1} - \Delta R_{t0}) \cdot e^{-dt / \text{Tau}} \quad (2.3.4.5)$$

$\Delta R_{t0}, \Delta R_{t1}$	Transient change in radius at start and end of time interval
ΔR_{ss1}	Steady state change in radius at end of time interval
dt	Time interval
Tau	Time constant

The transient growths of the casing, blades, and discs are then used to calculate the instantaneous operating clearances for each component in the engine, which are then used to modify component efficiency, flow, surge margins, etc. in the whole engine performance code. This method relies wholly upon the use of empirically defined constants.

The GTTrans engine performance code (Pilidis 1983) calculates not only primary gas path heat soakage but also transient blade tip and seal clearance changes, by way of simple thermomechanical models of the engine's compressors and turbines. Each compressor and turbine is represented as a single equivalent stage found by averaging the geometry of each stage within a particular component. Thermal masses and wetted areas used in heat transfer calculations are not averaged but kept at a size that represents the whole component. These single averaged stages are given separate thermal and mechanical treatment (c.f. figure 2.3.4.2).

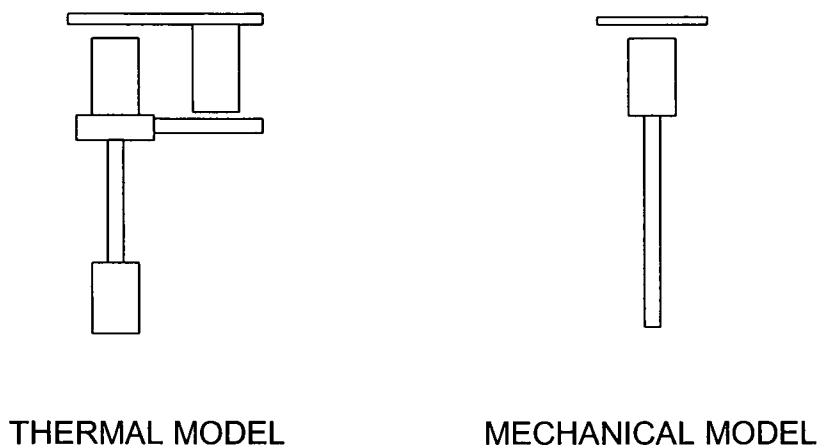


Figure 2.3.4.2

GTTrans turbomachinery component representation using a single average stage approach.

Transient heat transfer calculations in the GTTrans method use a lumped capacitance approach; the primary gas path, casing, blades, and disc rim metal, each have their own unique temperature. Blade cooling is accounted for by the inclusion of a blade cooling effectiveness term in the calculations. The disc is divided into a rim, a diaphragm, and a cob, each with a unique temperature. Heat transfer coefficients for the disc are calculated using correlations for either Nusselt number for a disc rotating close to a stationary surface, or Grashof number for rotating disc chambers (it is assumed that this flow condition resembles natural convection in a high gravity field). The bleed in the disc cavity is assumed to behave as a temperature reservoir at the same temperature as the bleed point from which it originated. The transient temperatures of the casing, blades, rim, diaphragm, and cob are all used in the tip clearance calculation. Thermal growths of the blade and case are calculated directly from their temperature, geometry, and material thermal expansion coefficient. A mass weighted average of rim, diaphragm, and cob temperatures is used to represent the temperature of the disc. At this point mechanical growth calculations are performed. The mechanical growth of the casing is calculated using the pressure differential between primary gas path and casing cavity pressures. For the blade, mechanical growth is calculated from centrifugal stress assuming blades to be prismatic members loaded axially by their own weight, and also the weight of shrouds were applicable. Mechanically, the disc is represented as a single disc of a constant thickness that is equal to the thickness of the diaphragm. Disc growth is calculated from centrifugal stress including the pull of the blades. Disc growth is multiplied by a scaling factor found from a parallel study using finite element analysis (a factor of 1.3 was used in the Pilidis 1983 study). Summing transient thermal and mechanical growths GTTrans is able to estimate tip clearance changes during transient manoeuvres. Within GTTrans transient seal clearance changes are also calculated, using a modified version of the blade tip clearance method (i.e. no blades).

Fiola (1993) also created a heat soakage, tip clearance and seal clearance algorithm based on single average stage representation (c.f. figure 2.3.4.3).

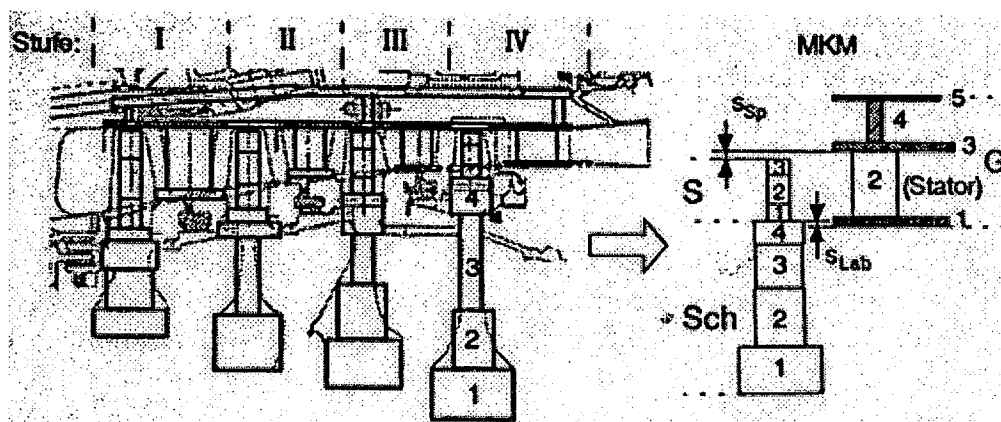


Figure 2.3.4.3 Single averaged stage representation using one dimensional discretisation (Fiola 1993)

Fiola (1993) went one step further than Pilidis (1983) with his thermomechanical model. Instead of a relatively simple representation of discs, blades, and casing, Fiola used a finite difference approach to model the component to a finer degree of discretisation and also included the effect of conduction in the calculations. He started out with 2-dimensional discretisation of the single averaged stage but in the version incorporated into the whole engine performance code he used 1-dimensional discretisation (c.f. figure 2.3.4.3), to speed up computation.

Moll et al (2003) present a method in which detailed finite element analysis models of engine components, which are created in the normal course of the engine development process, are used to provide the source data for a state space model. Data from finite element analysis is used in conjunction with the Matlab® systems identification toolbox to produce a series of charts which can be used to characterise the tip clearance behaviour of the components of the engine throughout the whole of its operational envelope. During that process, output data from the finite element analyses is iteratively compared against output from the state space model and the state space model is modified until its output matches that of the finite element analyses to within a specified tolerance. The state space model is built into the whole engine performance code. At run time the model uses these tip clearance characteristic maps produced off line, to calculate the instantaneous tip clearances on line, which are then used to modify the performance of the engine component when it is called from within the whole engine performance code.

A similar but less sophisticated approach is used by the RRAP subroutine BR083 (see Grainger-Allen et al 2000). To calculate instantaneous tip clearance BR083 uses previously prepared look up charts of steady state rig tip clearance versus non-dimensional speed, casing and rotor growths versus a given engine parameter, and user defined time constants for casing and rotor response. It then uses the difference between transient and steady state tip clearance and a look up chart of tip deltas versus efficiency and flow defect to modify the performance of the engine component when it is called from the whole engine performance code. The obvious problem with this method is the chart in which steady state rig tip clearance is plotted against non-dimensional speed. Clearly tip growths are more dependent upon actual speed than they are on non-dimensional speed.

The Lewis method (see Naylor 2002e) is similar to the Moll (2003) method in that it too is based upon output from a detailed finite element analysis model of an engine component (a Rolls-Royce finite element programme (SC03) model of the Rolls-Royce Trent 500 engine high-pressure compressor). Unlike Moll (2003) however, the Lewis method does not use graphs and charts, but characterises the tip clearance behaviour of the engine component using a set of equations, in a similar manner to the method described by Khalid & Hearne (1980). The Lewis method has been validated against the SC03 model, which has been validated against test data but only for the first stage due to a high test sensor mortality rate (Anderson 2000). The method accounts for asymmetrical tip clearances which occur due to static pressure field changes that take place when the compressor bleeds are switched on and off. The input data required by the Lewis method for each stage of the compressor is as follows: for the casing, radius, total stabilised growth at idle and max, time constants at idle and max, and a correction delta; for the disc, the same as for the casing; for the blades, the same again but with radius replaced by blade height, and blade

temperatures at idle and max. Thermal growth is assumed to decay exponentially and so time constants are found taking the time required for thermal growth to change by $1/(1-e)$ of its steady state value. 'Tip closure' means the change in tip clearance from the cold strip clearance value.

The Lewis method uses this input data to calculate mechanical and thermal growth at idle and max, temperature at idle and max, and an X factor (fraction of component temperature rise) at idle and max, according to equation 2.3.4.6 calculated at idle and max for each subcomponent of each stage.

$$\begin{aligned} A &= 2 \text{ for disc, } 6 \text{ for casing} \\ \text{MGi} &= C \cdot \text{Ni}^A / (\text{Nm}^A - \text{Ni}^A) \\ \text{MGm} &= C \cdot \text{Nm}^A / (\text{Nm}^A - \text{Ni}^A) \end{aligned}$$

$$\begin{aligned} \text{TG} &= D - \text{MG} \\ T &= \text{TG} / (R \cdot \text{Alpha}) + 288 \\ X &= (T - T_{26}) / (T_{30} - T_{26}) \end{aligned}$$

(2.3.4.6)

C	Correction factor
D	Total growth
MG	Mechanical growth
N	Speed
TG	Thermal growth
T	Temperature of subcomponent
R	Radius
Alpha	Coeff of therm expansion at the appropriate metal temperature
X	X factor (fraction of component temperature rise)

Suffixes:

i,m	Idle and max
26, 30	compressor inlet and outlet

Once all of these parameters have been found, the steady state tip closures corresponding to the instantaneous component operating point are calculated from equation 2.3.4.7. Mechanical blade growth is assumed to be zero.

$$\begin{aligned} A &= 2 \text{ for disc, } 6 \text{ for casing} \\ \text{MG} &= \text{MGm} \cdot (\text{N}/\text{Nm})^A \\ X &= X_i + (X_m - X_i) \cdot (\text{N} - \text{Ni}) / (\text{Nm} - \text{Ni}) \\ T &= T_{26} + X \cdot (T_{30} - T_{26}) \\ \text{TG} &= \text{TGm} \cdot (T - 288) / (T_m - 288) \end{aligned}$$

(2.3.4.7)

MG	Mechanical growth
N	Speed
TG	Thermal growth
T	Temperature of subcomponent
X	X factor (fraction of component temperature rise)

Suffixes:

i,m	Idle and max
26, 30	compressor inlet and outlet

Once the steady state tip closures corresponding to the instantaneous component operating point have been found, then instantaneous tip closures can be calculated. Instantaneous mechanical growths are assumed to be the same as for steady state, however instantaneous thermal growths are calculated using equation 2.3.4.8. The thermal response of the blades is assumed to be instantaneous.

$$\begin{aligned} F &= (N-17445)/(-26324.1) \text{ for the rotor} \\ F &= (N-9646.2)/(-8240.6) \text{ for the casing} \\ T &= T_i + (T_m - T_i) \cdot ((N - N_i)/(N_m - N_i)) / (1 - F \cdot (A/A_h)) \\ TG1 &= TG0 + (TG_{ss} - TG0) \cdot (1 - e^{-dt/T}) \end{aligned} \quad (2.3.4.8)$$

F	Intermediate variable
N	Speed
TG0	Thermal growth from previous time interval
TG1	Thermal growth at current time interval
TGss	Steady state thermal growth
T	Time constant for the subcomponent
dt	Time interval
A	Current altitude
A _h	Reference altitude (high)
Suffixes:	
i,m	Idle and max

Once transient tip closures have been found then the transient tip clearances can be found from knowledge of the component strip clearances. The change in tip clearance at the top, bottom, port, and starboard due to the switching on and off of the compressor bleeds are found from experimental measurements. The instantaneous tip clearance at top, bottom, port, and starboard are found using equation 2.3.4.9

$$\begin{aligned} Z &= (P30 - P26) / (P30_m - P26_m) \\ D &= C_s + C + B \cdot Z \end{aligned} \quad (2.3.4.9)$$

Z	Intermediate variable
D	Instantaneous clearance
C _s	Strip clearance (i.e. found from engine strip down)
C	Instantaneous closure
B	Change in tip clearance due to presence of bleed at either top, bottom, port or starboard
P26, P30	Instantaneous compressor inlet and outlet pressure
P26 _m , P30 _m	compressor inlet and outlet pressure at max power

Knowing the instantaneous tip clearance at top, bottom, port, and starboard, we can calculate the instantaneous maximum, minimum, and average tip clearance for each stage. Maximum tip clearances are used for surge margin calculations, minimum tip clearances are used for rub calculations, and average tip clearances are used to calculate modifications to component efficiency and flow (see Graf et al 1998).

TRANSIENT COMPONENT CHARACTERISTICS

Turbomachinery maps characterise the steady state performance of the component over a range of different operating conditions. These maps are traditionally comprised of flow, pressure ratio, speed, and efficiency.

A number of researchers have published relationships between tip clearance and component efficiency. Lakshminarayana (1970) gives a semi-empirical relationship for efficiency change versus tip clearance, as does Hesselgreaves (1969). Robinson (1982) proposed a wholly empirical relationship following a series of compressor studies, and Kacker & Okapuu (1982) proposed a general relationship which when tested against 33 turbines gave accuracy to within 1.5%. Storer & Cumpsty (1994) present an equation for tip clearance efficiency in a single stage compressor as a function of basic design parameters. Their equation is based on assumptions of low Mach number flow without heat transfer. Ro & Cho (1999) report a numerical and experimental single stage axial compressor study in which they give two alternative expressions for the reduction in efficiency with tip clearance, against which they compare the Lakshminarayana (1970) correlation (c.f. figure 2.3.4.4).

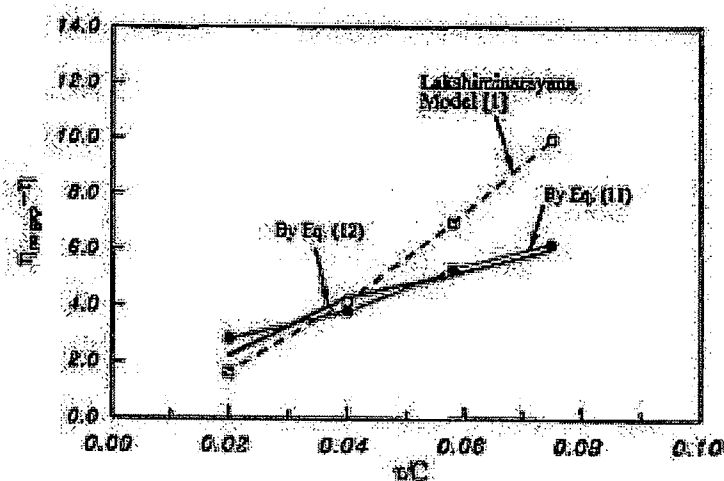


Figure 2.3.4.4 Comparison of relationships for reduction in efficiency with tip clearance. Ro & Cho (1999).

Ameri et al (1999) conducted a numerical study of a GE-E3 first stage turbine and reported a linear relationship between efficiency and tip gap, suggesting about 1.5 % efficiency loss per 1% gap clearance. Szanca et al (1974) performed experiments on a single stage axial turbine and found total component efficiency to change 1.7 % for every 1 % of blade annular passage height. Walsh & Fletcher (1998) suggest that a 1% change in non-dimensional tip gap will produce a 1% change in efficiency in turbines, and a 1 to 2% change in compressors. Clearly the choice of which of these correlations to use will depend on the actual piece of turbomachinery being studied.

Walsh & Fletcher (1998) also state that tip clearance can have a large effect upon compressor map surge lines. They say that the amount depends upon the particular compressor design and must be determined by utilising design codes or more

accurately via rig test. They say that the exchange rate will be in the range of a 1% increase in rms tip clearance reducing the surge line by between 2% and 15% of surge margin.

Tip clearance also affects component flow (Walsh & Fletcher). Exchange rates for tip clearance changes against flow changes are of the same order of magnitude as those for tip clearance changes against efficiency changes (Curnock 2002).

Without reference to tip clearances, Rowe (2000) looked at transient compressor map representation by splitting a compressor map in to stages, added primary gas path heating/cooling, and then restacked the map. The maps only accounted for bulk fluid heating.

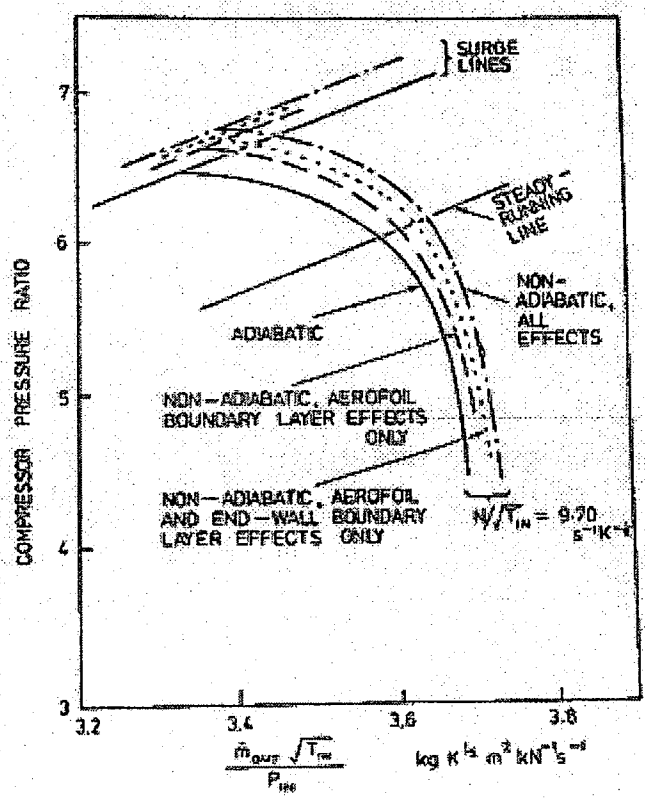


Figure 2.3.4.5 High-pressure compressor characteristic at the 6th second of a sea-level acceleration (Maccallum 1982)

Some of the greatest contributions to transient compressor map representation have come out of Glasgow University. Of the many publications Maccallum (1982) and Maccallum & Pilidis (1985) give perhaps the best summaries of the work. They outline methods for transient modifications to steady state performance maps accounting not only for bulk heating of the primary gas path fluid, but also changes in blade tip clearances, changes to aerofoil boundary layer thickness, and changes to end wall boundary layer thickness. Figure 2.3.4.5 shows an example of the effect that some of these effects had on a compressor map at during a sea level acceleration (Maccallum 1982).

Maccallum & Pilidis (1985) gives a series of expressions for changes in physical speed and pressure ratio as a result of these various phenomena. These expressions are called from within whole engine performance codes during transient simulations and the results are used to instantaneously update the maps. The coefficients used in those expressions are found by experiment, and so are engine specific.

2.3.5 SECTION CONCLUSION

The purpose of this literature review was to outline the information available in the open literature that is relevant to this project.

The first part of this section very briefly introduced gas turbine performance modelling, and then went on to discuss a selection of whole engine performance codes described in the open literature. The second part of this section described the types of physical phenomena that are associated with gas turbine transient manoeuvres and heat soakage. The final part of this section outlined methods used by previous researchers to model the various heat soakage phenomena.

2.4 PROJECT REQUIREMENTS

The EngD programme requires that research engineers address not only an engineering problem, but also management issues associated with it in the broader business environment. Engineering doctorate projects therefore typically have both engineering and management requirements.

ENGINEERING

The requirements for the engineering part of this project came from a number of Rolls-Royce sources. At a meeting between Rolls-Royce and Cranfield to discuss methods of turbomachinery tip clearance effect modelling for performance synthesis calculations and the part the Cranfield UTC might play in developing a new algorithm it was concluded that (Campbell 1999):

- 1) The method should handle both tip clearance and heat soakage

Within the Rolls-Royce Performance Function Improvement Programme document (Martin 2001) it states that methods are required which enable the thermal response of engine hardware to be predicted at a level of complexity consistent with the following requirements:

- 2) The ability to predict the response of untested hardware (i.e. to be physics based rather than simply a numerical line fit type method).
- 3) To be accurate enough to adequately predict component performance effects.
- 4) To be able to carry out the calculations in a time frame consistent with performance code calculations, i.e. at or near real time.

- 5) To be capable of being fine-tuned using engine test data.

With regard to requirement 3, no numerical limits were specified. However, in a meeting between the author and Rolls-Royce compressor thermals experts (see Naylor 2002b) to discuss the time constants based Lewis method for tip clearance modelling, accuracy targets were established for tip clearances predictions coming from methods produced by the author, i.e. tip clearances predictions in the Trent 500 high-pressure compressor should:

3a) Be within 0.025 mm of Lewis method predictions steady state

3b) Be within 0.1mm of Lewis method predictions transiently

MANAGEMENT

At the 4th quarterly EngD steering group meeting for UTC work package 7: heat soakage (Naylor 2002a), it was stated that technology transfer from UTC to Rolls-Royce would be the management topic associated with this EngD project, more especially the business questions surrounding the usability of the output from Cranfield UTC projects and how an understanding the technology transfer issues are likely to affect the approach researchers take to UTC project tasks.

2.5 RESEARCH GAP

Traditionally research activities centre on identifying and filling a gap in the general body of knowledge, i.e. a knowledge gap. Sometimes this gap is not so much a knowledge gap as it is a capability gap.

ENGINEERING

As part of their ongoing programme of self-improvement, Rolls-Royce have identified that their current heat soakage and tip clearance methods fail to meet their perceived requirements. Table 2.5.1 lists the Rolls-Royce and other methods described in the literature review contained in this chapter. The table shows to what extent the author feels each method meets the requirements laid down for this project. The column representing the requirement for the method to be capable of being calibrated has a yes on every row. The methods that rely upon either experiment or heavyweight numerical analyses for their input data by definition can be easily calibrated, and those methods which are physics based can be calibrated to a degree by manipulation of their input data. The methods that do not run at or close to real time are those which are based upon iterative numerical routines. The accuracy of each method is related to whether or not it relies upon experimental or heavyweight numerical analyses for input data. By definition, those that do are likely to be more accurate than those based solely upon simple physical models of components and hardware. The yes/no in the GTTrans row of the third requirements column is there because the tip clearance and heat soakage models within GTTrans are physics based, whereas the correlations it uses for transient updates of component maps are based upon empirical data.

We can see from Table 2.5.1 that none of the methods described in the literature review meet all of the requirements laid down by the project sponsor. There is evidently a heat soakage and tip clearance modelling capability gap.

METHOD	REQUIREMENT				
	Both Tips and heat soakage	Physics based, i.e. predictive	Accurate	Run close to Real time	Able to be Calibrated
BR081, Elder	NO	YES	?	YES	YES
GTTrans (Pilidis & Maccallum)	YES	YES/NO	?	YES	YES
Fiola	YES	YES	?	NO	YES
Riegler	NO	NO	YES	YES	YES
NPSS	YES	YES	YES	NO	YES
Lewis	NO	NO	YES	YES	YES
Moll et al	NO	NO	YES	YES	YES
BR083	NO	NO	YES	YES	YES
Khalid & Hearne	NO	NO	YES	YES	YES

Table 2.5.1 Methods described in the literature versus project requirements

MANAGEMENT

In addition to this tip clearance and heat soakage capability gap, Rolls-Royce have identified a gap in the current Cranfield UTC to Rolls-Royce technology transfer arrangements, i.e. novel methods developed within the UTC do not always successfully make the transition over to Rolls-Royce working practice.

The work described in this thesis outlines the author's attempt at filling these two research gaps.

2.6 PROJECT STRATEGY AND SCOPE

ENGINEERING

Table 2.5.1 shows that none of the methods listed in the table fully meet all of the requirements laid down for this project. The only three methods that can handle both tip clearance and heat soakage are GTTrans, the Fiola method, and NPSS. An NPSS type approach is clearly too large an undertaking for a single individual within the time scale allowed for an EngD degree, therefore the two remaining appropriate approaches are the GTTrans and Fiola methods. Both of which are capable of being calibrated, but a question remains over their predictive capabilities.

Both the GTTrans and Fiola methods use a single averaged stage to represent multistage components. This begs the question: how different is the tip clearance prediction from a single averaged stage to the average of multistage tip calculations? The answer to this question has a significant effect on the size and complexity required of component tip clearance models in whole engine performance codes. Moreover, single averaged stage component models are likely to be much faster to run than multistage component models. To address this issue the author has included in this thesis is a study comparing predictions made by a single averaged stage against those made from averaging individual stage values.

The Pilidis (1983) method is very fast but excludes the effects of conduction. Conduction can play a very significant part in heat transfer calculations when metal components are thick (e.g. disc cobs). The Fiola (1993) method includes conduction but is very slow, and likely to be slower than the 'at or near real time' running requirement outlined in the section above. This is due to iterative nature of its thermal and mechanical strain calculations. The strategy adopted in the methods developed by the author has been to account for conduction using a discretised component approach (like Fiola) but to separate thermal and mechanical models (like GTTrans). In this way the author has attempted to reduce iterations to a minimum and thus increase the speed of heat soakage and tip clearance calculations.

The key areas to focus upon in the development of the methods were laid out in the Rolls-Royce Performance Function Improvement Programme document (Martin 2001) as heat soakage, tip clearances, secondary flows (i.e. bleeds), and compressor surge.

Surge is a very complicated aerodynamic phenomenon, the onset of which is very difficult to confidently predict. Empirical correlations have been developed (e.g. Maccallum & Pilidis 1985) but these depend upon engine specific constants found from experiment. Because no such experimental data has been available in this project the author has decided that surge in this study would only be addressed in so far as it is affected by excursions of the high pressure compressor transient working line. Modelling of transient surge line excursions has been excluded from this study.

Primary gas path aerodynamics are extremely complicated and interdependent. It is very difficult to say without recourse to experiment of computational fluid dynamics whether blade passage heating changes one blade passage flow structure any more than it does another, or that a change in one flow structure has a larger effect on component performance than a change in another. For this reason primary gas path aerodynamics and component characteristic map modifications have been excluded from this project. Only the effect of changes in tip clearance has been included, accounted for using simple exchange rates against efficiency and flow.

Only low frequency transient manoeuvres between idle and maximum power on a sea level test bed were investigated in this study.

The Trent 500 (c.f. figure 2.6.1) entered service in August 2002 on Virgin Atlantic's Airbus A-340 aircraft. Because the timing of the launch of this new engine coincided with the author's research programme, the author chose this engine to be the base upon which to develop the required heat soakage and tip clearance methods.

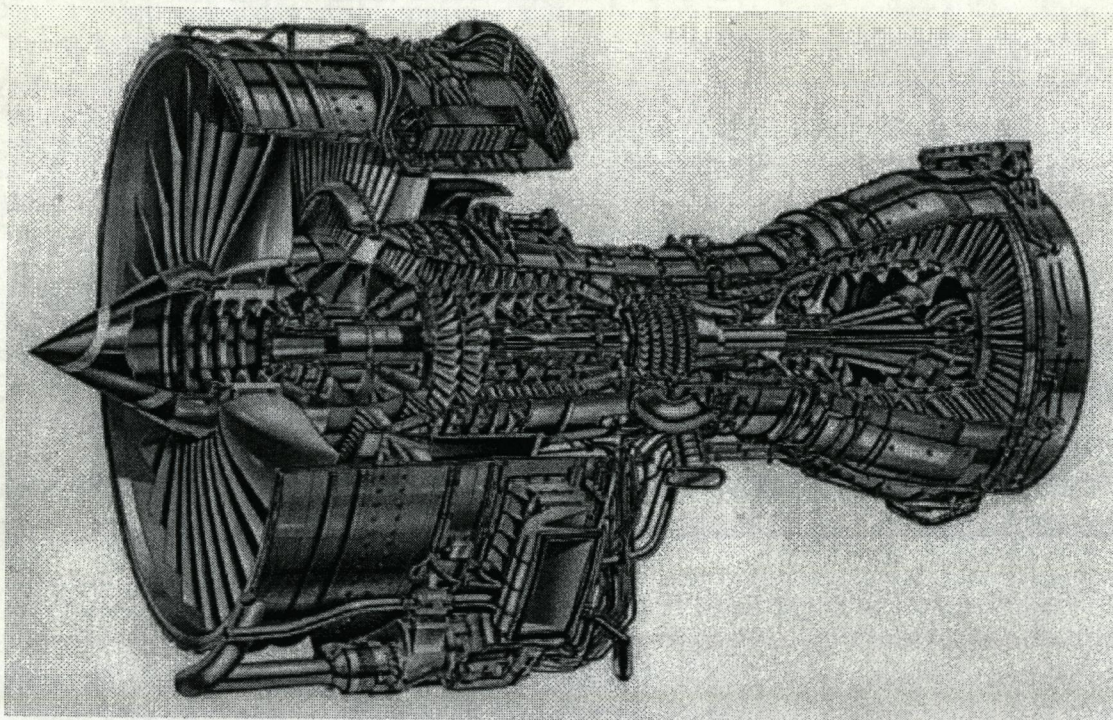


Figure 2.6.1 The Rolls-Royce Trent 500 (Rolls-Royce 2001)

At the 5th quarterly EngD steering group meeting for UTC work package 7: heat soakage (Naylor 2002a), following a discussion of a number of different alternatives, it was decided that the Rolls-Royce Aero-engine Performance (RRAP) code would be used as the computing platform upon which the author's heat soakage and tip clearance routines would be developed. The main reason for this decision was that developing new methods on a RRAP platform would facilitate the technology transfer process.

MANAGEMENT

The scope of the management component of this project could have easily ballooned into a PhD project all of its own. The EngD is more of an engineering qualification than a management qualification and therefore the scope of the technology transfer study carried out in relation to this thesis was restricted to a survey of relevant literature, an investigation into the current UTC situation, and the recommendation of possible ways to improve the current technology transfer process (see chapter 5).

2.7 CHAPTER CONCLUSION

The purpose of this chapter was to present and justify the scope and objectives chosen for the project outlined in this thesis

Section 2.2 of this chapter began with a description of the business case that led to sponsorship of this research project, i.e. the potential to save engine development time and cost via an enhanced heat soakage and tip clearance modelling capability.

Section 2.3 presented a review of literature relevant to heat soakage and tip clearance modelling within the context of whole engine transient performance codes. In the subsection on whole engine performance modelling, the basic principles underlying steady state and transient performance codes were briefly described, followed by a review of a number of existing whole engine performance codes reported in the literature. The next subsection contained a review of phenomena associated with heat soakage as they affect the geometry, aerodynamics, and thermodynamics of a gas turbine engine. The next subsection gave brief descriptions of nine or so existing methods used to model such phenomena within the context of a whole engine transient performance code.

Section 2.4 listed the engineering requirements for heat soakage and tip clearance methods laid down by the sponsor, i.e. to be able to handle both tip clearance and heat soakage, to be physics based, to meet tip clearance accuracy targets, to run at or close to real time, and be capable of being calibrated. The section also laid out the management topic requirement, i.e. to address the issues relating to the Cranfield UTC to Rolls-Royce technology transfer process.

Section 2.5 showed that none of the methods described in the literature review fully meet project sponsor requirements, also that not all of the methods produced by the Cranfield UTC make their way into Rolls-Royce working practice. The section ended by saying that these two issues constitute the research gaps associated with this project.

Section 2.6 described the strategy adopted by the author in an attempt to meet project requirements, i.e. essentially further development of the GTTrans method within the context of a Rolls-Royce Trent 500 performance code. It also describes the decisions on project scope made by the author in an attempt to restrict the engineering and management topic tasks to size appropriate to a single person's EngD project.

The author's contribution toward meeting engineering requirements is discussed in chapters 3 and 4. The author's contribution toward meeting management topic requirements is given in chapter 5.

Chapter 3: METHOD

3.1 CHAPTER INTRODUCTION

The purpose of this chapter is to present the heat soakage and tip clearance methods developed by the author in an attempt to meet the project requirements, and to describe the incorporation of these methods into a whole engine transient performance code.

The sponsor's requirements for heat soakage and tip clearance methods are that they should be able to handle both heat soakage and tip clearance, be physics based and therefore have some predictive capability, be accurate, fast, and have the capacity to be calibrated against more authoritative data as and when it becomes available. The strategy adopted by the author has been to further develop the GTTrans method of hardware modelling within a Rolls-Royce Trent 500 performance code. In order to account for conductive heat transfer the author decided to use a discretised volumes approach. For speed the author separated thermal and mechanical calculations so that component elasticity could be dealt with more simply with a minimum of iteration, and conduction calculations were performed only once for each sub-volume during each time interval. In this way the author hoped to approach the accuracy of finite difference and finite element approaches, but without the slowness associated with iterative whole matrix operations (e.g. as in the finite element method). Models always contain less physics than the real world and necessarily involve simplifying assumptions. Because of this the author decided that modifications to those assumptions would provide the means by which individual component models could be calibrated.

In this chapter the author develops the approaches used to represent convection, conduction, and radiation in the heat soakage and tip clearance models. He then goes on to explain how the methods account for thermomechanical growth within individual engine components. Finally the author explains how the heat soakage and tip clearance methods are incorporated into the BD31 version of the Rolls-Royce whole engine performance code, RRAP.

3.2 CONVECTION

3.2.1 TRANSIENT CONVECTION

Consider in isolation the transient heat exchange between a hot fluid and cold solid of similar thermal mass and specific heat capacity. Assume that both solid and fluid behave as point thermal masses, i.e. each has but a single representative temperature. During each time interval a quantity of heat ' q ' will pass from the fluid to the solid, the fluid will cool down and the solid will heat up (see figure 3.2.1.1). Eventually both fluid and solid will share a common temperature and a steady state will have been reached.

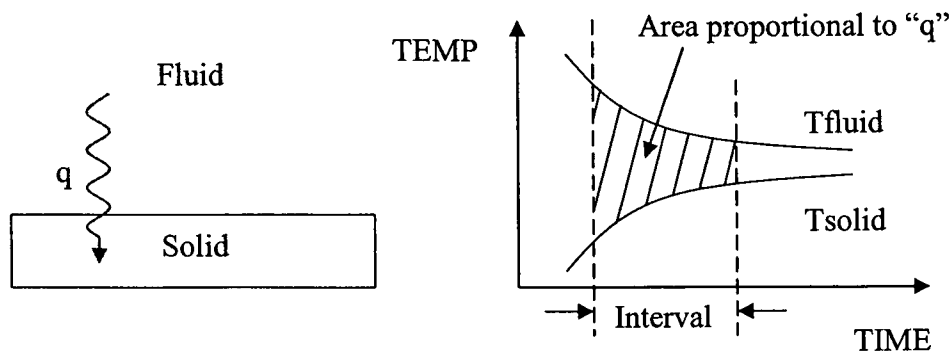


Figure 3.2.1.1 One-dimensional transient convective heat transfer

The equation used to describe the heat flux between fluid and solid at any given instant is:

$$dq/dt = h.A.(T_{\text{fluid}} - T_{\text{solid}}) \quad (3.2.1.1)$$

q	quantity of heat
t	time
h	convective heat transfer coefficient
A	wetted area
T _{fluid}	temperature of the fluid
T _{solid}	temperature of the solid

The equation used to describe the change in temperature of a body receiving or rejecting a quantity of heat (at constant pressure) is:

$$\Delta T = \Delta q / M.C_p \quad (3.2.1.2)$$

T	temperature of the body
q	quantity of heat
M	thermal mass of the body
C _p	specific heat at constant pressure of the body

Integration of equation 3.2.1.1 with respect to time over a given interval will yield the magnitude of the quantity of heat passed from fluid to solid. Knowing the amount of heat transferred during the time interval we can calculate the new temperature of the solid and the fluid using equation 3.2.1.2.

The simplest way to integrate equation 3.2.1.1 would be to assume that during the time interval both solid and fluid temperatures remained constant at perhaps their start values, or their average values, as shown in figure 3.2.1.2. Indeed this is the approach used in BR081.

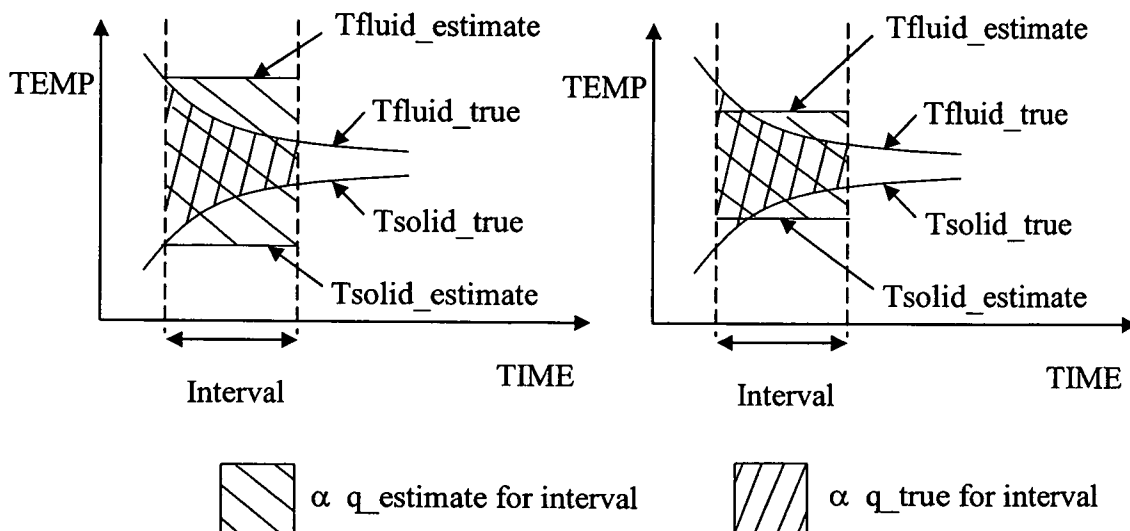


Figure 3.2.1.2 Heat quantity estimation using fixed temperatures

Slightly more sophisticated is the lumped capacitance method, which assumes that one of the bodies has a constant temperature during the time interval, but that the other changes according to an exponential decay (see figure 3.2.1.3).

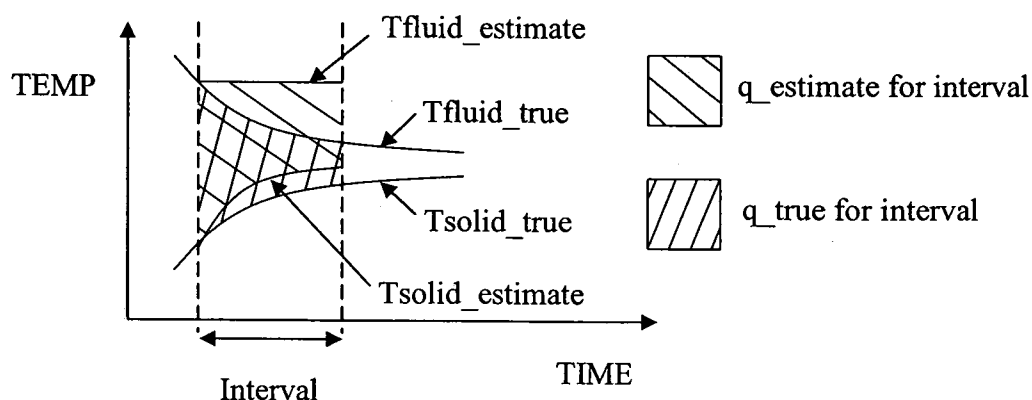


Figure 3.2.1.3 The lumped capacitance approach.

The lumped capacitance method is derived from equations 3.2.1.1 and 3.2.1.2 using integral calculus to produce equation 3.2.1.4 for temperature change of the solid (in this example) during the time interval.

$$\tau = M.C_p / (h.A)$$

$$(T_{st1} - T_f) / (T_{st0} - T_f) = e^{-\Delta t / \tau} \quad (3.2.1.4)$$

T_{st1}	temperature of the solid at the end of the interval
T_{st0}	temperature of the solid at the start of the interval
T_f	temperature of the fluid
Δt	time interval
τ	time constant
M	mass of the solid
C_p	specific heat at constant pressure of the solid
h	convective heat transfer coefficient
A	wetted area

An important assumption within the lumped capacitance method is that the thermal conductivity of the solid is infinite. This makes the lumped capacitance method only good for low Biot numbers (discussed later in this section).

If we assume that at least one of the bodies has a fixed temperature during the time interval then this can lead to an overestimation of the quantity of heat transferred during the time interval (c.f. figures 3.2.1.2 and 3.2.1.3). This is particularly so when the time interval is large relative to the system response time. This can become a big problem when calculations are repeated in series during a given investigation period. Overestimation of heat transferred in the first interval will result in an overestimation of temperature response in both fluid and solid. As a result, at the start of the next interval the temperature difference between fluid and solid will be smaller than it should be, this then results in an underestimation of heat transferred for that time interval. The cycle repeats and the result is system oscillation. If the oscillations become large enough they can become unstable and calculations no longer represent the system being modelled. Reducing the time interval relative to the response time of the system reduces the problem of overestimation and instability. However, reducing time interval increases computational time.

To help deal with the conflict between stability and computational time, the author took the lumped capacitance method one stage further by removing the constant temperature assumption allowing both body temperatures to change according to an exponential decay, giving greater realism to the answer and thus a lower propensity toward numerical instability. In addition, the author introduced the effect of a non-infinite solid thermal conductivity into the convective heat transfer calculation.

Consider figure 3.2.1.4 below. As before both fluid and solid are treated as point thermal masses, and during a particular time interval a quantity of heat is passed from the hotter fluid to the cooler solid. This time however we assume that the fluid flows past the solid and we calculate the mass of fluid ' M_f ' interacting with the solid during the time interval to be equal to the fluid mass flow rate ' W_f ' multiplied by the time interval ' Δt '. In addition, the transfer of heat is obstructed not only by the boundary layer at the fluid solid interface, but also by the thermal resistance of the solid, as it makes its way from the centre of mass of the fluid to the centre of mass of the solid.

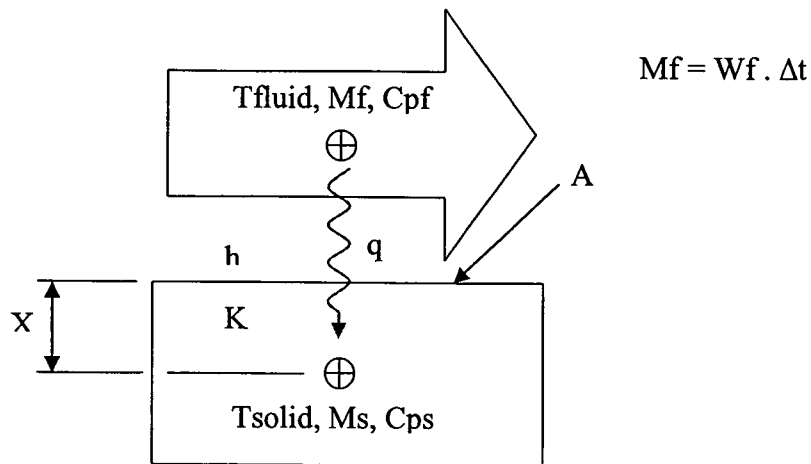


Figure 3.2.1.4 Transient convective heat transfer

Equation 3.2.1.5 for convective heat transfer shows that for a given difference between fluid temperature and solid temperature, at any given instant heat flux from fluid to solid is proportional to the term ' $h \cdot A$ '.

$$dq/dt = h.A.(T_{\text{fluid}} - T_{\text{solid}}) \quad (3.2.1.5)$$

Equation 3.2.1.6 is an expression of Fourier's conduction law, used to describe instantaneous conductive heat flux along a negative temperature gradient. We can see from equation 3.2.1.6 that for any given temperature difference between two point in a solid, assuming a linear temperature gradient at any given instant heat flux within a solid is proportional to the term ' $K \cdot A / X$ '.

$$dq/dt = - K \cdot A / X (T_1 - T_2) \quad (3.2.1.6)$$

q	quantity of heat
t	time
K	thermal conductivity of the solid
A	conduction area normal to the flow of heat
T1	temperature at point 1 in the solid
T2	temperature at point 2 in the solid

Given a set of resistances acting in series, their combined effect, i.e. the total resistance, can be calculated simply by summing the individual resistance values. However, the terms $h.A$ and KA/X are more akin to conductance, the inverse of resistance, and so to sum the terms $h.A$ and KA/X in series we must use the following equation.

$$1/U_{\text{total}} = 1/U_1 + 1/U_2 + 1/U_3 \dots \quad (3.2.1.7)$$

U_{total}	total conductance
U_1, U_2, U_3	individual conductances

Using equations 3.2.1.1, 3.2.1.2, 3.2.1.6, and 3.2.1.7, integrating with respect to time once to find an expression for transient fluid and solid temperatures as a function of time, and then integrating with respect to time again, we obtain equation 3.2.1.8 below, i.e. an expression for the quantity of heat exchanged during the interval in terms of the initial temperatures and a quantity 'Z' representing the thermal characteristic of the system.

$$\begin{aligned}
 U &= 1/(1/(h.A) + 1/(K.A/X)) \\
 G &= Wf. \Delta t. Cpf \\
 M &= Ms. Cps \\
 Z &= -(U/G + U/M) \\
 q &= U.((Tf0-Ts0)/Z).(e^{Z \Delta t} - 1)
 \end{aligned}
 \tag{3.2.1.8}$$

h convective heat transfer coefficient
 A wetted area
 K thermal conductivity of the solid
 X normal distance from wetted area to solid centre of mass
 Wf fluid mass flow
 Ms mass of solid
 Δt time interval
 Cpf, Cps specific heat (constant pressure) of fluid and solid
 q quantity of heat transferred during time interval
 Tf0, Ts0 temperature of fluid and solid at start of time interval

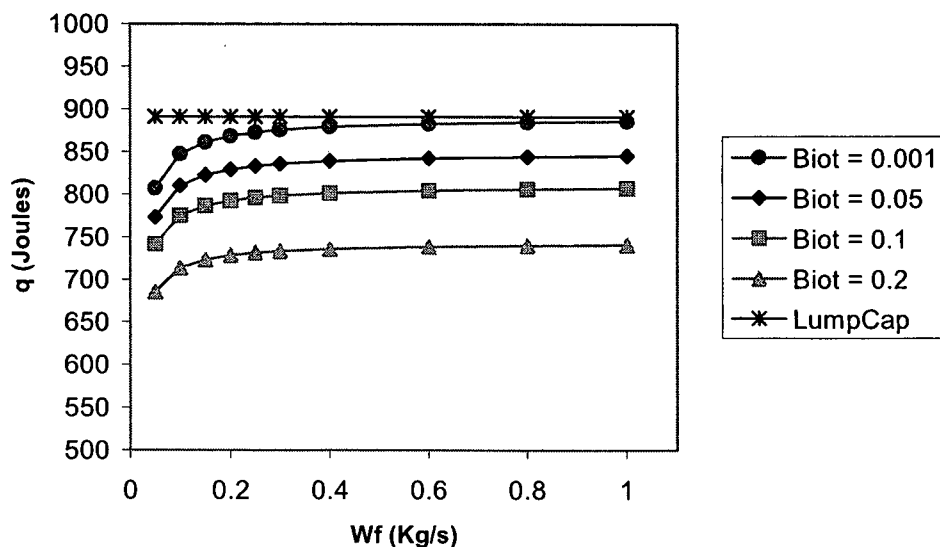


Figure 3.2.1.5

Equation 3.2.1.8 versus lumped capacitance method for a range of fluid mass flows and Biot numbers. Lumped capacitance method significantly overestimates heat transfer at high Biot numbers and low fluid mass flows.

Ms = 1kg Cps = 500J/kgK Ts0 = 10K
 A = 0.1m² Cpf = 1000J/kgK Tf0 = 100K
 h = 100 W/m² K Δt = 1 second.

By using equation 3.2.1.8 we are less likely to encounter numerical instability from overestimation of heat transfer per time interval. In addition, equation 3.2.1.8 performs better (see figure 3.2.1.5) than the lumped capacitance method in situations where neither fluid or solid can be assumed to be a temperature reservoir and where conduction is significant, i.e. where fluid mass flows are relatively low and where Biot numbers are relatively large - situations common in bleed paths in gas turbine engines, or where primary gas path components have thermal barrier coatings.

Essential to any kind of convective heat transfer calculation are representative values of heat transfer coefficient for each wetted surface. Heat transfer coefficients can be inferred from test measurements, calculated from such tools as computational fluid dynamics packages, or estimated using correlations found empirically for heat transfer occurring under specific flow conditions for specific hardware geometries. In this project the latter method has been used. Three different heat transfer coefficient correlations were used: that for flow through a duct, that for forced convection from a rotating disc, and that for natural convection from a flat vertical surface (c.f. appendix D). In this study, heat transfer in the primary gas path was calculated using the correlation for flow through a duct. In the disc cavities heat transfer was calculated using the correlation for forced convection from a rotating disc. Heat transfer in the casing bleeds cavities was calculated using the correlation for flow through a duct in the most part, but in some places (e.g. bleed path dead ends) the correlation for natural convection from a flat vertical surface was used. The correlations used in this project were provided by Rolls-Royce. The decision to use heat transfer coefficient correlations that Rolls-Royce already use as opposed to those available in the open literature was made as it was thought that this would make the methods developed in this project more acceptable to Rolls-Royce, thus facilitating the technology transfer process.

In addition to convective heat transfer affecting fluid and solid temperatures, when parts rotate at high speed there is also the important effect of viscous losses heating up the fluid to take account of (i.e. windage). The equation used for windage heating is also given in appendix D.

3.2.2 THE STREAMS METHOD

The streams method is essentially the application of equation 3.2.1.8 to geometries more complicated than those of subsection 3.2.1. Consider a long solid as shown in figure 3.2.2.1. The streams method can be used to estimate the transient temperature distribution of the fluid and the solid as well as the transient heat flow between fluid and solid. To do this the streams method involves dividing the solid up into a number of discrete solids each with its own wetted surface (see figure 3.2.2.1). Then, equation 3.2.1.8 is used once on each surface in sequence during the time interval. The temperature of the fluid leaving surface 1 is used as the temperature of the fluid entering surface 2, and so on, until the fluid leaves the solid completely. At this stage, the assumption is that no conductive thermal exchange occurs between any of the four sections. By repeating this procedure for every interval throughout a given period of interest, the streams method gives an estimate of the changing temperature profiles along the length of the solid and fluid, and hence transient heat flows. Figure 3.2.2.2 shows the streams method predictions of fluid and solid temperature profile along the

four surfaces of the solid shown in figure 3.2.2.1, as compared to those using the lumped capacitance method.

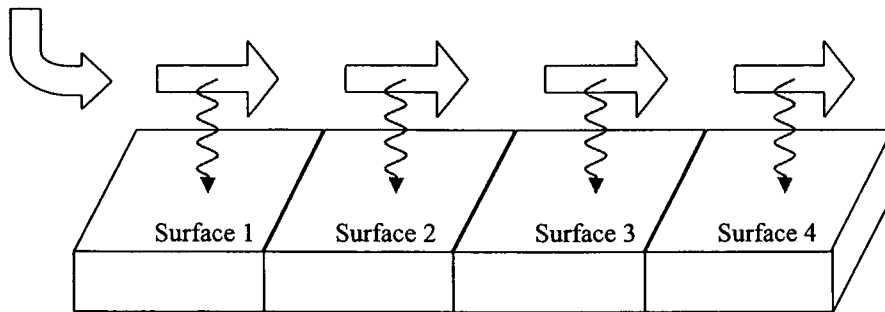


Figure 3.2.2.1 A long solid divided into sections, receiving heat from a fluid flowing along its length.

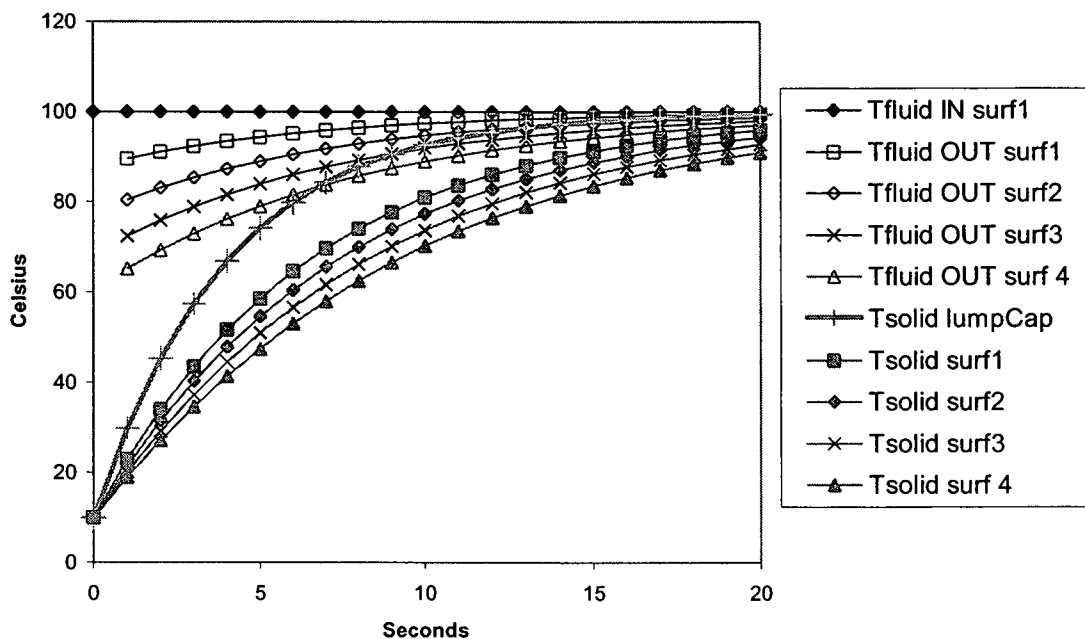


Figure 3.2.2.2 Streams method approximation of fluid and solid temperature profiles (see figure 3.2.2.1, and equation 3.2.1.8) as they change with time, compared to the lumped capacitance (LC) method. All surfaces have the following: $A = 0.1\text{m}^2$, $M_s = 1\text{kg}$, $C_{ps} = 400\text{ J/kgK}$, $X = 0.01$, $K = 20\text{W/mK}$, and for the fluid the following: $W_f = 0.5\text{kg/s}$, $h = 1000\text{ W/m}^2\text{K}$, and $\Delta t = 1\text{ second}$

In the study that produced figure 3.2.2.2, the time interval was varied from 0.1 to 10 seconds. It was found that the method remained stable, and only very small changes occurred to the shapes of the curves, i.e. the curves shifted to the right slightly with larger time intervals. Use of the streams method for more complex geometries is an improvement upon the lumped capacitance method in that it estimates solid and fluid temperature profiles and heat flows as they change with time, as well as accounting for the effect of solid thermal conductivity.

So far we have shown the streams method used on surfaces in series. Figure 3.2.2.3 shows how it can also be used in parallel, and in series and parallel together.

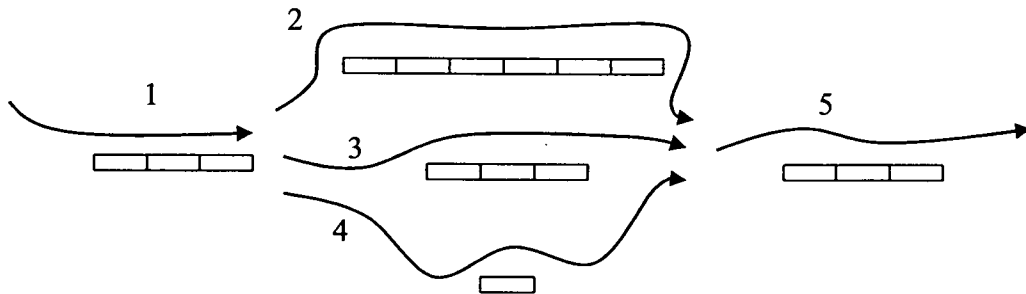


Figure 3.2.2.3 An arrangement of streams in series and in parallel.

Within the streams method, the flow over a set of surfaces in series is called a stream, and a set of streams in series and/or parallel is called a stream assembly. Figure 3.2.2.3 shows a stream assembly. After the fluid leaves stream 1, its mass flow is divided into three, and fluid inlet temperature for streams 2, 3, and 4 equal to the fluid exit temperature from stream 1. The inlet temperature to stream 5 is found from the average, weighted by mass flow, of the exit temperatures of streams 2, 3, and 4.

Figure 3.2.2.4 shows how the streams method might be applied to the disc cavity of a multistage turbomachinery component, with the assumption that all geometry is axisymmetric.

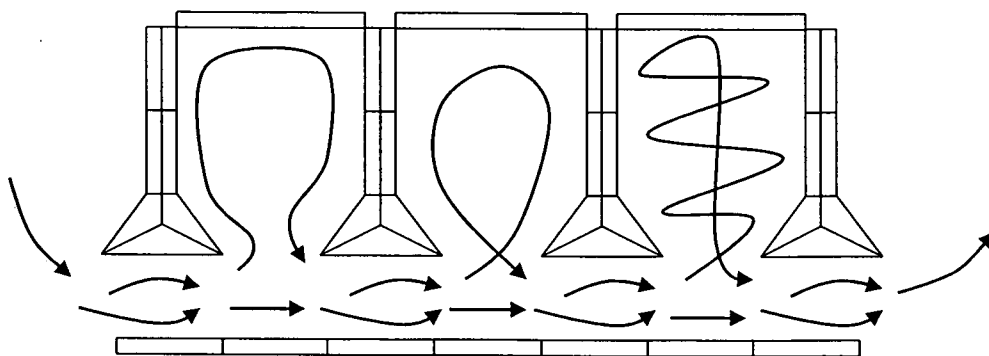


Figure 3.2.2.4 A streams method model of a multistage disc cavity.

Knowledge of the transient temperature profiles of the bleed air and of the discs can be used to help estimate disc growths (and thence blade tip gap changes), and also transient bleed exit temperatures.

There are of course problems with this approach. To name but a few, as figure 3.2.2.4 suggests, a degree of artistic licence is required when deciding flow patterns because it is always uncertain how much air is flowing where inside a gas turbine multistage disc cavity. In addition there are limits to the applicability of the underlying streams method assumption that all of the fluid entering contact with the solid interacts with the solid and then leaves contact with the solid entirely within the time interval. For example, if in figure 3.2.2.1 we imagine the solid to be very long then the fluid that first entered into contact with the solid at the start of the time interval would never in reality reach the last section of the solid by the end of the time interval. Also, in figures 3.2.2.3 and 3.2.2.4 we assume the flows channelled along the parallel arms of the streams assemblies all reach their recombination junctions together at exactly the same time, which is unlikely to be the case in reality.

The streams method does not model flow to a high level of sophistication, however, with a little effort and engineering judgement it can give a fast and fairly representative first estimate of transient convective heat transfer and temperature profiles in moderately complicated systems. Better estimates are likely to require more rigorous investigations using CFD, FEA, and/or experimentation. These alternatives are not always practical in the early stages of novel gas turbine engine development, which is when forewarning of heat soakage and blade tip gap issues has the potential to result in the greatest time and cost savings.

3.2.3 STEADY STATE CONVECTION

For any numerical method involving division of the system into a large number of constituent elements or volumes (e.g. finite element analysis, computational fluid dynamics) a series of iterations is required before a set of solutions can be obtained. Iterations are repeated every interval until convergence. The streams method is no different, and unfortunately the only practical way the streams method can be used to estimate steady state gas temperature profiles and convective heat transfer is by running it with constant inputs for an extended number of time intervals until the whole system being modelled reaches stabilisation. This is particularly true when convective heat transfer calculations are combined with conductive heat transfer calculations (see next section).

3.3 CONDUCTION

3.3.1 TRANSIENT CONDUCTION

The streams method accounts, to a degree, for the influence of thermal conductivity of the solid on convective heat transfer in a system. However, thus far the streams method has assumed that the solid material associated with each convecting surface does not thermally interact with the solid material associated with adjacent surfaces.

This is clearly not the case in reality. To improve the situation the author developed a finite volumes style approach to estimate conductive heat flows within the solid parts of streams method models. Consider the one-dimensional steady state heat flow through a solid as shown in figure 3.3.1.1.

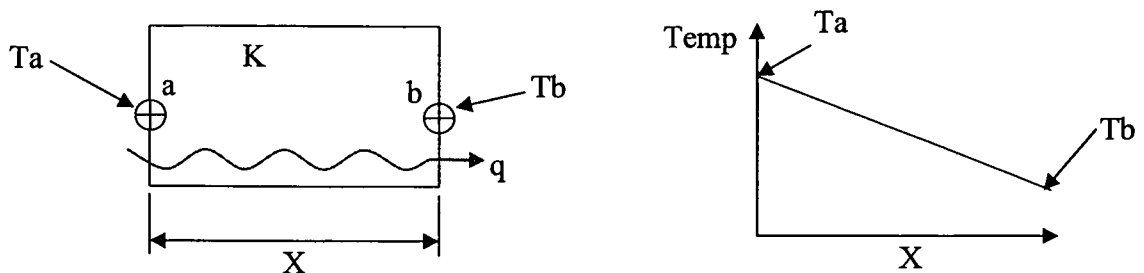


Figure 3.3.1.1 One dimensional steady state conductive heat flow through a solid.

Under steady state conditions the one-dimensional heat flux through the solid is given by Fourier's conduction law, as expressed by equation 3.3.1.1.

$$dq/dt = -(K \cdot A / X) \cdot (Ta - Tb) \quad (3.3.1.1)$$

q quantity of heat
t time
K thermal conductivity
A normal area of conduction
X linear distance between a and b
Ta temperature at a
Tb temperature at b

From equation 3.3.1.1 we can see that steady state heat flux is proportional to the conductance term ' $K \cdot A / X$ '. Let us consider the case shown in figure 3.3.1.2.

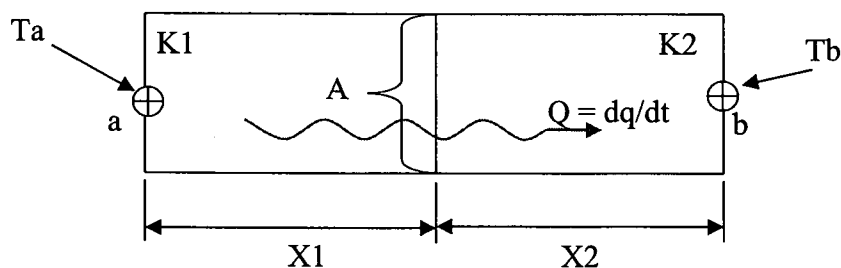


Figure 3.3.1.2 One dimensional steady state conduction through a sectioned solid

Figure 3.3.1.2 shows the case where one part of the solid is made of a different material to the other part of the solid. Equation 3.3.1.1 still applies, however the conductance term ' KA/X ' must be recalculated to account for the difference in material properties associated with each part of the solid. Resistances in series can simply be summed to find total resistance, however, conductance is the inverse of resistance and so to find our total conductance we must use the following equation.

$$1/U_{\text{tot}} = 1/U_1 + 1/U_2 \quad (3.3.1.2)$$

U_{tot} = the value of $K.A/X$ to use in equation 3.3.1.1

U_1 = $K_1.A/X_1$

U_2 = $K_2.A/X_2$

We can extend this approach up to more than one-dimension. Consider the section of the three dimensional axisymmetric solid shown in figure 3.3.1.3. By discretising the solid into a finite number of smaller axisymmetric volumes we can perform heat transfer calculations for each volume in turn. Dividing the solid into volumes with four sides, each volume will exchange heat with up to four neighbours.

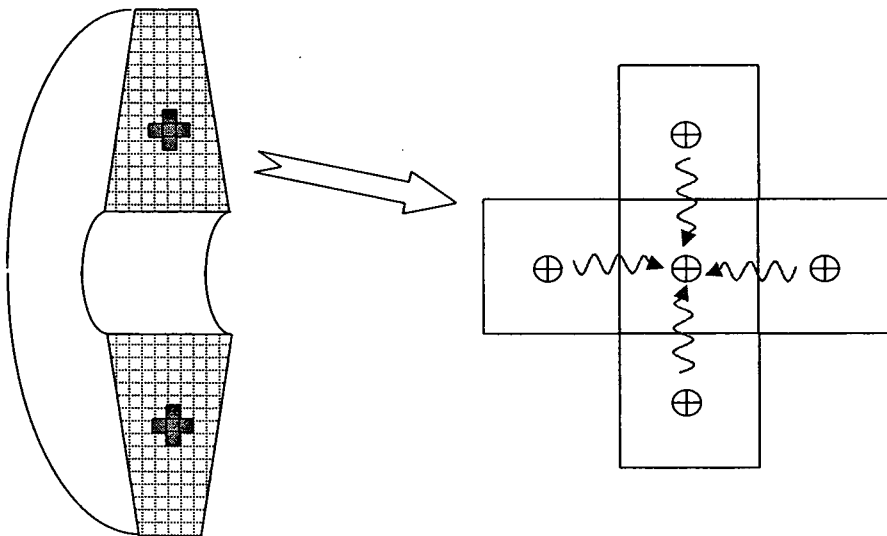


Figure 3.3.1.3 **finite volumes conductive heat transfer**

We can extend this approach to transient conductive heat transfer calculations. Let us consider the view on the right hand side of figure 3.3.1.3, and assume that the mass of each volume is concentrated on a circular line passing through the centre of area of the cross section shown. The temperature distribution of the solid can thus be defined in terms of the temperatures of the concentrated masses of each of its volumes. At each time interval if we know the temperature of the concentrated mass of each volume, the thermal conductivity of each volume, and the conduction area between a volume and each of its neighbours, then we can apply equations 3.3.1.1 and 3.3.1.2 to find the instantaneous heat exchange between a volume and each of its neighbours.

Integrating this value with respect to time we can find the amount of heat absorbed by the volume during the time interval. We can then use equation 3.3.1.3 to find what the temperature of the volume should be at the end of the time interval.

$$\Delta T = \Delta q / M.C_p \quad (3.3.1.3)$$

T	temperature of the volume
q	quantity of heat
M	thermal mass of the volume
C _p	specific heat at constant pressure of the volume

If we repeat this procedure for each of the volumes in turn (assuming the neighbouring volumes' temperatures remain constant at their starting value during the interval), for each time interval throughout the period of interest, the result will be an estimate of the transient conductive heat flows and temperature profiles within the solid.

The result of integrating equations 3.3.1.1, and 3.3.1.2 for the situation shown in figure 3.3.1.3 is the following equation for the amount of conductive heat transfer to each volume from each neighbour.

$$\begin{aligned} T_{ss} &= \text{Sum}KT / \text{Sum}K \\ \text{Tau} &= (\text{Mass} \cdot C_p) / \text{Sum}K \\ q &= -KAQX * ((T_{ss} - T_n) * t + ((T_{ss} - T_m) * \text{Tau}) * (\text{EXP}(-t/\text{Tau}) - 1)) \end{aligned} \quad (3.3.1.4)$$

T _{ss}	Pseudo steady state temperature of the volume
SumKT	Sum of the product of conductance term (see eqn 3.3.1.2) multiplied by the temperature for each of the volume's neighbours
SumK	Sum of the conductance terms (see eqn 3.3.1.2) for each of the volume's neighbours
Mass	Mass of volume
C _p	Specific heat of the volume (at the appropriate temperature)
q	Quantity of heat passed to volume from one of its neighbours
KAQX	Conductance term between volume and one of its neighbours (see eqn 3.3.1.2)
T _n	Temperature of neighbour at start of interval
T _m	Temperature of volume at start of interval
t	Time elapsed

The third line of equation 3.3.1.4 contains a term before the "+" sign and a term after it. If the system is left to stabilise, then the second term exponentially decays to zero whereas the first term will persist. At which point the first term will represent the steady state value of heat passed per time interval between any particular neighbour and the volume itself. The 'mask' of five volumes visits each of the volumes in turn until all of the volumes in the solid have been visited before the end of time interval.

This 'mask' of five volumes is a simplification. In more rigorous methods, such as finite element analysis, a stiffness matrix for the whole solid is constructed and a whole matrix solution is found during every time interval. Figure 3.3.1.4 shows the streams transient conduction method model and the ANSYS® (finite element software) model of a test solid used to compare the similarity of results produced by the two methods.

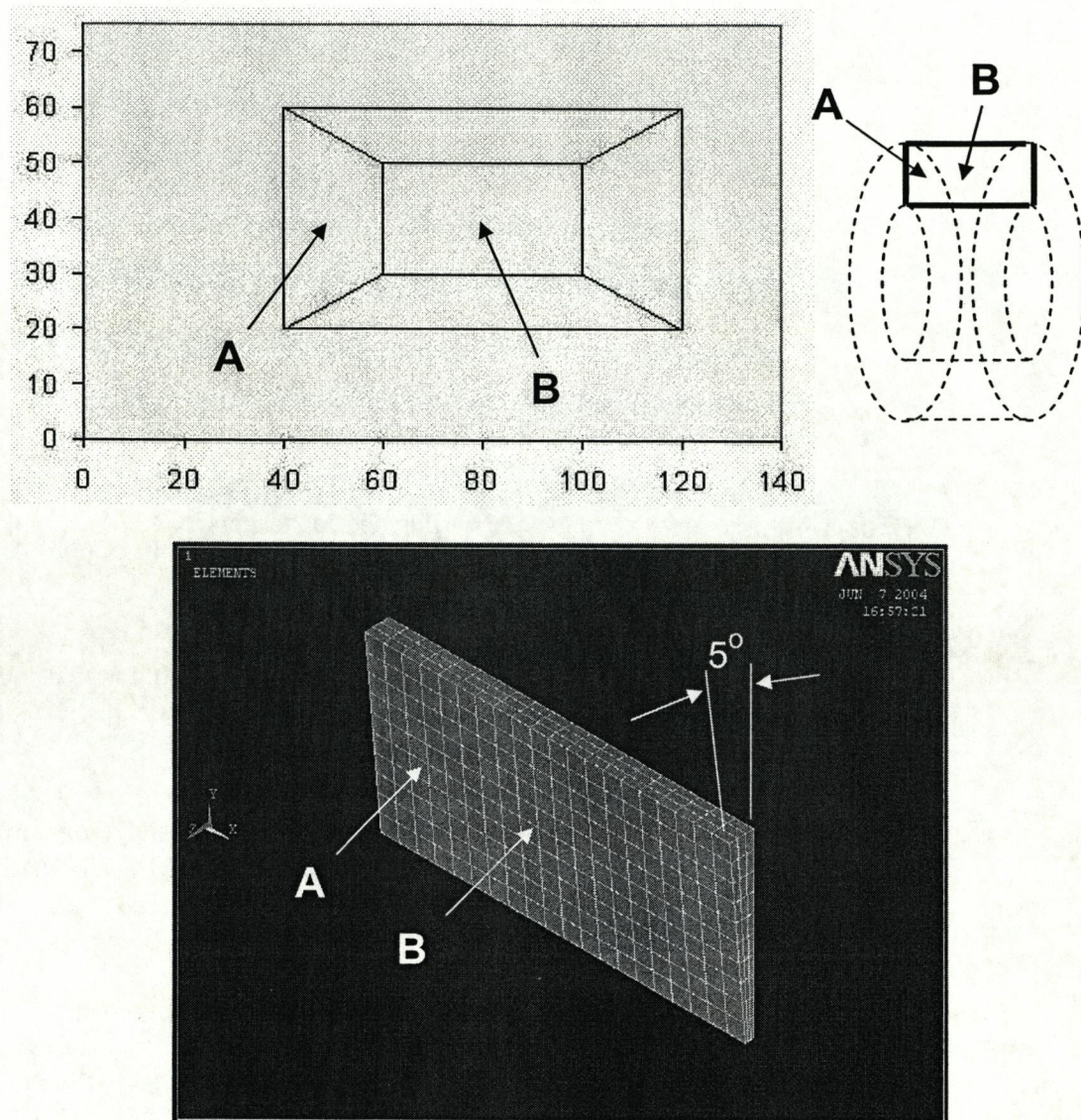


Figure 3.3.1.4 Streams model (top) and ANSYS® model (bottom)

The test solid they both represent is a rectangular sectioned ring. Figure 3.3.1.4 (top) shows the ring cross section, and figure 3.3.1.4 (bottom) shows the five degree slice of the whole solid used in ANSYS®. Just 5 subvolumes were used to represent the streams conduction method model, whereas the ANSYS® model used a much larger number of elements. The objective was to see how well a coarse streams method representation would approximate a finer more rigorous method. This was because in

future streams method models of engine components, only a very coarse discretisation would be used in order to increase computational speed and reduce set up times.

Both the streams and ANSYS® models had the same geometry, material properties, and boundary conditions. The transient temperatures at points A and B (see figure 3.3.1.4) were taken from both models and the results are shown in figure 3.3.1.5 compared to results of applying the same boundary conditions to the lumped capacitance method.

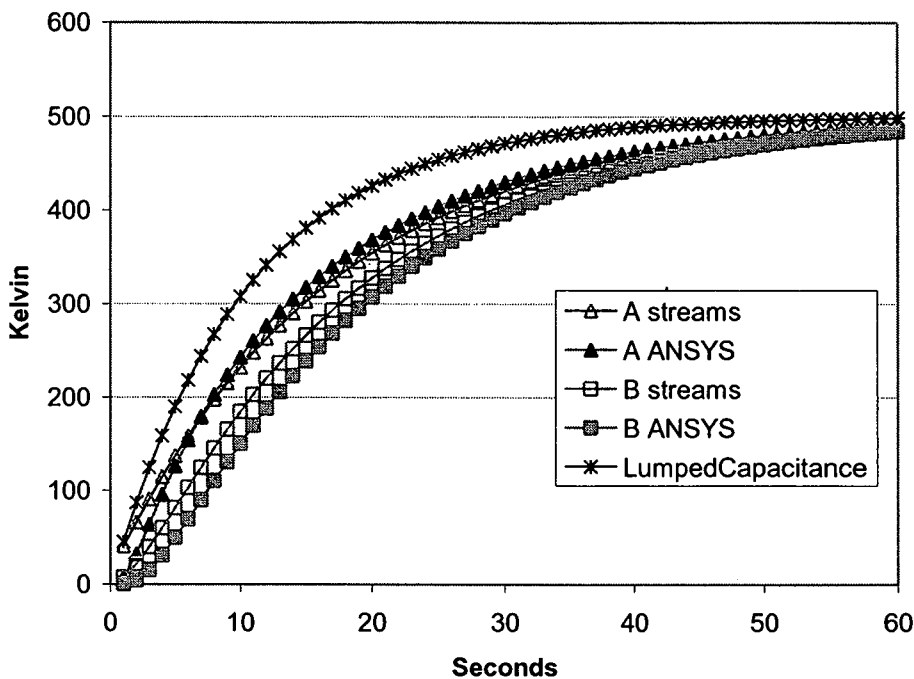


Figure 3.3.1.5 Streams model (top) and ANSYS® model (bottom) and lumped capacitance method, for Mass = 7.16, wetted area = 0.0603, $h = 5000$, $cp = 440$. $Bi = 2.2$

Taking results from the ANSYS® software to be authoritative, we can see that the streams conduction method model performed much better than the lumped capacitance model, not only capturing the thermal delay due to the effect of conduction, but also producing an internal temperature distribution not dissimilar to that produced by ANSYS® despite the coarseness of the streams method discretisation.

In addition to the mask versus full matrix approach of the streams method, two other factors have a significant effect on its performance. The first is the fineness of the discretisation, and the second is the squareness of the subvolumes. The finer the discretisation the more accurate the transient temperature profile, however the slower the model is to run and to set up, and for the streams method speed of use is an important requirement. Furthermore, the further a subvolume's section is away from rectangular then the more the streams method over-represents thermal resistance in

the conductive heat transfer calculations. This can be a problem when solids are being modelled that are not convenient to rectangular discretisation.

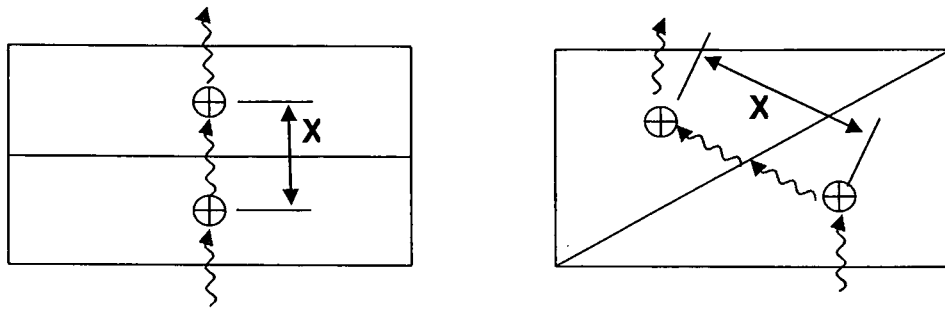


Figure 3.3.1.6 Conduction calculation inaccuracy occurring due to non-rectangular subvolumes

Figure 3.3.1.6 shows how non-rectangular subvolumes over-estimate the thermal resistance of a system. In the streams method, the distance X (see figure 3.3.1.2) is calculated as the distance from a volume's cross sectional centroid to the midpoint of the line representing the relevant conducting surface. When subvolumes are rectangular, this produces a good representation of the distance X , however, we can see that when they are not rectangular, the distance through which heat is calculated to flow can end up being significantly longer (see figure 3.3.1.6).

3.3.2 STEADY STATE CONDUCTION

The scheme for predicting the steady state temperature distribution of a solid using the streams method discrete volumes approach for a fixed set of boundary conditions is more simple than it is for transient temperature distributions, and consists of essentially a reduced form of equation 3.3.1.3. Unfortunately, boundary conditions for the systems modelled in this study are generally set by convection calculations, and because steady state convection cannot be calculated using the streams method (it can only be done by running the system and then waiting for it to stabilise), this means that there is little point doing separate steady state calculations for solid temperature distribution.

3.4 RADIATION

Radiation within a gas turbine can be thought of as occurring by two separate mechanisms, i.e. non-luminous radiation, and luminous radiation. Non-luminous radiation is essentially spectral radiation and occurs when combustion products emit energy at wavelengths corresponding to particular chemical species. Luminous radiation is essentially black body radiation and occurs in proportion to the level of soot that exists in the engine. According to (Lefebvre 1999) in modern gas turbines the flame is characterised by a predominance of luminous radiation, i.e. that occurring due to the presence of soot. Soot formation takes place in the primary zone of the combustor, and soot burnout takes place in the intermediate and dilution zones of the

combustor. In light of this, radiation can be assumed to be very much a combustion chamber issue.

Alves (1989) presents the results of a study in to the effect of combustor heat soakage on transient gas turbine performance. His study included radiation, convection, conduction, and also fuel evaporation and heat release dead time and delays. In his conclusion he reports that "The effect of heat storage in the combustor has no significant effect, even at static, high altitude, or during a step fuel flow transient." Similar experience was found elsewhere (Fiola 2002). Moreover, according to a chief combustion specialist at Rolls-Royce (Jones 2003), the huge amount of fuel energy release that takes place within the Trent 500 combustor means that the combustor's thermal response is near instantaneous, and a transient thermal model of the combustor is unlikely to be a useful exercise. Consequently, the author excluded any kind of combustor thermal model (radiation or otherwise) from the present study, however, such a course of action is unlikely to be justifiable for land based engines which typically have combustion chambers of very significant thermal mass.

3.5 THERMOMECHANICAL GROWTH

The effect of heat transfer to and from, and also within the engine carcass is to change the temperature of its various components. When the component parts of a piece of turbomachinery change temperature their geometry also changes through thermal expansion or contraction. In addition to thermal changes, turbomachinery components undergo geometry changes due to pressure imbalance and to centrifugal effects. Two types of geometry change that can have a significant effect upon engine performance (excluding variable geometry of course) are alterations to labyrinth seal clearances and blade tip clearances.

In the early work carried out by Maccallum (1979) and Pilidis (1983) changes in labyrinth seal gaps due to thermal and centrifugal effects were found to have quite a significant effect upon bleed mass flows, and thus the transient performance of a gas turbine engine. Of course, the effect of seal clearance changes is different for different types of engine, and is specific to the geometry and layout of its hardware. In some more modern engines, hardware designs have been developed whereby this sensitivity is better controlled. Flow through an orifice is far more predictable than flow through a labyrinth seal sensitive to temperature and rotational speed changes. In the Trent 500, critical bleed flows are controlled by strategically located orifices rather than labyrinth seals (Rowe 2000). For this reason, models of transient seal clearance changes have been excluded from this study.

Unfortunately, blade tip clearances are more difficult to control. Not only do blade tip clearances change radially due to centrifugal growth of the rotor, but hade angle in the casing can also result in an axial component to blade tip clearance changes. In addition to thermal axial growths, the lift generated by compressor blades results in a nett blade force in the direction of the front of the engine, whereas in the turbines the nett force is toward the rear of the engine, and consequently, shafts within the engine experience substantial mechanical stretching. The engine's internal cavity geometry is often designed such that the pressures exerted by various bleed flows can be used to help balance the nett axial forces experienced by the fixed bearings, which

can be quite considerable especially for the fan and low-pressure spool. To mitigate the effects of axial components of tip clearance in the Trent 500 the internal casing of the high pressure compressor is built with zero hade. Also, turbine blades are shrouded to reduce the effect of tip clearance, and turbine casing liners are built with zero hade. In addition, the arrangement of the fixed and floating shaft bearings is such that fixed bearings are located in between the intermediate and high pressure compressors, thus minimising the axial shaft movement they experience (compared to their associated turbines). Given these design features, and the small size of the axial component of tip clearance relative to its radial component, and the complexity associated with calculating axial movements of rotors relative to casings, it was decided that in this study only radial changes in tip clearance would be modelled.

The subsections that follow describe the methods used to calculate the radial components of changes in blade tip clearance. The effect of tip clearance on the steady state performance of a component is captured in the component's characteristic map. During engine manoeuvres a component's transient departure from these steady state tip clearances affect the transient performance of the engine. For this reason the focus for this project has been to calculate transient changes in tip clearance rather than the clearances themselves. If we assume that the cold build clearance for each stage of each component is zero, we can perform a steady state tip closure calculation, and also a transient tip closure calculation, and then subtract the two to find the transient change in tip clearance without ever needing to know if our assumption of zero cold build tip clearance was true or not. Thus we achieve our goal with a minimum of complication. An alternative to this approach would be to calculate absolute tip clearances rather than relative tip closures and use an equation such as that suggested by Lakshminarayana (1970), to find instantaneous efficiency defects. However, using the approach chosen by the author obviates the need to know cold build tip clearances, which will be different for different engines of the same type due to manufacturing tolerances and time since last overhaul.

3.5.1 CASINGS

With reference to the streams method (above) we can represent the casing of a piece of turbomachinery using a collection of small volumes (see figure 3.5.1.1)

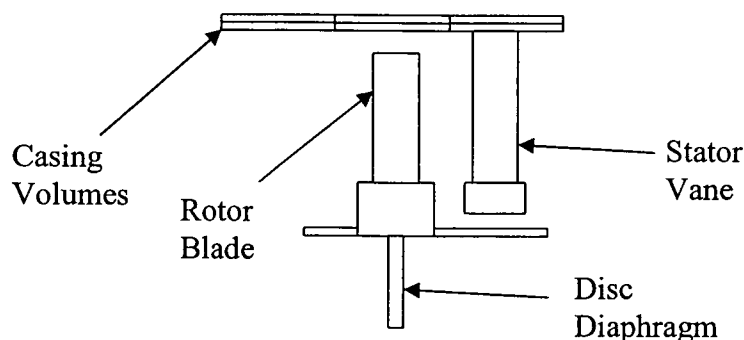


Figure 3.5.1.1

Streams method representation of the casing of a piece of turbomachinery

The two volumes directly above the rotor blade of figure 3.5.1.1 are of the greatest interest. The conduction algorithm of the streams method produces estimations of temperatures for each subvolume at each time interval. By taking the mass weighted average of the two (or more) subvolumes directly above the rotor blade, then we can find a single temperature that we assume is representative of the casing. At this point we are in a position to estimate the thermal growth of the casing at a particular stage within a component. Thermal casing growth at each stage within a component can be estimated from equation 3.5.1.1

$$\Delta R = R \cdot \alpha \cdot (T_c - T_d) \quad (3.5.1.1)$$

R	Internal casing radius
α	thermal expansivity coefficient of the casing material at T_c
T_c	temperature of the case
T_d	temperature of the case under datum conditons.

To find the mechanical growth of the casing at each stage, we need to know the pressure acting on either side of a particular section of casing. The pressure acting on the exterior side of the casing can be estimated from the bleed air pressure. The pressure acting on the interior side of the casing can be estimated from the primary gas path air/gas pressure. If a component has multiple stages then the pressure on the interior side of section of casing above the rotor blade of a particular stage can be estimated by assuming a linear pressure rise along the length of the annulus of the component. In this study matters were further simplified by using total pressures instead of static pressures, which would have been more realistic. Once we have an estimation of the pressure on each side of a particular section of casing, treating the casing as a simple pressure vessel we can calculate mechanical casing growth from equation 3.5.1.2.

$$\Delta R = (P \cdot R^2) / (E \cdot t) \quad (3.5.1.2)$$

R	Internal casing radius
P	Pressure difference between interior and exterior surfaces
E	Young's Modulus for the material at the appropriate temperature
t	Effective mechanical thickness of the casing

In this study, thermal and mechanical growth of the casing were calculated as described above and then simply added together to find the total thermomechanical growth for a particular stage of a piece of turbomachinery.

3.5.2 BLADES

The simplest way to model a blade using the streams method is to represent the blade with a single elemental volume. Following convection and conduction calculations, an estimate of the blade temperature will have been made. The simplest way to model

a cooled blade using the streams method is to represent the blade with two contiguous elemental volumes: one associated with its interior and the other associated with its exterior. During the convection calculations primary gas can be specified to interact with the exterior volume, and cooling bleed air can be specified to interact with the interior volume. Taking a mass weighted average we are able to again find a single temperature for the blade. With this single blade temperature we can use equation 3.5.2.1 to calculate the thermal growth of the blade.

$$\Delta L = L \cdot \alpha \cdot (T_b - T_d) \quad (3.5.2.1)$$

L	Blade length
α	Thermal expansivity coefficient of the blade material at T_b
T_b	Temperature of the blade
T_d	Temperature of the blade under datum conditions.

The mechanical growth of the blade can be estimated by assuming each blade is a simple prismatic member, is constructed of an elastic material of constant density. We also assume that it is attached to a disc at its root, and is centrifugally loaded by its own weight as it rotates with the disc. Using the basic engineering equations for stress, strain, and centripetal force, and using integral calculus to sum the instantaneous strain along the length of the whole member, we can arrive at equation 3.5.2.2 for mechanical growth of a shroudless blade.

$$\Delta L = (\rho \cdot \omega^2 / 2E) \cdot (((2R^3 + r^3)/3) - R^2 \cdot r) \quad (3.5.2.2)$$

L	Blade length
ρ	Density of blade material
ω	Angular velocity of the blade
E	Young's modulus of the blade material at the appropriate temperature
R	Radius of rotation at the tip of the blade
r	Radius of rotation at the root of the blade

To model the mechanical growth of a shrouded blade, we can use equation 3.5.2.3 to calculate the additional blade stretch due to the presence of the shroud.

$$\Delta L = L \cdot ((M \cdot R \cdot \omega^2) / A) / E \quad (3.5.2.3)$$

L	Blade length
M	Mass of a single shroud
R	Radius of rotation of the shroud centre of mass
ω	Angular velocity of the shroud
A	Cross sectional area of a single blade
E	Young's modulus of the blade material at the appropriate temperature

In this study thermal and mechanical blade growths were calculated as described above and simply summed to find total thermomechanical blade growth for each stage of each piece of turbomachinery.

3.5.3 DISCS

Gas turbine components feature a large variety of disc shapes. Simple models for thermomechanical disc growth may not always give representative answers. Indeed, Pilidis (1983) found that centrifugal disc growths calculated by assuming constant disc thickness had to be scaled up in his study by 30% to bring them in line with results from more rigorous finite element analysis using more realistic disc geometries. Figure 3.5.3.1 shows a schematic of one type of dovetail blade root fixing common on axial compressors. As well as disc featuring axially oriented dovetails, some discs accommodate blade dovetail roots in a circumferential orientation, and turbine blades are often attached to turbine discs using firtree rootings. It is assumed in this study that dovetails and firtrees cannot bear disc hoop stress, and therefore effectively act as dead weight.

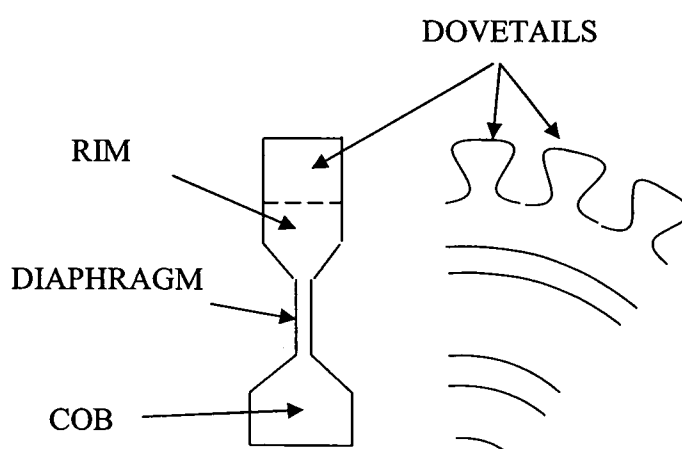


Figure 3.5.3.1 Section of a disc, showing dovetails at rim.

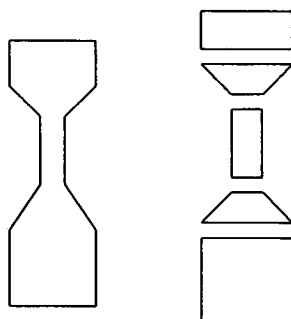


Figure 3.5.3.2 Section of a disc divided into rings.

Figure 3.5.3.2 shows how a disc can be divided into rings. Instead of trying to perform thermomechanical strain calculations on the whole disc in one go, by

performing thermomechanical strain calculations one at a time on the individual rings we can eventually come to an estimate of the total thermomechanical growth of the disc at its rim. This approach is referred to hereafter as the rings method. Young (1989) contains equations for the radial stress and hoop stress at a given radius in a homogeneous circular disc of hyperbolic section (c.f. figure 3.5.3.3) rotating about its own axis.

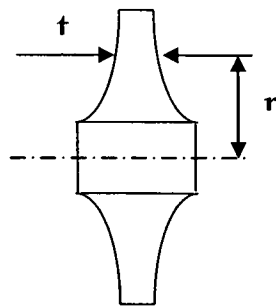


Figure 3.5.3.3 Disc of hyperbolic section

The equation $t = c r^a$ defines the section of the disc, where if t_1 is the axial thickness of the disc at radius r_1 , and t_2 is the axial thickness at radius r_2 , then (for SI units):

$$\begin{aligned}
 a &= \ln(t_1/t_2) / \ln(r_1/r_2) \\
 \ln(c) &= \ln(t_1) - a \ln(r_1) = \ln(t_2) - a \ln(r_2) \\
 F &= -(1-\nu^2) \cdot \rho \cdot \omega^2 / (E \cdot (8 + (3+\nu) \cdot a)) \\
 m_1 &= -(a/2) - ((a^2/4) - a\nu + 1)^{1/2} \\
 m_2 &= -(a/2) + ((a^2/4) - a\nu + 1)^{1/2} \\
 \sigma_r &= (E/(1-\nu^2)) \cdot ((3+\nu) \cdot F \cdot r^2 + (m_1+\nu) \cdot A \cdot r^{m_1-1} + (m_2+\nu) \cdot B \cdot r^{m_2-1}) \\
 \sigma_h &= (E/(1-\nu^2)) \cdot ((1+3\nu) \cdot F \cdot r^2 + (1+m_1 \cdot \nu) \cdot A \cdot r^{m_1-1} + (1+m_2 \cdot \nu) \cdot B \cdot r^{m_2-1})
 \end{aligned}
 \tag{3.5.3.1}$$

a, c, F, m_1, m_2	Intermediate variables
ρ	Density of the disc material
ω	Angular velocity of the disc
ν	Poisson's ratio of the disc material
E	Young's modulus at the appropriate temperature
r	Radius
σ_r	Radial stress at radius r
σ_h	Hoop stress at radius r

A and B in equation 3.5.3.1 are constants found by setting either radial stress or hoop stress to its known or assumed values at the inner and outer perimeters and solving the last two lines of equation 3.5.3.1 simultaneously for A and B.

Discs, or rather, rings, of constant thickness were not used as a large number of them would have been required to produce a sensible representation of the disc profile. Young (1989) gives equations for rotational and hoop stress at any radius in a conical disc. A small number of conical rings can be used to represent real disc cross section very well, but unfortunately the equations shoot off to infinity when presented with a disc of constant axial thickness. The equations for discs of hyperbolic section are much better behaved and so it was decided that they would form the building blocks of the rings method, even though for some discs a set of rings of hyperbolic section may not represent real disc geometry as well as a set of conical rings.

In the current discussion we assume that stresses in the axial direction are negligible. Once the stress state of each ring has been established, we are then in a position to calculate radial strains. In reality each ring is not free but rather attached to its neighbours. This being the case we assume that the rings are connected by interference fits. Equation 3.5.3.2 (Rees 1990) gives the radial interface strain of a two discs.

$$\Delta R/R = (\sigma_{h1} + \nu_1.P)/E1 - (\sigma_{h2} + \nu_2.P)/E2 \quad (3.5.3.2)$$

R	Interface radius
σ_{h1}, σ_{h2}	Hoop stress at interface for outer and inner discs respectively
ν_1, ν_2	Poisson's ratio for outer and inner discs respectively
P	Common interface pressure (radial stress)
E1, E2	Young's modulus of discs at the appropriate temperature

From equations 3.5.3.1 and 3.5.3.2 we can see that if we know the radial pressure at the inner and outer surface of each of our rings, then we can calculate the hoop stress at the inner and outer surface of each of our rings. If we know both radial and hoop stress for both surfaces, then we can calculate the radial mechanical strain of each ring at both inner and outer surfaces.

But as well as mechanical strain, the rings of our disc also experience thermal strain. To calculate the thermal strain of each ring we first need to calculate the temperature of each ring. Using the streams method we represent components such as discs as a collection of elemental subvolumes (c.f. figure 3.3.1.3). By suitably allocating specific subvolumes to specific disc rings, we can find a single representative temperature for each ring by calculating the mass averaged temperature of the subvolumes associated with each ring. Once we have a single representative temperature for each ring we can then calculate the thermal strain at its inner and outer surfaces by using equation 3.5.3.3.

$$\Delta R = R \cdot \alpha \cdot (T_b - T_d) \quad (3.5.3.3)$$

R	Radius
α	Thermal expansivity coefficient of the ring material at T_r
T_r	Temperature of the ring
T_d	Temperature of the ring under datum conditions.

The rings method uses an iterative approach to guess all of the interface pressures. It then calculates inner and outer mechanical strain for each ring, to which it adds thermal strain to find total inner and outer strain for each ring. It then checks to see how big the gaps between the rings are and uses the size and sign of these gaps to reguess values for interface pressures. Convergence is achieved when the gaps between rings are within a predefined tolerance, which in this study was one millionth of the disc rim radius. Interface pressures at the innermost ring's inner surfaces is zero (or it can be set to equal bleed pressure, but the difference it makes is tiny), and interface pressures at the outermost ring's outer surface is calculated from the centrifugal pull of the dovetails and blades.

In this study to shorten the time needed for the rings method to converge a Newton Raphson type method was used and the equations rearranged so that as much of the calculation as possible took place outside of the iterative loop. In addition, interface pressure distributions from each time interval were carried forward as starting guesses for the calculations of the next time interval. Appendix C contains the FORTRAN source code for the rings method.

The author conducted a study to assess the accuracy of the rings method, in which he created a variety of disc models using the rings method, the ANSYS[®] finite element package, and the GTTrans single disc approach. The author ran each model to speeds and boundary conditions not unrepresentative of conditions within a gas turbine, i.e. 1000 rad/s rotational speed, 400 MPa blade pull, 1000 K outer rim temperature, 500 K disc cavity temperature, 5000 W/m²K primary gas path heat transfer coefficient, and 300 W/m²K disc cavity heat transfer coefficient. At the start of each run a uniform 288 K was set for the disc temperature. The material properties used were that of a Nickel alloy, and were looked up during calculation from charts of property versus temperature.

Figure 3.5.3.4 shows transient disc rim growth versus time for three different disc geometries as calculated by ANSYS[®], a GTTrans style single disc approximation method, and the rings method (the number in parentheses denotes the number of rings in the model). Figure 3.5.3.5 shows radial disc temperature distributions when stabilised at the end of the transient, as calculated by the three different methods.

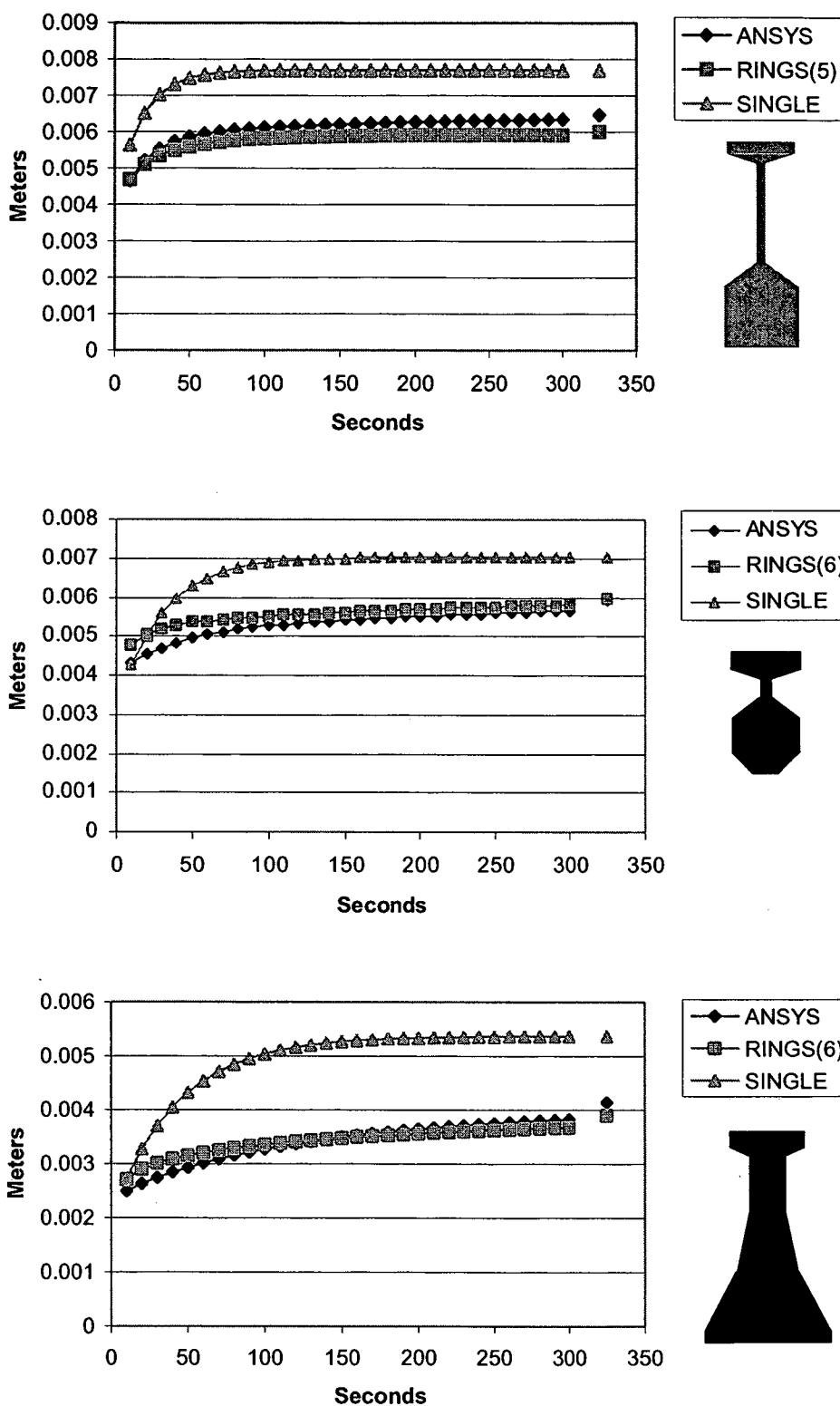


Figure 3.5.3.4 Transient disc rim growth as calculated by ANSYS®, a GTTrans style single disc, and the rings method

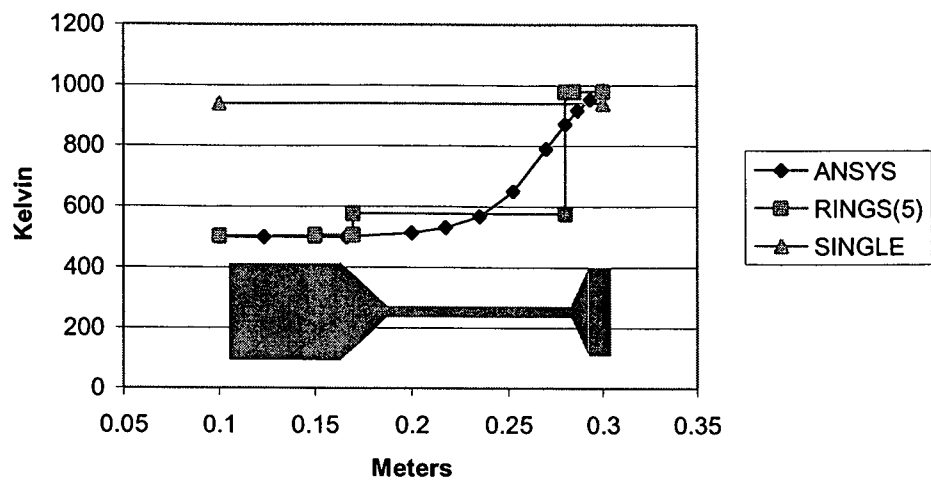
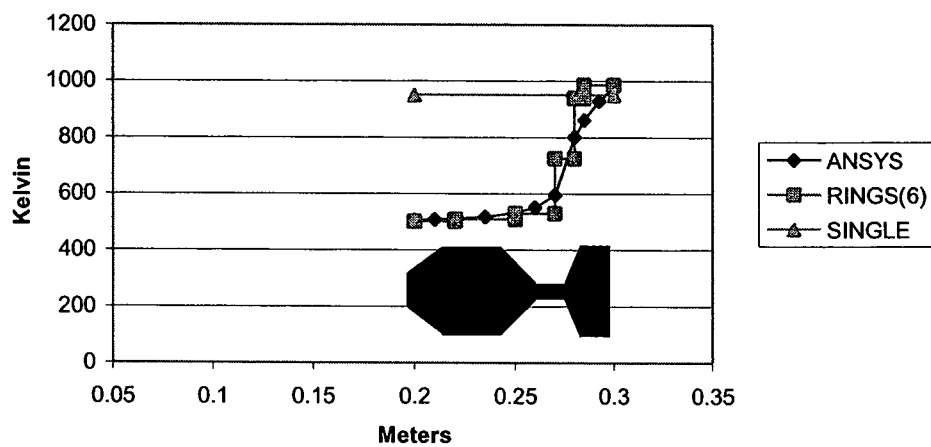
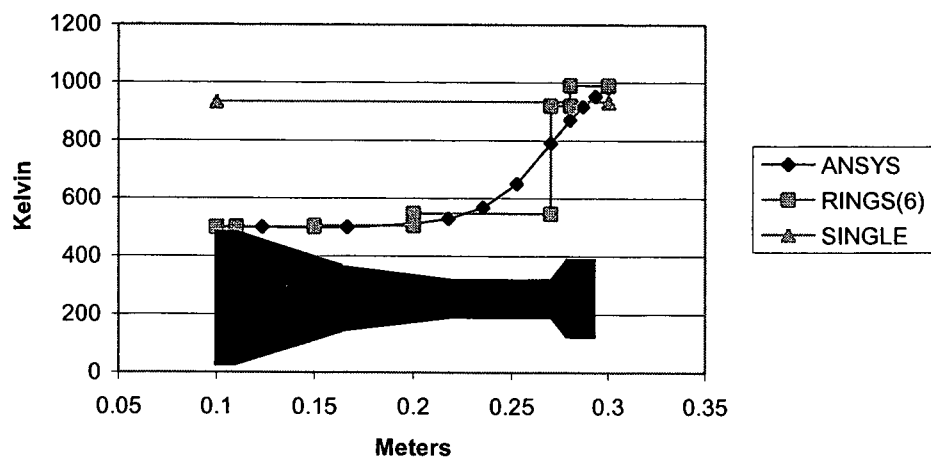


Figure 3.5.3.5 Stabilised radial temperature distributions as calculated by ANSYS®, a GTTrans style single disc, and the rings method

We can see from figures 3.5.3.4 and 3.5.3.5 that the rings method gives answers that are much closer to the ANSYS® results than the GTTrans style single disc approximation for both transient disc rim growth predictions, and stabilised radial temperature distributions. Clearly there are still differences between the rings method predictions and ANSYS® predictions. The rings method is only a one-dimensional thermomechanical model and is very coarsely discretised. However, for only a little more computational burden, the rings method gives us considerably greater accuracy than a GTTrans style single disc approach. It is also considerably faster than a full finite element analysis approach.

Before moving on there is the phenomenon of dovetail slip to address. Dovetail slip occurs when the rim radius of a disc increases as it is spun up to working speed. When rim radius increases the pitch spacing of the dovetails mounted on the disc rim also increases, and when this occurs blade dovetails slip outward. If we assume that each dovetail is a rigid body of fixed dimensions then by similar triangles the equation derived by the author for dovetail slip is

$$\delta = (\sin(720/n) \cdot \Delta R) / \tan(\theta) \quad (3.5.3.4)$$

δ	Dovetail slippage
$\sin()$	Sine function using degrees
n	Number of blades on the disc
R	Radius of disc rim, excluding dovetails
θ	Dovetail angle (from a tangent to the disc) in degrees

3.6 INTEGRATION OF STREAMS INTO RRAP

The methods outlined above were incorporated into a set of FORTRAN subroutines so that they could be easily integrated with the FORTRAN source code for the BD31 version of RRAP. These subroutines are hereafter referred to as the streams method subroutines. As well as the streams method subroutines, streams method input files for each of the components to be modelled had to be created. Input files for the Trent 500 intermediate pressure compressor (IPC), high pressure compressor (HPC), high pressure turbine (HPT), intermediate pressure turbine (IPT), and low pressure turbine (LPT) were created. No input file for the combustor was created for reasons outlined in section 3.4, and no input file for the fan was created because the fan sees much smaller temperature excursions during operation (and thus less heat soakage) than the other components, and fan blades are too complex to be modelled as simple prismatic members. Input files were created using Microsoft Excel spreadsheets that contained: subvolume labels and geometries; material specifications; blade, casing, and disc subvolume allocations; the order in which subvolumes were called to interact with gas flows in convection calculations; etc. Subvolume geometries for each component of each piece of turbomachinery were scaled from a general arrangement drawing of the Trent 500. Tables of material properties versus temperature were provided by Rolls-Royce. Intelligent estimates were made where other information was unavailable.

The creation of input file spreadsheets and subroutines to take input file data and prepare it for use by the streams method subroutines was a task of considerable size and complexity, and so too was the process of integrating the streams method subroutines into the RRAP source code.

The source code of the BD31 version of RRAP is written in FORTRAN 66. The architecture of the programme makes extensive use of common blocks and so is quite open, which made integrating the streams subroutines into it a little easier. During operation the programme calls what is known as the engine routine, which is a subroutine written for a specific type of engine, e.g. Trent 500. The engine routine calls a series of other subroutines as it performs steady state and transient aero-thermal performance calculations. The two subroutines of particular importance to this project are BR081 and BR064 (BR083, see page 27, is not used). BR081 is a subroutine that updates the temperature and pressure at the outlet of a component during transients to account for the effect of primary gas path heat soakage. BR064 is a subroutine that comes in two parts, A and B. Part A deals with the extraction of bleed air from the primary gas path, and part B deals with the return of bleed air to the primary gas path. BR064 is extremely numerical and complicated and can handle up to 80 bleeds from up to 40 different bleed points, it can also handle 8 different types of bleed (e.g. bleed as a fixed percentage of flow, bleed defined by a schedule, switching bleeds, etc.). BR064 has the ability to read bleed temperatures and pressures from a chart, or to call user defined subroutines that calculate bleed temperatures or pressures. During the process of integrating the streams method subroutines into RRAP, the author made changes to the engine routine, BR081, and BR064. The engine routine was changed so that instead of calling the generic BR081 for each component, calls were made to ipcBR081, hpcBR081, hptBR081, iptBR081, and lptBR081, written by the author for each specific piece of turbomachinery individually. Within each of the new BR081 subroutines engine data relating to primary and secondary gas paths were fed into streams method subroutines for each stream assembly (see section 3.2.2), which returned convective heat flows and temperature updates for each stream assembly. Stream assembly heat flows within each specific component were summed and then used to update the component's delivery temperature and pressure using the equations for non-adiabatic compression and expansion. Changes in non-dimensional blade tip clearance were also calculated, and these tip deltas were used to calculate changes in the efficiency and flow for the component in question using 1% for 1% exchange rates. The facility within BR064 for the user to create user-defined subroutines to set bleed temperatures was used to good effect. The author created a subroutine and included it in the engine code in which the outlet temperature of each secondary gas path stream assembly was accessed and used to update the temperature of the bleed to which it corresponded.

The process of integrating the streams subroutines into the BD31 version of RRAP, and performing the studies presented in chapter 4 was not without its problems. One of the problems resulted from the fact that the calculation of non-dimensional transient tip deltas requires both an instantaneous tip closure value, and a corresponding steady state tip closure value. The Lewis method uses the component's instantaneous station vector data (i.e. inlet and outlet temperatures, pressures, speeds, etc.) to calculate both the instantaneous and the equivalent steady state blade tip closures for each stage. From this position we should be able simply to subtract one from the other to get a transient tip delta for the component at any given time interval.

Unfortunately, the Lewis method approach gives steady state tip closures that change with time, i.e. are not steady state. This is because even at the end of the speed transient, engine temperatures, pressures, and mass flows (i.e. station vector data) still change with time, due to heat soakage, bleed temperature delays, and changing tip clearances. An alternative approach would be to first run the whole engine model to a number of steady state points to give a set of fully stabilised tip clearances that correspond to particular engine handle settings, and that do not change with time. Then engine model would be run again but this time through the transient, at each time interval calculating instantaneous tip closure and comparing it to the steady state tip closure at the corresponding handle setting, the resulting tip deltas could then be applied to component efficiency and flow. This latter approach is the one chosen by the author. Fuel flow was chosen as the engine handle because this is the only transient handle available to the BD31 of RRAP as delivered to the author.

Another problem was this. In steady state mode RRAP switches off BR081 because it assumes that no primary gas path heat flow takes place under steady state conditions. The steady state matching procedure involves iteratively guessing various engine parameters, putting them through the engine model and then recording the values of an equal number of checking errors. This information is then used to construct a matrix of partial derivatives that is used to make a whole new set of guesses. The process continues until whole engine convergence is reached. RRAP with the streams method incorporated into it calculates non-zero steady state heat flows. Unfortunately this disturbs the steady state engine matching procedure, changing the goal posts at each call of the engine routine and preventing steady state matching of the engine model. The consequence is that streams method heat soakage must be turned off when RRAP is in steady state mode. This means that a steady state run gives different engine performance to that of a stabilised transient run at the same fuel flow, and steady state tip closures are never the same as stabilised transient tip closures. This leads to non-zero residual steady tip deltas that affect steady engine performance. To get around this problem, the author only ran the engine model in transient mode. In this way he was able to have all of the heat soakage models turned on all of the time. To find steady state tip closures, instead of running a set of steady state points, the author ran a transient which involved incrementing fuel flow through a series of steps and waiting a while for the model to stabilise at each step before reading tip closure and moving on with the next fuel increment.

Another problem was that bleed temperatures are updated in BR064, and BR064 is called in sequence as it occurs in the engine routine of the BD31 source code. Because of bleed system architecture this meant that bleed temperatures that are calculated at one time interval might not be used until the next time interval. Fortunately in the Trent 500 this only affected the IPC re-circulation bleed (see appendix A).

One more problem was that of RRAP convergence in transient mode. RRAP uses the inter-component volumes method for transient calculations. When integrating the streams method subroutines into RRAP it was discovered that in order to prevent programme failures RRAP's inbuilt transient convergence criteria had to be relaxed a little. As will be seen in chapter 4, this has sometimes lead to noisy output.

3.7 CHAPTER CONCLUSION

The purpose of this chapter was to present the heat soakage and tip clearance methods developed by the author in an attempt to meet the project objectives, and describe the incorporation of these methods into a whole engine transient performance code.

Section 3.2 of this chapter describes how basic convective heat transfer theory has been used by the author to develop the streams method. Using the streams method, non-trivial surfaces are segmented and flow is passed over each segment in series during each time interval. The fluid exchanges energy with the first surface via convective heat transfer and then its temperature is updated. This fluid with its new temperature then exchanges thermal energy with the next surface, and so on until heat transfer over the whole surface has been calculated. The temperature of flow leaving parallel surfaces can be estimated from the mass weighted average of the temperatures of the parallel flow. The only way in which the streams method can be used to calculate steady state convective heat transfer distributions is by running a model with fixed boundary conditions until the whole system stabilises.

Section 3.3 of this chapter describes the finite volumes approach used by the author to account for conductive heat transfer within models of the solid parts of an engine component. The method performs one pass of all subvolumes using a five element mask to calculate heat flows between neighbours, and then performs another pass to update the temperature of each volume.

Section 3.4 of this chapter argues that radiation is very much a combustion chamber issue. It also describes why a streams method model of the Trent 500 combustor was not created, i.e. three independent references reported that the effect of heat soakage in the combustor has only a very small effect on the transient performance of an aero gas turbine engine.

Section 3.5 of this chapter describes the thermomechanical growth algorithms developed by the author and used in chapter 4 of this thesis. It describes the simple treatment given to turbomachinery casing and blades, and the slightly more sophisticated rings method used to calculate the transient radial thermomechanical growth of discs. This section also shows how results of rings method calculations are much closer to results produced by full finite element disc models created in ANSYS® than are those produced by the GTTrans method.

Section 3.6 of this chapter describes the problems encountered by the author during the process of integration of the streams method heat soakage and tip clearance subroutines into the FORTRAN source code of the BD31 version of the sponsor's whole engine transient performance code RRAP.

Studies carried out using the updated version of BD31 are described in the next chapter.

Chapter 4: RESULTS AND DISCUSSION

4.1 CHAPTER INTRODUCTION

The main purpose of this chapter is to establish the extent to which the heat soakage and tip clearance methods developed by the author meet the project requirements laid down by the sponsor. Other relevant studies are also included. This chapter is very long.

Section 4.2 of this chapter describes a study in which the new heat soakage and tip clearance methods developed by the author were used to construct a multistage model of a Rolls-Royce Trent 500 high-pressure compressor. In this section the transient heat soakage and tip clearance change predictions produced by this multistage model were compared to those produced by a more authoritative multistage method, i.e. the Lewis method (c.f. chapter 2).

Section 4.3 of this chapter describes a study in which the new heat soakage and tip clearance methods were used to construct a single averaged stage model of the Rolls-Royce Trent 500 high-pressure compressor. The single averaged stage approach is an approach commonly used by previous researchers (c.f. chapter 2). In this section the predictions produced by this single averaged stage model were compared to the averages of the individual stage predictions produced by the multistage model, and also Lewis method model.

Section 4.4 of this chapter describes a study in which the new methods were used to construct multistage models of the other major pieces of turbomachinery in the Trent 500 engine, with the exception of the fan, which of all engine components sees the least variation in transient temperature. The models and methods were then incorporated into the Trent 500 transient performance code provided by the sponsor (c.f. chapter 3). This section presents a comparison of transient performance predictions produced by RRAP as received by the author, RRAP with the streams method and models incorporated into it, and a set of cold stab acceleration test data provided by the sponsor. Also, the effects on whole engine performance of switching on and off heat soakage phenomena models individually and in combination are presented in this section.

Section 4.5 of this chapter describes the possibility of extracting steady state engine performance information from transient test data, in an attempt to realise order of magnitude test bed time and cost savings. The section contains a discussion of the relevant issues, and suggests a possible approach to such a method.

The chapter concludes with a summary of how the various studies help assess the extent to which the heat soakage and tip clearance methods developed by the author have met the project requirements laid down by the sponsor.

4.2 HPC MULTISTAGE MODEL

4.2.1 SECTION INTRODUCTION

The purpose of this section is to demonstrate the extent to which the streams method can be used to predict transient heat flow and tip clearance changes that occur in the Trent 500 high-pressure compressor (HPC).

The test case transient manoeuvres used in the study were a slam acceleration and a slam deceleration, both at sea level and at zero flight Mach number. A streams method model of the Trent 500 HPC was prepared and the data it contained was used by the streams method subroutines. The stream method subroutines were called directly from within the BD31 version of RRAP. The values of heat flow and blade tip clearance changes produced by the streams method subroutines were not fed back into RRAP for this particular study.

The only transient handle available on the user-deck version of BD31 is fuel flow. The slam acceleration and deceleration test case manoeuvres took place between arbitrarily defined 'idle' and 'maximum power'. During the test case acceleration, fuel flow started at 2,000 lbs/hr (idle) and rose parabolically to 22,000 lbs/hr during a period of 5 seconds, after which fuel flow decayed exponentially with a time constant of 5 seconds until it reached a steady state 20,000 lbs/hr (max power). The test case deceleration started at 20,000 lbs/hr and fell parabolically with a soft landing to 2,000 lbs/hr during a period of 5 seconds, after which it remained at a steady 2,000 lbs/hr. The temporary acceleration fuel overshoot was included so as to represent the fuel spike typically occurring during accelerations in engines with a control system schedule to exhaust pressure ratio for example, as it is in the Trent 500.

This section begins with a brief description of the Trent 500 HPC hardware and follows with a description of the streams method model used to approximate it. Heat flow calculations are then presented and compared against those produced by the RRAP heat soakage subroutine BR081. This is followed by a presentation of predictions of transient changes in blade tip clearance and these are compared against those produced by a FORTRAN version of the Lewis method constructed by the author and called from within RRAP. The results of a brief sensitivity study of the streams method are then presented, followed by a summary of streams method set up and run times. The section ends with a short conclusion.

TRENT 500 HPC

The Trent 500 HPC is a six stage compressor (see figure 4.2.1.1). Its discs are made from a variety of materials: stages 1 to 4 are made from two types of titanium alloy, and stages 5 and 6 are made from a nickel alloy. The rotor blades for stages 1 and 2 are made from two types of titanium alloy, and the blades for stages 4 to 6 are made from a nickel alloy. The stator blades for stages 1 to 3 are made from steel, and from a nickel alloy for stages 4 to 6. The casing is made from a variety of titanium alloys and also contains abradable lining materials.

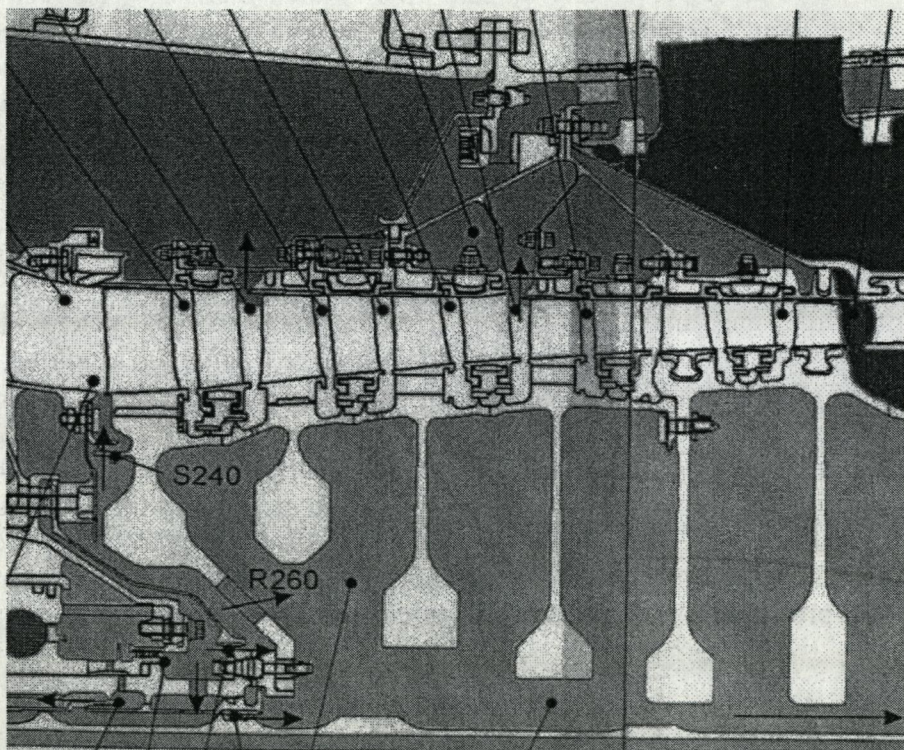


Figure 4.2.1.1 Trent 500 HPC basic geometry and bleed paths.

The disc profiles are different for each stage. The stage 1 diaphragm is much thicker than those of the other stages because it not only restrains its own blades, but also transmits torque to/from the engine's gearbox and axially restrains the HPC drum. The nature of the blade dovetail joints varies throughout the component: stages 1 to 4 are axially oriented and stages 5 and 6 are circumferentially oriented. The casing is a highly complicated structure possessing a number of different joints, cavities, and load bearing features. The near-zero lead angle of the case and the close proximity to a fixed shaft bearing mean that rotor blade tip gap changes occurring due to axial expansions are kept to a minimum.

Air ventilating the disc cavities is bled from the exit plane of the IPC. After leaving the HPC disc cavities this air goes on to ventilate the cob of the HPT disc and also the discs at the front of the LPT. Flow in the primary gas path is often transonic and extremely complicated due to a variety of secondary and leakage flows. Flow in the

casing cavities is also very complicated. The casing cavities contain the variable handling and customer bleeds taken off from behind stages 1 and 3 (hereafter HP1 and HP3 bleed).

STREAMS METHOD REPRESENTATION

Use of the streams method starts with simplification of the geometry and flows associated with the compressor. Figure 4.2.1.2 below shows how the Trent 500 HPC has been represented in the streams method input files used for this study.

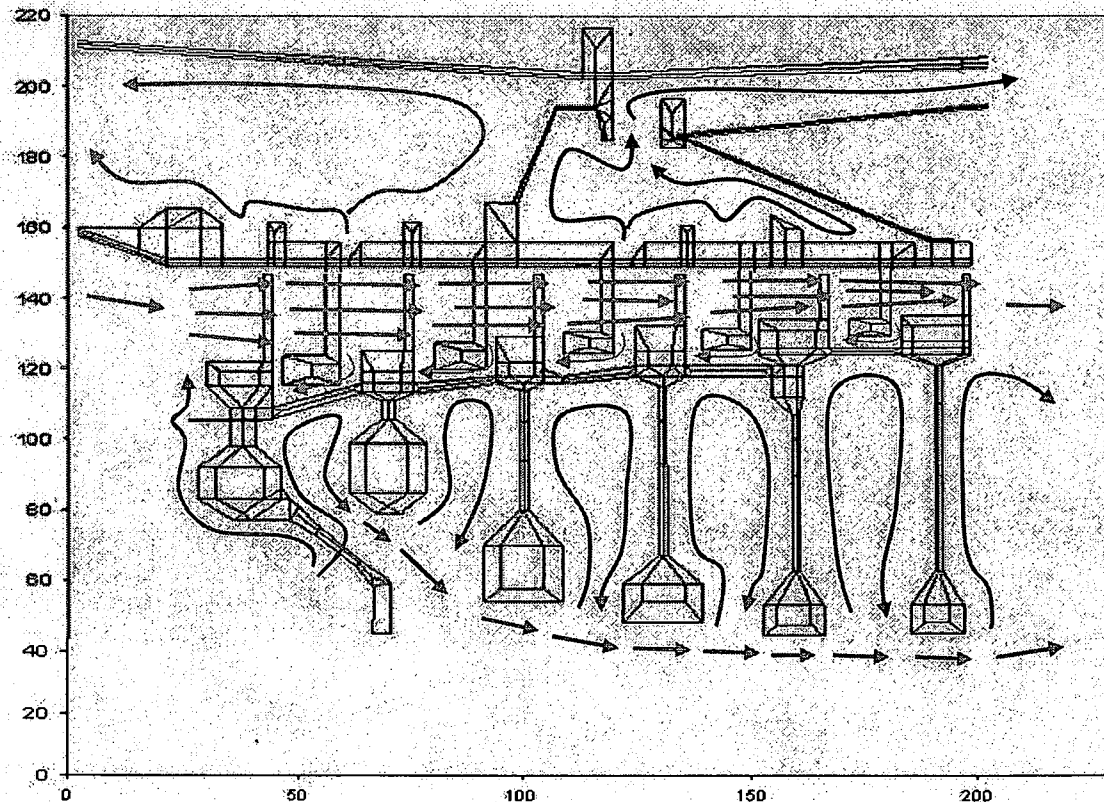


Figure 4.2.1.2 Streams model representation of the Trent 500 HPC.

The internal airflows used in this study have been derived from an early air-systems diagram (the source of figure 4.2.1.1), whereas the hardware geometries have been scaled from a slightly different general arrangement drawing of the engine (notice the different stage 5 & 6 disc profiles). Tables of material properties versus temperature for the various materials used in the HPC were provided by Rolls-Royce. Primary and some secondary gas path mass flows, temperatures, and pressures were taken from the BD31 version of RRAP. Information regarding some other bleed flows was provided by engineers at Rolls-Royce. The exact paths followed by air in the bleed systems is difficult if not impossible to know and the author made intelligent estimates whenever recommendations regarding flow paths were not available.

4.2.2 HEAT FLOW CALCULATIONS

CASING

Figure 4.2.2.1 shows the streams method prediction of convective heat flows into and out of the HP1 and HP3 casing bleed mass flows during acceleration and deceleration.

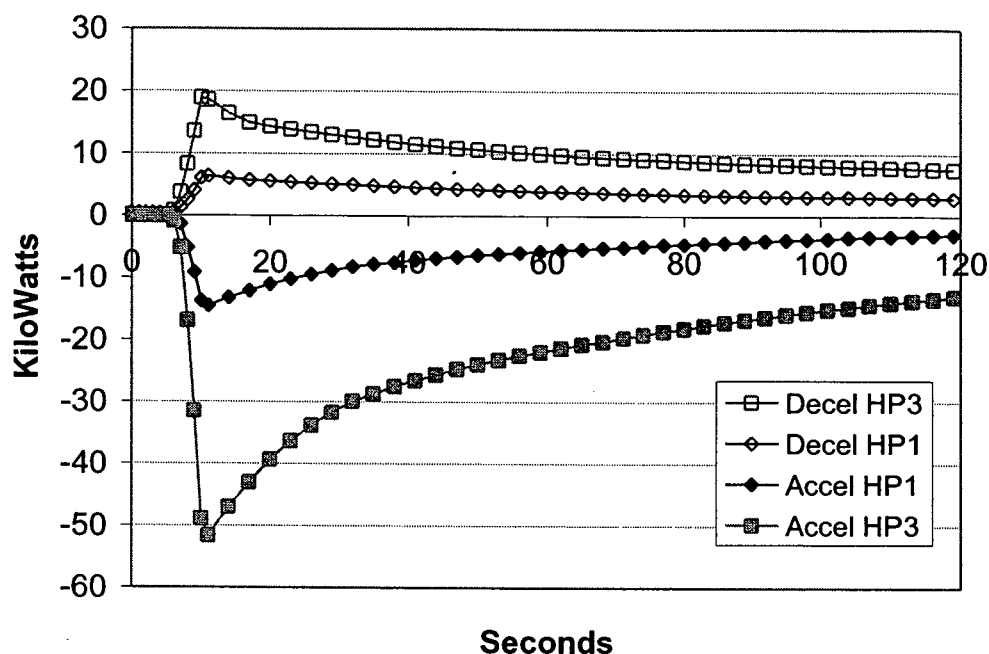


Figure 4.2.2.1 Streams method predictions for convective heat flows into HPC casing bleeds during acceleration and deceleration.

Under steady state conditions in a real engine, a small amount of heat is likely to flow steadily out of the primary gas path and into the casing bleed cavities. During transient acceleration manoeuvres however, heat flow is likely to change direction and flow out of the bleed cavities, because at higher engine settings bleed inlet temperatures are temporarily hotter than the casing cavity metal. Following the transient, when thermal stabilisation has been reached, heat again flows out of the primary gas path through the casing metal and into the bleeds. Figure 4.2.2.1 shows that the streams method model of the Trent 500 HPC behaved just as described. The peak total heat flow out of bleeds during the acceleration was approximately 65 kW, which was two orders of magnitude greater than the steady state heat flows experienced. During the deceleration the peak total heat flow into the bleeds was about 25 kW. During the acceleration, the exit temperature of the HP3 bleed troughed at 45 K below its bleed point temperature, and during the deceleration it peaked at 56 K above its bleed point temperature. A larger bleed temperature excursion was demonstrated during the deceleration manoeuvre even though deceleration heat flows were actually lower than acceleration heat flows. This is because mass flow rates

post-deceleration are lower than they are post-acceleration, and even small heat flows into very small mass flows can produce large temperature excursions.

Typical steady state values of heat transfer coefficient calculated for the forward bleed cavity were about $20 \text{ W/m}^2\text{K}$ at idle and $40 \text{ W/m}^2\text{K}$ at maximum, although peaks of up to $65 \text{ W/m}^2\text{K}$ occurred immediately after the acceleration and deceleration. These peaks occurred because the heat transfer correlation used in the forward casing cavity was that for natural convection on a vertical plate. In natural convection buoyancy plays a significant role and the peaks occurred when the difference between gas and metal temperatures were at their greatest, i.e. when the buoyancy forces were likely to be greatest. The forward bleed cavity is the source of customer bleed air and during engine testing this can be blanked off. For this study it was assumed that the forward bleed cavity was essentially a dead end and therefore a natural convection heat transfer correlation was selected. Bleed air passing through the rearward casing cavity goes on to cool hot section components. Because of its through flow, the heat transfer coefficient used for the rearward casing cavity was that for forced convection through a duct. Typical values of heat transfer coefficient in the rearward casing cavity were predicted to be about $30 \text{ W/m}^2\text{K}$ at idle and $100 \text{ W/m}^2\text{K}$ at maximum power. These values were sensitive to mass flow rather than buoyancy, and therefore no transient heat transfer coefficient peaks occurred in the rearward casing cavity.

The effective time constants (i.e. time taken for the heat flow to change by 63% of the difference between its peak value and steady state value) for the total casing cavity convective heat flows were 70 sec for the acceleration and 150 sec deceleration.

Figure 4.2.2.2 shows the streams method prediction of total radial conductive heat flow through the casing during the acceleration and deceleration manoeuvres used in this study. The results were calculated by summing the conductive heat flow through the top of the primary gas path wetted layer of casing elements.

The figure suggests that conductive heat flow peaks are higher and shorter lived than those predicted for convective heat flows into the casing bleed (c.f. figure 4.2.2.1). This observation needs to be explained. Local conductive heat flow is a consequence of local temperature gradient, which means that length scale as well as temperature difference is important. In this instance we looked at the conductive heat flow through the top of a thin layer of streams method elements. If the casing in the streams model had more layers of elements, then the instantaneous heat flows we would find would depend very much on which layer of elements we examined – layers closer to the surface would be likely to have much quicker thermal responses than those away from the surface in a particular area of metal. By the same token, convective heat flows are generally calculated on the strength of bulk fluid and solid surface temperatures, furthermore, boundary layers are not addressed directly (they are absorbed into the calculations by empirical heat transfer coefficients). If we were to examine the boundary layers during the transient then we would find that local temperature gradients and heat flows in the boundary layer would be much greater than those suggested by the simple conventional approach using bulk fluid and wall temperature. Our conclusion must be therefore that the peak values and time response of transient conductive heat flows are clearly point location specific. Comparison of point specific conductive heat flows against bulk convective heat flows may or may

not be useful, however for reasons of completeness conductive heat flow predictions have been included in this section.

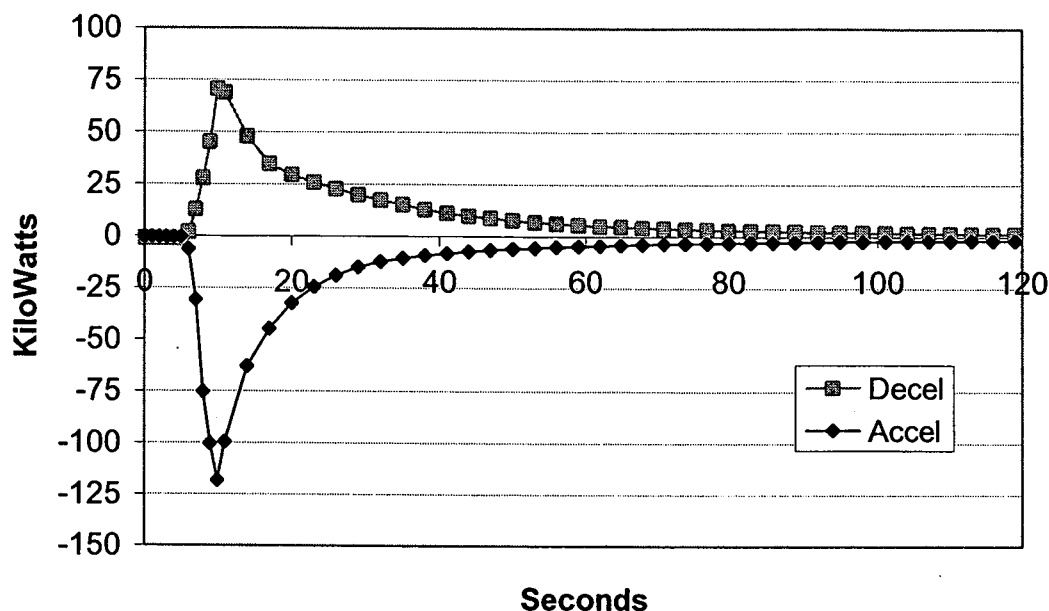


Figure 4.2.2.2 Streams method predictions for radial conductive heat flows through the HPC casing during acceleration and deceleration.

Axial conduction was investigated by interrogating the streams model at a set of locations along the length of the casing. The values were all quite similar to each other, which is to be expected due to the largely linear temperature gradient found in the primary gas path being reflected in the casing metal. Axial conduction areas are relatively very small and therefore most of the time axial heat flows at each location were much smaller than heat flows found elsewhere in the HPC, typically less than ten watts spread over the circumference.

PRIMARY GAS PATH

Figure 4.2.2.3 shows the streams method prediction of convective heat flows into and out of each stage of the primary gas path during the acceleration and deceleration.

Prior to the start of the transient (i.e. steady state) a small amount of heat leaked steadily out of the primary gas path. This was true for all stages except stage 1, where the nett heat flow was in the opposite direction (as a result of bleed air on the outside of the casing being warmer than primary gas path air on the inside of the casing). The amount of steady state heat flow associated with each stage increased toward the rear of the compressor. The reason for this was because temperature differences and heat transfer coefficients rise steeply along the length the primary gas path. This results in heat flows that grow from front to rear inside the compressor. During acceleration

and deceleration manoeuvres the trend still holds because rearward stages experience much larger swings in primary gas path temperature than the forward stages. The total steady state heat flux at idle out of the primary gas path was calculated to be about 10 kW. At maximum power it was approximately 40 kW. During acceleration the peak total heat flow out of the primary gas path was an order of magnitude larger than steady state values at around 500 kW. During deceleration the peak heat flow was in the opposite direction and was around 320 kW, lower than for the acceleration due to falling primary gas path mass flows and heat transfer coefficients that occur during a deceleration manoeuvre.

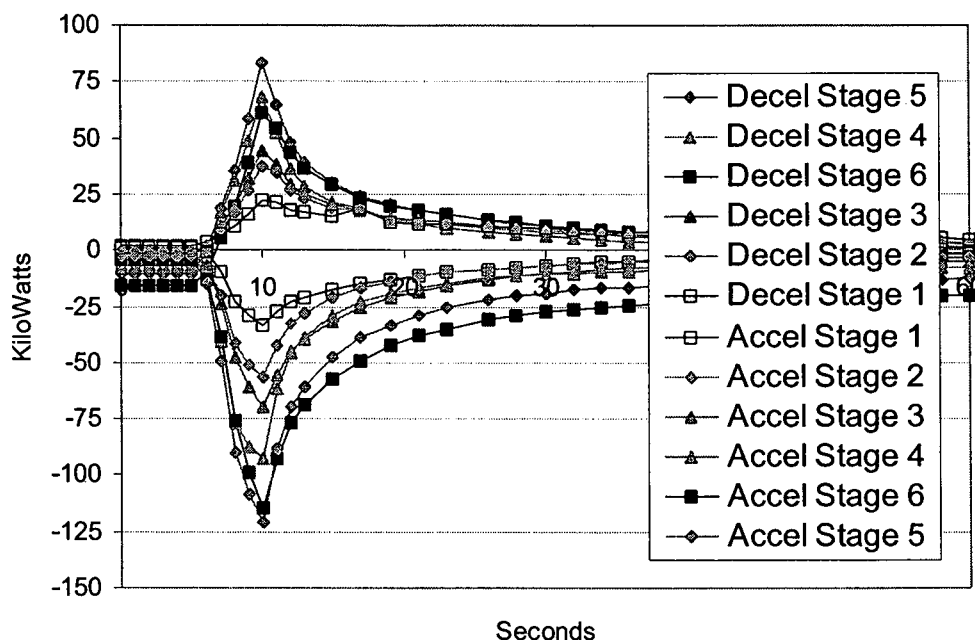


Figure 4.2.2.3 Streams method predictions for convective heat flows to/from HPC primary gas path during acceleration and deceleration (Deceleration graphs have positive peak values).

The heat transfer coefficient correlation used for the primary gas path was that for forced convection in a duct, in which mass flow and flow area are some of the important variables. The flow area at stage 1 is approximately double the flow area at stage 6, and therefore it is not surprising that heat transfer coefficients at the back of the compressor primary gas path are approximately double the value of those found at the front of the compressor. Also, for the manoeuvres used in this study, primary gas path mass flow at maximum was about four times that at idle. It is not surprising therefore that heat transfer coefficients at max power were about three and a half times those at idle. The heat transfer coefficients varied fairly linearly from front to back. At idle the heat transfer coefficients for stage 1 were about 1200 W/m²K, and for stage 6 about 2400 W/m²K. At maximum power for stage 1 they were 4000 W/m²K and for stage 6 about 8300 W/m²K.

The effective time constants (i.e. time taken for the heat flow to change by 63% of the difference between its peak value and steady state value) for the total primary gas path convective heat flows were approximately 5 sec for the acceleration and 7 sec deceleration. Heat flows had largely stabilised by about 3 minutes following the end of the speed transient.

Figure 4.2.2.4 shows the streams method predictions for heat flow into the primary gas path via conduction through the blades and vanes. The figure shows a steady state bias toward heat flowing in through the blades and vanes. This result was a little unexpected because the primary gas path is generally hotter than the bleeds and under steady state conditions heat ought to flow out of the primary gas path with blades and vanes acting like cooling fins. The reason for this unexpected observation lies in the way flow patterns had been prescribed in the streams method model. Inspection of the individual blades and vane heat flows revealed that under steady state conditions the streams model blades conducted heat into the primary gas path, whereas the vanes conducted heat out of it

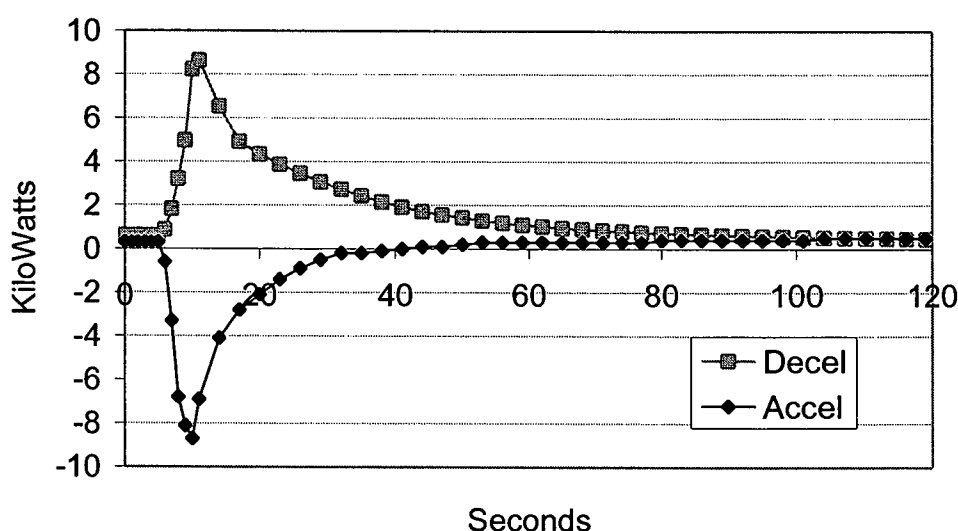


Figure 4.2.2.4 Streams method predictions of heat flow into the HPC primary gas path via conduction through the blades and vanes, during acceleration and deceleration.

In the HPC streams model constructed by the author stage temperature rise was set to occur at the exit plane of each rotor blade. Since temperature rises gradually through the blade passage and not suddenly at exit from the blade passage this meant that the blades themselves were cooler in the streams model than they would have been if a more linear temperature distribution had been assumed. In addition, the stator well leakage flows wetting the rear side of the rotor blade root (disc rim) originated from behind the stator vanes and had temperatures that did include stage temperature rise. This led to the blade root being hotter than the blade itself, and this in turn lead to steady state blade conduction into the primary gas path. The vanes were exposed to gas temperatures that did include stage temperature rise and so the vanes were at more elevated temperatures compared to the blades from the same stage. This resulted in

vane conduction in a direction out of the primary gas path. For this particular version of the streams method HPC model the balance of the sum of steady state blade and vane conductive heat flows happened to result in a nett heat flow into the primary gas path.

Figure 4.2.2.4 also shows that the 9 kW peak blade and vane conductive heat flow out of the primary gas path is small compared to the 500 kW convective heat flow out of the primary gas path.

The effect of heat flow out of the primary gas path is to reduce the compressor exit temperature. At the end of the speed transient, RRAP gave an HPC mass flow of very roughly 100 kg/s. If we assume the specific heat of air to be approximately 1000 J/kgK, then to change the compressor exit temperature by 1 K for one second would take 100 kJ. A heat flow of 500 kW convective heat transfer would equate therefore to about 5 K of temperature difference. The transient changes in exit temperature calculated by the streams method are shown in figure 4.2.2.5. The figure shows a peak temperature difference of about 7 K during acceleration, and 10 K during deceleration, which is not far off our rough calculation.

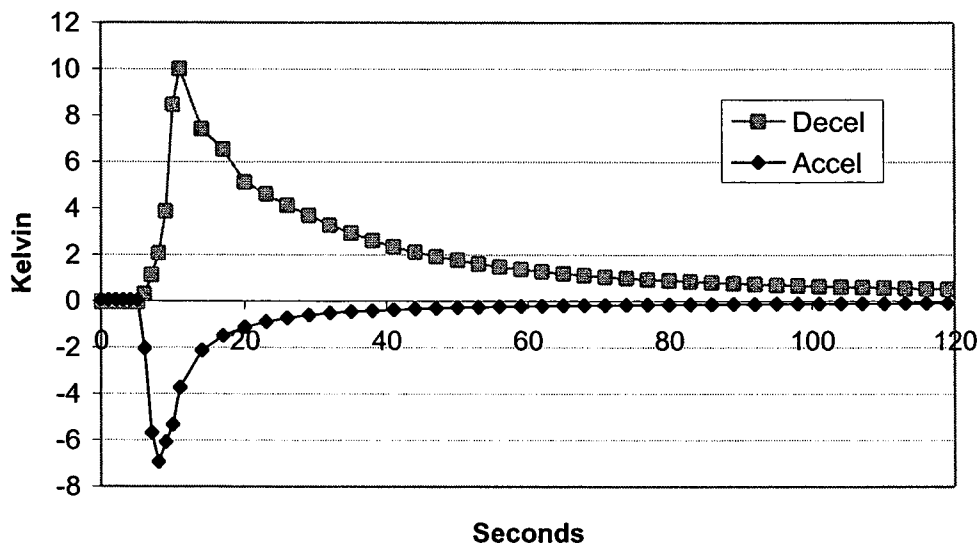


Figure 4.2.2.5 Streams method predictions of transient changes of HPC exit temperature during acceleration and deceleration.

Figure 4.2.2.5 shows that the deceleration compressor exit temperature excursion is larger and longer than it is for the acceleration, even though acceleration heat flows are larger. This is because just after deceleration mass flow has reduced by a larger fraction than that by which heat flow has reduced. This results in a larger compressor exit temperature excursion. In addition, heat transfer coefficients are lower post-deceleration than they are post-acceleration, and this slows the speed of thermal response.

When the effect of windage in the stator wells was included, it shifted compressor exit temperatures upward by 2 or 3 degrees (both transiently and steady state). However, the effect of windage in the stator wells has not been included in the results presented here. Effects such as stator well windage are accounted for in the component maps, and to include it again in the streams method primary gas path calculations would result in it being account for twice.

DISC CAVITIES

Windage has a major effect in the HPC disc cavities, but first let us consider figure 4.2.2.6, which shows streams method predictions for heat flow into the disc cavity bleed without accounting for the effects of windage. The figure shows steady state heat flows of 8 kW at idle and 35 kW at max power, and also a nett bias toward heat flow out of the relatively warm primary gas path and into the relatively cool bleed cavity. The figure also shows peak heat flows of about +50 kW during deceleration and -50 kW during acceleration. Bleed temperature pickup is about 8 K at steady state with peaks of 25 K during acceleration and 45 K during deceleration.

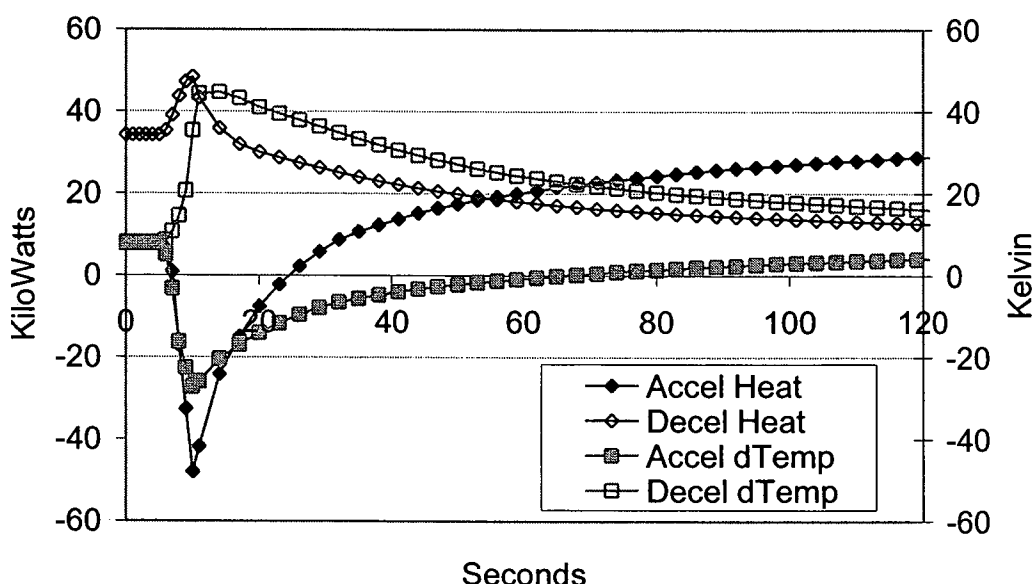


Figure 4.2.2.6

Streams method predictions for heat flows into the HPC disc cavities and bleed temperature pickup during acceleration and deceleration, not accounting for windage.

When the effects of windage are included in the calculations we get results that are significantly different. This can clearly be seen by comparing figure 4.2.2.6 to figure 4.2.2.7, which shows disc heat flows with windage included. Figure 4.2.2.7 shows a steady state heat flow with windage at idle of 30 kW, i.e. twice that predicted without windage, and a heat flow at max power of 200 kW, i.e. over 5 times that predicted without accounting for windage. Steady state temperature pick up is also much higher

when windage is included: 45 K at idle and 70 K at max power. Temperature pickup peaks at 90 K following deceleration, and following acceleration it dips to 22 K but still remains positive (unlike when windage is not included in the calculations). In this study of the Trent 500 HPC disc cavity, the amount of heat generated in the disc cavity by windage was larger than the amount of heat transferred to the disc cavity air by convection.

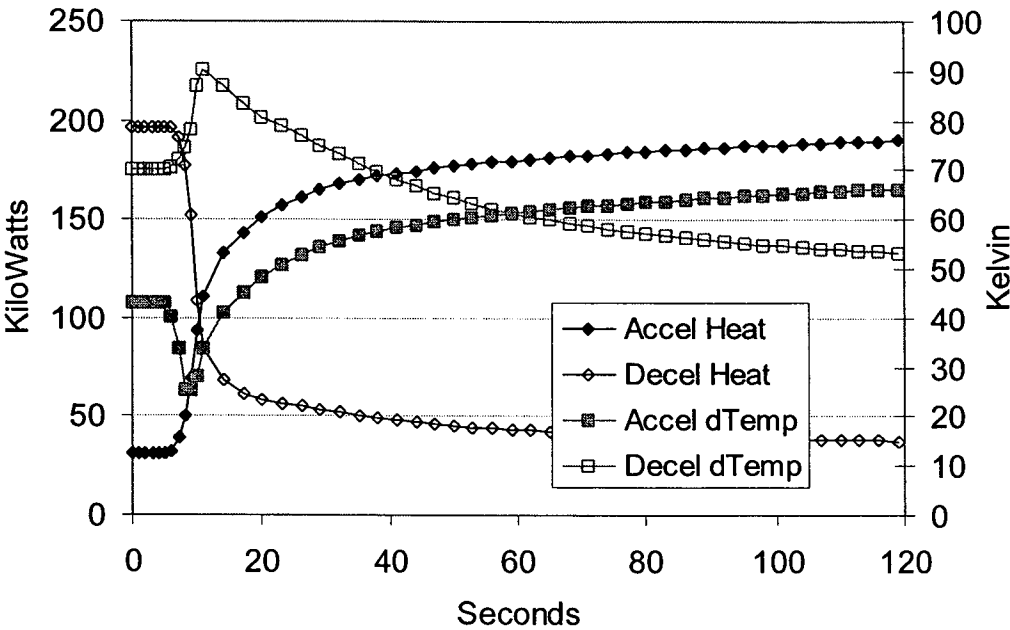


Figure 4.2.2.7 Streams method predictions for heat flows into the HPC disc cavities and bleed temperature pickup during acceleration and deceleration, with windage included.

The heat transfer coefficient correlation used for the disc cavity was that for forced convection over a rotating disc. In this study the heat transfer coefficients were calculated as:

W/m²K	Stage 1		Stage 6	
	Idle	Max	Idle	Max
Drum	635	2200	615	2145
Cob	510	1800	375	1270

Heat transfer coefficients for the drum were higher than at the cobs. This is because the heat transfer coefficient correlation used is positively related to radius (as well as speed and gas properties) and drums are generally at a larger radius than cobs. Heat transfer coefficients for the cobs were higher at stage 1 than at stage 6. The reason for this is that if we consider the geometry of the HPC discs then we can clearly see that the stage 1 cob is at a higher radius than the stage 6 cob. Contrary to this trend is the

observation that heat transfer coefficients for the drum at stage 6 are fractionally lower than for the drum at stage 1 even though they are at slightly larger radius. This is due to the change in gas properties of the bleed as its temperature increases as it flows from front to back inside the compressor.

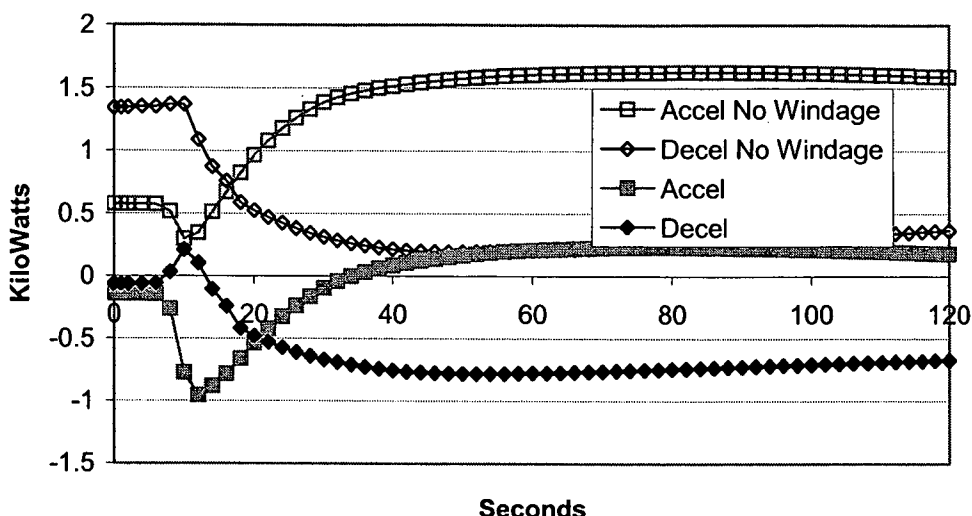


Figure 4.2.2.8 Streams method predictions for conductive heat flows into the top of HPC disc diaphragms during acceleration and deceleration, with and without windage.

Figure 4.2.2.8 shows the streams method prediction for total conductive heat flow into the top of HPC disc diaphragms during acceleration and deceleration, both with and without the inclusion of windage heating. These flows are very small, less than 2 kW.

Without windage, the conductive heat flows are as we might expect, i.e. always positive. The primary gas path is on average warmer than the disc cavity and so we would expect heat to flow via conduction into the top of the disc diaphragms. When we include windage into the streams method calculations the situation is quite different. Windage heating in the disc cavities raises the temperature of the disc bleed so much that its temperature at stages 1 and 2 can be higher than the temperature of the primary gas path, resulting in heat flow out of the top of the disc diaphragms at steady state. Conductive heat flow generally becomes more positive following acceleration and more negative following deceleration for calculations both with and without windage. This is because at max power the absolute temperature differences between primary gas path and disc bleed is higher than at idle.

We can see from figure 4.2.2.8 that conductive heat flow from the primary gas path in through the top of the disc diaphragms is small at less than 2 kW. From figure 4.2.2.6 we can see that the disc cavity total convective heat flow without windage into the disc bleed was 35 kW at maximum power. Interrogation of the streams model output showed that 30 kW was attributed to heat flowing from the drum, the remainder being due to convective heat exchange between the bleed and the discs themselves. With the exception of the disc cobs, most of the metal associated with the disc cavities is

quite thin, i.e. the discs are thin and the drum is thin. The thinness of these structures gives them large convective areas in one direction and small conductive areas in the perpendicular direction, therefore, convective heat flow into the disc cavities via the drum is much larger than radial conductive heat flow in through the top of the diaphragms. Furthermore, on one side of the drum wall is the primary gas path (or rather stator wells) and on the other is the disc bleed air. The result is a large temperature gradient across the drum wall, and relatively large heat flows through it. The disc diaphragms see only a small amount of heat conducted in from the primary gas path and have their large convective faces bathed in bleed air, thus heat flow into the discs is dominated by convection. Bulk temperature gradients experienced by the discs are smaller than those experienced by the drum, and for this reason the drum accounts for the larger part of convective heat flow into the disc cavities and the discs themselves accounts for a much smaller part.

Conduction in the disc assembly is very important. Conduction in through the tops of the diaphragms may be small, but conduction through the drum is not – under steady conditions it is of course equal to the convective heat flow. Biot number can be used as a measure of the importance of conduction. High Biot numbers occur when conduction within the solid is low compared to convection from the solid. Under such conditions conduction is the rate-controlling step, i.e. the bottleneck. So the higher the Biot number, the more care we need to take with conduction. Simple treatment of conduction (i.e. the lumped capacitance method) is acceptable for Biot numbers of up to 0.1. The streams method study of the Trent 500 HPC produced maximum power Biot numbers for the drum and diaphragms that were in the order of 0.2, and for the cob values that were greater than unity. It is clear therefore that conduction plays an important role in the disc assembly, especially in the cobs, the thermal growth of which has a large impact upon radial disc growth and thus blade tip clearances.

COMPARISON TO BR081

Figure 4.2.2.9 shows Trent 500 HPC heat flows calculated using the multistage streams model as compared with BR081 heat flows for the slam acceleration test case used in this study. No comparison can be made against BR081 bleed path heat flows because BR081 only calculates heat flows to and from the primary gas path.

During the slam acceleration used in this study, BR081 calculated the peak heat flows to be approximately 650 kW for the casing and 450 kW for the blades, totalling a flow of about 1100 kW out of the primary gas path. In contrast, we can see that the streams method predicted a peak heat flow of approximately 450 kW for the annulus and just 75 kW for the blades, bringing the total to 525 kW, i.e. just under half the value predicted by BR081. During deceleration the values were smaller all round. For BR081 peak heat flow out of the casing was about 250 kW and for the blades it was about 325 kW. For the streams method, peak heat flow out of the annulus was about 250 kW and for the blades about 70 kW. Again BR081 predicted total peak heat flow (575 kW) of about twice the value predicted by the streams method (320 kW).

BR081 response rates were notably slower following decelerations than they were following accelerations, whereas streams method response rates were quite similar following both types of transient manoeuvre. The effective time constants for BR081 during accelerations were 4 sec for the casing and 2 sec for the blades; and for deceleration they were 13 seconds for the casing and 3 seconds for the blades.

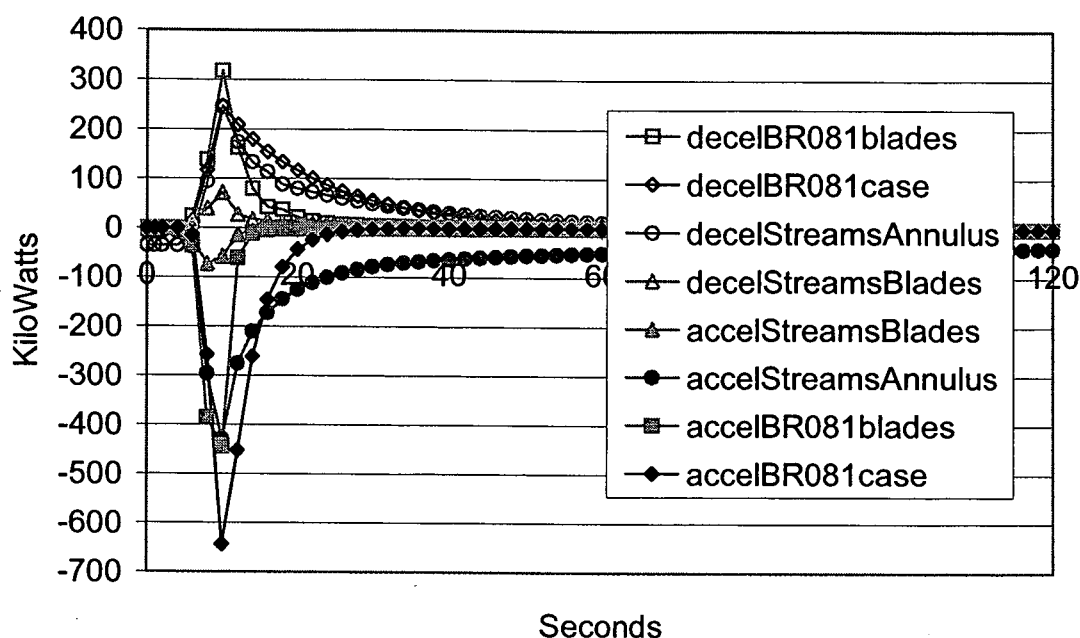


Figure 4.2.2.9 BR081 and streams method primary gas path heat flows.

For the streams method for both transient manoeuvres for annulus they were about 7 sec, and for the blades about 2 sec.

BR081 predicts zero steady state heat flows, whereas the streams method predicts non-zero steady state heat flows. Further processing of the results showed that for BR081 the sum of the integrals of casing and blade heat flow was the same for both acceleration and deceleration manoeuvres, i.e. heat absorbed by the metal equalled heat released by the metal. For the streams method this was not the case because of the different non-zero steady state heat flows at idle and maximum power attained by the system following thermal stabilisation.

For the streams method blade heat flows are small compared to annulus heat flows. They are also small compared to BR081 heat flows for either blade or casing. BR081 blade heat flows are smaller than BR081 case heat flows during accelerations, but larger during decelerations.

When we consider the nature of both the BR081 and the streams methods it is not difficult to see why they behave as described above. BR081 assumes the casing and blades are independent masses with zero thermal resistance and that they are in contact with nothing but the air in the primary gas path. Conduction is not accounted for in BR081 and so it might be expected that BR081 predicted a higher, less

impeded, total peak heat flux than the streams method. The streams method accounts for the fact that some of the compressor metal is in contact with the primary gas path air and some of it with casing and disc bleed air, and that conduction takes place within the metal components, and that materials have non-zero thermal resistances. It would be expected therefore that the streams method gave lower predictions of peak heat flux, slower decay times, and non-zero steady state heat fluxes.

If again we return to the nature of the two methods we can see that the whole of the BR081 casing and blade masses are directly exposed to the average primary gas path temperature, whereas in the streams method, upstream metal either preheats or precools the air before it reaches downstream metal. The gas to metal temperature difference predicted by the streams method is therefore much less than that predicted by BR081. This explanation was supported by an interrogation of blade data in the output files for the two methods, which showed that the peak average gas to blade metal temperature difference in the streams model was about 9 K, whereas for BR081 it was 60 K. This also explains the large difference between the values of blade heat flows predicted by the two methods.

Figure 4.2.2.10 shows acceleration and deceleration HPC exit temperature change due to heat soakage as predicted by BR081 and the streams method.

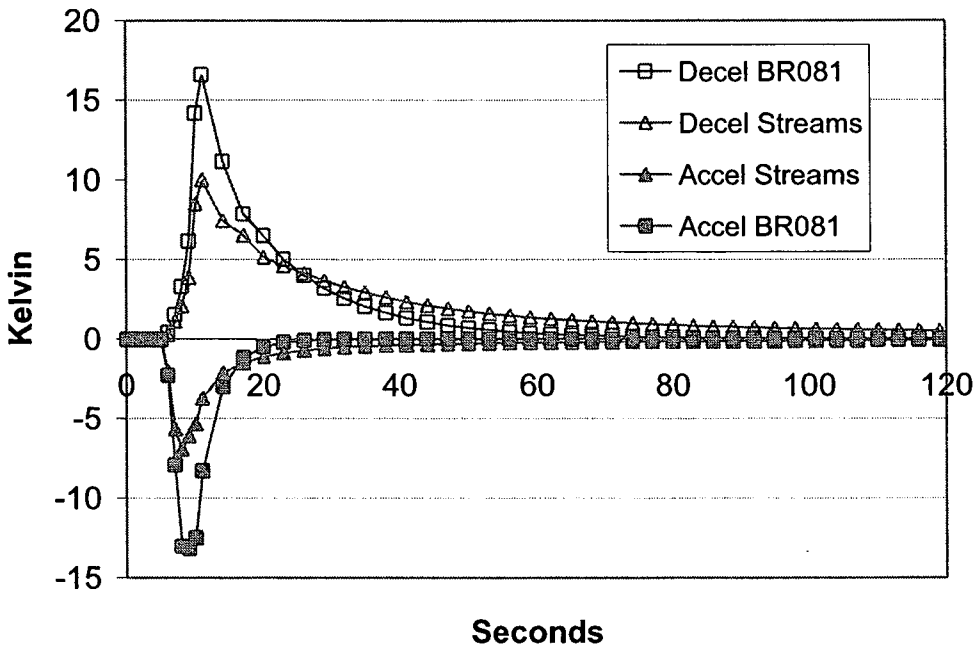


Figure 4.2.2.10 HPC exit temperature change during acceleration and deceleration as predicted by BR081, the streams method.

BR081 predicted an HPC exit temperature dip of down to -13 K during acceleration and a temperature rise of up to +17 K during deceleration. The streams method temperature changed -7 K during acceleration and +10 K during deceleration. This

corresponds pro rata to the ratio of streams method primary gas path heat flows to BR081 primary gas path heat flows.

4.2.3 TIP GAPS

MECHANICAL CASE GROWTH

Figure 4.2.3.1 shows predictions of casing mechanical growth for each stage as calculated by the Lewis method and by the streams method during the slam acceleration test case. Values for the mechanical element of casing growth during deceleration have not been shown because graphs would show essentially the same values as for the acceleration, but only in reverse.

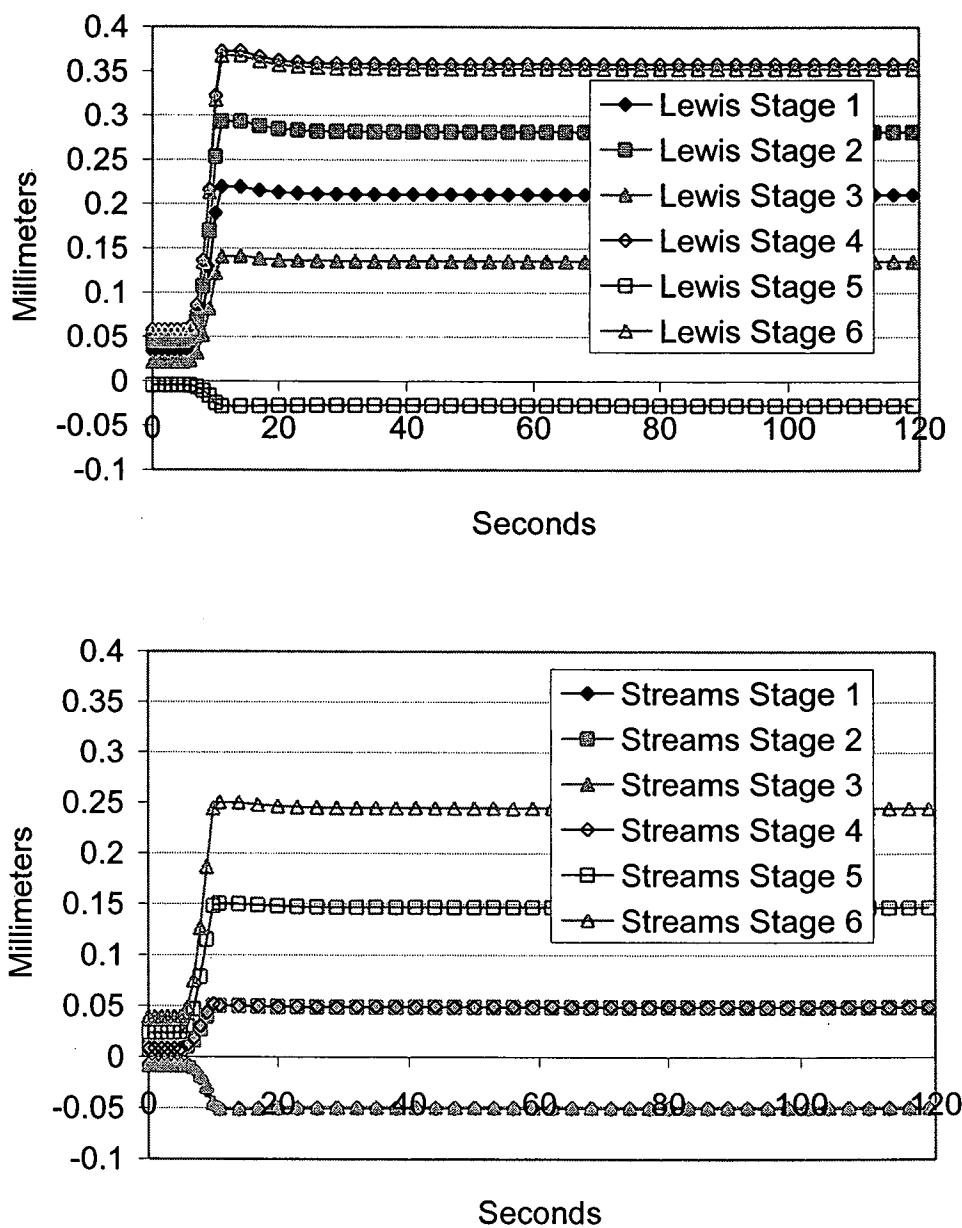


Figure 4.2.3.1 HPC mechanical case growth as calculated using the Lewis method (top) and streams method (bottom).

The streams method calculations gave results that were generally lower than the Lewis method results. The size order of stage growth was also different, and the growth for some stages was even of a different sign. The difference in nature of the two methods can be used to shed light on this situation. The streams method uses the total pressure distributions in the primary gas path and in the case bleed cavities as the basis for its mechanical growth calculations. For stages 1 and 3 due to the location of the bleed ports the pressure in the rotor casing was higher on the outside than the inside at those stages, and this resulted in negative mechanical case growths at those stages. In addition, the internal casing pressure was found by dividing the total compressor pressure rise by the number of stages, and the external casing pressure was assumed to be the same as at the relevant bleed point. The result of this assumption was that the sizes of the streams mechanical growths increase at regular intervals. In contrast the Lewis method gets its fundamental data from the output of the heavyweight SC03 finite element model. The SC03 model is based upon much more accurate hardware geometry and much more sophisticated stress and strain calculations than is the streams method. The Trent 500 HPC casing is structurally very complicated and for mechanical pressure growth static pressure distribution is more important than the total pressure distribution. This is particularly important when handling and customer bleeds are turned on and off. In addition to these factors, features such as case mountings introduce asymmetry into mechanical loadings. Many more of these factors are accounted for in the SC03 model than in the streams method model. It is hardly surprising therefore that there is such difference between the mechanical growths predicted by these two methods.

THERMAL CASE GROWTH

Figure 4.2.3.2 shows predictions of casing thermal growth for each stage as calculated by the Lewis method and streams method during the slam acceleration test case.

We can see from the figure that the thermal growth of the casing is approximately an order of magnitude greater than the mechanical growth of casing, but unlike mechanical growth it takes a significant period of time for the thermal growth of the casing to reach new steady state values. The thermal growths predicted by the streams method are generally of similar magnitude to those predicted by the Lewis method, however, the size order of stage growth is different. In addition, the thermal response times predicted by the streams method are generally shorter than those predicted by the Lewis method, especially at stages 5 and 6.

The growths predicted by streams at stages 1 and 2 appear to be in closest agreement with the Lewis method results. This corresponds to a region in the casing where the geometry is the simplest. The casing geometry for stages 3 to 6 is very complicated, and so are the interactions between the mechanical and thermal stresses. The streams method representation of the casing as a single skin is perhaps an over simplification. In reality the casing is a sophisticated double skin type structure with various enclosed subspaces. Under these circumstances convection is complicated and conduction

from the primary gas path wetted surfaces to the load bearing outer skin is likely to be slower than it would be for a simple single skin casing. These factors have better representation in the SC03 model in the Lewis method than they do in the streams method model, and so temperature distributions in the streams model are likely to be different than those upon which the Lewis model is based.

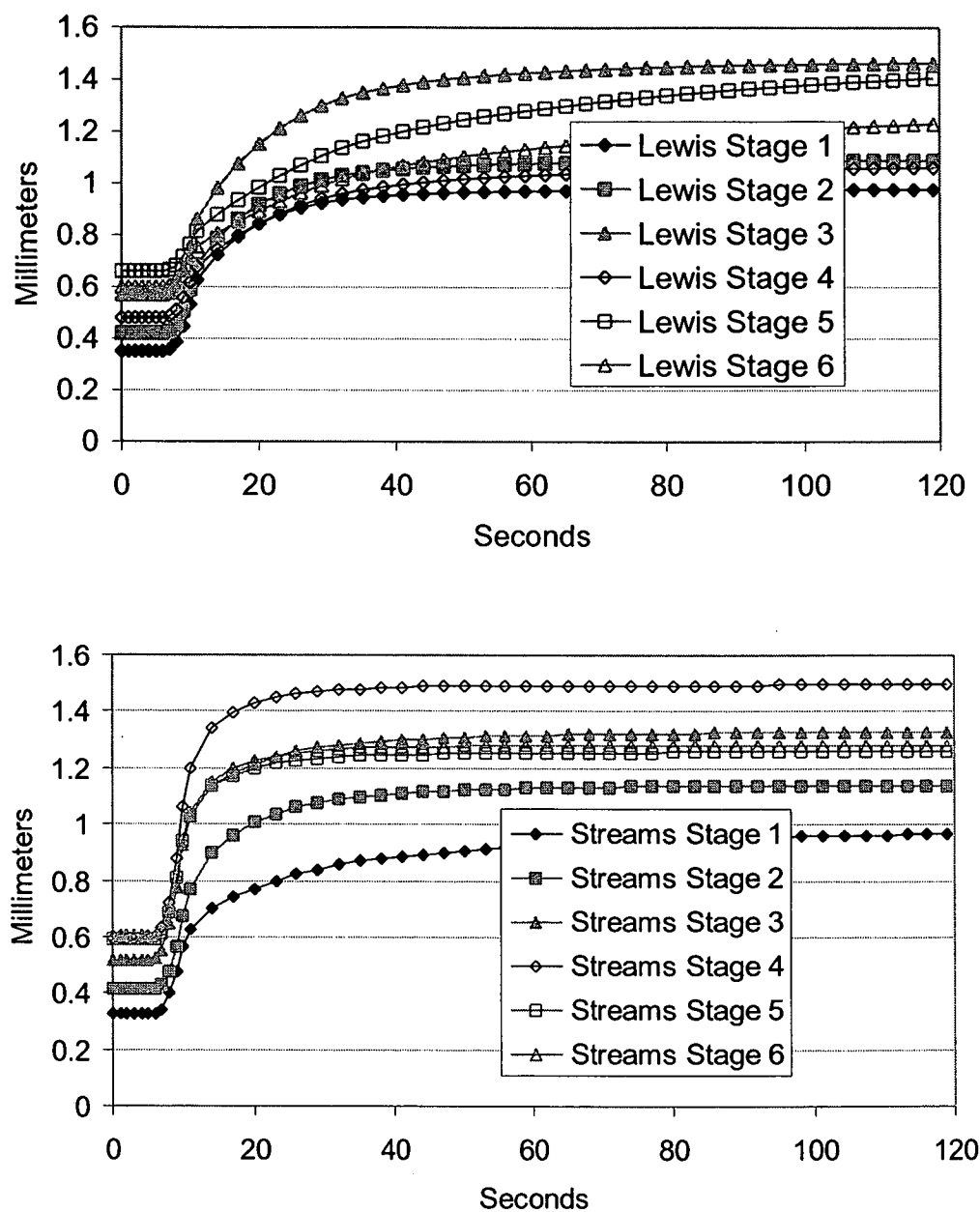


Figure 4.2.3.2 HPC thermal case growth during acceleration. Lewis method (top) versus Streams method (bottom).

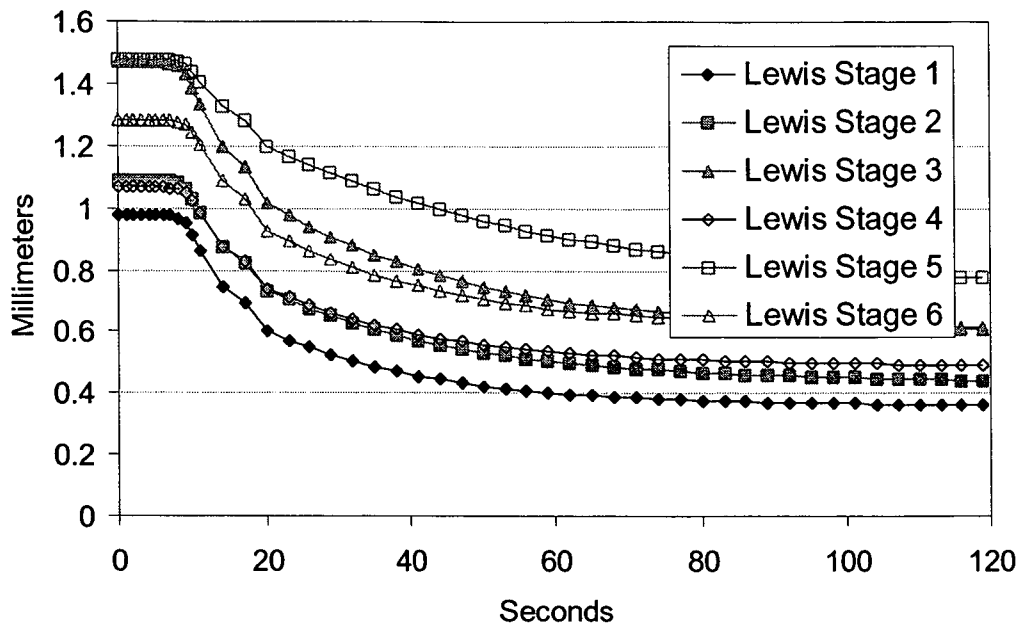
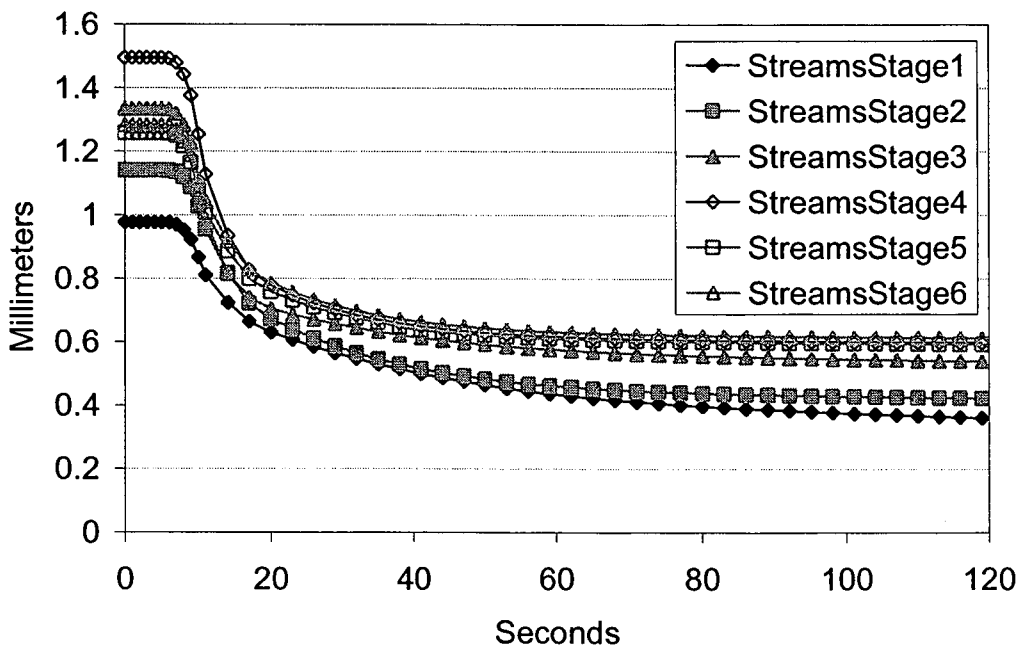


Figure 4.2.3.3 HPC thermal case growth during deceleration. Lewis method (top) versus Streams method (bottom).

Figure 4.2.3.3 shows thermal casing growth predictions during a deceleration manoeuvre. Again the streams prediction for stage 1 is quite close to the Lewis method prediction, however in general the streams method response rates are quicker

than those for the Lewis method and the size order of growths is quite different. The thermal responses for both methods are slower following the deceleration than they are following the acceleration. The reason for this is that heat transfer coefficients and mass flows are lower following a deceleration (c.f. section 4.2.2).

MECHANICAL BLADE GROWTH

Figure 4.2.3.4 shows predictions of rotor blade mechanical growth for each stage as calculated by the streams method during the slam acceleration test case.

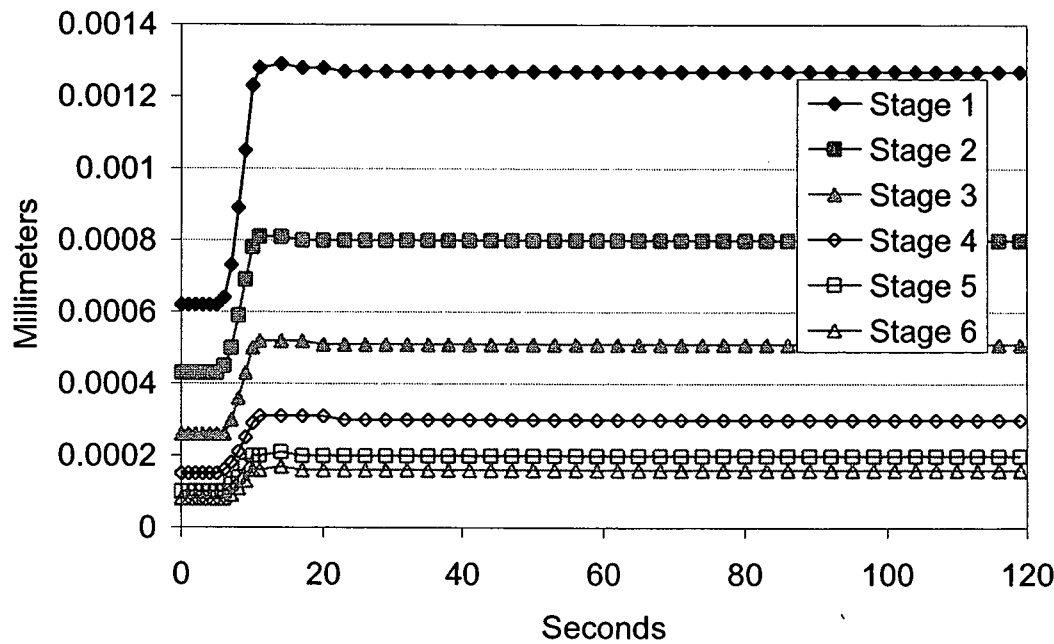


Figure 4.2.3.4 HPC mechanical blade growth – predictions made using the streams method.

The streams method calculates the 1-D radial centrifugal growth of the aerofoils assuming they are prismatic members of constant cross section. As might be expected mechanical blade growth is proportional to the size of the blade, i.e. it is largest for stage 1, and least for stage 6. The centrifugal mechanical growths predicted were of the order of one micron, i.e. three orders of magnitude smaller than the thermal growths experienced by the casing. The Lewis method assumes aerofoil mechanical growths to be negligible and omits them from the calculations. From the scale of the y axis on figure 4.2.3.4 it is not difficult to see why.

THERMAL BLADE GROWTH

Figure 4.2.3.5 shows predictions of rotor blade thermal growth for each stage as calculated by the streams method and Lewis method during the slam acceleration test

case. Streams method blade thermal response times were very quick (much less than one second), and in the Lewis method the thermal response of the blades is assumed to be instantaneous. For this reason only the acceleration graphs are shown, as deceleration values would be the same as acceleration values, only in reverse.

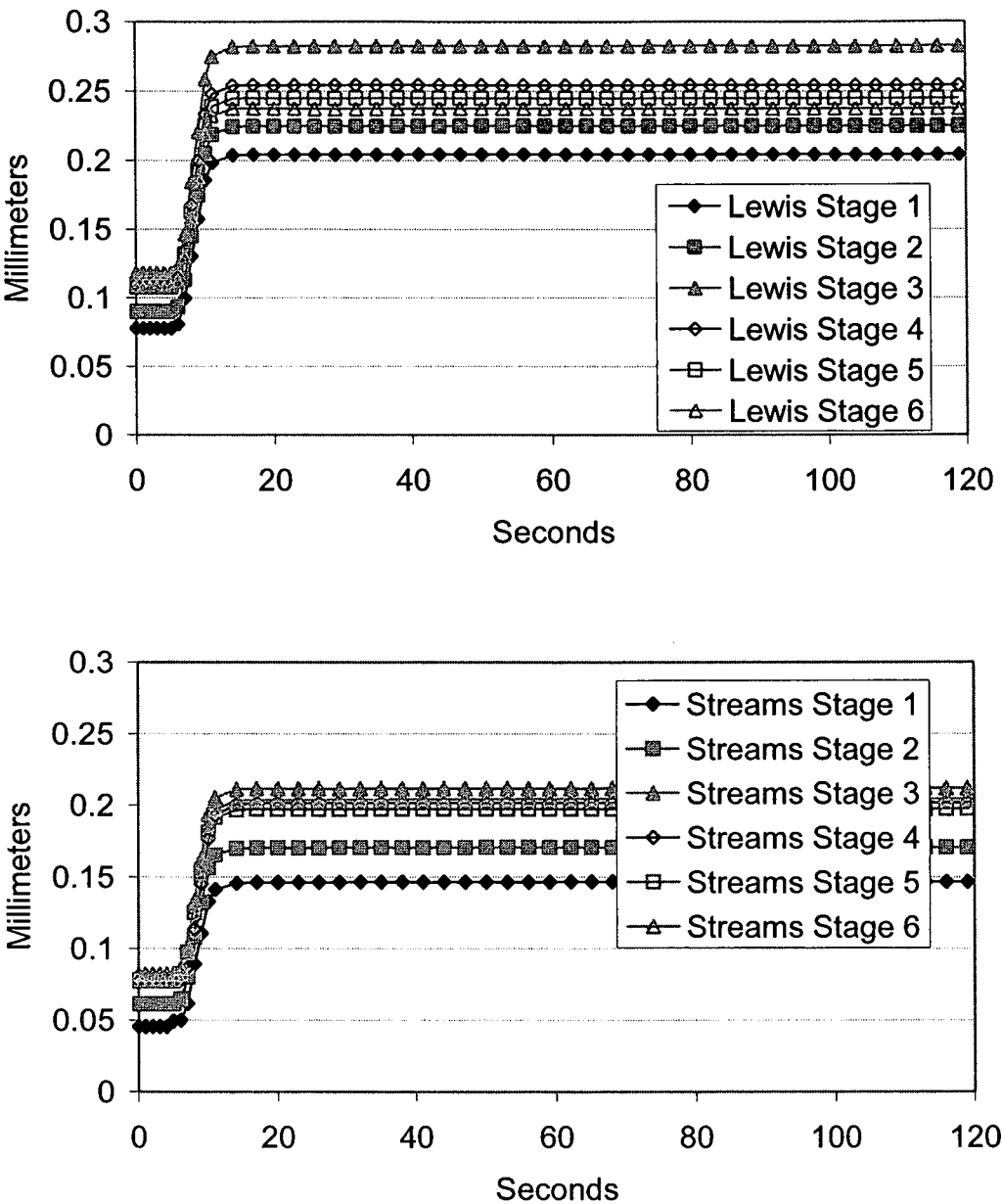


Figure 4.2.3.5 HPC thermal blade growth. Leo Lewis method (top) versus Streams method (bottom).

We can see from the figure that the streams method growths are smaller than Leo Lewis method growths. Upon investigation it was discovered that the Leo Lewis

method predicted blade temperatures between 40K at idle and 70K at max higher than those predicted using the streams method. Upon further investigation it was discovered that the Leo Lewis method input file (and the spreadsheet from which it was derived) contained blade temperature specifications that appear quite high. The temperatures for the blades started at 26 percent of compressor temperature rise at stage 1, and finished with 105 percent at stage 6 - making the stage six blade hotter than the exit plane of the compressor. In contrast, the approach used by the streams method is simply to divide the compressor temperature rise and by the number of stages, then add this value to the primary gas path temperature as the flow leaves each stage. In this way the streams method does not give blade temperatures in excess of the compressor exit temperature, but by the same token does not give blade temperatures that agree with SC03 and Lewis method blade temperatures.

The size order of the growths for each stage appears not to be too dissimilar for both the streams method and the Lewis method. The size of growth increases slightly from stages 1 to 3, and then decreases slightly for the later stages. The reason for this becomes clear if we consider temperature rises and blade lengths. Temperature rises along the compressor and so it would be reasonable to assume that thermal blade growth increases as we move from stage 1 to stage 6. Indeed this is what we can observe in figure 4.2.3.5 for stages 1 to 3. However, as we move from front to rear blade length decreases. And the nett result is that absolute thermal blade growth rises from stage 1 to stage 3 and then falls slightly for the later stages.

THERMOMECHANICAL DISC GROWTH

The combined nature of the streams method thermomechanical disc growth algorithm makes it difficult to separate thermal from mechanical strain for comparison against Lewis methods predictions. To get around this problem, runs were carried out with disc temperatures set to room temperature in the hope that mechanical growth would be calculated without being contaminated by thermal growth. Figures 4.2.3.6a and 4.2.3.6b show the results for the acceleration test case.

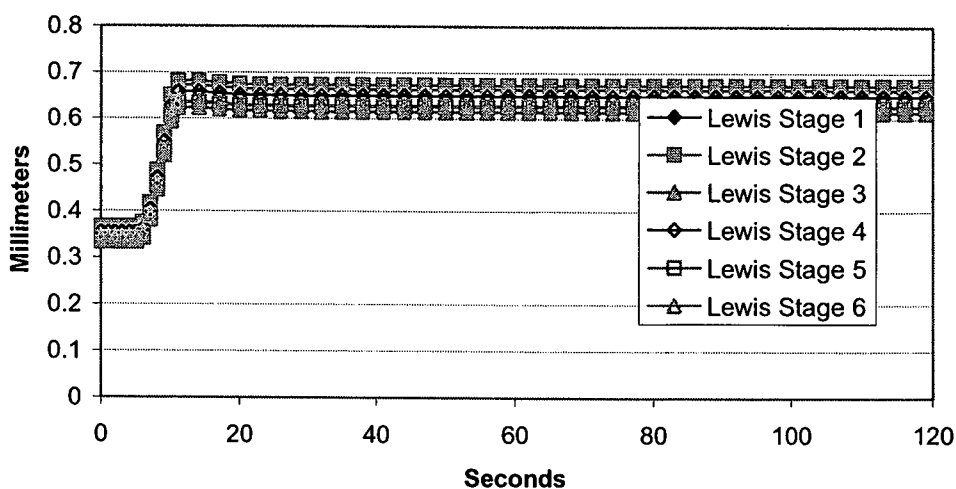


Figure 4.2.3.6a HPC mechanical disc growth during acceleration. (Lewis method)

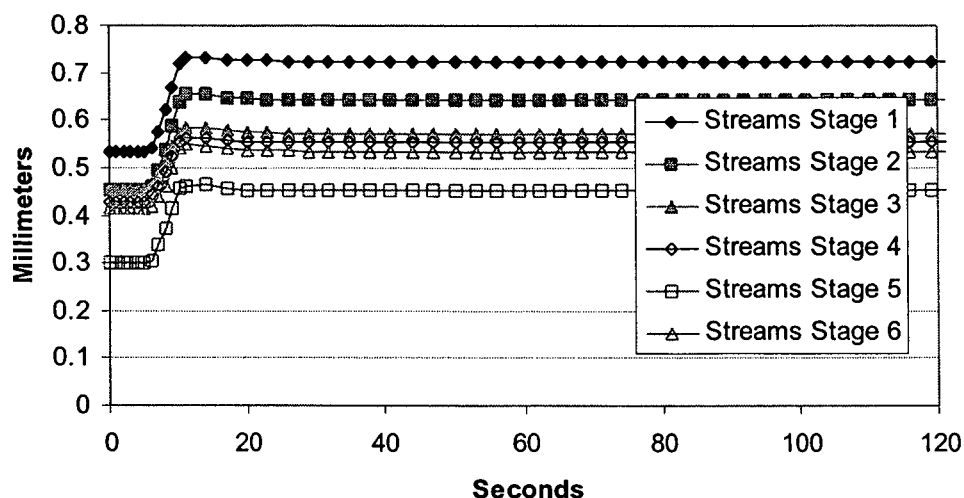


Figure 4.2.3.6b HPC mechanical disc growth during acceleration. (Streams method)

The figures show that a much larger spread in values is predicted for mechanical disc growth by the streams method as compared to the Lewis method, and also that the size order of stage growths are different. Unfortunately, such a straightforward comparison is not really very accurate because thermal and mechanical stresses are often coupled, and the mechanical behaviour of a disc depends on its material properties, which in general vary with temperature. Nonetheless, figure 4.2.3.6 is useful in that it gives us an indication of the size of the mechanical component of disc growth as compared to total thermomechanical disc growth. An example of how strain related material properties vary with temperature can be seen in figure 4.2.3.7, which shows the variation of Young's modulus with temperature for the materials used in HPC discs.

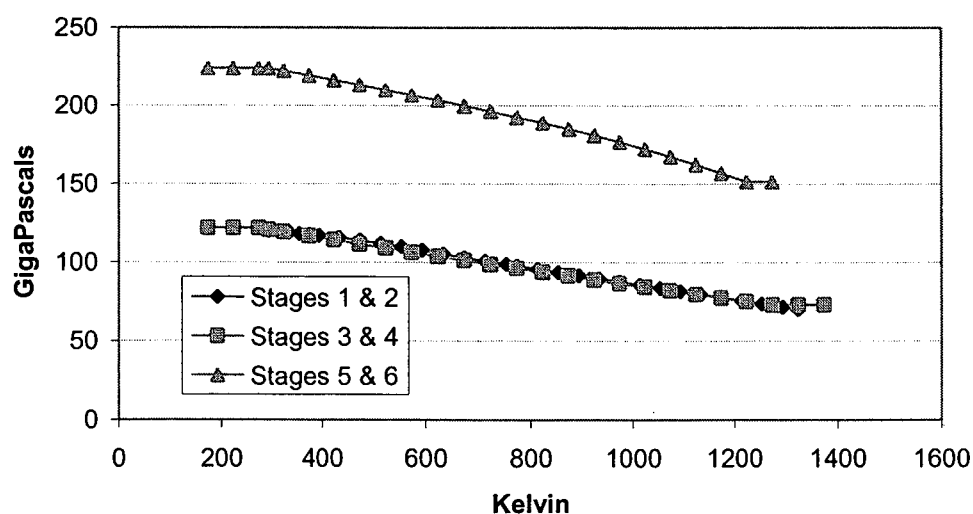


Figure 4.2.3.7 Young's modulus versus temperature for HPC disc materials

Figure 4.2.3.8 shows predictions for combined mechanical and thermal disc rim growth for each stage as calculated by the Lewis method and streams method during the slam acceleration test case. A comparison of figures 4.2.3.1 to 4.2.3.8 shows that streams method predictions are closer to Lewis method predictions for disc rim growth than they are for casing or blade growth in terms of absolute size, speed of response, and also size order of growth for each stage. The likely reason for this is the higher degree of sophistication associated with the thermomechanical growth model of the discs (i.e. the rings method) compared to those used for the casing and blades.

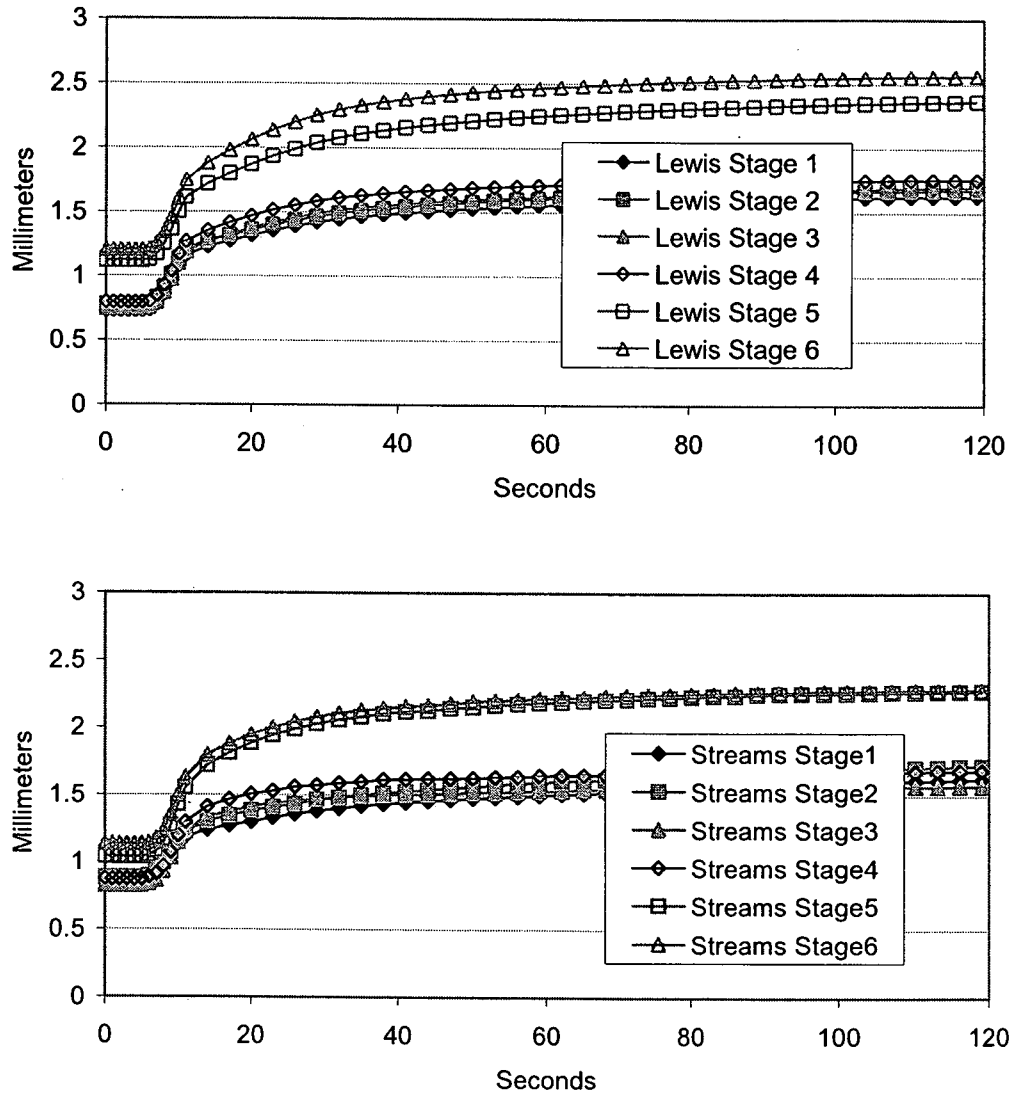


Figure 4.2.3.8 HPC thermo-mechanical rotor growth during acceleration. Lewis method (top), streams method (bottom)

Figure 4.2.3.9 shows Lewis and streams method total disc rim growth predictions during the deceleration test case. For both methods, thermal response times are slightly slower than they are during the acceleration test case.

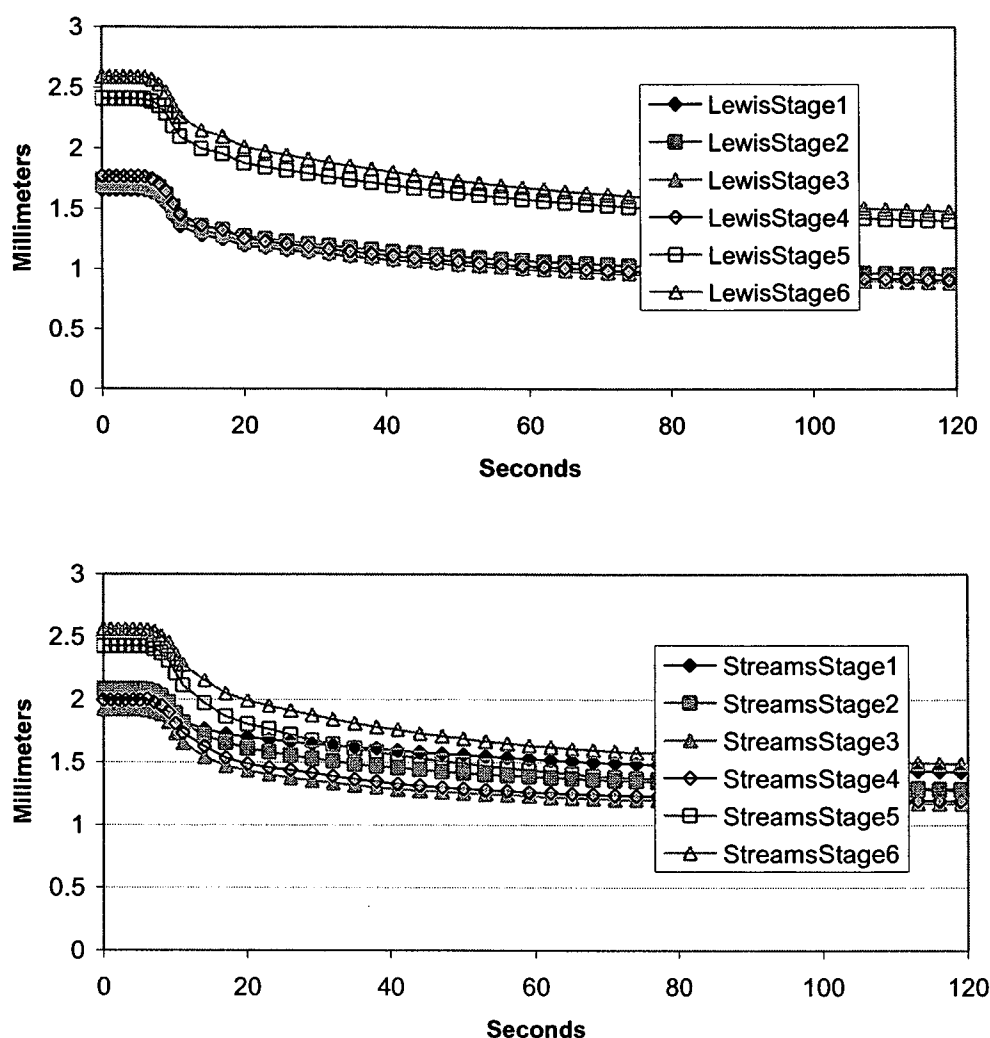


Figure 4.2.3.9 HPC thermo-mechanical rotor growth during deceleration. Lewis method (top), streams method without dovetail slip (bottom)

Figures 4.2.3.8 and 4.2.3.9 show that at steady state idle the streams method slightly overpredicts growth (compared to the Lewis method) for stages 1 to 4, and that overprediction also occurs at steady state maximum power. In contrast, growths for stages 5 & 6 are underpredicted at idle, but at max power they match quite well. Transiently, the streams method produces a larger spread of response speeds than the Lewis method, especially following deceleration. The differences between results produced by the two methods are likely to be due to a number of reasons. Differences between the two methods in geometrical representation of the hardware is likely since the SC03 model upon which the Lewis method is based is based upon true dimensions, whereas the streams method input file relies upon dimensions scaled with

a ruler from a general arrangement diagram. Also, much more simplification of geometry is associated with the streams method input file. Also, mass flows and heat transfer coefficient factors across the two methods are unlikely to be identical (engineers using the SC03 model have access to much more test data than the author of the streams method). In addition, the streams method algorithms are far less sophisticated than the SC03 algorithms (particularly for thermomechanical strain). There is also the possibility of error being introduced into the Lewis method representation during its curve fitting procedure, which forces disc and casing responses to fit simple exponential decay. However, despite of all of these opportunities for difference the streams method based on scaled geometry and estimated flow patterns was still able to achieve 0.25 mm disc rim growth accuracy on disc rim radii of approximately 250 mm.

Within the streams model used in this study, angles of 45 degrees were assumed for dovetail blade footings. The inclusion of dovetail slippage increased streams method disc rim growth predictions by 15% or more. It was found that the inclusion of dovetail slippage at such an angle lead to estimation of disc rim growths that were slightly over those predicted by the Lewis method (without dovetail slip the streams method were under those from the Lewis method, c.f. figure 4.2.3.8). The SC03 finite element model upon which the input data for the Lewis method input file were based did not include dovetail slippage. On the strength of this the author decided that for fairness of comparison dovetail slippage would not been included in comparisons against Lewis method results.

Of particular interest is the observation that stage 5 and 6 disc growths for both the Lewis and streams methods are much larger than for the other stages. Figure 4.2.3.10 shows that the coefficient of thermal expansion of the disc material used at stages 5 and 6 is up to 40% higher than for the material used at stages 1 to 4. It is to be expected therefore that stage 5 & 6 disc growth is up to 40% larger than that of stages 1 to 4.

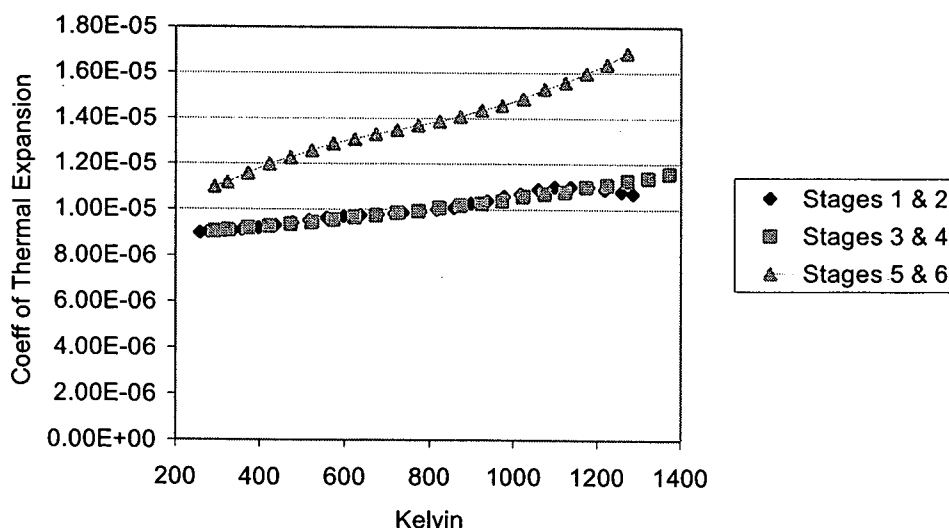


Figure 4.2.3.10

Coefficient of thermal expansion versus temperature for HPC disc materials

The effect of windage heating in the disc cavities cannot be ignored. Figure 4.2.3.11 shows the effect of switching streams method windage calculations on and off.

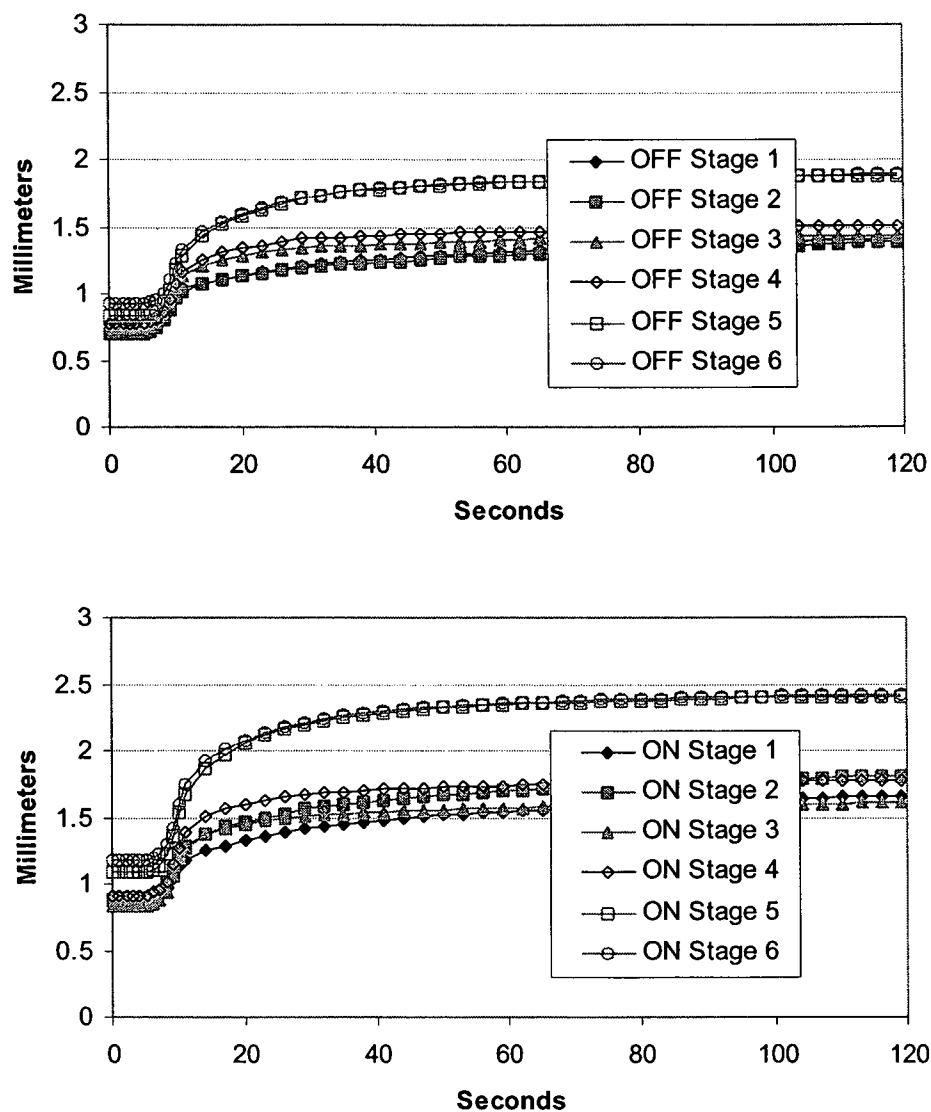


Figure 4.2.3.11 Streams method predictions for radial disc rim growth during acceleration with windage heating switched off (top) and on (bottom)

In this study the inclusion of windage heating increased disc rim growth by up to 25%. The effects were felt much more at the rear than they were at the front of the compressor, since not only do rear stages generate their own windage heating, but they are also subject to the windage heating of the bleed air accumulating in all of the previous stage disc cavities.

SUMMARY OF STREAMS METHOD CASING, BLADE, AND DISC GROWTHS

Figure 4.2.3.12 shows a summary of streams method growth predictions at idle and maximum power. At both idle and max disc and casing growth dominates at all stages, and thermal growth is generally much larger than mechanical growth. The exception to this is in the discs where centrifugal growth is relatively large, especially at the cooler temperatures of idle when thermal growths are relatively small.

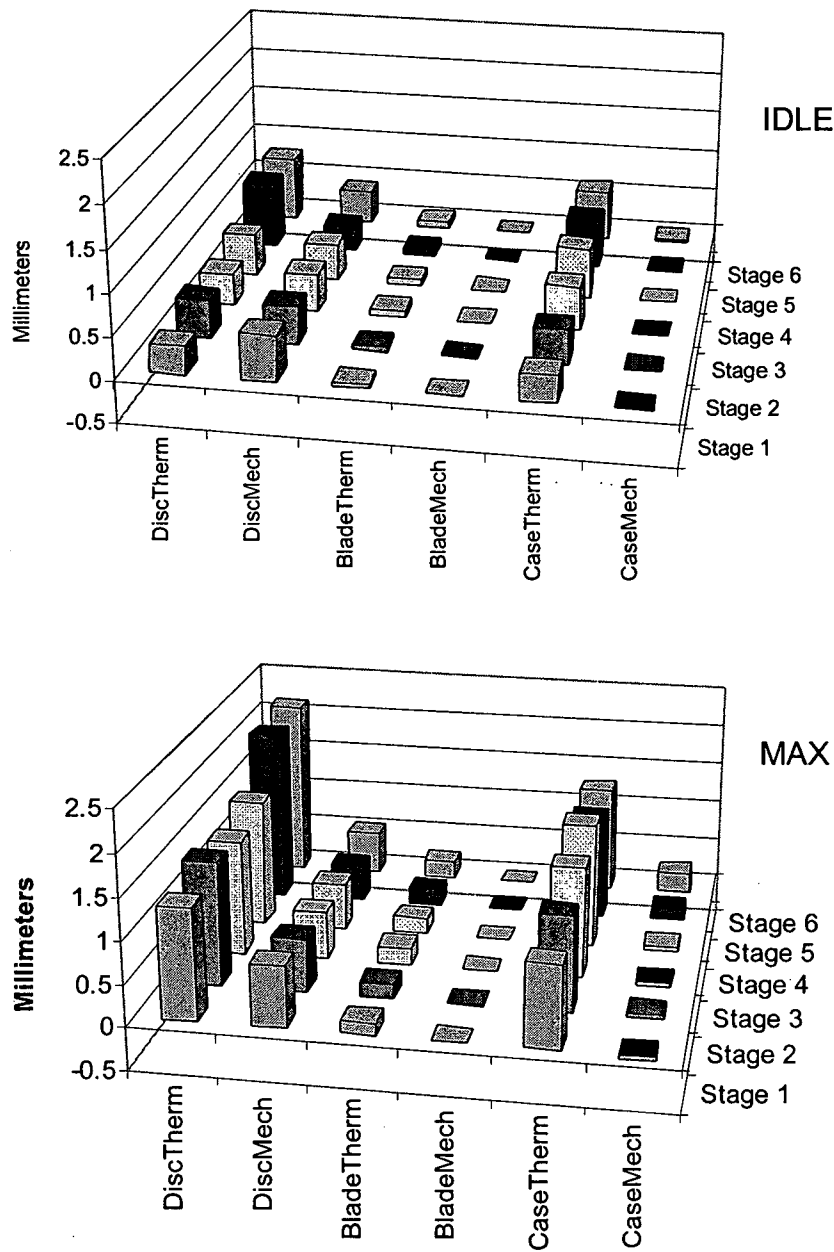


Figure 4.2.3.12 Streams method growth predictions at idle (top) and maximum power (bottom)

DELTA TIP GAP CALCULATIONS

Figure 4.3.2.13 shows Lewis method and streams method predictions of changes in blade tip clearance, compared to a cold non-rotating engine, for each stage during the slam acceleration test case.

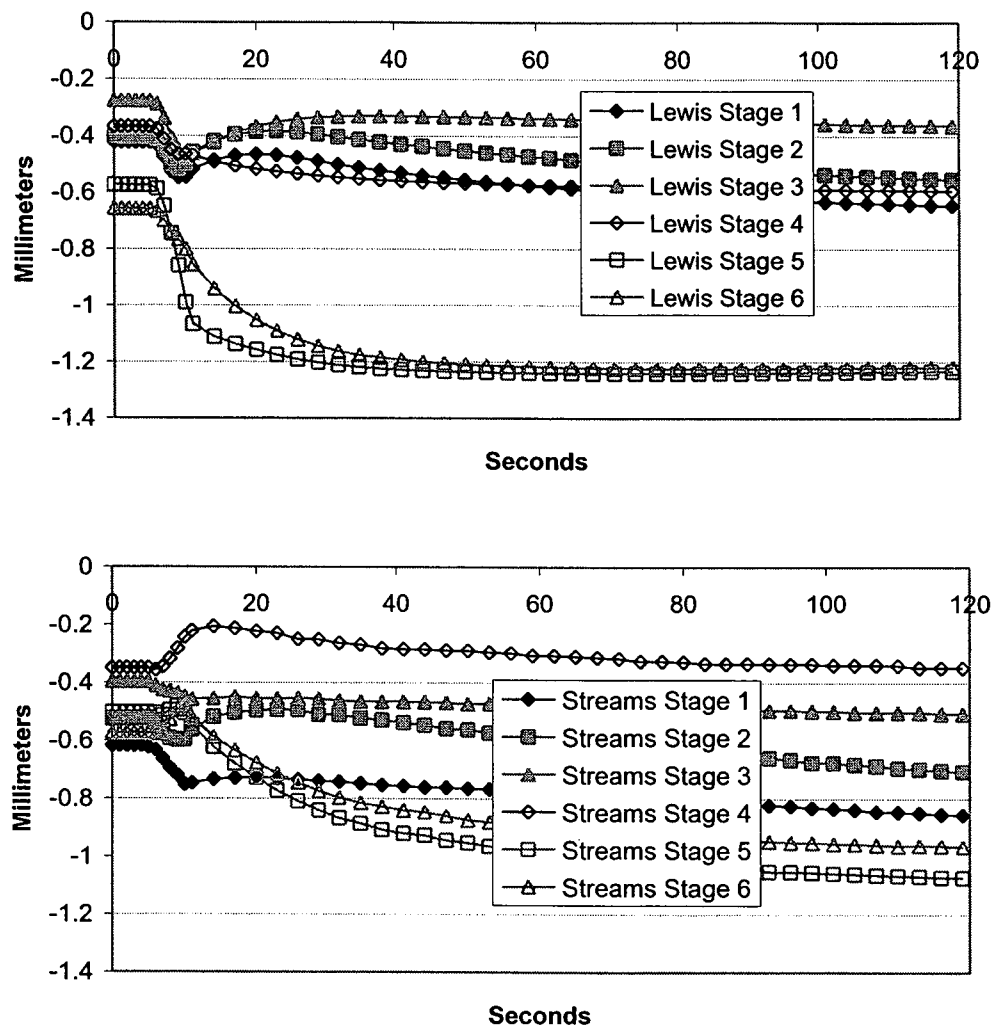


Figure 4.2.3.13 HPC delta tip gaps during acceleration: Lewis method (top) versus Streams method (bottom)

The Lewis method is based upon an SC03 finite element model, which has been validated against test data for stage 1 of the Trent 500 HPC. From the figure we can see that significant discrepancy exists between the streams method stage 1 delta tip gap prediction and that produced by the validated-at-stage-1 Lewis method. In general, the magnitudes of the graphs are comparable but the traces produced for each individual stage and the size order of the stage tip gap deltas are markedly different. This is hardly surprising given the variation we have seen between the Lewis and

streams method predictions of each of the constituent elements of delta tip gap, i.e. case thermal growth, blade mechanical growth, etc.

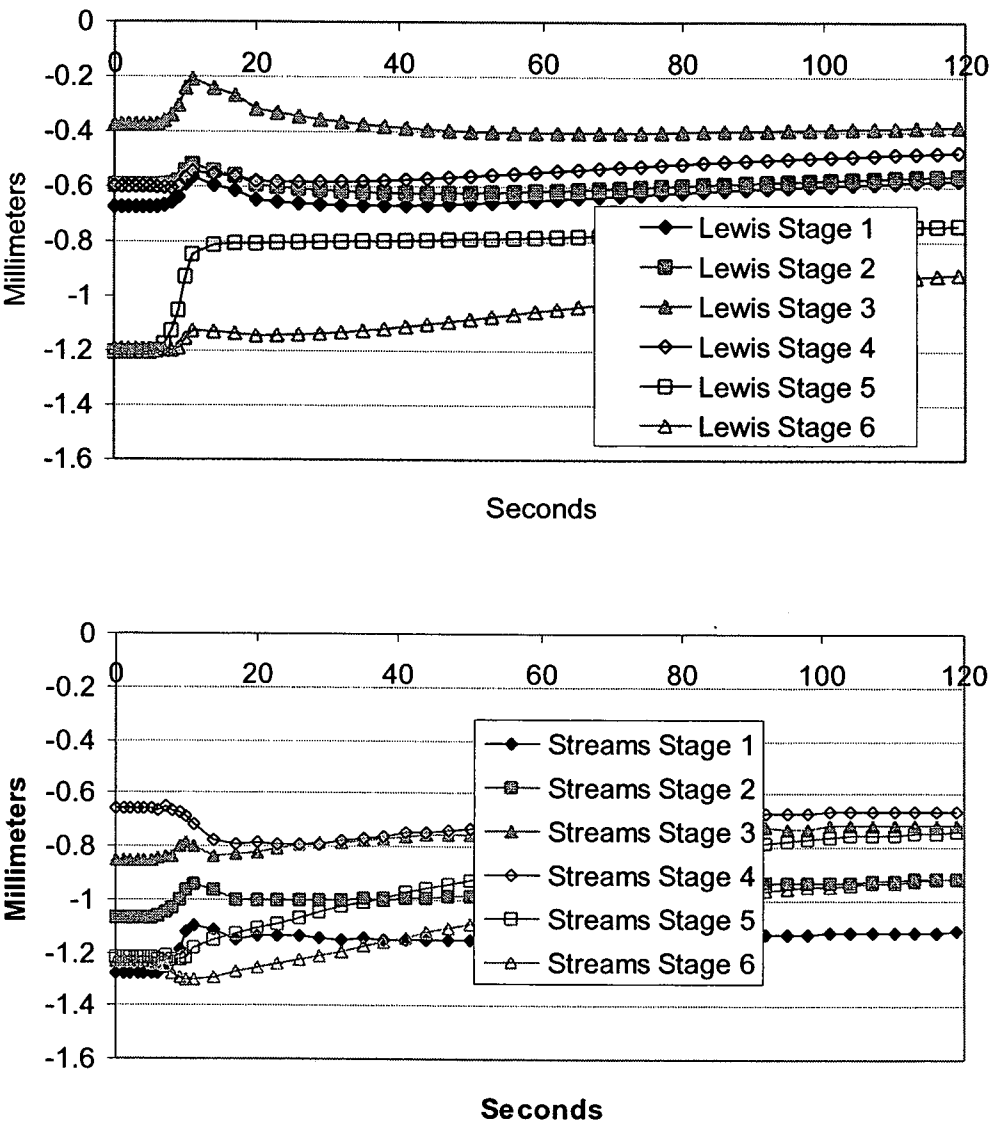


Figure 4.2.3.14 HPC delta tip gaps during deceleration: Lewis method (top) versus Streams method (bottom)

Figure 4.3.2.14 shows Lewis method and streams method predictions of changes in blade tip clearance during the slam deceleration test case. The same comment applies as before, i.e. differences in delta tip gap between the two methods are not surprising in light of the differences between predictions of the components of delta tip gap.

From figure 4.2.3.13 and 4.2.3.14 we can see that use of the streams method to predict rubs or surge initiation at a particular stage would probably identify the wrong stage, however the results for all stages together might be used to estimation whole component average tip gap, and thence used to calculate efficiency and flow capacity changes for the whole component during and following transient manoeuvres. Figure

4.2.3.15 shows a comparison of stage averaged tip clearance changes as predicted by the Lewis and streams methods. Chapter 2 outlined the sponsor's tip clearance accuracy requirements, i.e. within 0.025mm of the Lewis method steady state, and 0.1mm during transients. From figure 4.2.3.15 we can see that the streams method predictions are outside both of these targets. Interestingly, the difference between mean and RMS values of Lewis method predictions also misses the steady state target, and almost misses the transient target.

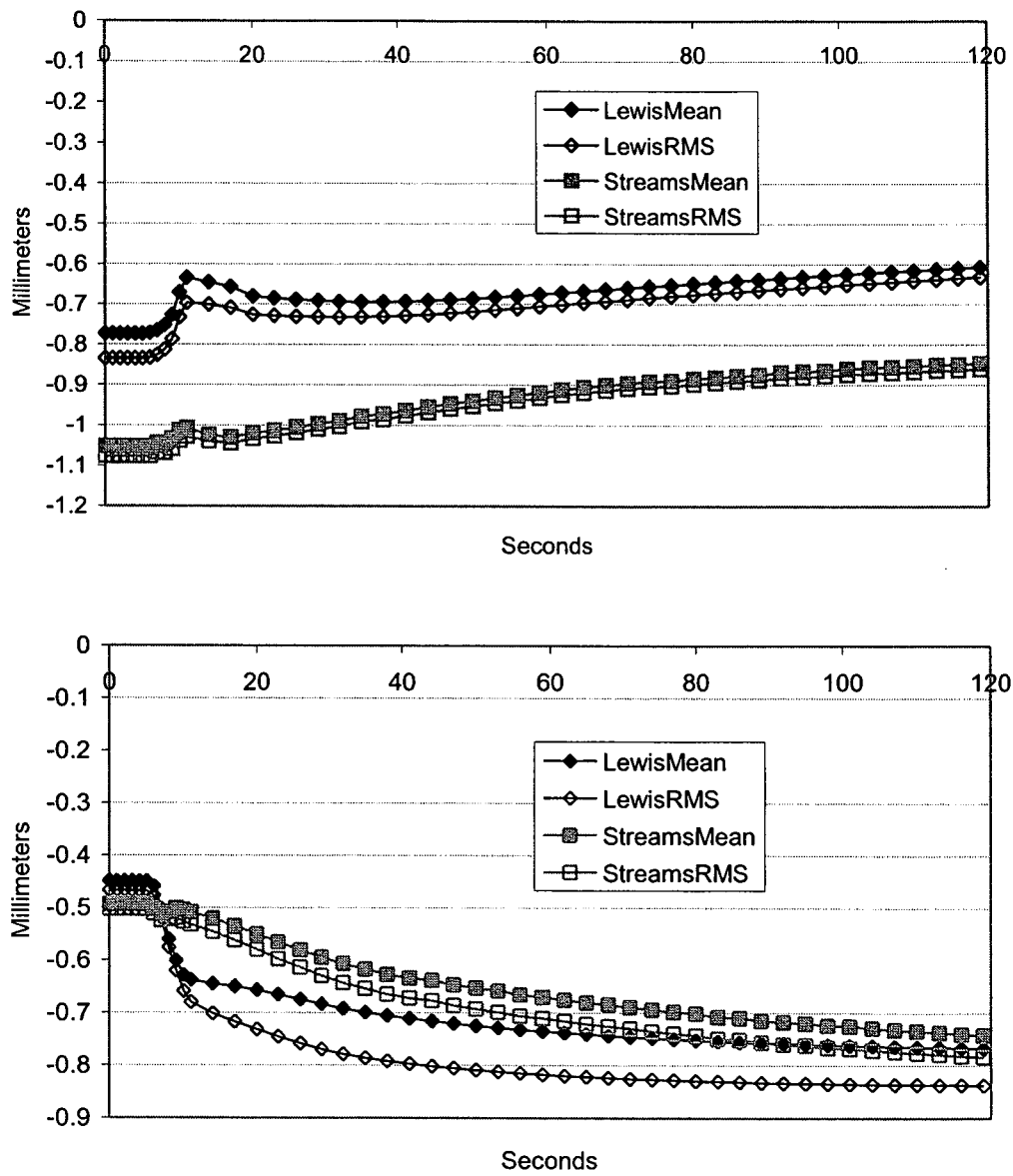


Figure 4.2.3.15 Stage averaged HPC delta tip gaps predicted by the Lewis method and Streams method during acceleration (top) and deceleration (bottom)

4.2.4 SENSITIVITY ANALYSIS

The thermal and tip clearance behaviour of a component is dependent upon a number of factors: the geometry of its hardware, the properties of its materials, and the properties and patterns of the gas flowing through and around it. These can be divided into what we might call discretionary and non-discretionary variables.

Non-discretionary variables are the givens, i.e. hardware geometry, material properties, and station vector data. Discretionary variables are those which involve a large amount of uncertainty and which are difficult for people to argue against, i.e. factors on heat transfer coefficients derived from standard correlations, and flow patterns in the disc cavities. The Trent 500 HPC can be thought of as three distinct regions: the casing, the primary gas path, and the disc cavities. A sensitivity study can easily become a mammoth task, especially when the system contains a large number of variables. To keep things simple (for himself and the reader) the author presents here a study of the effect of changes in only the discretionary variables on the behaviour of the three HPC regions. In this study, factors of 0.5, 1.0, and 1.5 were applied to heat transfer coefficients (h) in each region. Mass flow interaction fractions (MF) were originally 0.5 in the casing, 0.8 in the primary gas path, and 0.1 in the disc cavities, however other values (see figures below) were also used in this study. The test case acceleration manoeuvre was used to produce all of the data presented in this section.

CASING

Figure 4.2.4.1 shows the effect of discretionary variable changes on streams method predictions of heat flows in the Trent 500 HPC casing bleed cavity.

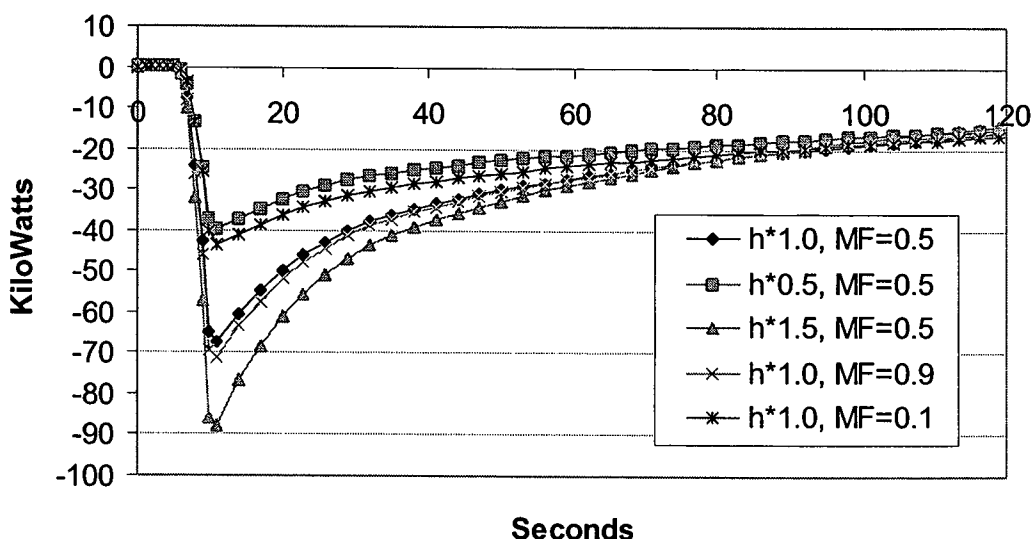


Figure 4.2.4.1

Variation of streams method HPC casing bleed cavity heat flows with changes in heat transfer coefficient and mass flow interaction fraction

Figure 4.2.4.1 shows that peak heat flow predictions were moderately sensitive to h , and less sensitive to MF. MF for the casing is defined as the portion of bleed mass flow assumed to interact with the casing metal. Peak heat flows were more sensitive to low values of h and MF than they were to high values. At very low mass flows bleed temperature rapidly approaches metal temperature, and before very long equals metal temperature. The amount of heat that it is possible for the bleed to absorb/release thus becomes governed by mass flow. As mass flow increases bleed temperatures change less dramatically and heat absorption and release rates begin to be governed by heat transfer coefficient. For this reason peak heat flows are more sensitive at a given heat transfer coefficient to low MF than they are to high MF. When h is small, bleed temperature doesn't converge that quickly to metal temperature and temperature differentials are maintained along the bleed path. In contrast, when h is large, bleed temperature converges (proportionally) more quickly to metal temperature, and even though still large, average temperature differentials are smaller than they otherwise might be. The result is that peak heat flows become less sensitive to increases in h as the increase becomes higher.

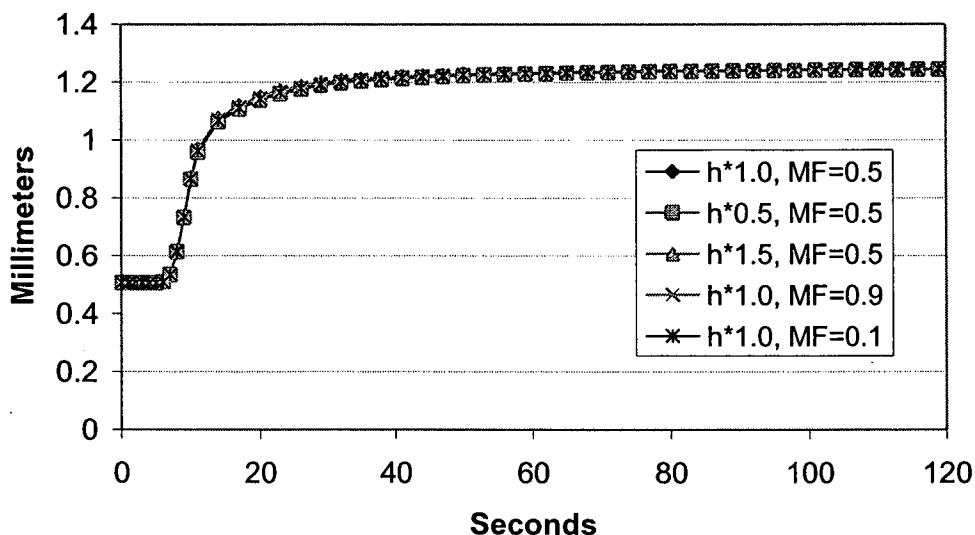


Figure 4.2.4.2 Variation of streams method prediction for average HPC casing growth with changes in heat transfer coefficient and mass flow interaction fraction

Figure 4.2.4.2 shows the variation of streams method prediction for average casing growth with changes in h and MF. In spite of the relatively large variations in heat flows demonstrated in figure 4.2.4.1, casing growth appears to be quite insensitive to any of the changes made as part of this study. Casing temperature governs casing growth. In the bleed cavities heat transfer coefficients were calculated to be a maximum of $65 \text{ W/m}^2\text{K}$, whereas in the primary gas path they were calculated to be up to $8000 \text{ W/m}^2\text{K}$. It is no surprise therefore that the temperature of the casing followed the temperature of the primary gas path more than that of the casing bleed cavity. Even though the percentage change in casing cavity values was quite large,

they were still tiny values when compared to conditions in the primary gas path. Thus casing growth was found to be quite insensitive to casing discretionary variables.

PRIMARY GAS PATH

Figure 4.2.4.3 shows how streams method primary gas path heat flows change with variations in primary gas path discretionary variables. MF for the primary gas path is defined as the portion of primary gas path mass flow assumed to interact with the blades, the remainder being split evenly to interact with the casing and rotor.

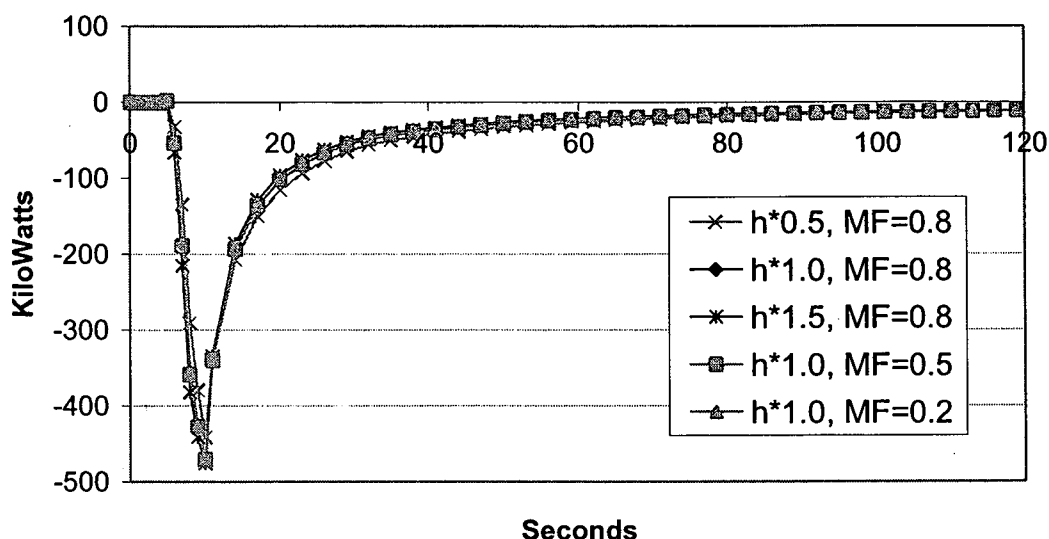


Figure 4.2.4.3 Variation of streams method HPC primary gas path heat flows with changes in heat transfer coefficient and mass flow interaction fraction

The figure shows that primary gas path heat flows are largely insensitive to changes in the discretionary variables. In the primary gas path heat transfer coefficients are relatively large, and this makes Biot numbers quite large. When Biot numbers are large then conduction plays a significant role in the heat transfer process. The higher the Biot number is past unity, the more heat transfer is restricted by conduction. The heat transfer coefficients in the primary gas path were relatively very high resulting in Biot numbers of about 3. Clearly conduction plays a significant restrictive role in the heat transfer process.

Changes in h of plus and minus 50 % produce Biot numbers that are still greater than unity. Only when h is reduced by 50 % is there any visible change in the results. Primary gas path mass flows are relatively large, such that it is h rather than mass flow or MF that is the rate-controlling factor in the heat transfer process. We can thus see why primary gas path heat flows appear to quite be insensitive to variation of the discretionary variables.

Figure 4.2.4.4 shows how average disc rim and casing growth vary with changes in the primary gas path discretionary variables.

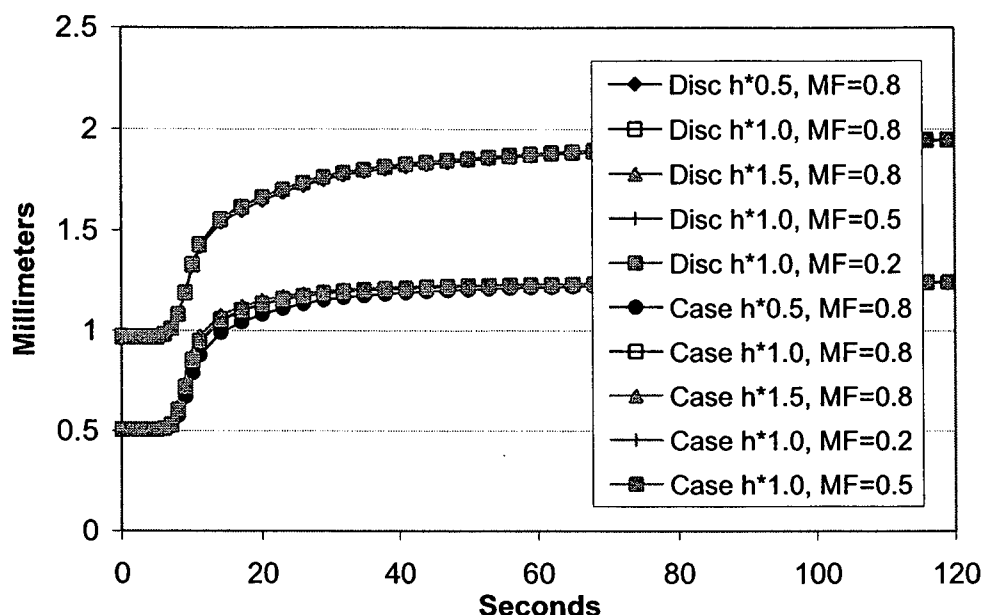


Figure 4.2.4.4 Variation of streams method HPC average disc rim and casing growth predictions with changes in primary gas path heat transfer coefficient and mass flow interaction fraction

The figure shows that disc growth was quite insensitive to changes in MF and h , whereas casing growth was insensitive to MF but slightly sensitive to h . Smaller h appears to delay the thermal response of the casing but not of the disc. Disc growth is governed largely by the temperatures of the diaphragm and cob, whereas casing growth is more dependent upon primary gas path conditions, but due to high primary gas path Biot numbers the effect of h variation on the case is somewhat muted. Blade thermal response was almost immediate no matter how the discretionary variables were changed.

DISC CAVITIES

Figure 4.2.4.5 shows how streams method predictions for HPC disc cavity heat flows change with variations in the discretionary variables. MF for the disc cavities is defined as the portion of bleed flow assumed to interact with the disc sides and drum inner surface, as opposed to flowing straight through (c.f. figure 4.2.1.2).

The figure shows that in the disc cavities heat flows are sensitive to both h and MF. As h gets lower peak heat flows also gets lower and vice versa. When MF is high the temperature pickup of the bleed is low, and therefore the average gas to metal temperature differential is maintained more than for when MF is low and temperature

pickup is high. The result is higher heat flows for higher MF and vice versa. Figure 4.2.4.6 shows how these phenomena are reflected in the streams method disc rim growth predictions. We can see that changes in the discretionary variables can alter disc rim growth by up to about a third of a millimetre, which on a 50mm long blade equates to 0.6 % of non-dimensional blade height.

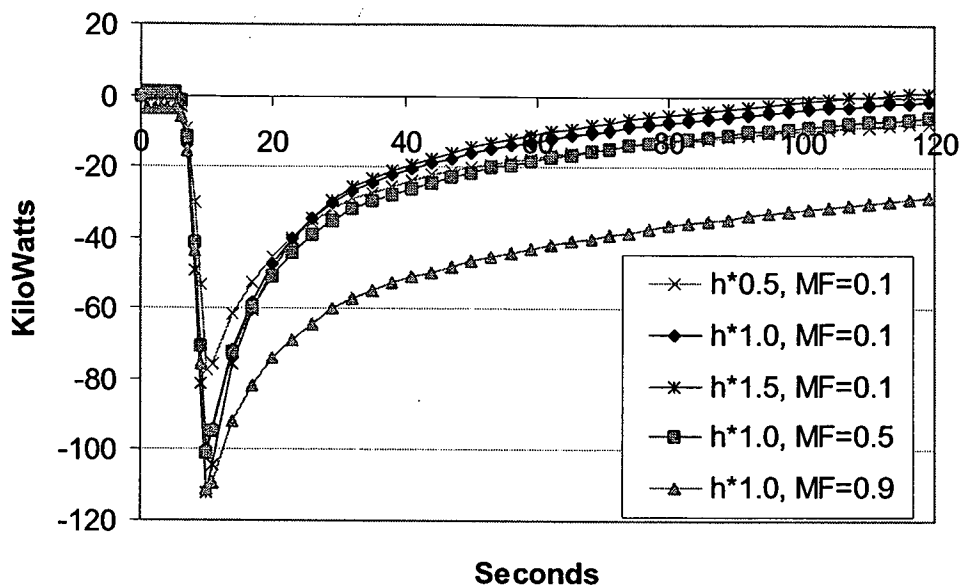


Figure 4.2.4.5 Variation of streams method HPC disc cavity heat flows with changes in heat transfer coefficient and mass flow interaction fraction

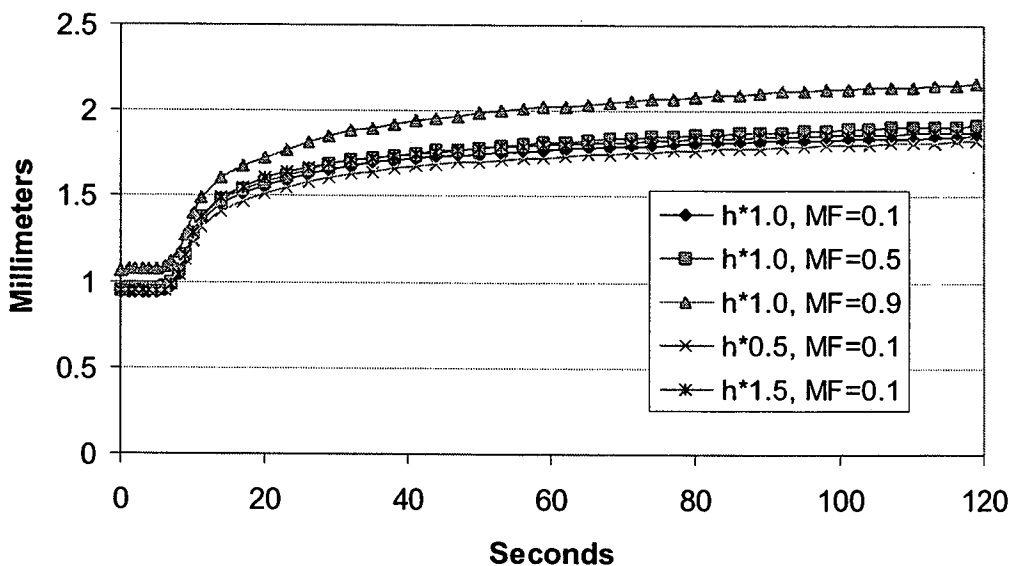


Figure 4.2.4.6 Variation of streams method HPC average disc rim growth predictions with changes in primary gas path heat transfer coefficient and mass flow interaction fraction

Originally disc cavity flow was set as shown in figure 4.2.1.2, however, as part of this sensitivity study the direction of flow circulation within the disc cavities was reversed. The resulting changes in heat flow and disc growth were very small, i.e. changes occurred only at the third significant figure.

4.2.5 USAGE TIMES

As the old adage goes, time is money. The time it takes to use the streams method can be divided into set up time and run time, a description of each is as follows.

The set up time required to create the Trent 500 HPC streams model input file was quite significant. As with all finite volume based methods, grid generation is the bottle neck in the process, and without the aid of any kind of CAD package the streams method relies upon the user to specify one at a time each individual x and y coordinate for each elemental volume used to represent the HPC geometry. Once this task is complete, the user must then specify which of the volume's surfaces is the convecting surface. In addition he/she must specify each volume's material, heat transfer coefficient correlation, and sometimes other information depending on the heat transfer coefficient correlation used. After this, the user must specify strings of volume numbers that will represent each stream, and also stream feeders and mass flow interaction fractions. Finally the user must specify which volumes represent the constituent rings of each disc, the load bearing casing, and the blades, as well as things such as numbers of blades for each blade row. The author took approximately 2 days to prepare the Trent 500 HPC streams model input file, assisted by some purpose built Excel spreadsheets.

Run time for the streams method can be divided into two categories, viz. the time required to converge the model at a series of steady state operating points (used for later calculation of steady state to transient tip gap deltas), and after this the time required to run the model through the transient manoeuvre. When RRAP was run with only the streams method model of the Trent 500 HPC incorporated into it on the author's 1.6 GHz desktop computer it took approximately 8 minutes to converge upon twenty steady state operating points from idle to maximum power and back to idle. Once that was completed it took approximately 40 seconds to run a 120 second transient. The steady state runs only need to be calculated once for each particular point in the operational envelope, so long as the transients performed are within the range of engine handle used for the steady state runs.

4.2.6 SECTION CONCLUSION

The purpose of this section was to demonstrate the extent to which the streams method can be used to predict transient heat flow and tip clearance changes that occur in the Trent 500 HPC.

Subsection 4.2.1 introduced the Trent 500 HPC and described the streams representation of it, and also the test case acceleration and deceleration manoeuvres used in the study.

Subsection 4.2.2 presented the streams method predictions for convective and conductive heat flow and temperature pickup in the casing bleed cavities, the primary gas path, and the disc cavities. Important observations were that large gas temperature excursions occur when mass flows are small, that windage has a larger effect than convective heat transfer in the disc cavities, and that heat flows are mass flow dependent at low mass flows but become heat transfer coefficient dependent at higher mass flows. Comparison of the streams method results against those produced by BR081 showed streams method predictions to be significantly lower, especially for the blades. This is because unlike in BR081, in the streams method upstream metal preheats or precools air before it interacts with downstream primary gas path metal.

Subsection 4.2.3 presented streams method predictions for transient thermal and mechanical growth of the casing, blades, and discs of the Trent 500 HPC. Comparison of the results was made against those produced using the Lewis method. It was shown that values for individual stages were quite different, and that the staged averaged tip clearance change predictions were up to 0.25mm different from the Lewis method predictions, outside of the target accuracies of 0.025mm at steady state and 0.1mm transiently. Interestingly, the difference between RMS and mean Lewis method values also missed the steady state target, and almost missed the transient target. It was shown that the simple thermomechanical treatment of the casing and blades yielded less accurate results than the more sophisticated thermomechanical treatment of the discs. It was also shown that in the Trent 500 HPC, changes in blade tip clearance are dominated by casing thermal growth and disc thermal and centrifugal growth.

Subsection 4.2.4 presented the results of a simple sensitivity study investigating how changes in the discretionary variables of heat transfer coefficient factor and mass flow interaction fraction effected heat flows and tip gaps. Since casing temperatures were dominated by flow conditions in the primary gas path casing heat flows were sensitive to variations in the casing cavity but the casing growths were not. Primary gas path heat flows were insensitive to changes due to the high Biot numbers that occur there. Disc cavity heat flow and disc growth on the other hand were sensitive to both sets of discretionary variables. Changing the direction of flow in each disc cavity made very little difference to predicted heat flows and growths. This study showed that the streams method has a degree of capacity to be calibrated.

Subsection 4.2.5 described the usage times associated with the streams method, i.e. that set up times are in days, that steady state run times are in minutes, and that transient run times are quicker than real time for RRAP running the HPC model only.

From this study of the Trent 500 HPC, we can see how well the streams method meets the requirements laid down by the sponsor. The streams method calculates both heat soakage and tip clearance changes, is physics based, runs faster than real time during transients, can be calibrated to a degree, but is outside of the tip clearance target accuracies.

4.3 SINGLE AVERAGED STAGE REPRESENTATION OF MULTISTAGE COMPONENTS

4.3.1 SECTION INTRODUCTION

The purpose of this section is to assess how well a multi-staged component (the Trent 500 HPC) can be modelled using single averaged stage representation during the acceleration manoeuvre described in section 4.2.

If the model performs well then this means that single averaged stage models might justifiably be used in place of multistage models where components have more than one stage. This would result in reduced computational burden and a reduced amount of work preparing input files. This is the primary reason why previous researchers focussed on single averaged stage representations. In addition, calibration of a single stage model to bring answers in line with test data is likely to be simpler and quicker to do than it would be for the corresponding multistage model.

The single averaged stage approach is not without its challenges. In the Trent 500 HPC eight different materials are used for its six stages of blades, discs, and casings. This makes the decision regarding what material to assign to the disc, blades, and casing for the single averaged stage a little tricky. Another complication is that the casing of the Trent 500 HPC is ventilated by two separate bleed systems each with its own different temperature, pressure, and mass flow. Using a combined averaged single value for casing bleed temperature, pressure, and mass flow may or may not yield representative results. The averaging process produces a single manageable value for component tip clearance, but it can result in a loss of information regarding important phenomena such as rub and stall (for multistage components a rub or stall occurring at an individual stage may go unnoticed if the average stage tip gap appears to be behaving well).

Using the single averaged stage approach causes another challenge when it comes to calculating heat flows in and out of the primary gas path. A six stage component is likely to have primary gas path wetted area and thermal mass approximately six times that of a single averaged stage. This means that the single averaged stage model is likely only be able to absorb about one sixth of the thermal energy, and so it might be expected that calculated heat flows might also be one sixth as large. One way used by previous researchers to manage this problem is to simply multiply wetted areas and masses for the single averaged stage by the number of stages in the component, such that they are equal those of the whole component. Unfortunately the streams method does not lend itself to this approach. In the streams method wetted areas are fixed by specified model geometry, and factoring up the density of disc materials to get the right mass would result in very large mechanical components of disc growth. An alternative approach might be to factor up specific heat to increase energy absorption, however, this would change transient temperature distributions and thus transient tip clearances. An approach more suited to the streams method would be to simply perform heat flow and tip gap calculations for the single averaged stage, take the tip gap calculations at face value, and multiply the heat flows estimated for the single

averaged stage by the number of stages in the whole component. Indeed, this is the approach that has been adopted in this study.

Figure 4.3.1.1 shows the single averaged stage representation of the Trent 500 HPC used in this study. Geometry for the disc, blades, and casing was derived by taking the arithmetic mean of the geometry of each of the six stages of the Trent 500 HPC. The casing was represented differently than in the multistage model: it was assumed to have a double skin: the outer skin to bear any loads, and the inner skin to form the end wall in the primary gas path. It was assumed that a void separated the two skins. The materials used in the single averaged stage model were assumed to be those of the middle two stages of the Trent 500 HPC. The casing bleed was assumed to be the average (weighted by the number of stages ventilated by each bleed) of the two bleeds that service the Trent 500 HPC casing. The disc bleed was assumed to be IP8 air. The mass flow, temperature, and pressure for the primary gas path were assumed to be the arithmetic mean of those found at the inlet and the outlet of the Trent 500 HPC. Stator well flow was assumed to be 1% of primary gas path flow. 10% of primary gas path air was assumed to interact with each of the end walls, and 80% with the aerofoils. The amount of bleed air interacting with the disc surfaces (as opposed to passing straight through, see figure 4.3.1.1) was assumed to be approximately 10%.

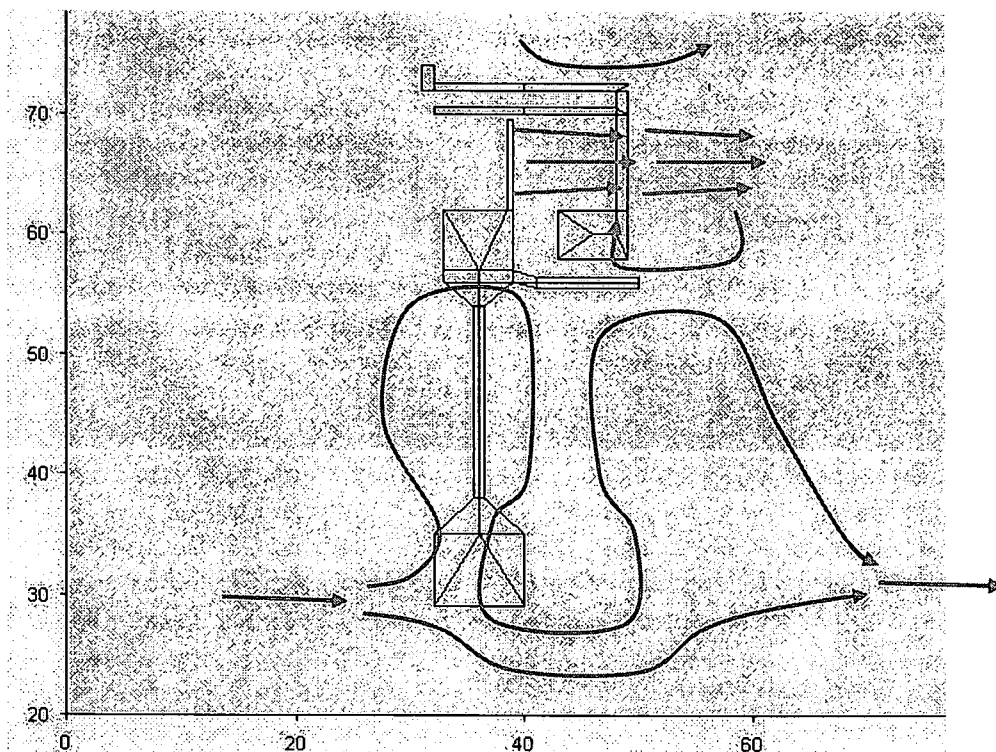


Figure 4.3.1.1

Streams method model of Trent 500 HPC using single averaged stage representation.

4.3.2 HEAT FLOWS

Figure 4.3.2.1 shows primary gas path heat flows calculated by BR081, the multistage streams model, and the single averaged stage streams model. It also shows the multistage model heat flow divided by six (the number of stages in the Trent 500 HPC), and the single averaged stage heat flow multiplied by six. We can see from figure 4.3.2.1 that simply multiplying the single averaged stage heat flow by the number of stages does not produce a value that is consistent with the heat flow predicted for the whole component using the multistage model. An interesting observation is that the multistage model predicts whole component heat flows that are lower than those predicted by BR081, whereas the single averaged stage value multiplied by the number of stages predicts heat flows that are higher than those predicted by BR081. At the time of writing the author did not have access to any real engine heat flow data, therefore it is impossible to say which of the methods gave a more realistic answer, especially when such a large spread exists in predicted values from each of the methods.

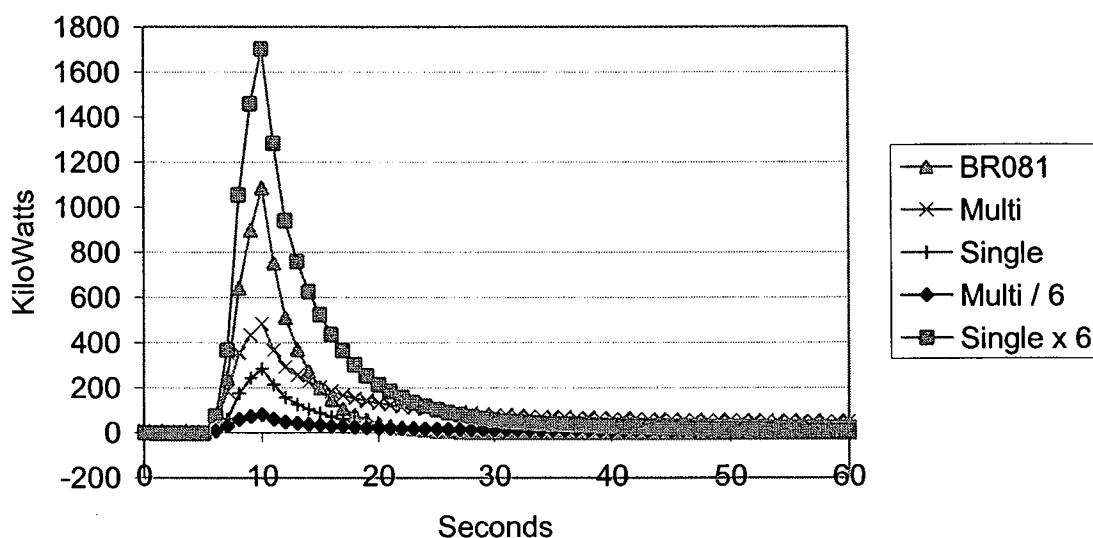


Figure 4.3.2.1 Primary gas path heat flows predicted using BR081, a multiple stage streams model, and a single averaged stage streams model.

Another interesting observation is that the heat flow decay response exhibited by the single averaged stage model appears to be similar to that of the multistage model. Thermal time constants depend on the ratio of thermal capacity to heat transfer rate. The heat transfer coefficients for both single averaged stage and multistage models were approximately the same, as were material specific heats and the ratio of wetted area to thermal mass. It is therefore not surprising that the heat flow decay time response for the single averaged stage model worked out to be very similar to that of the multistage model.

The high heat flow predictions of the single averaged stage are likely to be due to the existence of larger differences in gas and metal temperature than in the multistage model. During the acceleration manoeuvre, the forward stages experience lower swings in temperature than the rearward stages, which means that the forward stages experience lower than average heat flows. In addition, during acceleration manoeuvres the forward stages precool primary gas path air and preheat disc bleed air so that the temperature differences between the gas and metal in rearward stages is reduced. The single averaged stage model features no such precooling or preheating, and so gas metal temperature differences and thus heat flows are higher. Evidence to support this argument can be found by interrogating each of the streams models for temperature difference between gas and metal. It was found that for the casing, the peak temperature difference for the single averaged stage was a little lower than the averaged values from the multistage model (perhaps due to the double skin increasing thermal resistance). However, for the aerofoils, temperature differences were approximately double the average of values predicted by the multistage streams model (see figure 4.3.2.2). Temperature differences were also double in areas associated with the disc rims. Single averaged stage heat flows were larger than BR081 heat flows probably due to differences in assumed thermal mass.

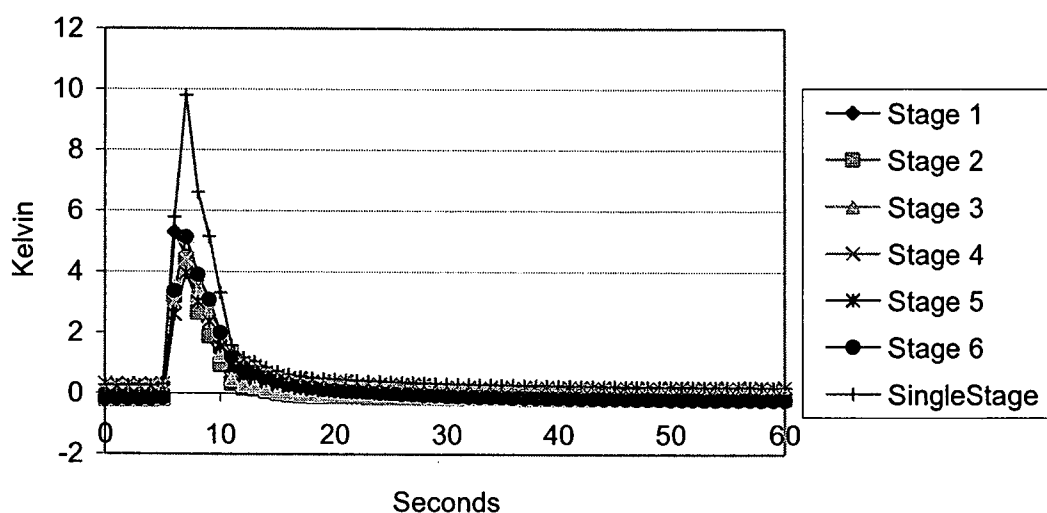


Figure 4.3.2.2 Primary gas path temperature differences ($T_{\text{gas}} - T_{\text{metal}}$) for aerofoils, multiple stage streams model versus single averaged stage streams model.

4.3.3 TIP GAPS

Figure 4.3.3.1 shows Lewis method (c.f. chapter 2) predictions for changes in tip gap from cold static for individual stages of the Trent 500 HPC. The spread between values for each stage is quite large. Figure 4.3.3.2 shows the stage averaged values from the Lewis method, from the multistage streams model, and from the single averaged stage streams model. It is interesting to note that the spread in values on figure 4.3.3.2 is smaller than the spread in values on figure 4.3.3.1.

An important observation is that the process of calculating six individual stage tip gap deltas and then finding their average gave a very different result to that produced by a process of first averaging the stages to get a single averaged stage and then doing the tip gap delta calculation (see figure 4.3.3.2), i.e. the single averaged stage answers were not equal to the average of the individual single stages, and the difference was relatively large.

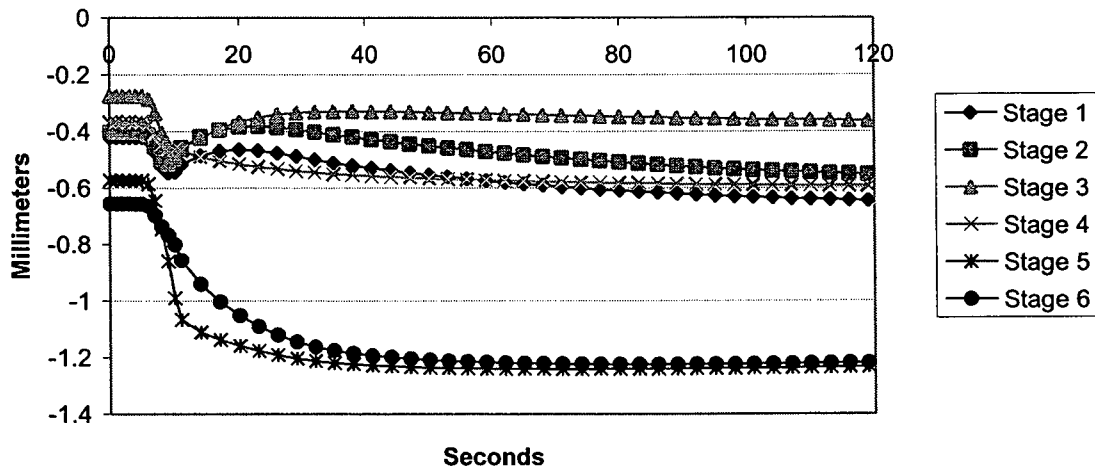


Figure 4.3.3.1 Transient delta tip gaps for each stage of the Trent 500 HPC produced using the Lewis method.

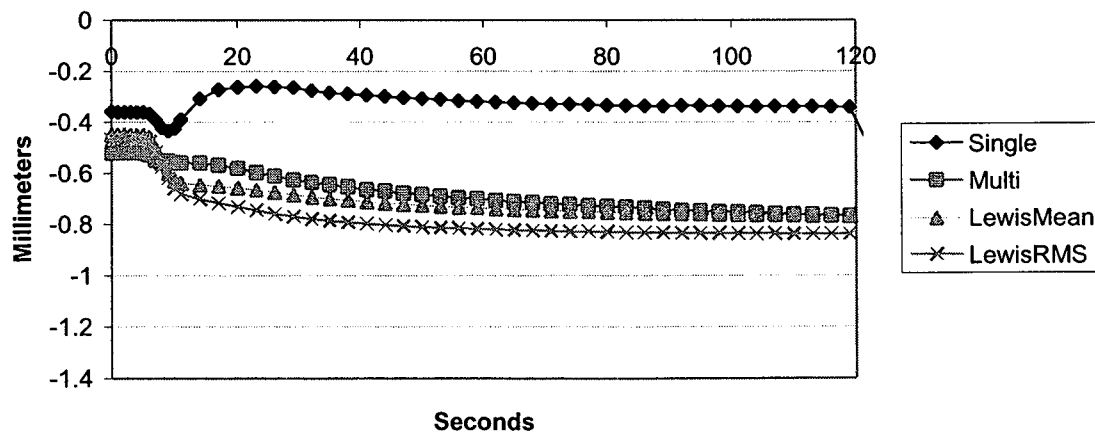


Figure 4.3.3.2 Average (mean and RMS) Lewis method delta tip gap versus average multistage streams model delta tip gap and single averaged stage streams model delta tip gap.

The Trent 500 HPC Lewis method has been validated against a detailed finite element model, that itself has been validated against test data. For this project the target has been to develop a method capable of predicting tip gap changes that are within 0.025 mm of the Lewis method predictions at steady state and 0.1 mm during transients.

Upon inspection of figure 4.3.3.2 the multistage streams model appears to make predictions that are much closer to the Lewis method compared to the single averaged stage model. Interestingly the difference between the arithmetic mean and the RMS average of the individual Lewis results approaches 0.1 mm (see figure 4.3.3.2), i.e. by simply changing the method of averaging almost all of the allowable tolerances have been used up.

Which averaging method is the right one to use? Rolls-Royce usually use the RMS value, but the RMS averaging process does not preserve information regarding the sign of the value (i.e. it always comes out positive and had to be made negative for figure 4.3.3.2). The RMS process also gives extra weight to outlying values. Is this any more accurate than using the arithmetic mean, or geometric mean, or harmonic mean or any other kind of mean? Whichever averaging method we use we are likely to get different answers, and the question remains of whether or not any physical justification exist for using one averaging method over another.

For a multistage component the maximum tip clearance exhibited by any individual stage is useful for stall predictions, and the minimum tip clearance useful for rub predictions, but this information is likely to be lost in the averaging process. All we are left with is a single value that we hope represents the whole multistage component. But how do the effects of individual stage tip gaps combine to affect the performance of the component? Average tip gap deltas are used to help estimate changes in component efficiency and flow function, but can the effects of individual tip gaps be treated as independent or does the tip gap on an upstream blade influence the efficiency effect of tip gaps on downstream blades? Can we accurately combine them using a simple mean, or should we use a more involved averaging process such as RMS? Without recourse to experiment or CFD investigation the answer is likely to remain a matter of opinion.

Figure 4.3.3.3 shows the arithmetic mean of stage disc growths, thermal and mechanical casing growths, thermal blade growths, and delta tip gaps calculated using the Lewis method. It compares these answers to what is essentially a first attempt prediction of Trent 500 HPC transient tip clearances using a single averaged stage streams model based upon scaled geometry from a general arrangement drawing, an uncertain material specification, station vector data, and estimated bleed flow conditions. The delta tip gap predicted by the single averaged stage model is outside of the target tolerances of 0.025mm and 0.1mm, as are most of the components of tip clearance predicted by the single averaged stage model.

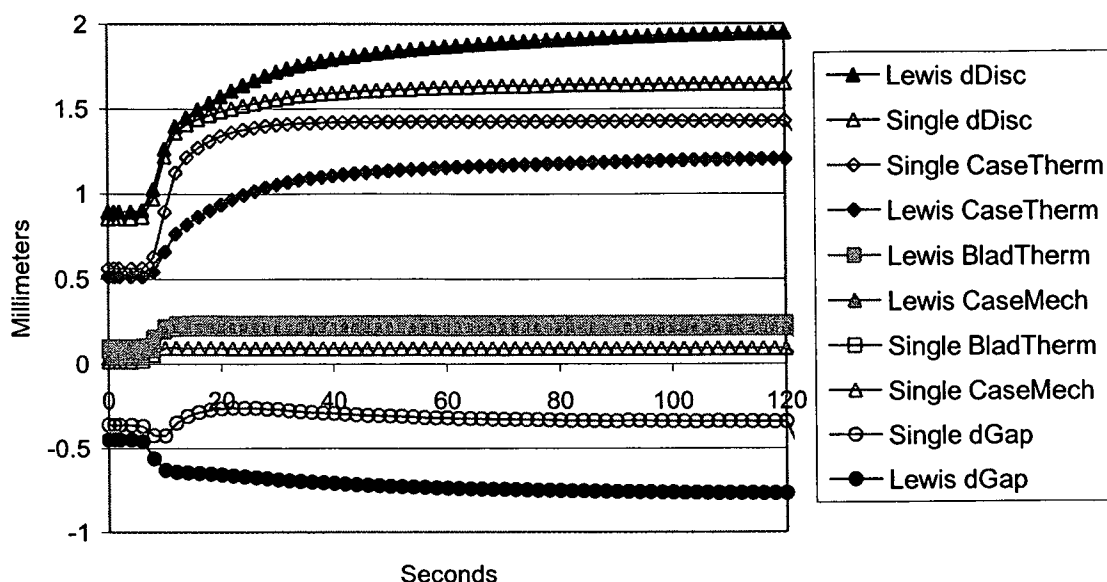


Figure 4.3.3.3 Lewis method delta tip gap components versus single averaged stage streams model.

4.3.4 SENSITIVITY ANALYSIS

The heat flow and tip clearance behaviour of a component is dependent upon the geometry of its hardware, the properties of its materials, and the properties and patterns of the gas flowing through and around it.

The construction of a multistage streams method model involves simplification of component geometry. A single averaged stage streams model involves even greater simplification of component geometry. It was demonstrated above that a result derived from averaged stage geometry does not equal the mean of results derived from a set of individual stage geometries. Nevertheless, for the purpose of this sensitivity analysis it is assumed that the single averaged stage streams model geometry correctly represents the multistage component. Discretionary variables within the streams method are heat transfer coefficients (h) and fractions of bulk flows that interact with associated wetted surfaces, i.e. mass flow interaction fractions (MF). They are discretionary variables because numbers chosen to represent them are difficult to substantiate, and therefore difficult for people to argue with. The sensitivity study presented here focuses on the effect of making changes to these discretionary variables.

For the purposes of this sensitivity study the single averaged stage has been divided into casing, primary gas path, and disc. Variations in discretionary variables in each of these three sections have been carried out in isolation.

CASING

Figure 4.3.4.1 shows that higher heat transfer coefficients in the casing bleed result in higher steady state and peak heat flows into and out of the casing bleed air. Figure 4.3.4.2 shows that higher casing bleed heat transfer coefficients also result in a lower steady state casing thermal growth, under these conditions the bleed air is likely to cool the case more effectively. Higher heat transfer coefficients also speed up the thermal response of the casing (see figure 4.3.4.2). For the casing, both heat flow and growth appear to be moderately sensitive to changes in heat transfer coefficient.

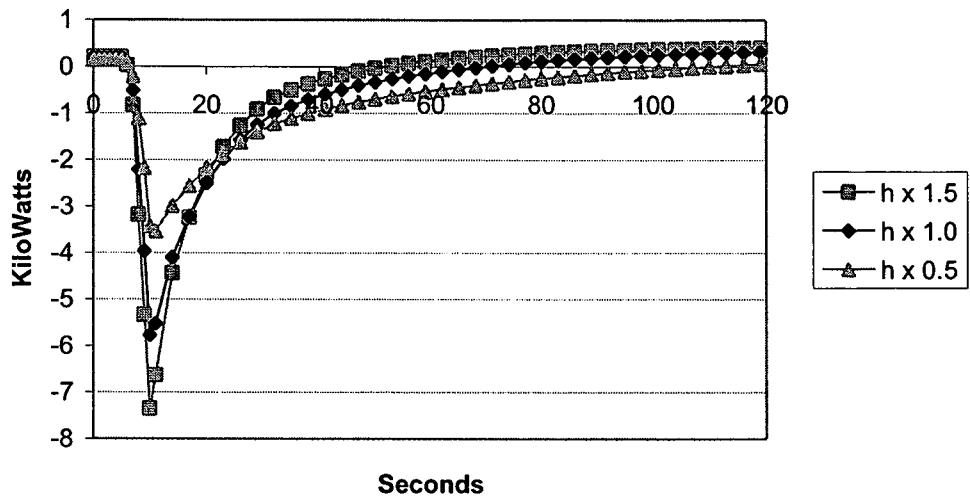


Figure 4.3.4.1 Variations heat flow into the casing with variation in casing bleed heat transfer coefficients.

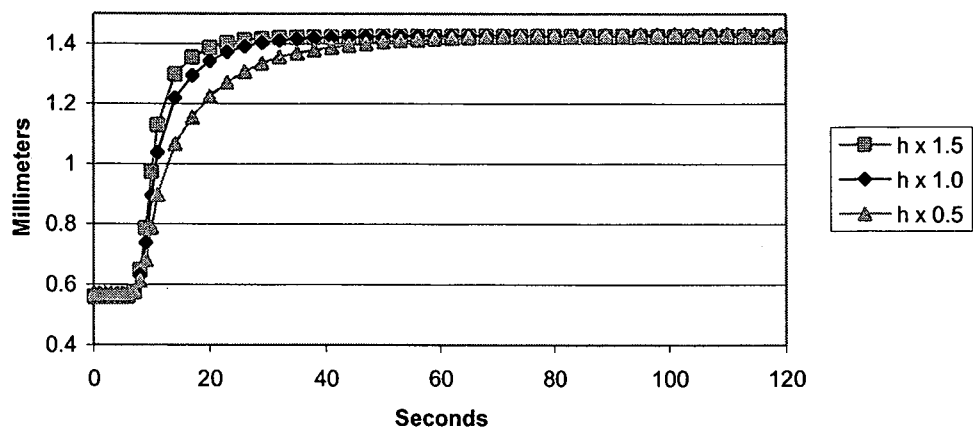


Figure 4.3.4.2 Variations in casing thermal growth with variation in casing bleed heat transfer coefficient.

Figures 4.3.4.3 and 4.3.4.4 show the variation of casing heat flow and thermal growth that occurred as a result of varying the value of mass flow interaction fraction assumed for the casing bleed (initially 50%). Only the effects of large changes are noticeable. It appears that the casing model is relatively insensitive to mass flow interaction fraction changes. Further investigation showed that bleed air temperature changes were only a few degrees and so it appears as though in the casing heat transfer coefficients are more dominant than the thermal capacity of the bleed air.

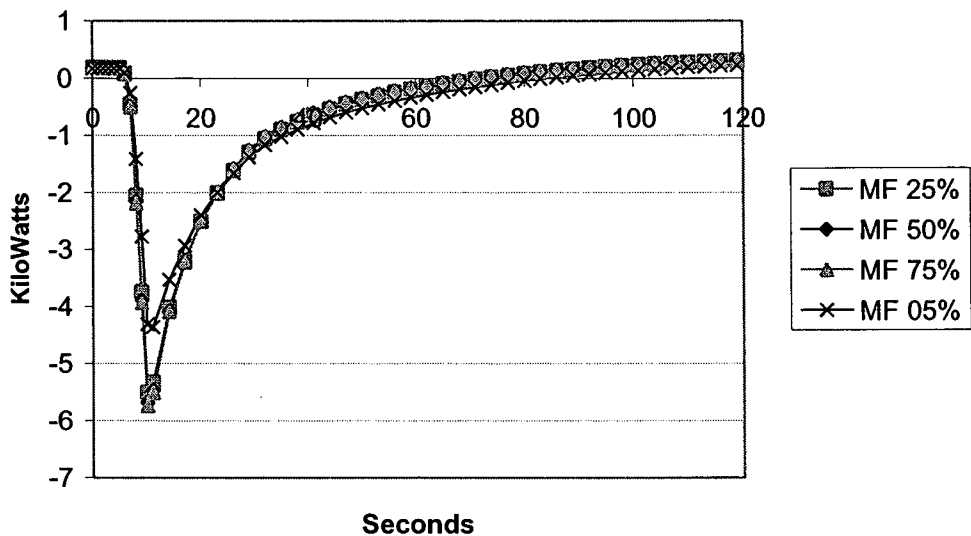


Figure 4.3.4.3 Variations in casing cavity heat flow with variation in casing mass flow interaction fraction.

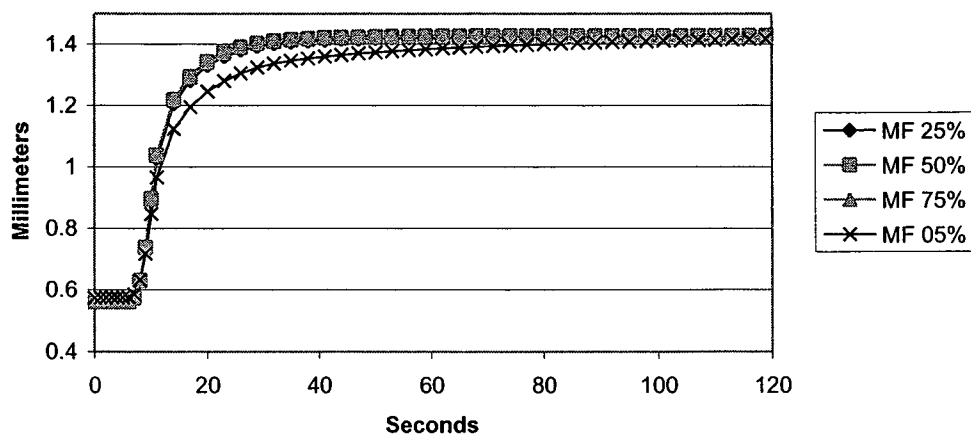


Figure 4.3.4.4 Variations in casing thermal growth with variation in casing mass flow interaction fraction.

PRIMARY GAS PATH

Figure 4.3.4.5 shows the variation in primary gas path heat flow that occurred as a result of factoring all heat transfer coefficients associated with the primary gas path. Figure 4.3.4.6 shows what effect this had on delta tip gap calculations. We can see from figure 4.3.4.5 that primary gas path heat flow is only mildly sensitive to changes in heat transfer coefficient, and from figure 4.3.4.6 we can see that delta tip gap calculations are not very sensitive at all. This might suggest that transient changes in tip clearance are more sensitive to changes in the bleed paths than changes in the primary gas path.

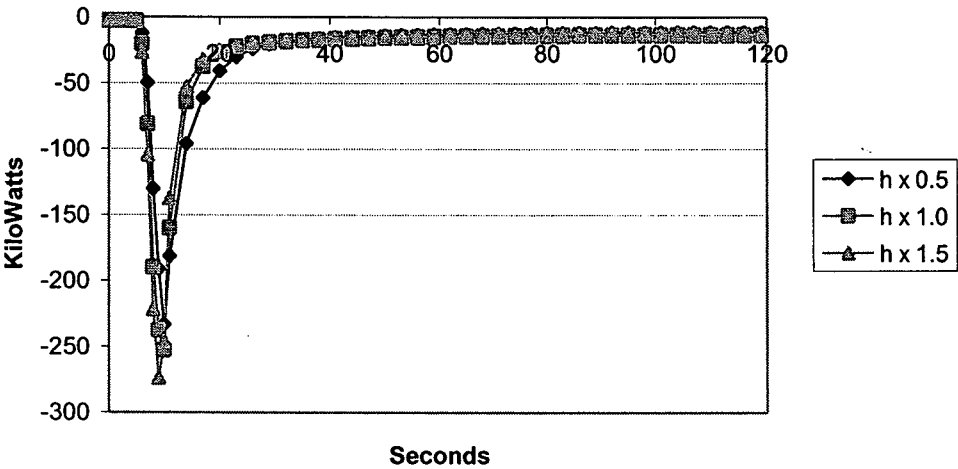


Figure 4.3.4.5 Variations in primary gas path heat flow with variation in primary gas path heat transfer coefficient.

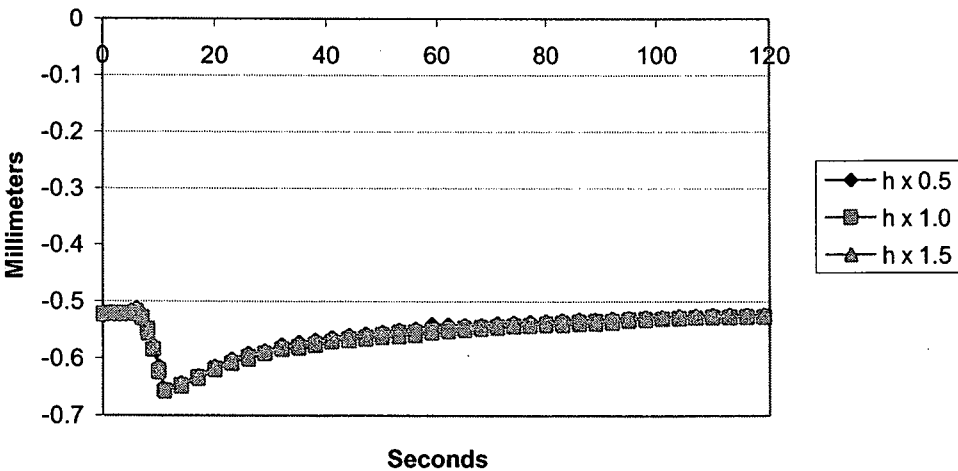


Figure 4.3.4.6 Variations in delta tip gap with variation in primary gas path heat transfer coefficient.

Figure 4.3.4.7 shows the variation in primary gas path heat flow that occurred as a result of varying the mass flow interaction fraction associated with the primary gas path aerofoils (the remainder of the air is assumed to thermally interact with the end walls). Figure 4.3.4.7 suggests that primary gas path heat flow is insensitive to aerofoil interaction fraction. Figure 4.3.4.8 shows what effect these variations had on delta tip gap calculations. We can see from figure 4.3.4.8 that delta tip gap calculations are also not sensitive to changes in aerofoil interaction fraction.

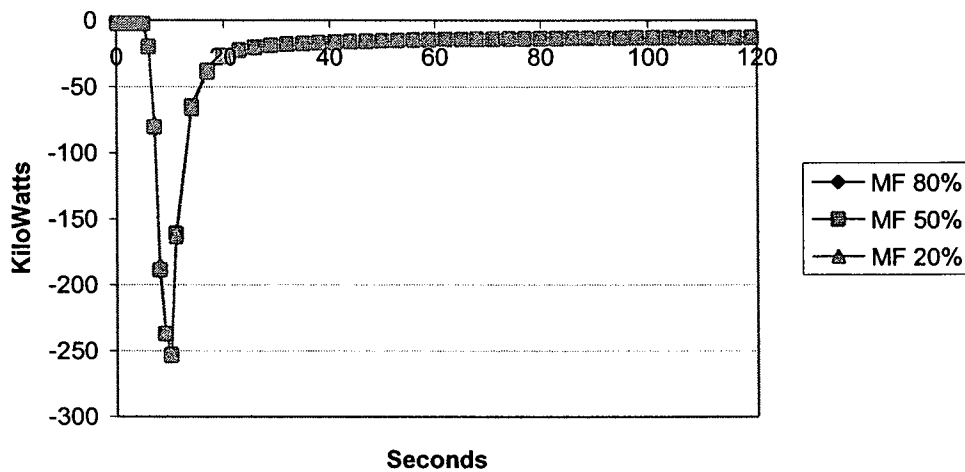


Figure 4.3.4.7 Variations in primary gas path heat flow with variation in primary gas path aerofoil mass flow interaction fraction.

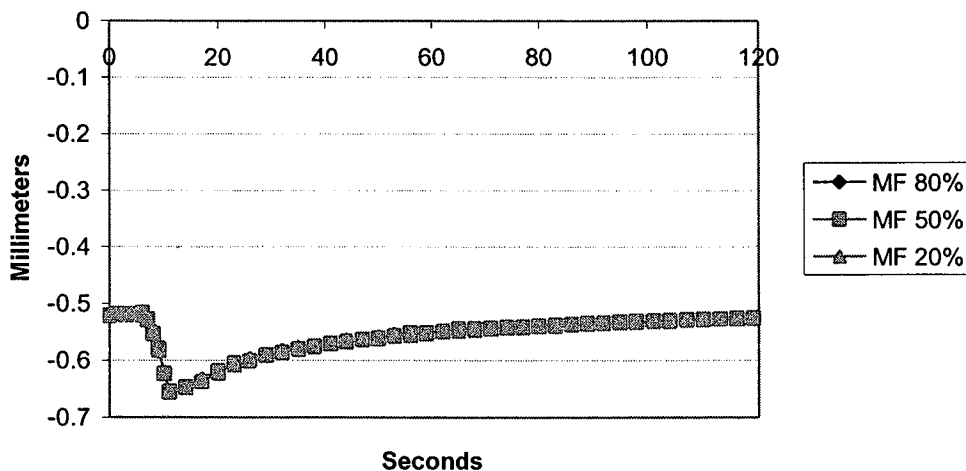


Figure 4.3.4.8 Variations in delta tip gap with variation in primary gas path mass flow interaction fraction.

DISC

Figure 4.3.4.9 shows how the predicted flow of heat into the disc cavity changed as disc cavity heat transfer coefficients were varied. Heat flow in and out of the disc cavity appears to be moderately sensitive to heat transfer coefficient changes, however figure 4.3.4.10 shows that disc growth is only slightly sensitive. This may at first seem contradictory, however we must remember that heat transfer coefficients in the disc cavity are significantly lower than in the primary gas path, and because they are lower, what appears to be a large percentage change in heat flow equates to not a very large absolute change in heat flow in terms of the thermal capacity of the discs. Thus thermal disc growth sees no significant change.

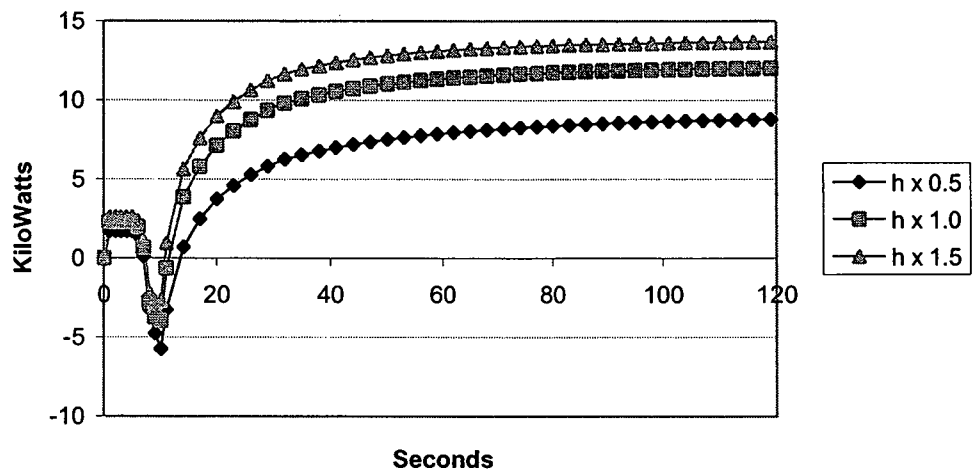


Figure 4.3.4.9 Variations in heat flow into the disc cavity with variation in disc cavity heat transfer coefficient.

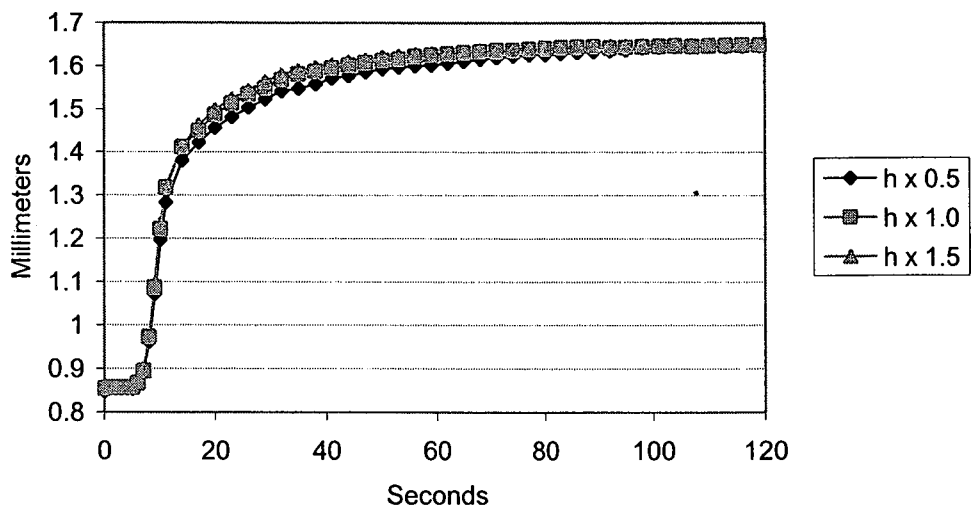


Figure 4.3.4.10 Variations in disc thermal growth with variation in disc cavity heat transfer coefficient.

Figure 4.3.4.11 shows how heat flow in and out of the disc cavity was affected with changes in the fraction of disc bleed mass flow assumed (initially 10%) to go up into the disc cavity, as opposed to passing straight through the component. We can see that heat flow is quite sensitive to mass flow interaction fraction, more sensitive it seems than either the casing or primary gas path. Figure 4.3.4.12 shows that disc growth is also sensitive to this fraction (comparatively speaking).

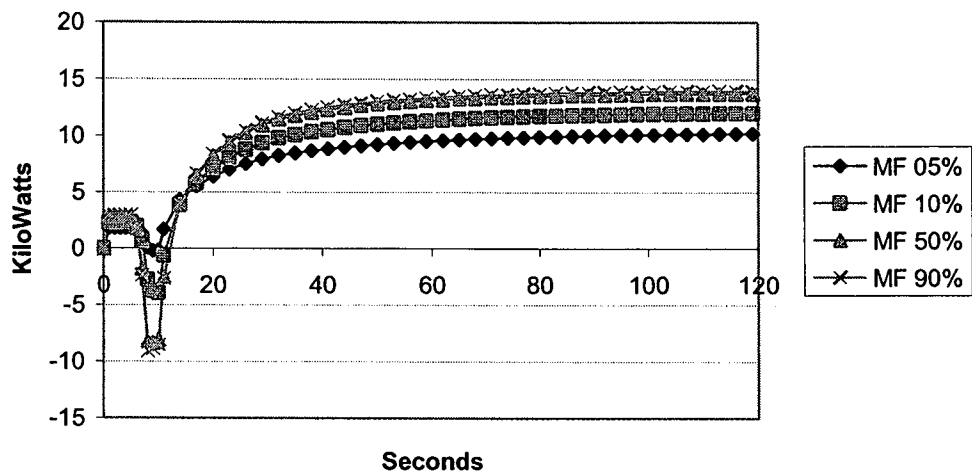


Figure 4.3.4.11 Variations in heat flow into the disc cavity with variation in disc mass flow interaction fraction.

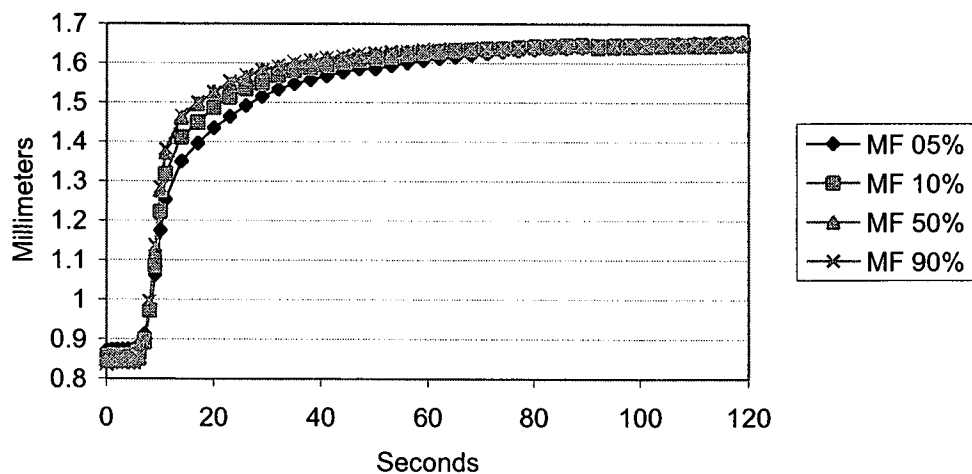


Figure 4.3.4.12 Variations in disc thermal growth with variation in disc mass flow interaction fraction.

It is interesting to see from figures 4.3.4.11 and 4.3.4.12 that as the mass flow interaction fraction approaches the larger value, the system becomes less sensitive (the 50% and 90% lines are practically coincident), and vice versa. When disc bleed air is in short supply its temperature will rise relatively quickly as it passes over the disc and the temperature difference between it and the disc will be smaller, thus restricting heat flow. However as bleed mass flow increases its temperature will rise less due to its larger thermal capacity, until a point is reached where gas temperature gradient is relatively stable along the bleed flow path, and the only way to further raise the heat flow is to increase heat transfer coefficient.

From this study of the streams method single averaged stage model of the Trent 500 HPC, it appears that the casing heat flow and thermal growth is sensitive to heat transfer coefficient, that the primary gas path heat flow is only slightly sensitive to heat transfer coefficient, and that disc cavity heat flow and disc growth are sensitive to both heat transfer coefficient and to mass flow interaction fraction.

4.3.5 MODEL CALIBRATION

Armed with knowledge about the sensitivity of the single averaged stage model to variation in the discretionary variables, we are now in a position to attempt to manipulate the first estimate prediction input file calibrating the answers it produces to those produced by the Lewis method.

Figure 4.3.5.1 shows the arithmetic mean of various growths calculated using the Lewis method for the Trent 500 HPC compared with the same growths predicted using the first attempt single averaged stage streams model. Figure 4.3.5.2 shows the result of streams model input file manipulation of only the discretionary variables, i.e. heat transfer coefficient factors and mass flow interaction fractions.

Inspection of the graphs in the sensitivity study and comparison of figures 4.3.5.1 and 4.3.5.2 shows that the transient response of the system is far more sensitive to the discretionary variables than is steady state behaviour of the system. In this first set of manipulations (discretionary variables only) heat transfer coefficients were finally factored by 0.2 in the casing, 3 on the primary gas path wetted surfaces of the disc, and 0.25 in the disc cavity.

In the second set of manipulations the author made changes not only to the discretionary variables but also to the model itself i.e. geometry and bleed conditions. Figure 4.3.5.3 shows the results of the second set of manipulations. In the second set of manipulations the effective mechanical thickness of the casing was reduced by almost two thirds, and the bleed to the casing was assumed to be HP1 bleed only (as opposed to a combination of HP1 and HP3 air). In addition, the disc diaphragm was further discretised from one ring into two rings, and the dovetail angle was reduced from 45 degrees to 30 degrees.

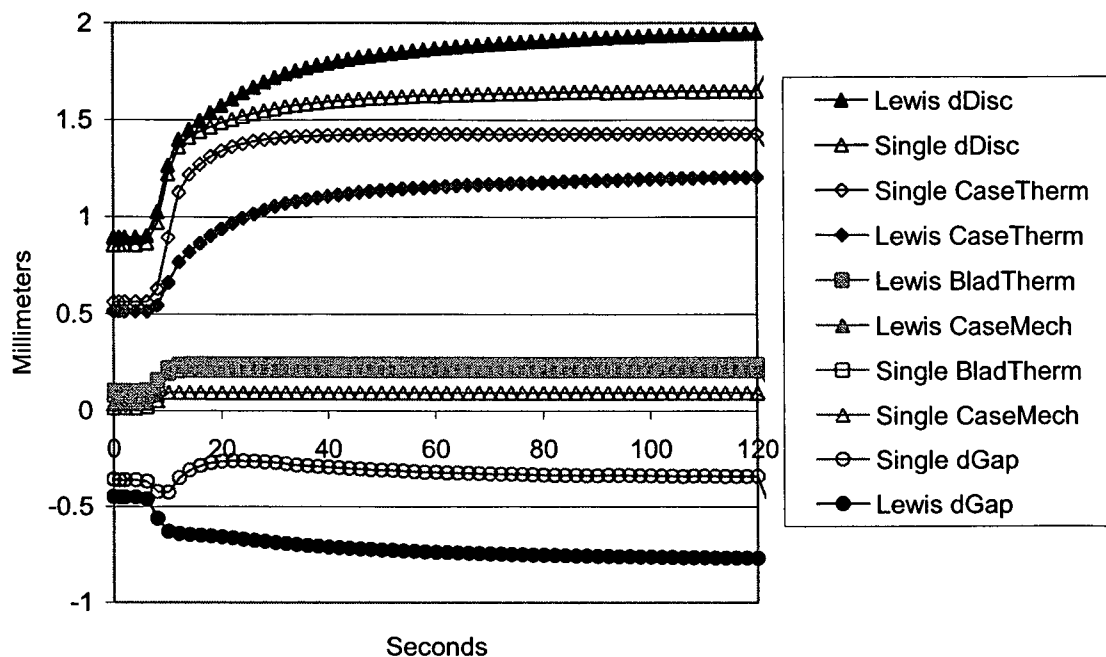


Figure 4.3.5.1 Lewis method delta tip gap components versus single averaged stage streams model.

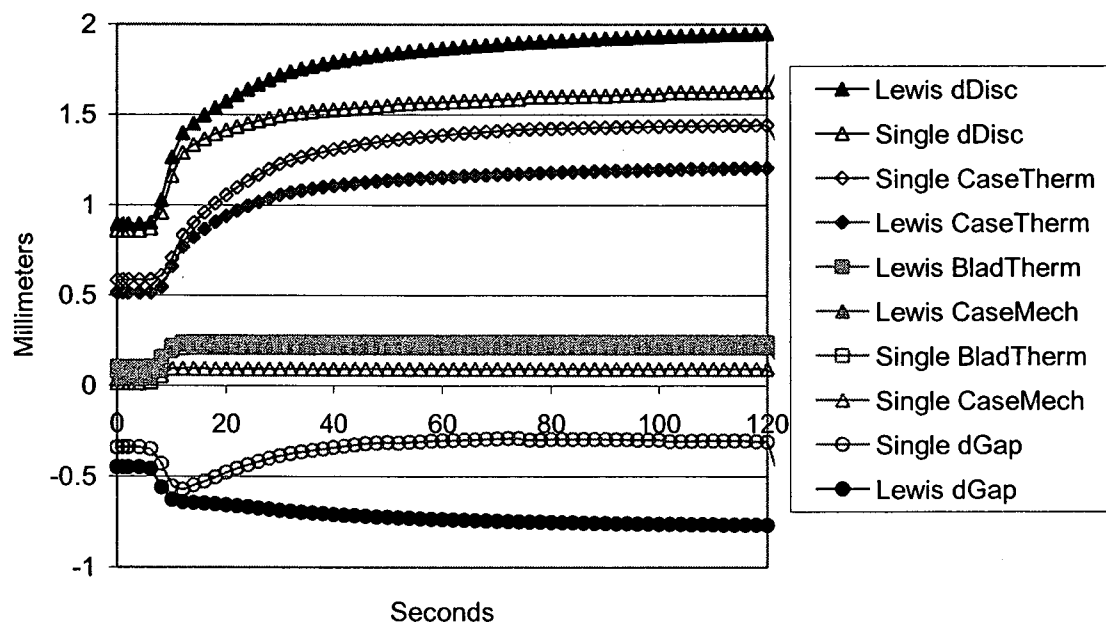


Figure 4.3.5.2 Lewis method delta tip gap components versus single averaged stage streams model with manipulation of only the discretionary variables.

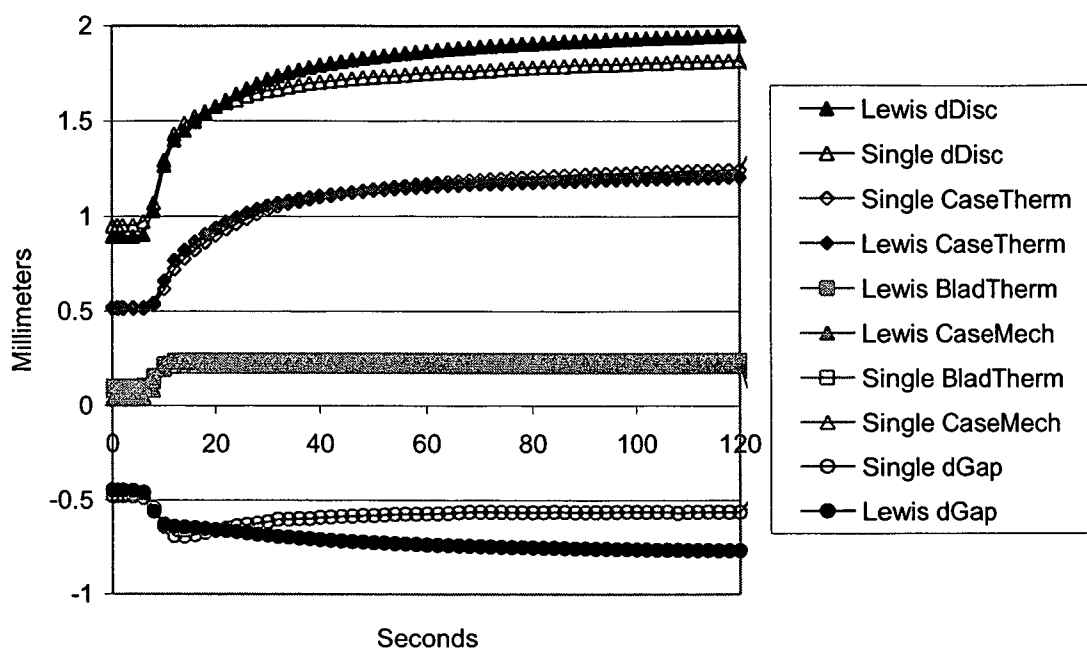


Figure 4.3.5.3 Lewis method delta tip gap components versus single averaged stage streams model with manipulation of the whole model.

After about half a day of manipulation there still existed significant difference between the mean Lewis method delta tip gap and that predicted by the manipulated single averaged streams method (i.e. outside of the project target tolerances). Most of the difference occurred in disc growth. It was difficult to find a single disc geometry and flow that behaved thermally similar to the average of 6 discs all of very different geometry. The casing, being modelled very simply in the streams method, was easier to manipulate and quite good agreement was eventually achieved (see figure 4.3.5.3). The more complicated disc however, took a much longer period of time and at the end was still the major reason for the final discrepancy in delta tip gap.

Figure 4.3.5.4 shows the values produced by the manipulated single averaged stage model for the components of delta tip gap (and average Lewis values) during a slam deceleration. It is interesting to note that the model manipulated to match acceleration data and it does not necessarily match well for a deceleration (or perhaps for other locations within the operational envelope). Figure 4.3.5.4 shows that although the manipulated model gives good deceleration values for the case, the deceleration values for the disc are less impressive, and the value for delta tip gap appears to be diverging from the mean Lewis value as time passes. An interesting observation is that in figure 4.3.5.3 single averaged stage disc growth is below the Lewis value after two minutes, but in figure 4.3.5.4 its stabilised maximum level at the start of the deceleration manoeuvre is above the Lewis value. This re-emphasises the difficulty in trying to match the behaviour of a single stage to the average of a number of stages, since disc thermal growth is controlled as much by conduction as it is by convection.

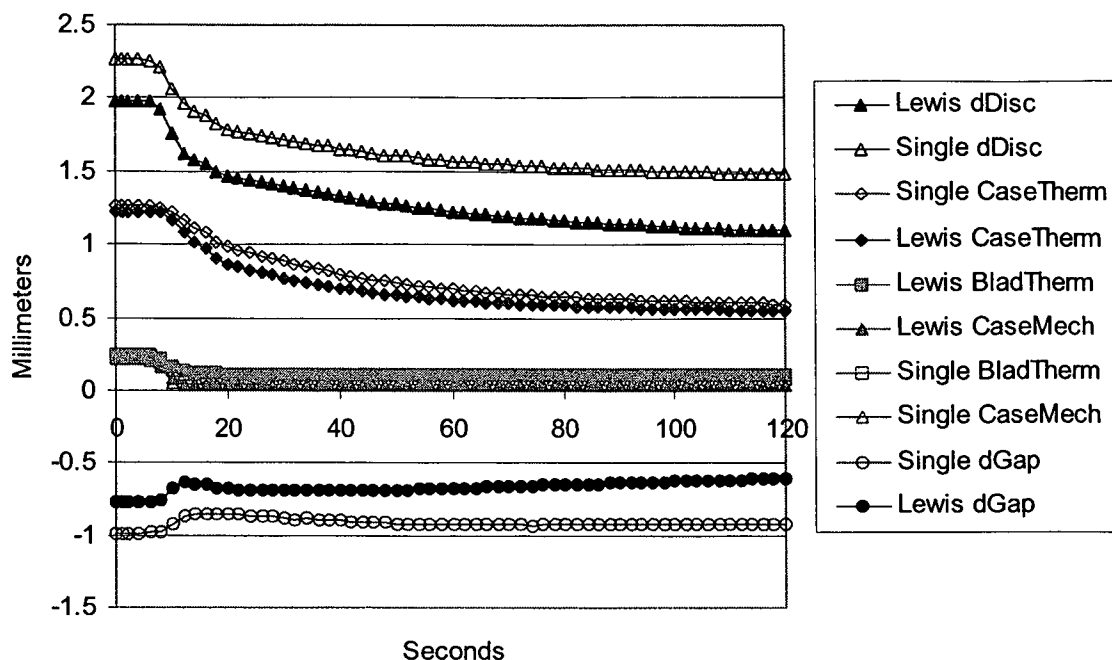


Figure 4.3.5.4 Lewis method delta tip gap components versus manipulated single averaged stage streams model under slam deceleration conditions.

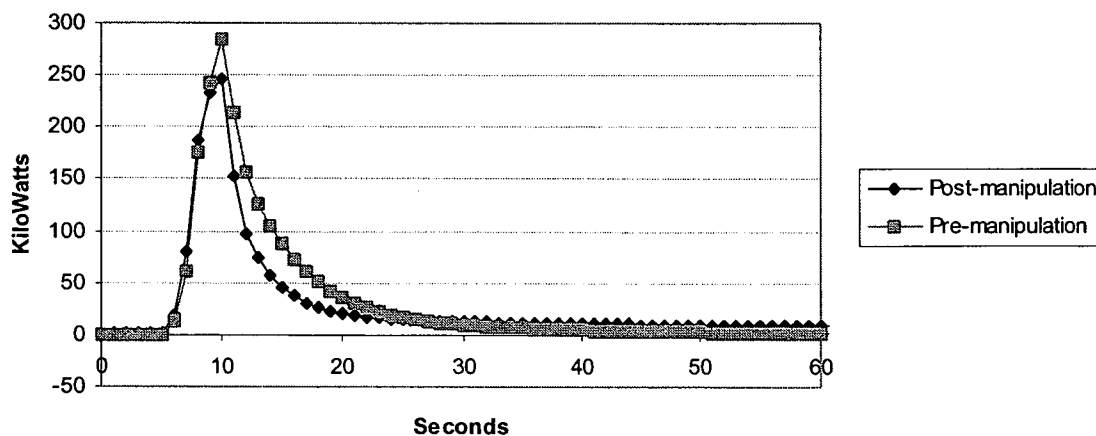


Figure 4.3.5.5 Primary gas path heat flows: first attempt model versus manipulated single averaged stage streams model.

Figure 4.3.5.5 shows graphs for primary gas path heat flow as calculated using the single averaged stage model both before and after manipulation. The changes made to the model during the process of manipulation centred around the secondary gas paths rather than the primary gas path. It is no surprise therefore that changes in primary gas path heat flow calculations as a result of the model manipulations were

not particularly large, even when we consider that a factor of three was used on the heat transfer coefficient for the primary gas path washed parts of the disc. This is further evidence to support the notion that conduction plays just as important a role as convection (i.e. can become a dominant factor) in the transient thermal behaviour of turbomachinery. It also suggests that manipulation of the primary gas path heat flows to match for example test data is likely to be quite difficult since such manipulation is likely to upset the changes made to make the model match up with tip gap data.

4.3.6 USAGE TIMES

Set up and run times for the single averaged stage model of the Trent 500 HPC were not surprisingly approximately one sixth of those associated with the multistage streams model. Perhaps more important than the initial input file set up time was the model calibration time. Half a day's manipulation of the single averaged stage model yielded only a modest improvement in delta tip gap accuracy and an uncertain improvement in the accuracy of heat flow calculations. To match a multistage model would probably require much more than half a day, especially if we were trying to match for tip gaps and heat flow for a variety of transients and over a variety of points within the operational envelope. The manipulation time involved might be tolerable if it only had to be done once, however during new engine development the various components undergo many revisions, and each revision is likely to require a rematch of the model. Under such circumstances the amount of time and effort required to keep models of each of the components updated may become prohibitive.

4.3.7 SECTION CONCLUSION

The purpose of this section was to assess how well a multi-staged component (the Trent 500 HPC) can be modelled using single averaged stage representation.

Subsection 4.3.1 described the challenges encountered constructing a single averaged stage streams. Subsection 4.3.2 showed that single averaged stage heat flow predictions multiplied by the number of stages gave total component heat flows much larger than those predicted by either BR081 or the multistage streams model. Subsection 4.3.3 showed that tip gap predictions from the single averaged stage model were further away from the authoritative Lewis method values than values from the multistage streams model. Subsection 4.3.4 showed that the single averaged stage model was moderately sensitive to its discretionary variables. Subsection 4.3.5 showed that half a day's manipulation of the model itself, as well as the discretionary variables, reduced the error significantly but was still not enough for the model to meet the target tip clearance tolerances. Subsection 4.3.6 stated that set up and run times were fast but warned that long calibration times may become prohibitive.

In this study we have seen that a single averaged stage input file based on assumptions that any engineer might make has produced respectable but not particularly accurate predictions, and that manipulation of the input file to match up with more reliable data has been quite difficult and time consuming. Can a single averaged stage model represent a multistage component? The answer to the question is likely to be yes but only if model set up and calibration times are not important.

4.4 WHOLE ENGINE MODEL

4.4.1 SECTION INTRODUCTION

The purpose of section 4.4 is to demonstrate the effect that the author's heat soakage and tip clearance methods have on whole engine transient performance predictions.

The author constructed multistage streams method model input files of the Trent 500 IPC, HPC, HPT, IPT, and LPT (appendix A), and incorporated the streams method subroutines into an early Rolls-Royce Trent 500 engine performance code, the BD31 version of RRAP. During runs of the code the streams method subroutines were called from within RRAP. The subroutines then fed heat soakage, tip gap change, and bleed temperature updates back into RRAP at each time interval during transient manoeuvres.

In this section three sea level static transients were performed using the version of RRAP with the streams method subroutines incorporated into it. So as to determine whole engine sensitivity to each of the transient phenomena modelled by the streams method (i.e. primary gas path heat soakage, transient tip deltas, and bleed temperature updates), runs were performed with either none, all three, or just one of the transient phenomena models activated. Throughout this section, in order to isolate engine physics from the effect of the presence of a control system, fuel flow to the engine code was scheduled against time rather than to a combination of engine parameters. The effect of the presence of a control system is discussed in appendix B.

The three transient manoeuvres were all run as if the engine was on a test bed, i.e. zero Mach number and zero altitude. The three manoeuvres were a slam acceleration, a slam deceleration, and a hot re-slam. The high power setting was a fuel flow of 22,000 lbs/hr, at which the high-pressure spool ran at approximately 95% speed. The low power setting was a fuel flow of 2,250 lbs/hr, at which the high-pressure spool ran at approximately 70% speed. Handling bleeds were switched off, i.e. shut. Fuel varied parabolically rather than linearly between low and high, and high and low settings, to help avoid numerical instability problems. In the studies presented in this subsection fuel flow changes for the slam acceleration and slam deceleration start at the 5th second and end at the 10th second. For the re-slam the fuel flow was identical to the slam deceleration until the 10th second, and from there until the 15th second it was identical to the acceleration fuel flow profile.

A limited amount of real engine test bed data for a Trent 500 was made available to the author. This section also presents a comparison between test data, RRAP as received by the author, and RRAP with the inclusion of the streams method subroutines. The two versions of RRAP were run to the same day conditions and fuel flow versus time as the engine on the test bed. Further details are given in subsection 4.4.5 below.

During the process of incorporating the streams method subroutines into RRAP, the author met with a few difficulties. Feeding streams answers back into RRAP caused slight problems with stability. Following consultation with Rolls-Royce, the stability

criteria within BD31 were relaxed to get round the problem. Consequently some of the results demonstrate a degree of noise.

4.4.2 SLAM ACCELERATION

The F parameter is defined as the ratio of the heat flow out the fluid to the work into the fluid. Figure 4.4.2.1 shows the F parameter as calculated by RRAP with the streams method subroutines incorporated into it, during the slam acceleration manoeuvre. We can see from the figure that non-zero F values were calculated for the steady state conditions prior to and following the manoeuvre, accounting for steady state heat loss from the primary gas path.

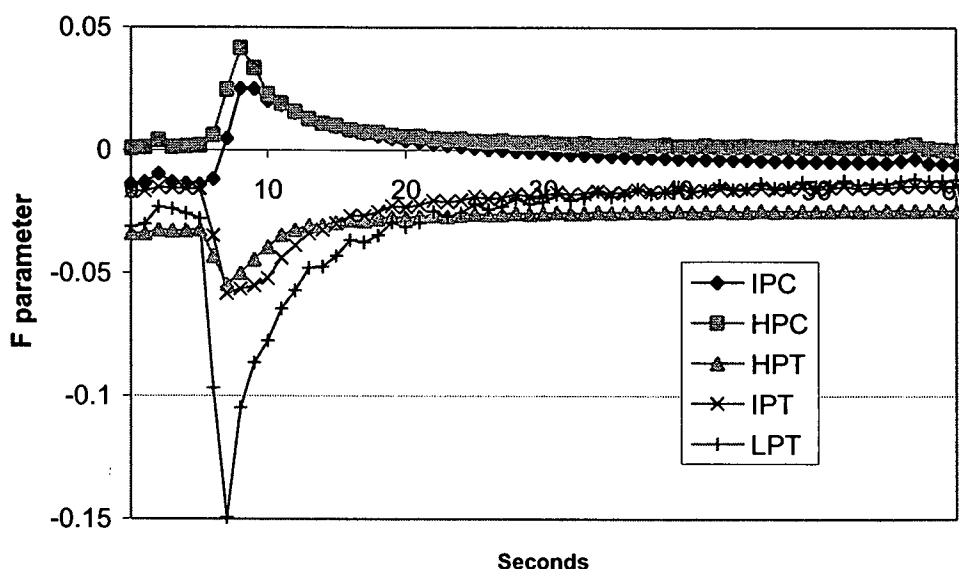


Figure 4.4.2.1 F parameter values as calculated by RRAP with streams for the slam acceleration

In the turbines, steady state heat flow is out of the primary gas path and steady state work flow is out of the working fluid, thus the steady state F value is negative. The HPT demonstrates the largest steady state F values. HPT blade-cooling within the streams method model is the reason for this. Interestingly, just like the turbines, the IPC demonstrates a negative steady state F value. The IPC casing and rotor is ventilated by air drawn from its rearward stages (c.f. appendix A), thus under steady state conditions IPC heat flow is into the primary gas path. This combined with the fact that in the compressors work flow is into the fluid gives the IPC negative steady state F values. The HPC demonstrates a relatively small positive steady state F value. The HPC rotor is ventilated by IPC air, and the casing is ventilated by air bled from behind the first and third of the HPC's six stages (c.f. appendix A). The net result is a positive steady state F value. During the acceleration turbine F values become more negative, whereas compressor F values become more positive. This is because for the turbines work flow is out of the fluid, whereas for the compressors work flow is into

the fluid. The relative sizes of peak F values correspond to the thermal mass, wetted area, and primary gas path temperature swing experienced by each of the components.

In this chapter the tip delta for each component is defined as the transient stage averaged non-dimensional blade tip gap change from cold static minus the steady state stage averaged non-dimensional blade tip gap change from cold static, at the same fuel flow.

Figure 4.4.2.2 shows the non-dimensional blade tip deltas calculated during and following the slam acceleration. Positive values in the figure relate to instantaneous tip gaps that are larger than their equivalent steady state values, whereas negative values relate to tip gaps that are tighter. Using straight exchange rates of delta tip clearance versus component efficiency and flow, the values shown in the figure also equate to efficiency defects in all components, flow reductions in the compressors, and flow increases in the turbines.

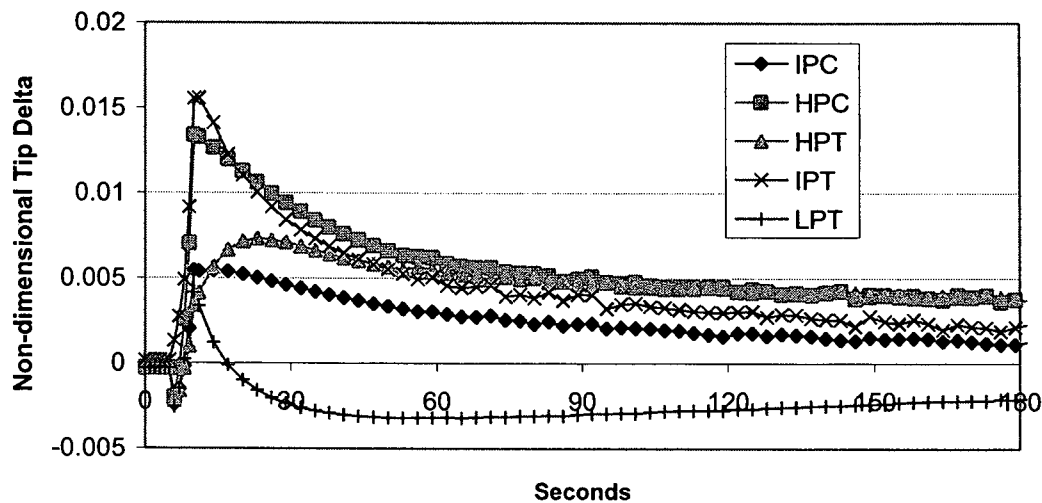


Figure 4.4.2.2 Non-dimensional mean blade tip gap deltas (transient less steady state) during and following the slam acceleration

Figure 4.4.2.2 shows that at the start of the transient tip gaps get tighter for a brief period, and then open up for a number of minutes following the transient. This is due to the time it takes for rotors that have stabilised at the low power setting to warm up and expand to their high power thermal growths. The exception is the LPT whose tips appear to tighten up shortly after the transient. The tip gaps of individual pieces of turbomachinery behave according to the combination of each stage's individual components of tip gap, i.e. rotor growth, blade growth, and casing growth. Appendix A shows charts of components of tip gap for each of the pieces of turbomachinery modelled in this study. A point to note is that for a slam acceleration primary gas path heat soakage is much shorter lived than changes relating to blade tip gaps. F values reach their steady state values in just a few seconds, whereas the blade tips can take 10 minutes or more to reach their steady state values.

Figure 4.4.2.3 shows the streams method calculations for bleed temperature rise from point where the bleed is taken to the point where the bleed is returned to the primary gas path for a small selection of bleeds (the BD31 version of RRAP has 34 separate bleeds). We can see from the figure that for some bleeds the temperature rise is larger and takes longer than it does for others. The IP8 bleed to the LPT entry demonstrates a particularly large temperature rise, over 100 Kelvin, that takes more than 2 minutes to reach its steady state value, whereas the HP6 bleed to the HPT entry varies very little along its journey from source to sink.

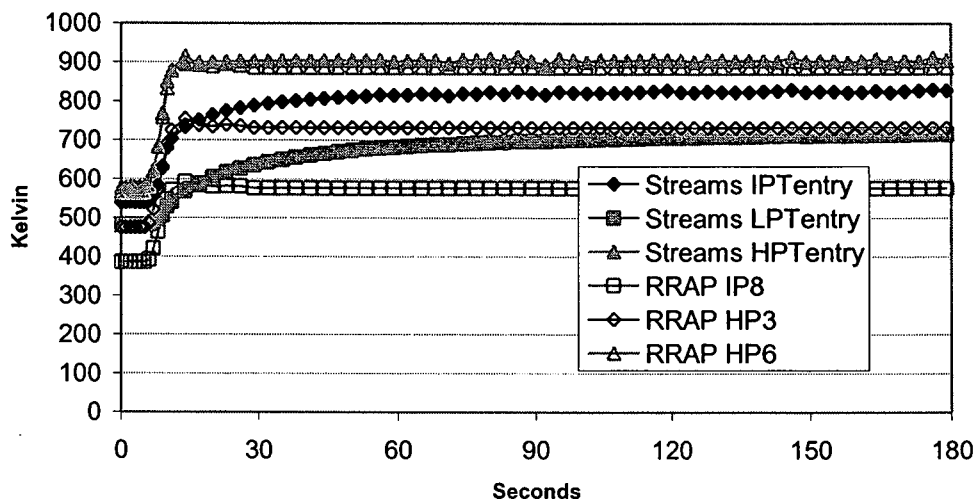


Figure 4.4.2.3 **Bleed inlet and outlet temperatures during and following the slam acceleration**

Figures 4.4.2.4 to 4.4.2.6 show how each of the transient phenomena modelled by the streams method subroutines (i.e. primary gas path heat soakage, blade tip deltas, and bleed temperature updates) affects spool speeds during the slam acceleration individually and in combination. The figures show results for spool speeds and also for rates of change of spool speed. A very important observation is that the inclusion of transient phenomena models significantly affects not only the transient behaviour of the code but also the steady state behaviour of the engine code. This is clearer in some parts of the figures than in others due to the large amount of noise present in some of the results. This finding sheds doubt upon the assumption frequently made in steady state performance calculations, that the compression and expansion take place adiabatically. Perhaps the gas turbine performance community might rethink what physics to include in some of its steady state performance simulation codes.

For the high-pressure spool the effect of bleed temperature updating appears to reduce speed at low power, but less so at high power, where the individual phenomena appear to have less of an effect than they do when they are combined. During the acceleration it appears that acceleration rates are affected again by bleed temperature updating, and by a combination of the phenomena.

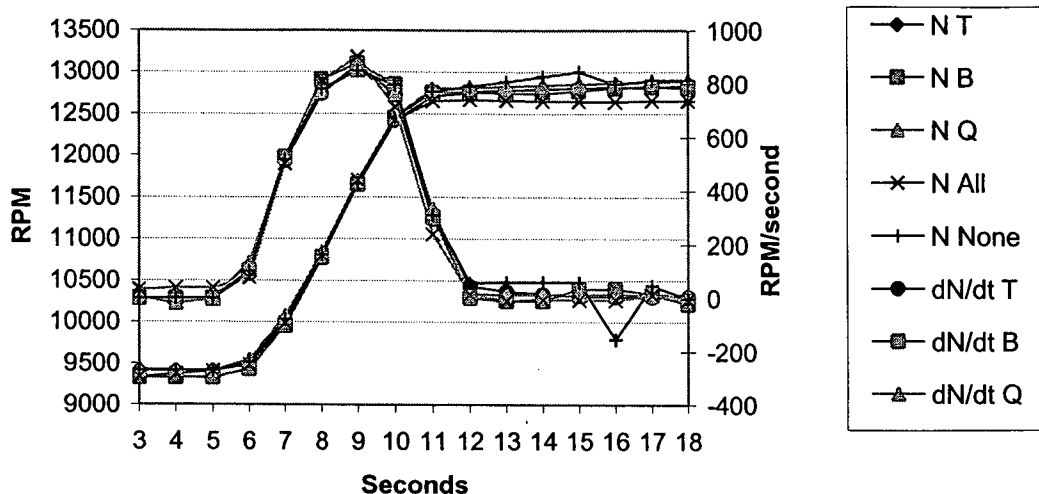


Figure 4.4.2.4 **HIGH PRESSURE SPOOL** speed and rate of change of speed during the slam acceleration. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

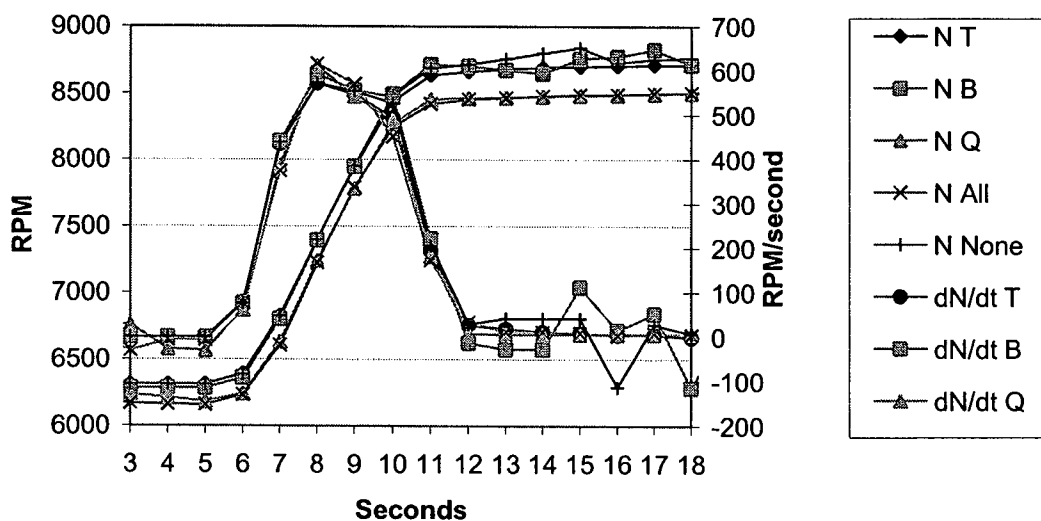


Figure 4.4.2.5 **INTERMEDIATE PRESSURE SPOOL** speed and rate of change of speed during the slam acceleration. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

For the intermediate-pressure spool primary gas path heat soakage has a very clear effect on speed, not only reducing speed at low and high power, but also reducing the rate of acceleration at the start of the manoeuvre. The effect of heat soakage appears to be so large as to dominate the response of the combined phenomena result, which is almost exactly the same as the result for heat soakage alone. The effect of tip gaps and bleed temperature updating are relatively small, although bleed temperature does show some speed reducing effect at low power.

The low-pressure spool shows a similar sensitivity to primary gas path heat soakage as the intermediate-pressure spool. It too dominates the combined phenomena response such that it is almost exactly the same as for heat soakage alone. Tip deltas and bleed temperature updating appear to make little change to the results, apart from to the acceleration rates toward the end of the manoeuvre.

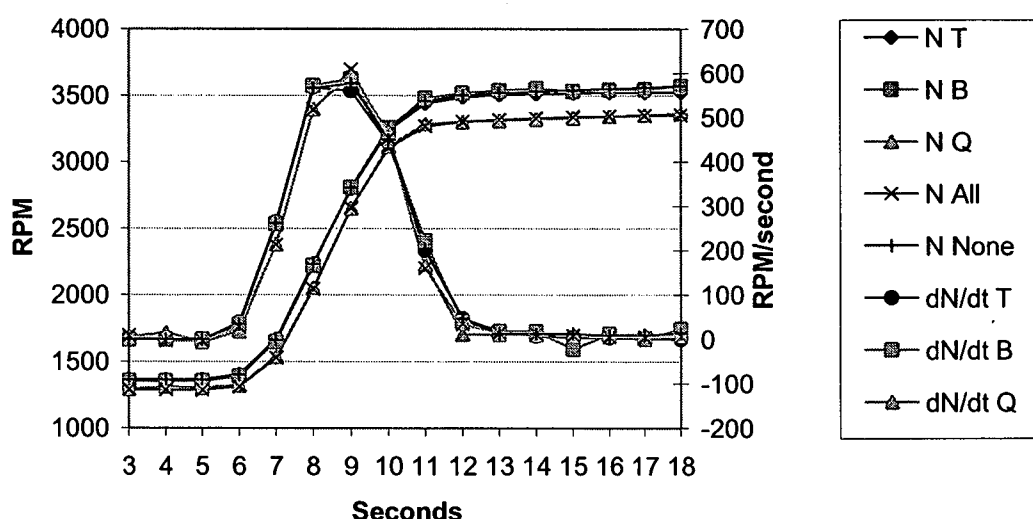


Figure 4.4.2.6

LOW PRESSURE SPOOL speed and rate of change of speed during the slam acceleration. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

It appears from the study of spool speeds that the high-pressure spool is sensitive to bleed temperature updating and the intermediate and low-pressure spools are sensitive to primary gas path heat soakage. The lack of sensitivity of the intermediate and low-pressure spools to bleed temperature updating is a little surprising given the large temperature rises occurring in bleeds returning to the primary gas path at the IPT and LPT entry. This may be due to the fact that despite the large temperature rise, these bleeds represent quite a small fraction of primary gas path flow. This is not the case in the HPT, where cooling bleed flows can be up to a few percent of primary gas path flow. In spite of the relatively low temperature rise shown for the bleed servicing entry to the HPT, its relatively large mass flow may be the reason for the high-

pressure spool speed's sensitivity to bleed temperature updating. Perhaps the least potent of the three phenomena in how it affects spool speed is the effect of transient blade tip deltas. But this is not necessarily the case when it comes to looking at the effect that each of the three phenomena has on the transient working line of the high-pressure compressor.

Figure 4.4.2.7 shows the effect that each of the transient phenomena modelled in the streams method has on the transient trajectory of the HPC working line, from the 5th second to the 15th second of the manoeuvre. Unfortunately a fair comparison is difficult because in addition to the noise in the results each phenomenon individually and in combination affects the starting and finishing steady state operating points of the compressor. Matching the starting and finishing operating points across the runs would require altering the starting and finishing fuel flow for each run. This could be done in an attempt to match spool speeds, mass flows, temperatures, and pressures, although achieving this result would be very unlikely. As a result, the transients manoeuvres would all have different starting and finishing temperature profiles for given spool inertias and would be unlikely to produce any more fair a comparison than a match on fuel flow, EPR, or any other single engine parameter.

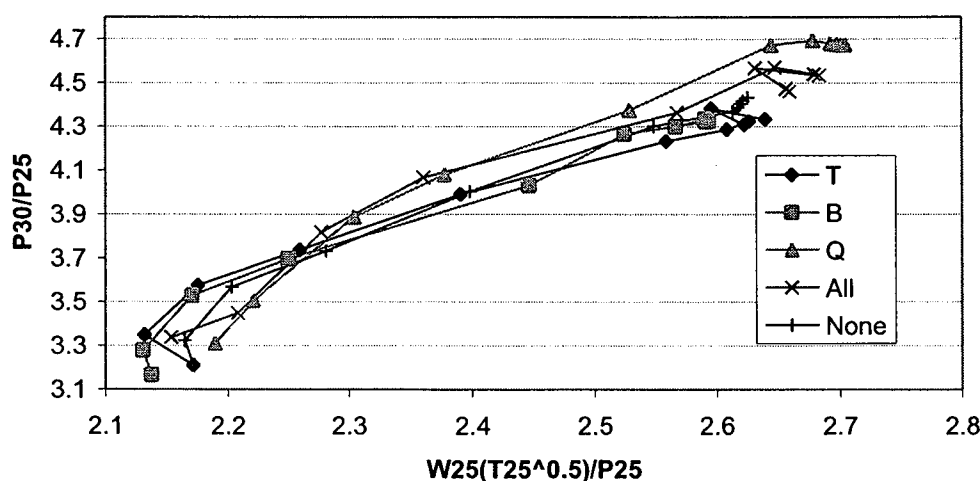


Figure 4.4.2.7 HPC transient working line trajectories during the slam acceleration, 5th second to 15th second. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

From figure 4.4.2.7 it appears that heat soakage suppresses the initial pressure ratio rise at the start of the transient. The second point on the 'Q' line and 'All' line of figure 4.4.2.7 is above and to the right of the first point, whereas for the other lines it is above and to the left. Perhaps therefore what is actually happening is an increase in non-dimensional speed, which is likely if HPC inlet temperature is suppressed. The effect of bleed temperature updating is less clear. It appears from the figure that the starting point is at a lower non-dimensional speed than the others. Indeed,

interrogation of the output files showed this to be the case. This is not a surprise because although at the start of the manoeuvre HPC inlet temperature was not particularly low, high-pressure spool speed was (c.f. figure 4.4.2.4). Toward the end of the manoeuvre the speed lines appear to move rightward more quickly than they do for the other runs (c.f. figure 4.4.2.4). An interesting feature of figure 4.4.2.7 is how the run with none of the transient phenomena modelled starts at the same point as the run with only tip deltas switched on. This is because at the start of the transient there should be no difference between instantaneous and steady state tip gaps, and therefore no transient tip delta effects. As the manoeuvre proceeds it appears that the tip deltas only run moves higher on its speed lines, i.e. at higher pressure ratios. Figure 4.4.2.8 shows why this may be so - for the first couple of seconds into the manoeuvre HPC tip clearances are tighter, and therefore the compressor is more efficient, than at steady state. And the same applies too for the HPT. A higher spool efficiency would imply a higher pressure ratio, and indeed that is what we see on figure 4.4.2.7. In addition, the converse appears to be true for the latter part of the manoeuvre - figure 4.4.2.8 shows that at the end of the acceleration a HPC efficiency defect of more than one percent exists. Although not modelled in this study the large tip openings relative to steady state are likely to lower the surge line quite significantly toward the end and immediately following the manoeuvre.

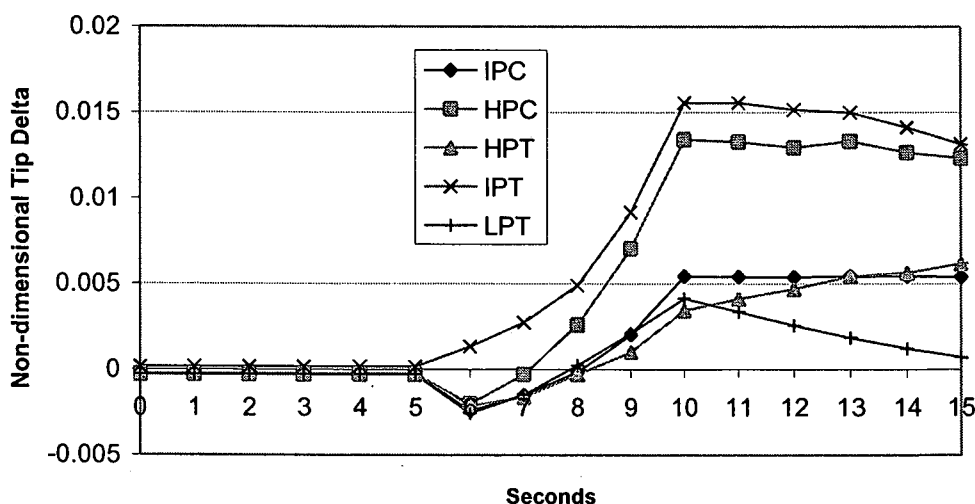


Figure 4.4.2.8 Non-dimensional mean blade tip gap deltas (transient less steady state) during the slam acceleration

4.4.3 SLAM DECELERATION

Figure 4.4.3.1 shows F values calculated during the slam deceleration manoeuvre. Although steady state values are of the same sign at the start of the slam acceleration, peak values are of opposite sign. This is due to the fact that during a slam deceleration heat pours out of the metal engine carcass into the working fluid, i.e. in the opposite direction to the heat flow occurring during a slam acceleration. Another

thing to note is that peak changes in F values are all much higher than for the slam acceleration. Furthermore, for the slam deceleration F values take much longer to decay than they do for the slam acceleration. This is because at the end of a slam deceleration, airflows are relatively small. Airflows are the means by which engine carcass thermal energy is convected away. Smaller airflows mean slower engine carcass cooling. Moreover, at the end of a slam deceleration component powers are relatively small, and so for a given amount of heat flow, F values are relatively large.

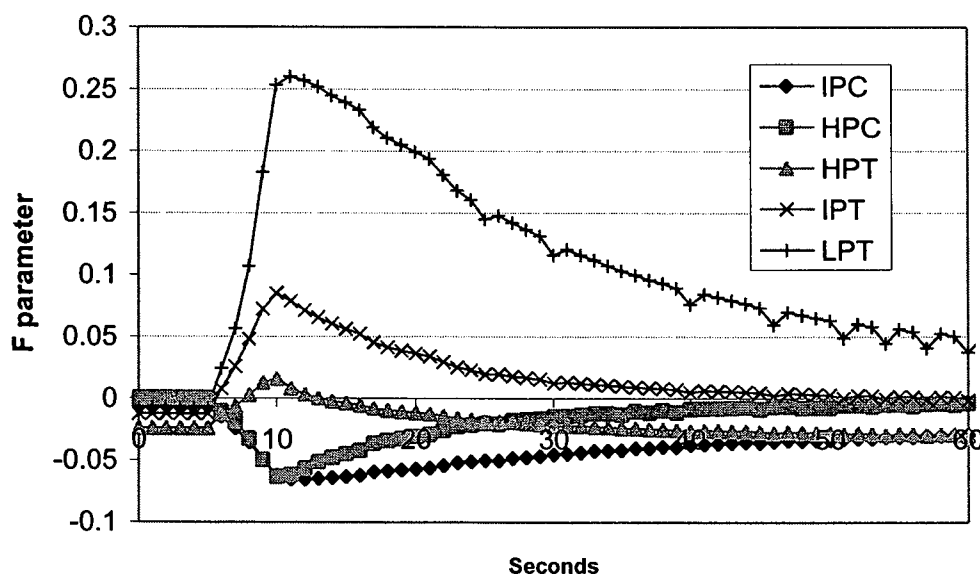


Figure 4.4.3.1 F parameter values as calculated by RRAP with streams for the slam deceleration

Figure 4.4.3.2 shows the non-dimensional tip deltas calculated during and following the slam deceleration manoeuvre. Response rates for the tip deltas are comparable to those during the slam acceleration, even if they are slightly slower. The peak values for each component are different from their values during the slam acceleration. Tip delta values are generally of opposite sign during the slam deceleration to the values calculated during the slam acceleration. In addition deceleration values do not demonstrate the spike at the onset of the manoeuvre that acceleration values do. The spike is due to centrifugal growth of the rotor, which occurs almost instantaneously as spool speeds are increased. Centrifugal growth occurs much faster than thermal growth during the acceleration and thus we see the brief tightening of the tip gaps relative to their steady state values in figures 4.4.2.2 and 4.4.2.8, following which the tip gaps open up for a while until the rotor reaches its steady state thermal growth. During deceleration even though centrifugal shrinkage takes place faster than thermal shrinkage and there is a physical opening of the tip gaps, the fact that the rotors are still warm means that tip gaps are actually tighter during and following the manoeuvre than their steady state values corresponding to the given instantaneous fuel flow. Figure 4.4.3.2 shows that following the slam deceleration, the streams method predicts a risk of blade tip rubbing, especially in the HPC and IPT.

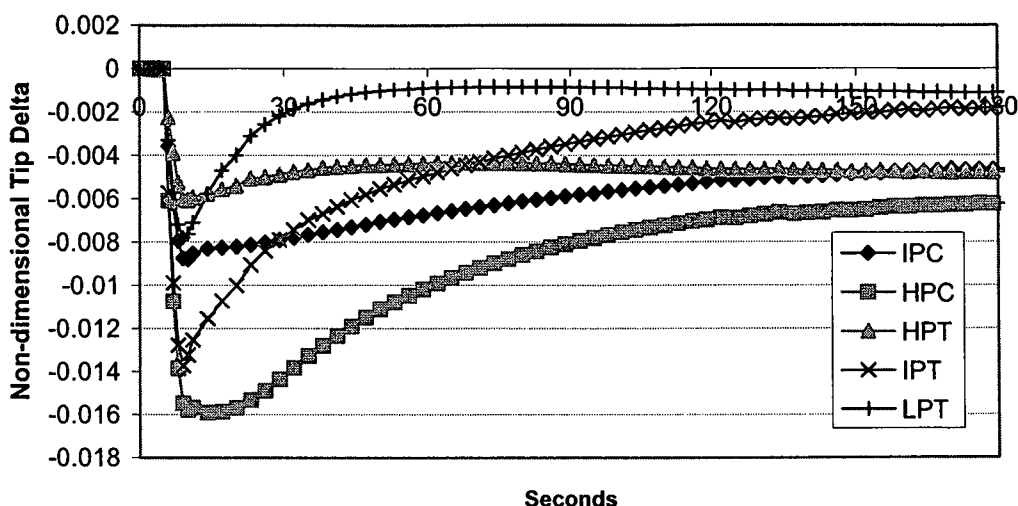


Figure 4.4.3.2 Non-dimensional mean blade tip gap deltas (transient less steady state) during and following the slam deceleration

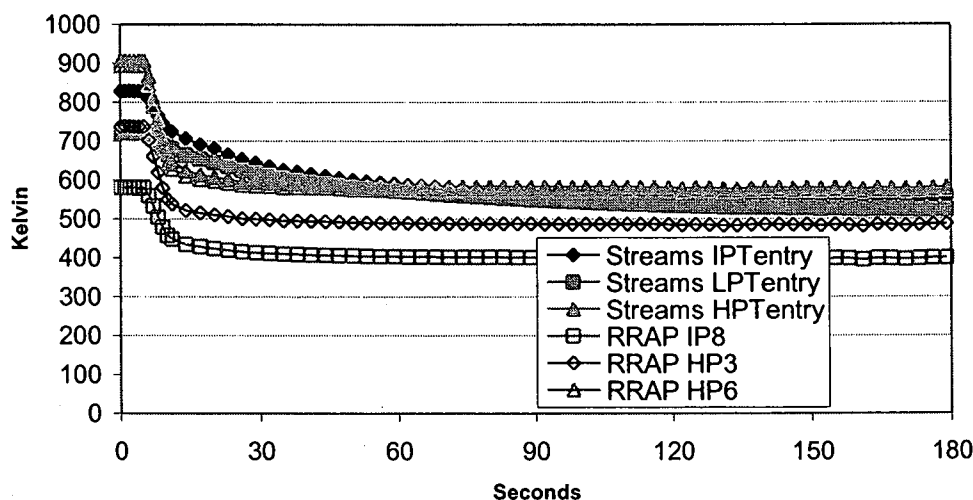


Figure 4.4.3.3 Bleed inlet and outlet temperatures during and following the slam deceleration

Figure 4.4.3.3 shows results of calculations for bleed temperature updates for three of the bleed flows in the engine model. We can see from a comparison against figure 4.3.2.3 that following the slam deceleration the response times of bleed temperatures are slightly longer than those following the slam acceleration. This is a result of the lower bleed mass flows as well as lower disc cavity rotational speeds and thus lower heat transfer coefficients, all of which combine to produce lower heat transfer rates.

Figures 4.4.3.4 to 4.4.3.6 show the spool speeds and rates of change of spool speed calculated for runs with each of the phenomena switched on and off and in

combination for the slam deceleration. We can see for the high-pressure spool that the effect of bleed temperature updating appears to have the largest effect on steady speed at high and low power, although its effect upon deceleration rate appears to be quite small. During the deceleration a more important effect seems to be primary gas path heat soakage, which also raises the level of spool speed at the end of the manoeuvre. The effect of tip delta appears to lessen deceleration rates during the early stages of manoeuvre. Spool speed prior to the manoeuvre exhibit quite a degree of noise, especially for the heat soakage run. The slow deceleration rate of the combined effect run is probably more due to noise than to any real interaction between the models. The figure shows that although fuel flow reaches its minimum level at the 10 second mark, high pressure spool speed does not, and still takes a few seconds longer to decelerate. During spool down the rotational energy of the rotor is absorbed by the compressor causing a spool power imbalance that decelerates the spool. However, the heat pouring out of the engine carcass during deceleration acts like extra fuel in the turbines and slows the process of spool deceleration, particularly at the end of the manoeuvre.

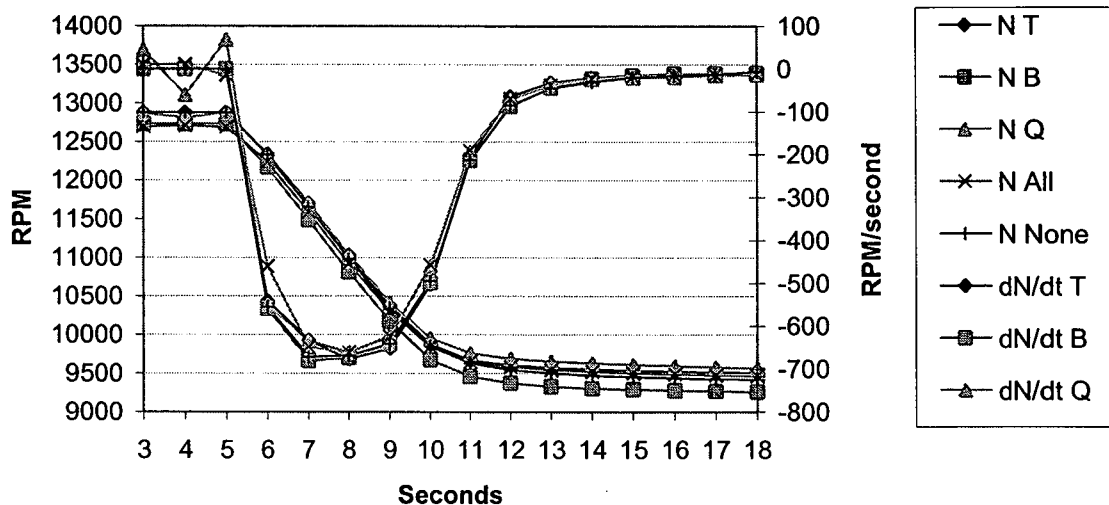


Figure 4.4.3.4 HIGH PRESSURE SPOOL speed and rate of change of speed during the slam deceleration. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

Figures 4.4.3.5 and 4.4.3.6 show that just as was the case for the slam acceleration, the largest effect on intermediate-pressure and low-pressure spool speed, both at steady state and transiently, is primary gas path heat soakage. However, for the acceleration the spool speeds were slower both before and after the manoeuvre, whereas for the deceleration spool speeds were lower before the manoeuvre but higher immediately following the manoeuvre. The smaller range of speed change

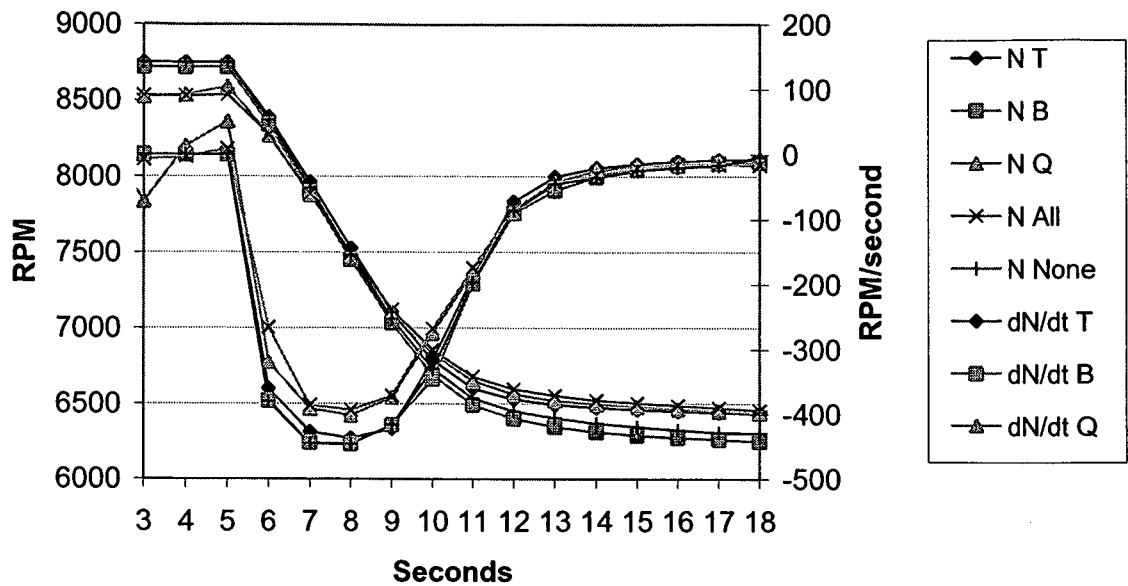


Figure 4.4.3.5

INTERMEDIATE PRESSURE SPOOL speed and rate of change of speed during the slam deceleration. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

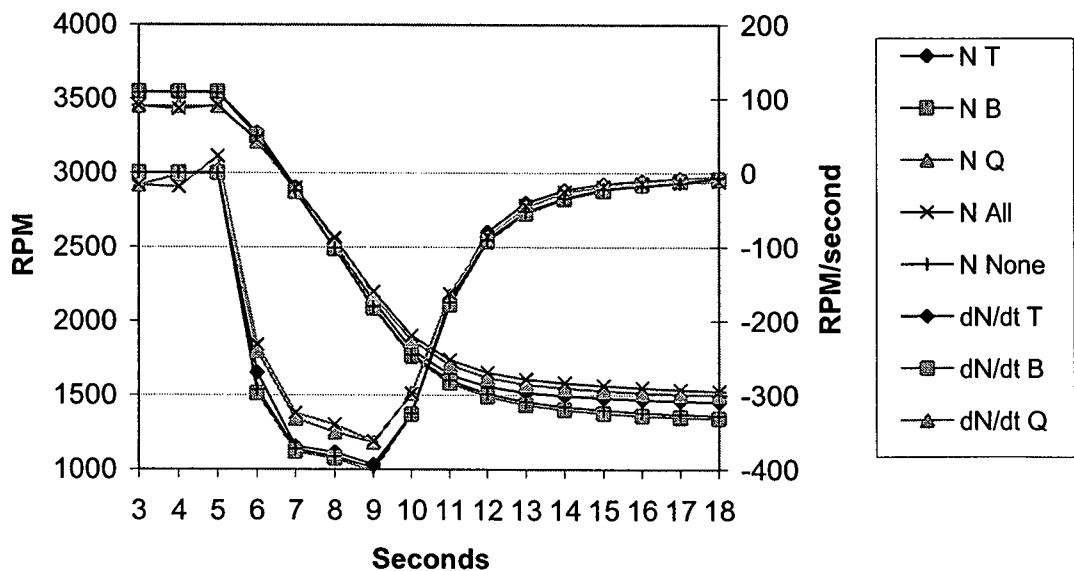


Figure 4.4.3.6

LOW PRESSURE SPOOL speed and rate of change of speed during the slam deceleration. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

due to sluggish spool down contributes to the smaller deceleration rates demonstrated for heat soakage runs for the intermediate and low-pressure spools. Heat soakage again dominates the combined phenomena results, however it appears for the deceleration that the effect of tip deltas also has a significant effect raising the post manoeuvre intermediate and low-pressure spool speeds for the combined phenomena run even higher than for the heat soakage run alone. Perhaps tighter tip clearances leading to higher component efficiencies are the reason for the boost in spool speed following the manoeuvre, especially in the LPT which sees what is effectively re-heat for more than a minute after the deceleration transient.

Figure 4.4.3.7 shows the transient working line for each run from the 5th second to the 15th second of the slam deceleration. For the same reasons as for the slam acceleration, starting and finishing points are all different. The primary gas path heat soakage trace starts higher and finishes lower than for any of the other runs, with the exception of the all-phenomena run, which includes heat soakage anyway. At the start of the manoeuvre HPC spool speeds were almost identical for the heat soakage run and the no-phenomena run, however heat soakage inlet temperature was about 15 K lower, the result being a higher non-dimensional speed. At the 15th second non-dimensional speed for the heat soakage run was below that of the no-phenomena run due to a 30 K inlet temperature overshoot lasting about a minute and a half. The bleed temperature update run showed very little difference to the no-phenomena run, apart from the fact that the whole line was shifted slightly down and to the left, due to its slightly lower physical speed (c.f. figure 4.4.3.4), but very similar HPC inlet temperature.

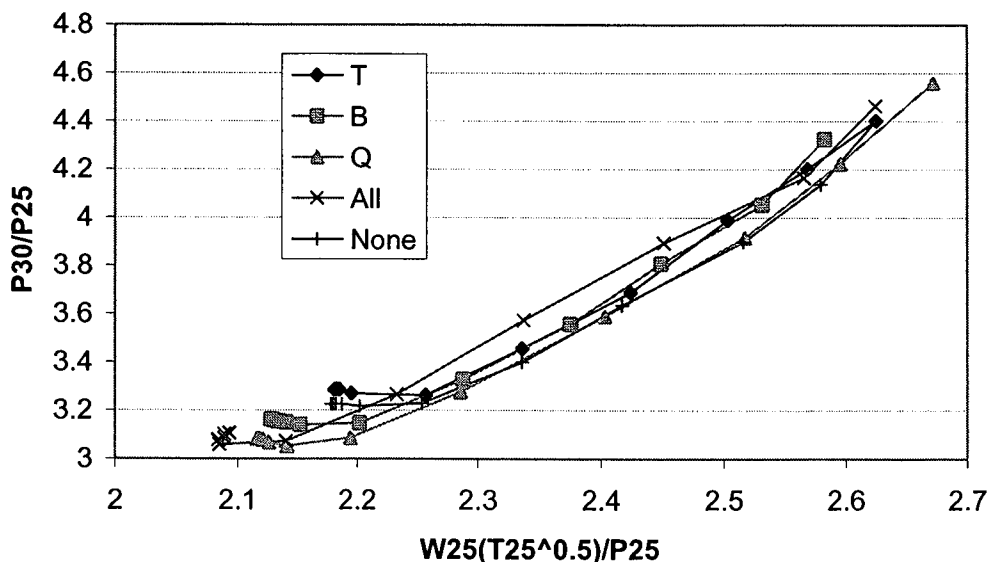


Figure 4.4.3.7

HPC transient working line trajectories during the slam deceleration, from the 5th to 15th second. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

The tip delta run was closer to the surge line than the no-phenomena run for the entire manoeuvre, particularly so just after the start of the deceleration. Tip clearances during the slam acceleration are tighter than their corresponding steady state values. The result is better component efficiencies, which is possibly why pressure ratios for the tip delta run are all higher than for the no-phenomena run. The all phenomena run shares the same relatively high pressure ratio characteristics as the tip deltas run, but also the relatively high and low starting and finishing points as the heat soakage run.

4.4.4 HOT RESLAM

Figure 4.4.4.1 shows intercooling factor F as calculated for each component during the hot reslam manoeuvre. The reader should remember that the hot reslam consisted of a slam deceleration from the 5th second until the 10th second, and a slam acceleration from the 10th second to the 15th second. The figure shows that F values calculated for the hot reslam are essentially identical until the 10th second as those produced for the slam deceleration, however, from the 10th to the 15th second they are very different to those produced for the slam acceleration. Firstly they are of opposite sign for the first part of the re-acceleration, and secondly they appear to stabilise more quickly than those of the slam acceleration. The reason for this becomes clear if we remember that during the reslam the engine metal is hotter than it is for the cold slam acceleration, and so during much of the re-acceleration heat flows in the opposite direction.

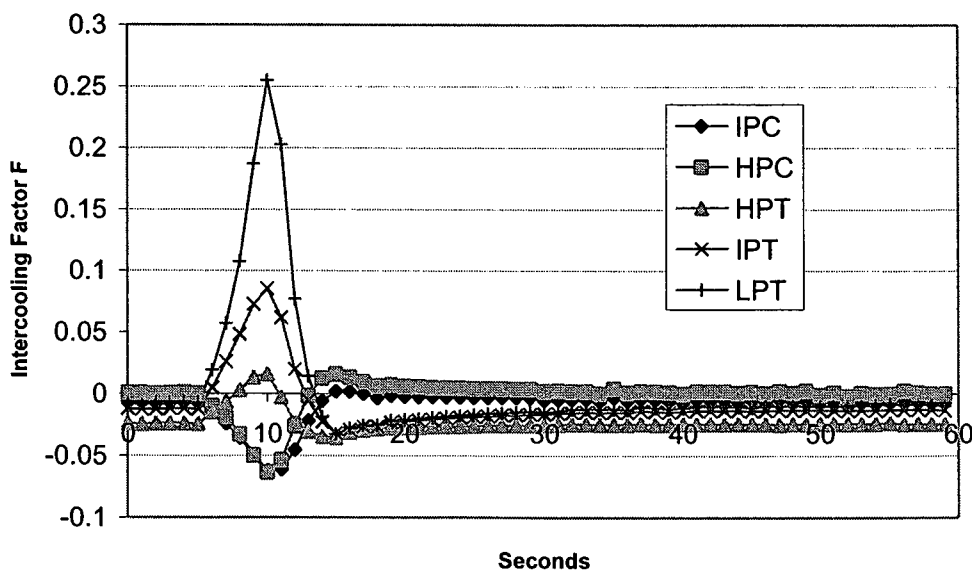


Figure 4.4.4.1

Intercooling factor values as calculated by RRAP with streams for the hot reslam

Figure 4.4.4.2 shows the non-dimensional tip deltas calculated for the hot reslam manoeuvre. As for the F values, tip deltas are essentially the same as the slam deceleration from the 5th to the 10th second, but are significantly different from the slam acceleration from the 10th to the 15th second. Comparing against figure 4.4.2.2, figure 4.4.4.2 shows that whereas cold slam acceleration tip deltas are relatively large positive values that persist for a number of minutes, hot re-acceleration tip deltas only reach modest positive values and only really at the end of the manoeuvre. For most of the manoeuvre they are significantly more negative. Also, hot re-acceleration tip deltas reach steady state values much more quickly than cold slam acceleration tip deltas. The reason for this is that rotor discs are already warm and so need less time to re-stabilise following the manoeuvre.

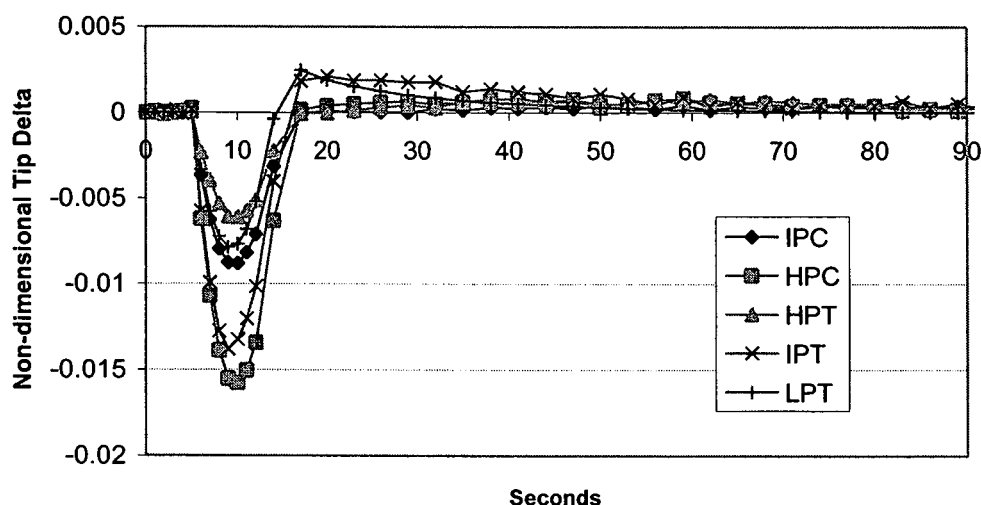


Figure 4.4.4.2 Non-dimensional mean blade tip gap deltas (transient less steady state) during and following the hot reslam

During the re-acceleration part of a hot reslam manoeuvre an engine is more likely to surge than it is during the equivalent cold slam acceleration. It is universally accepted that surge lines are affected strongly by tip clearance, the larger the clearance the worse the surge margin. However, from figure 4.4.4.2 we can see that during hot re-acceleration tip clearances are tighter than they are for the equivalent cold slam acceleration (c.f. figure 4.4.2.2). Wider than normal tip clearances are usually associated with surge, but since hot reslam tip clearances are relatively tight it appears that hot reslam surge is more to do with things other than tip clearance.

Positive F factors in the turbines effectively equate to reheat, whereas positive F values in the compressors effectively equate to intercooling, and vice versa. If we compare what happens to the F factors during cold acceleration and hot re-acceleration, we can see some important thermodynamic differences. During the cold acceleration both compressor and turbines are intercooled, but during the hot acceleration both compressors and turbines are reheated. This is a very obvious statement – a cold engine takes heat out of the primary gas path and a hot engine puts

heat into it. This means is that during a cold acceleration the compressors over-perform and the turbines under-perform, whereas during a hot acceleration the compressors under-perform and the turbines over-perform. Where the result of this effect ought to become apparent is in transient spool power imbalances and therefore spool acceleration rates. If we compare figures 4.4.2.4 to 4.4.2.6 for the cold acceleration to figures 4.4.4.4 to 4.4.4.6 (below) we can see that the effect of heat soakage should be, according to the argument above, to increase re-acceleration rates. And this is what we see. The effect is more clear for the high-pressure spool but less obvious for the intermediate and low-pressure spools, perhaps because the fan and IPC don't see any difference in inlet temperature with or without the heat soakage methods switched on whereas the HPC does.

Figure 4.4.4.3 shows that bleed temperatures during the deceleration part of the manoeuvre are essentially the same as for the slam deceleration, however during and following the re-acceleration part of the manoeuvre, those bleeds that would otherwise take perhaps minutes to reach steady state temperatures actually reach steady state almost instantaneously.

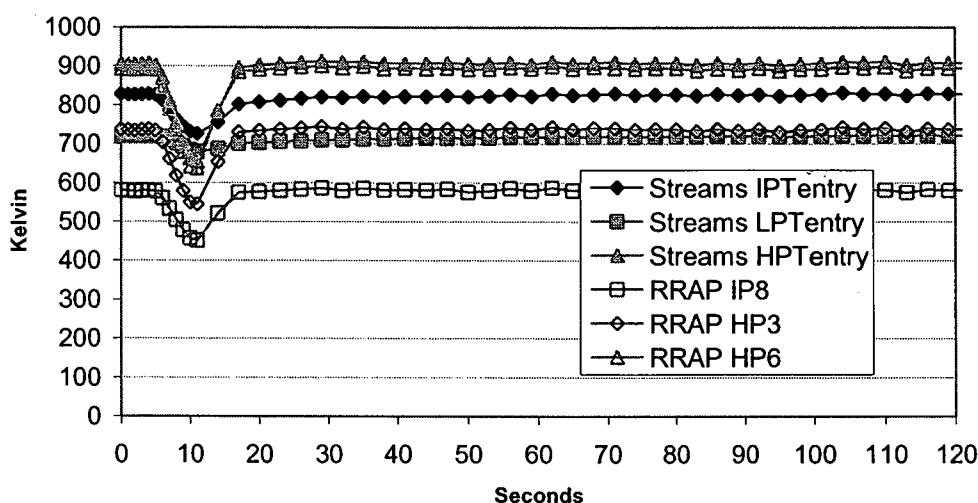


Figure 4.4.4.3 Selected bleed inlet and outlet temperatures during and following the hot reslam

In the studies for the slam acceleration and slam deceleration of subsections 4.4.2 and 4.4.3 we saw that the high-pressure spool was sensitive to bleed temperature updates and that the intermediate and low-pressure spools were sensitive to primary gas path heat soakage. This again is true for the hot reslam manoeuvre both in terms of speed and rate of change of speed.

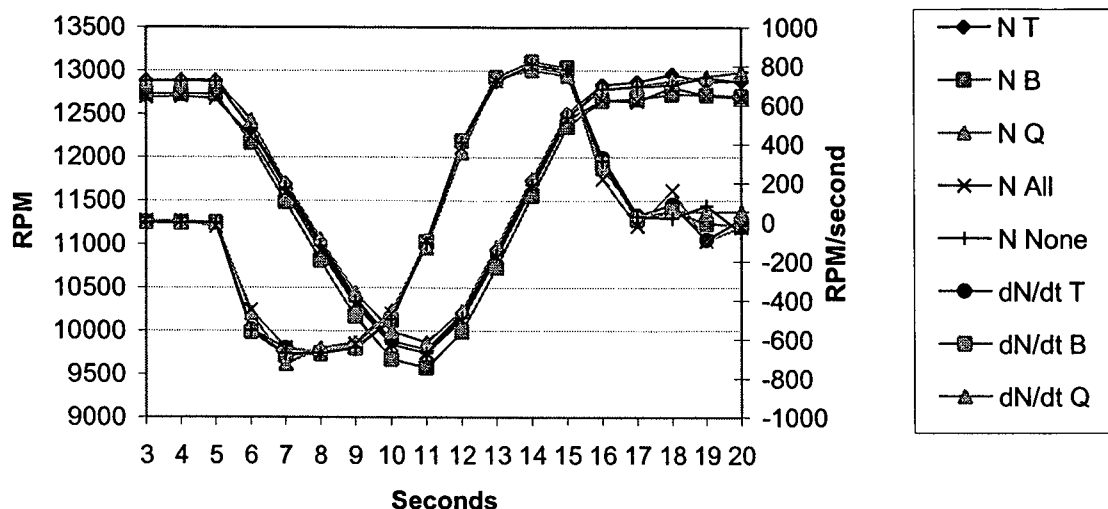


Figure 4.4.4.4 HIGH PRESSURE SPOOL speed and rate of change of speed during the hot reslam. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

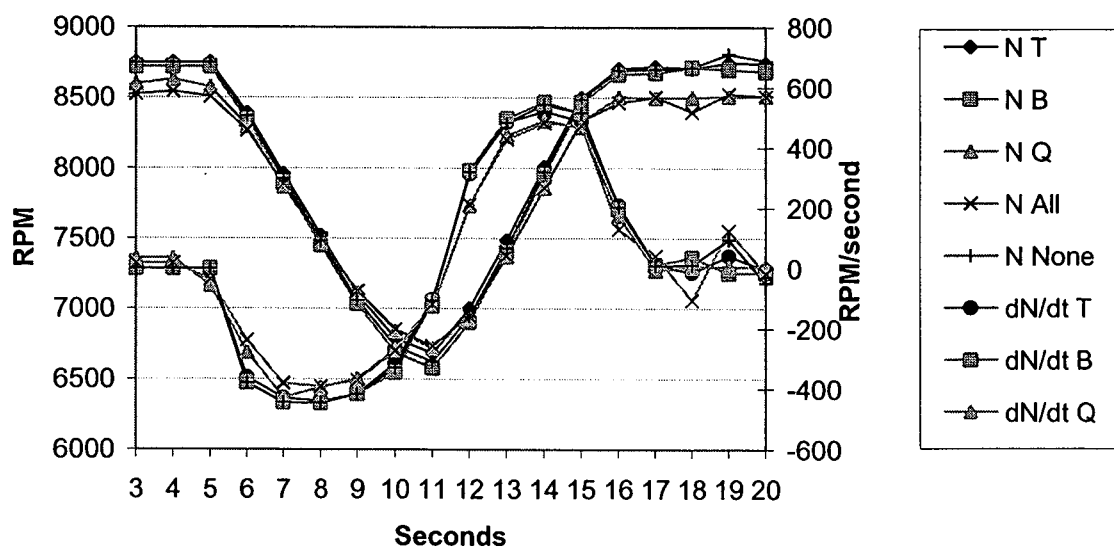


Figure 4.4.4.5 INTERMEDIATE PRESSURE SPOOL speed and rate of change of speed during the hot reslam. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

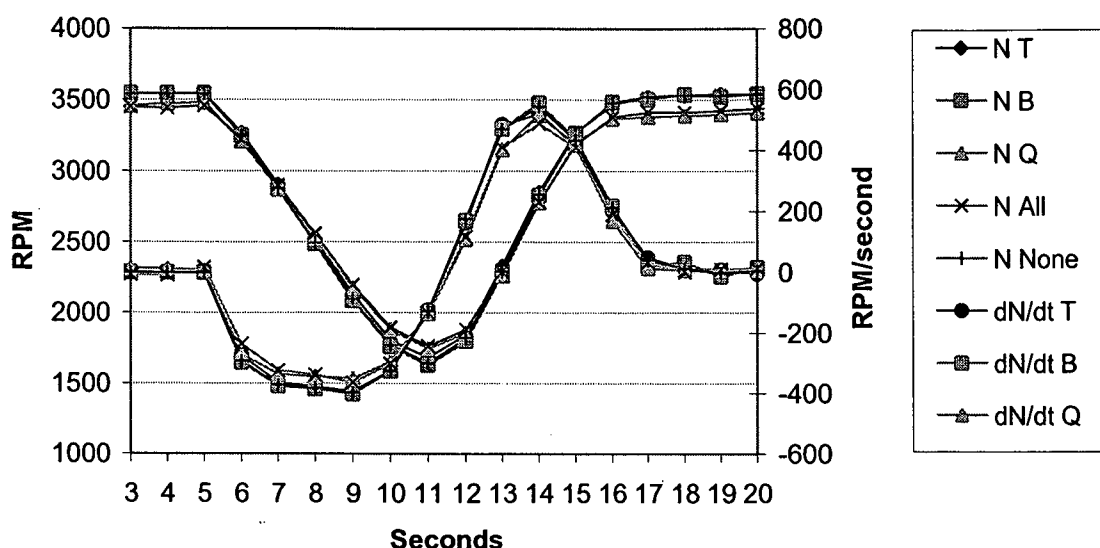


Figure 4.4.4.6

LOW PRESSURE SPOOL speed and rate of change of speed during the hot reslam. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

The figures for F values, tip deltas and bleed temperatures for the slam manoeuvres and the Bodie show one important difference, which is following the slam acceleration and slam deceleration manoeuvre, a significant period of time is needed for things to settle down and reach their steady state values, whereas during and following the hot reslam values appear to just bounce back to their pre-manoevre values in quite an elastic fashion. This is likely to be due to the fact that because all objects have thermal mass, which means that where heat transfer is concerned a period of time is required for temperature changes and thermal expansions, etc. to take place. Indeed, we have seen in previous subsections that the gas turbine transient phenomena of heat soakage, tip clearance changes, and bleed temperature changes can take significant time to complete. It is likely therefore that the elastic-like rebound described above is due to there being insufficient time for some thermal effects (more especially tip deltas and bleed temperatures) to make their mark.

Figure 4.4.4.7 shows the transient working line trajectories from the 5th to the 20th second for hot reslam manoeuvres for runs performed with the transient phenomenon switched on in isolation and combination. During the deceleration part of the manoeuvre there are no significant difference between the hot reslam and the slam deceleration working lines. However, some differences do become apparent when we comparing the hot reslam re-acceleration lines against the cold slam acceleration lines. Figure 4.4.4.8 shows how cold and hot acceleration working lines are affected by switching on the transient phenomena models individually, and figure 4.4.4.9 shows their combined effect.

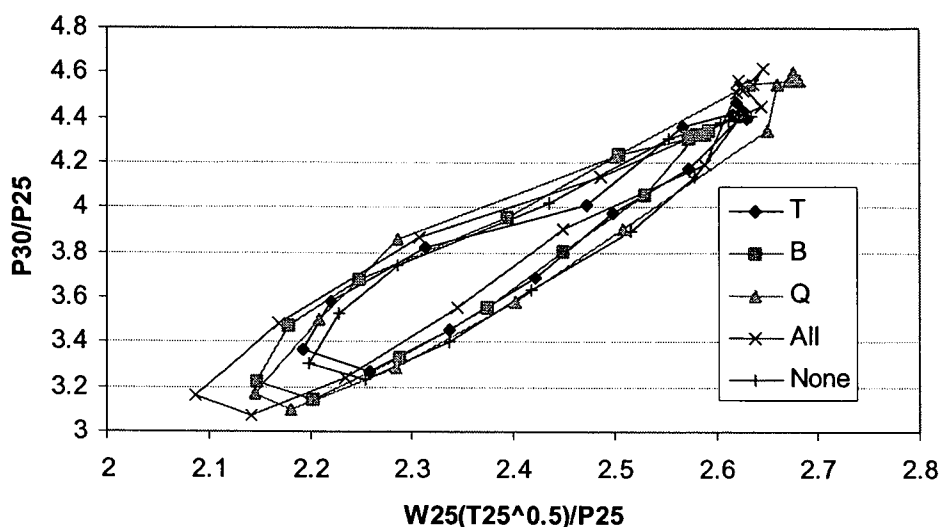


Figure 4.4.4.7 Transient working line trajectories during the hot reslam, 5th second to 20th second. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only, All = all effects, None = none of the effects.

We have seen from figures 4.4.4.4 to 4.4.4.6 that at the end of the deceleration (i.e. the 10th second), even though fuel flow has reached its minimum, spool speeds are still slowing down. Figure 4.4.4.8 shows that the result is relatively high non-dimensional speeds for hot runs compared to cold runs at the start of the acceleration period. The exception is the heat soakage run, in which the outpouring of heat in the IPC during the deceleration results in a higher HPC inlet temperature, and thus lower non-dimensional speed as compared to the tip delta and bleed temperature update runs. In the first few seconds of the hot re-acceleration the runs for the tip deltas and bleed temperature updates appear to give working lines below their cold acceleration equivalents. This is contrary to the expectation of a higher working line during hot reslam manoeuvres, however, in these two cases because the engine starts from a higher operating point compared to the cold acceleration the transient is less severe, so this may be the reason for the lower transient working lines. By contrast, the heat soakage run for the first few seconds of the hot re-acceleration is on a higher working line than that of the corresponding cold acceleration, which is more in line with what we might expect for a hot reslam. Interrogation of the output files revealed that for the hot engine heat soakage run, towards the end of the re-acceleration HPC inlet temperature was warmer than that corresponding to the cold acceleration resulting in the lower non-dimensional speed observed in figure 4.4.4.8. Comparing figure 4.4.4.9 to figure 4.4.4.8, we can see that the tip delta and bleed temperature update runs follow the shape of the adiabatic (i.e. 'none' line) run quite well, whereas the shape of the 'all' run is governed mostly by the shape of the heat soakage run. We can conclude therefore, in this study at least, that heat soakage has a bigger effect on hot reslam working line excursion than either tip delta or bleed temperature updating.

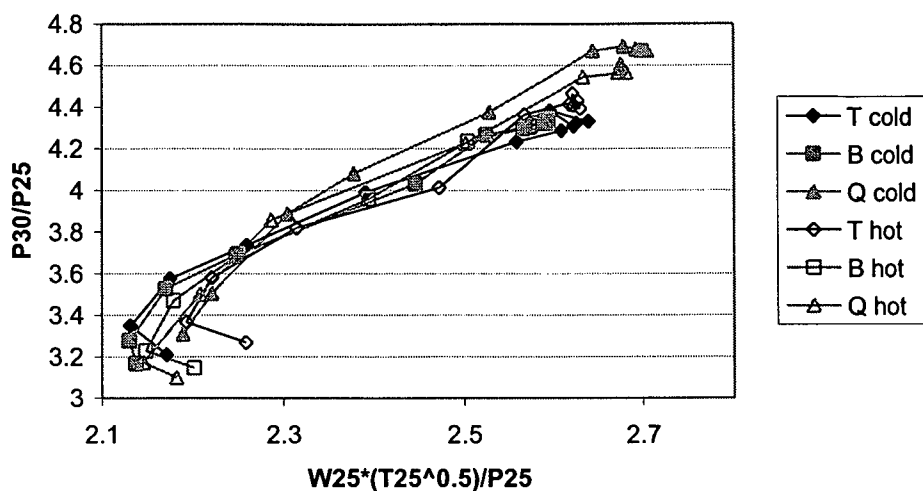


Figure 4.4.4.8

Transient working line trajectories during the cold slam acceleration, 5th to 15th second, and the hot reslam, 10th to 25th second. T = transient tip deltas only, B = bleed temperature updates only, Q = primary gas path temperature (and pressure) updates only

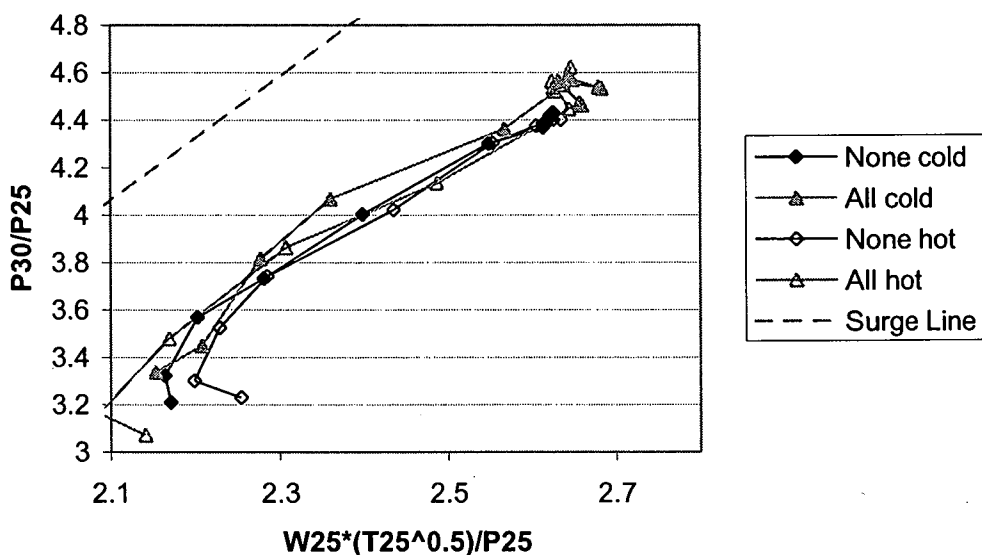


Figure 4.4.4.9

Transient working line trajectories during the cold slam acceleration, 5th to 15th second, and the hot reslam, 10th to 25th second. All = all effects, None = none of the effects.

At the bottom left of figure 4.4.4.9 we can see that at the cold engine 'none' and 'all runs are closer to each other than the corresponding hot engine runs. This suggests that inclusion of transient phenomena effects are important, especially for hot reslam manoeuvres.

4.4.5 COMPARISON AGAINST TEST BED DATA

At the time of writing the only test data available to the author was for a sea-level test bed 'cold stab' (cold slam) acceleration manoeuvre on an ISA -7.5 K day. The test data is shown here and is compared against runs of RRAP with and without the streams method subroutines. The RRAP runs were made to the same fuel flow versus time and at the same day temperature and pressure as the test data, and RRAP's internal handling bleed controls were switched on.

Fuel flow and spool speeds measured from the test bed cold stab acceleration are shown in figure 4.4.5.1. The manoeuvre started at a low idle and accelerated up to a high power setting in just under 8 seconds. The high-pressure spool speed started at about 65% of its high power value and the fuel flow increased by a factor of about 15. The test data available to the author included fuel flow, W24, W26, W30, T20, T24, T26, T30, P20, P24, P26, P30, NL, NI, NH, and thermocouple measurements at the inlet to the low-pressure turbine.

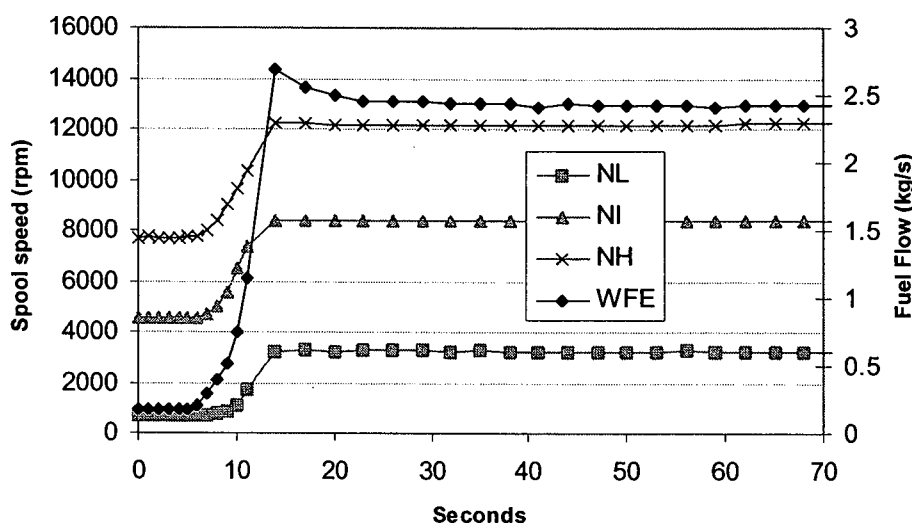


Figure 4.4.5.1 Spool speeds and fuel flow for sea level test bed cold stab acceleration

Before continuing the author wishes to remind the reader that the BD31 version of RRAP used in this study was created prior to the existence of a tangible Trent 500 engine, and is a relatively basic version of RRAP. Nevertheless, the sponsors of this project felt that BD31 would be sufficient for the current undertaking. This means that the various maps and component characteristics contained in BD31, i.e. those used in this study, may not necessarily correspond to the particular hardware build of the engine used in the test that produced the rig data presented here. Also, bleed fractions specified in BD31 may not be the same as those experienced in the test engine. Furthermore, the BD31 internal variable geometry scheduling does not necessarily represent that contained in the test engine control system algorithms.

Figure 4.4.5.2 shows how mass flows as measured from the test compare against mass flows calculated using the BD31 version of RRAP, with and without the inclusion of the streams method subroutines. From the figure we can see that at high power RRAP overestimates W25 but underestimates W30, and that the inclusion of streams lowers both the W25 and W30 estimates. The lower mass flows that occur with the inclusion of the streams method are likely to be a result of the lower temperatures and pressures that are a consequence of modelling steady state heat flow out of the primary gas path (see later). Comparison of mass flows is made difficult without knowledge of test bed bleed fractions.

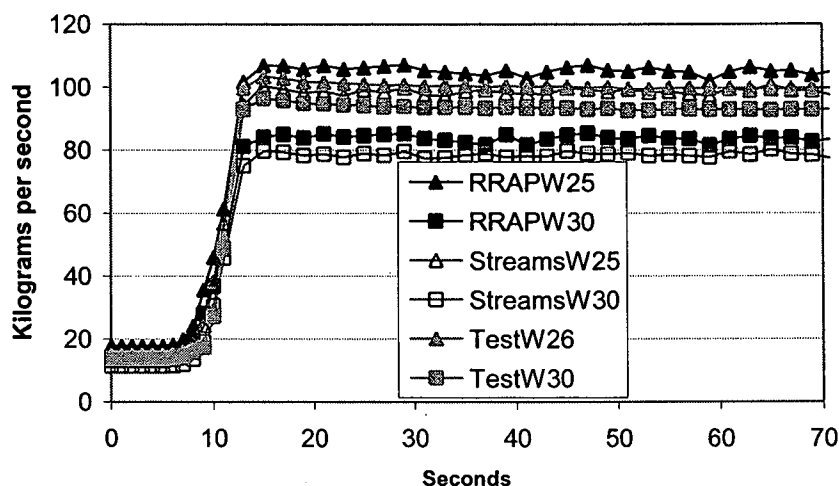


Figure 4.4.5.2 Test bed mass flows for sea level test bed cold stab acceleration versus those calculated using RRAP alone, and RRAP with streams.

Figure 4.4.5.3 shows how test bed pressures compare to RRAP, with and without streams. From the figure we can see that RRAP overestimates the test bed pressures, but that the inclusion of streams lowers estimates and introduces more time lag during the transient manoeuvre bringing them closer to test bed values.

Figure 4.4.5.4 shows how test bed temperatures compare to those predicted using RRAP with and without the streams method subroutines. We can see that RRAP with streams produces temperature estimates that are generally lower than those produced by RRAP alone, bringing them closer toward the test data. The exception is the temperature at inlet to the LPT in the seconds following the start of the transient

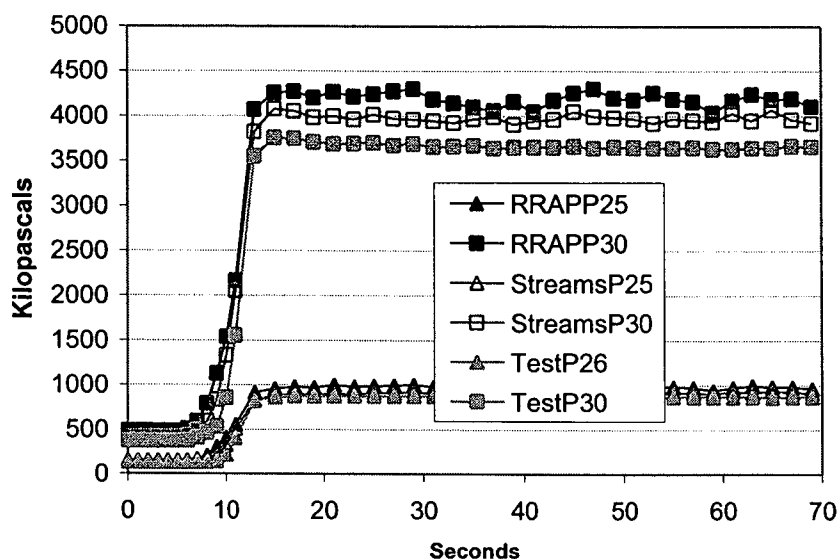


Figure 4.4.5.3 Test bed pressures for sea level test bed cold stab acceleration versus those calculated using RRAP alone, and RRAP with streams.

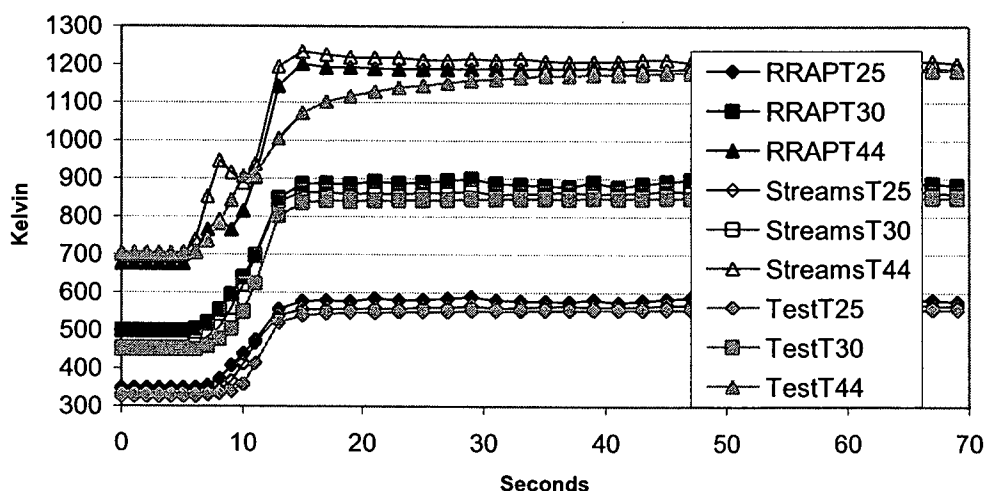


Figure 4.4.5.4 Test bed temperatures for sea level test bed cold stab acceleration versus those calculated using RRAP alone, and RRAP with streams.

The LPT inlet T44 test temperature significantly lags behind the temperatures predicted using both versions of RRAP. The likely reason for this is the way the test temperatures were measured. Thermocouples in the Trent 500 engine are located on the inside of the stator vanes at the inlet to the LPT. The lag apparent in figure 4.4.5.4 is likely to be due to time associated with the thermal stabilisation of the thermocouple fixings.

Figure 4.4.5.4 shows a distinct perturbation in T44 as measured from test and as calculated using RRAP both with and without streams. Figure 4.4.5.5 below shows the likely reason for this. Figure 4.4.5.5 shows the handling bleed mass flows and T44 during the acceleration as calculated by RRAP with streams. In the BD31 version of RRAP the IP8 bleed valve accounts for 14% of IPC inlet air, and the HP3 bleed accounts for 5% of HPC inlet air. This amounts to a relatively large amount of air and therefore it is not surprising that closure of these valves causes such a perturbation in T44 (and also T40 and T42, not shown) in both RRAP with streams and RRAP without streams. Figure 4.4.5.4 shows that the perturbation is larger for RRAP with streams than RRAP without streams, and is smallest for the test bed measurement. In the streams method, heat flow calculations include conduction. The effect of material thermal conductivity is generally to hinder heat flow, and as the acceleration progresses heat flow out of the primary gas path predicted by streams is lower than that predicted by RRAP BR081. Lower primary gas path heat flows mean that primary gas path temperatures are less modified, therefore the increase in mass flow coming from the compressors when the handling bleeds close results in a temporarily larger T44 reduction in RRAP with streams than in RRAP without streams. The perturbation in test bed T44 appears to occur later than in the RRAP runs. This may be due to a difference in bleed valve scheduling, but may also be a result of the time lag associated with the thermocouple fixings. Thermocouple time lag may also damp out the amplitude of the T44 perturbation, making it appear smaller than it might be in reality.

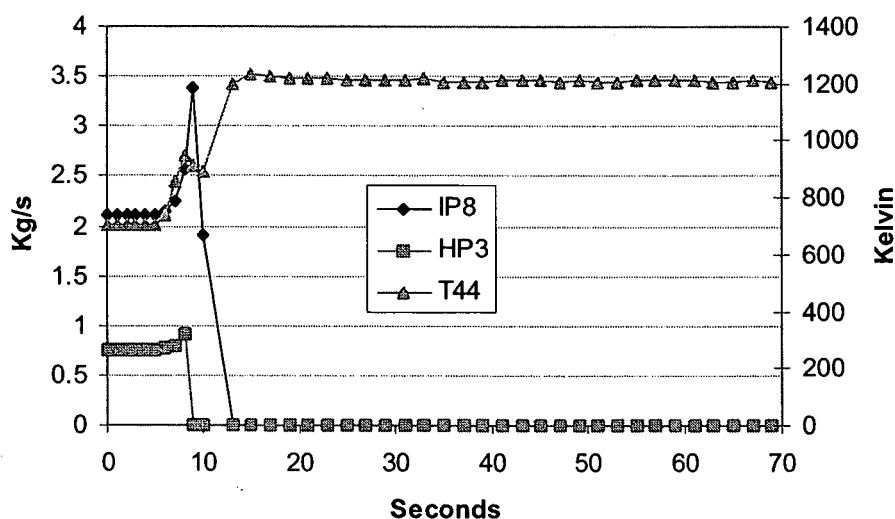


Figure 4.4.5.5

IP8 and HP3 handling bleed mass flows and T44 calculated using RRAP with streams for the test case acceleration.

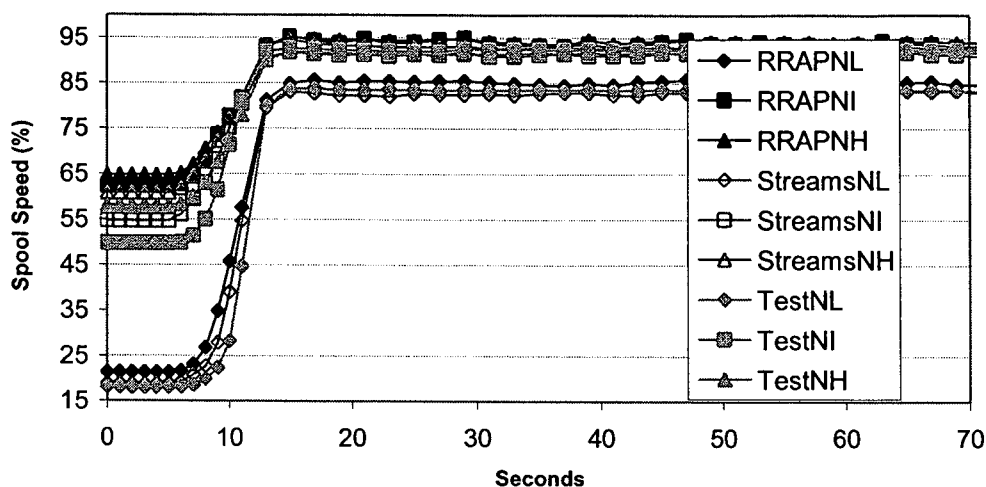


Figure 4.4.5.6 Test bed spool speeds for sea level test bed cold stab acceleration versus those calculated using RRAP alone, and RRAP with streams.

Figure 4.4.5.6 shows how test bed spool speed measurements compare against those predicted by RRAP with and without streams. RRAP alone overestimates spool speeds at both idle and high power, however it appears that inclusion of the streams method brings RRAP predictions closer to test measurements. The large differences in starting spool speeds distort plots of spool acceleration rates and are therefore not shown, however from the figure it is clear that the large lag in low-pressure spool speed is modelled better by RRAP with streams than by RRAP without streams.

In terms of thermal modelling, one important way that the streams method differs from RRAP is that it allows for non-zero steady state heat flow in or out of the primary gas path under steady state conditions. Figure 4.4.5.7 shows F values as calculated by RRAP with and without streams before, during, and after the cold stab acceleration manoeuvre. We can see from the figure that the F factors from RRAP start at zero and decay back to zero following the transient, whereas streams F factors start and end at non-zero values. At low idle the streams method F value for the LPT -13%, for the IPC, HPT, and IPT, it is approximately -5%, whereas for the HPC it is 0.1%. At high power the F value for the IPC lessens to about -0.5%, for the HPC it is 0.2%, for the HPT -2.4%, for the IPT -1.3% and for the LPT it is -1.2%. At the end of the acceleration shaft powers increase to more than ten times low idle values, and so as we can see from the results F values thus generally lower at high power than they are at low idle. This also explains the very high peak F values calculated during the manoeuvre - up to 70% for the streams method LPT. The very low engine setting from which the cold stab acceleration was initiated (65% high-pressure spool speed, and 1,346 lbs/hr fuel flow, is the reason why F values in this study are higher than those for the slam acceleration manoeuvre presented in subsection 4.4.2.

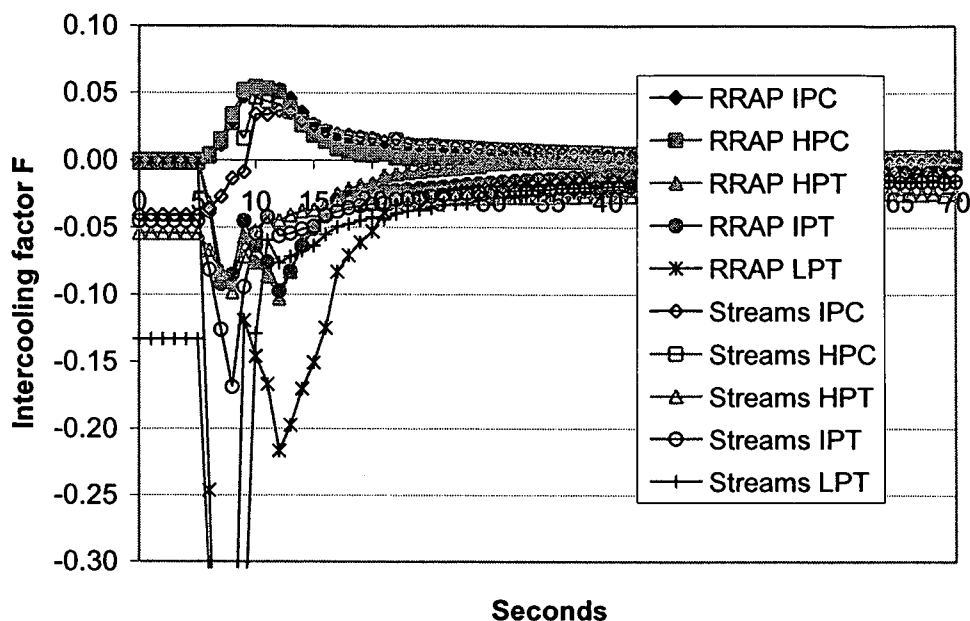


Figure 4.4.5.7 Intercooling factors for sea level test bed cold stab acceleration calculated using RRAP alone, and RRAP with streams.

One interesting feature of figure 4.4.5.7 is the double peak in F value that occurs during the transient manoeuvre. This double peak is demonstrated by both RRAP with streams and RRAP without streams and is due to the closure of the handling bleed valves during the acceleration manoeuvre.

RRAP with streams predicts lower engine mass flows than RRAP alone. As we have seen in previous subsections this is due to the effect of steady state heat soakage and bleed temperature updating. A consequence of this is that streams method shaft powers are lower, and hence F values higher than RRAP without streams. This is true for the earlier stages of the transient. However, the streams method generally predicts lower heat flows (due to the presence of a conduction model) than BR081, and so as shaft powers increase towards the end of the manoeuvre, smaller streams heat flows produce smaller F values. The nett result is that RRAP with streams F values have higher first peaks and lower second peaks as compared to RRAP without streams.

RRAP without the inclusion of the streams method subroutines does not model the effect of transient changes in component tip clearance from their steady state values. In order to calculate what these differences are the streams method calculates both steady state and instantaneous tip closures. Figure 4.4.5.8 shows the stabilised streams method stage averaged non-dimensional tip closures (from cold static) used in this cold stab test bed acceleration study. In figure 4.4.5.8 non-dimensional tip closures at the given point in the operational envelope are plotted against fuel flow.

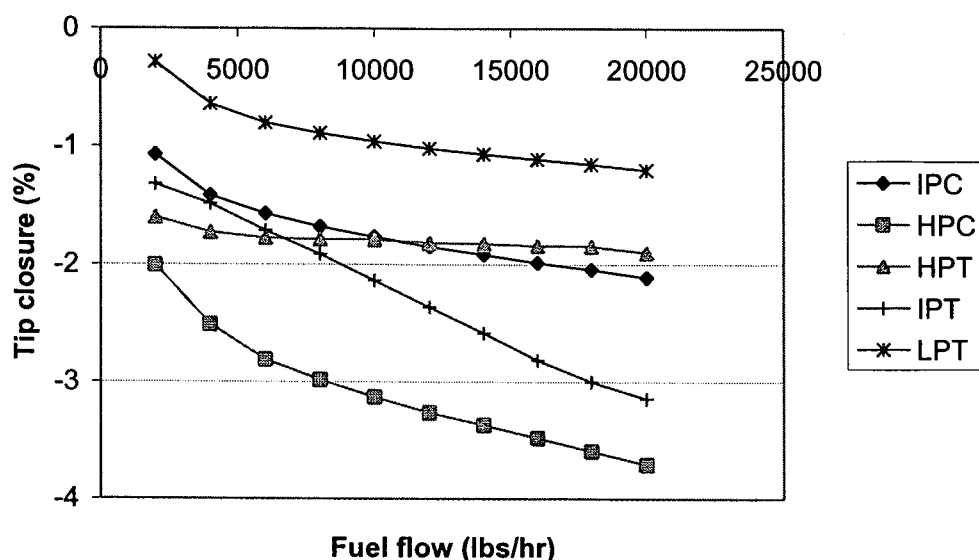


Figure 4.4.5.8 Stabilised non-dimensional tip closure (from cold static) as calculated for the Trent 500 using RRAP with the inclusion of the streams method subroutines

The plot is only representative of a single point in the operational envelope. If altitude, Mach number, day temperature, etc., were changed, then a new plot would have to be prepared. Figure 4.4.5.8 shows that for this particular engine tip closures behave non-linearly against fuel flow. The departure from linearity varies depending upon what the tip closures are plotted against. Against EPR the centre of curvature of the lines is generally above the lines, whereas against NL the centre of curvature is below the lines.

The values appear to be a little large compared to the 1-2% quoted in the literature (Walsh & Fletcher 1998). The worst offender appears to be the HPC. Section 4.2 of this chapter compares the performance of the streams method against the Rolls-Royce in house method (the Lewis method) for transient tip closure prediction and demonstrated that although individual stages were different the mean stage closure predicted by the streams method was quite close to the Rolls-Royce predictions. The Rolls-Royce method is validated against rig test data. On compressor rig tests bleeds are taken but only to speed thermal stabilisation and with no attempt to replicate bleed conditions in the real engine (Lewis 2003). The streams model in this study was set up using the bleed fractions already part of the BD31 version of RRAP, which were in many cases larger than those used in the section 4.2 study. This may explain why the tip closures of figure 4.4.5.8 are perhaps a little large.

Using this plot of steady state tip closures, transient tip deltas (and thence efficiency and flow defects) were calculated for the cold stab acceleration for each of the components modelled. Figure 4.4.5.9 shows the transient tip deltas calculated in this study. The tip deltas of figure 4.4.5.9 are very similar to those of figure 4.4.2.2, for the slam acceleration study of section 4.4.2. Figure 4.4.5.10 shows the effect that switching tip delta calculations on and off had on thrust predictions.

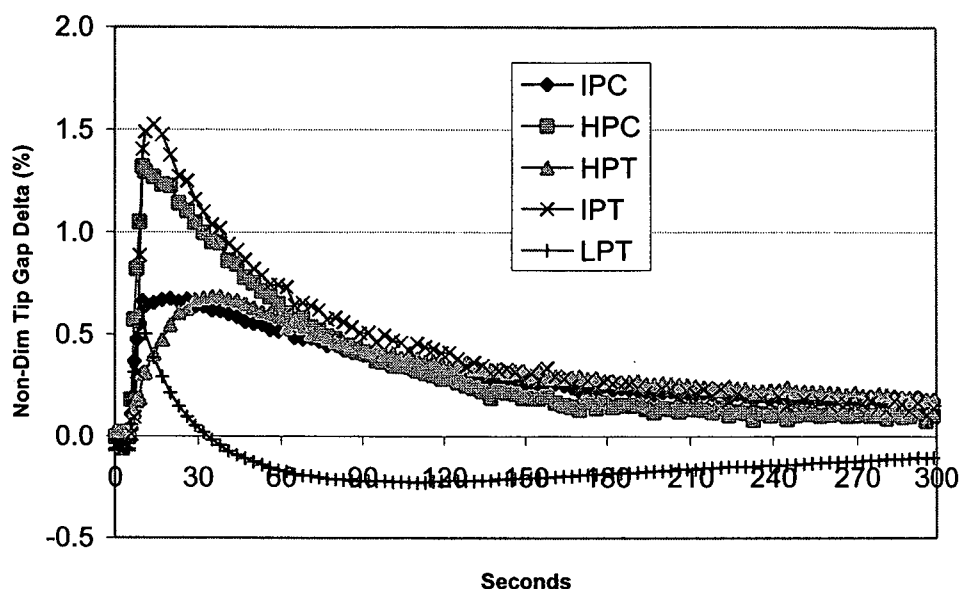


Figure 4.4.5.9 Transient non-dimensional tip gap deltas (transient closure minus steady state closure) as calculated for the Trent 500 using RRAP with the inclusion of the streams method subroutines

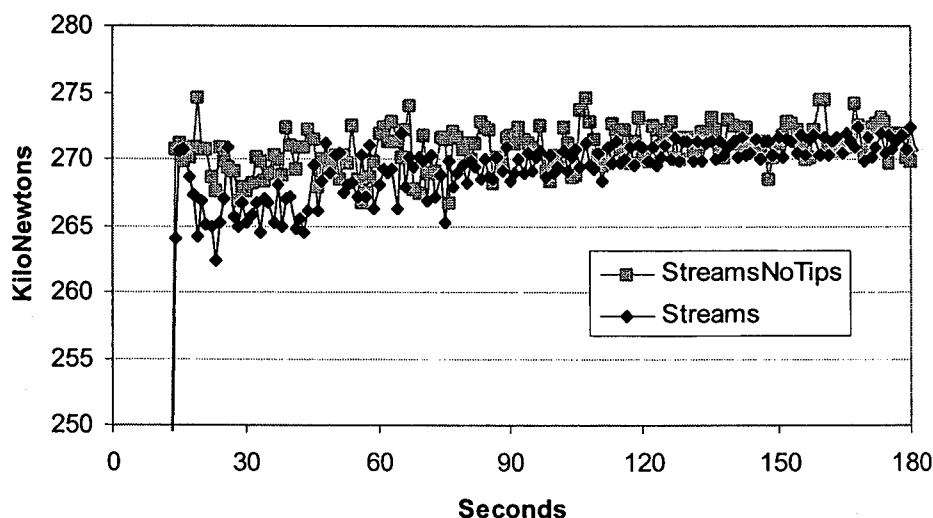


Figure 4.4.5.10 Thrust predictions produced for the cold stab acceleration using RRAP with streams and RRAP with streams but with tip clearance calculations switched off.

Figure 4.4.5.10 shows that the effect of tip deltas has the effect of reducing thrust by about 1.5% immediately following the end of the speed transient, but that this defect fades away after about three minutes. Figure 4.4.5.10 suggests that the negative LPT

delta appears to compensate somewhat for the positive deltas of the other components, so far as thrust is concerned, making the thrust defect unobservable after about three minutes, even though the tips gaps themselves don't fully stabilise until many minutes later.

Figure 4.4.5.11 shows a selection of bleed take off and re-entry temperatures as calculated by RRAP with and without streams for the cold stab acceleration. In the BD31 version of RRAP, the bleeds at re-entry are assumed to have the same temperature as the bleed point from which they originated. The figure shows that just like for the slam acceleration of subsection 4.4.2, by accounting for heat exchange in the secondary gas paths a significant amount of bleed heating can take place - over 100K for the IP8 bleed which ventilates the high-pressure spool and returns to the primary gas path at LPT entry. The figure also shows that bleed re-entry temperatures as calculated using the streams method can take a number of minutes to reach steady state values following the end of the speed transient. The figure shows an IP8 bleed, an HP3 bleed, and an HP6 bleed. Whether the bleed is heated or is cooled depends very much on the size and temperature of the parts of the engine with which it comes into contact. The figure shows that HP6 bleed serving the front of the HPT is actually cooled slightly before re-entry. The HPT disc is ventilated by three different bleed flows, two of which are cooler than HP6 air (c.f. appendix A). It is no surprise therefore that the disc removes heat from the HP6 bleed before it re-enters the primary gas path.

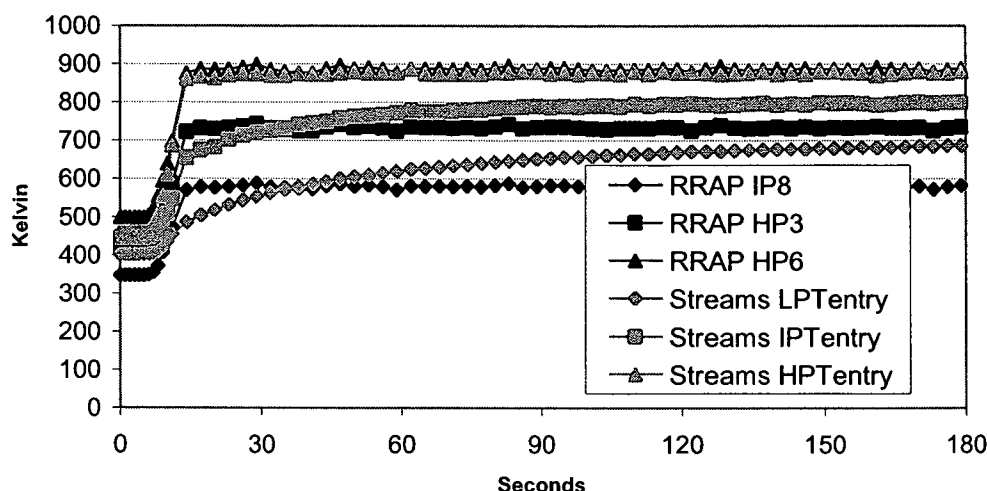


Figure 4.4.5.11 Selected bleed re-entry temperatures as calculated using RRAP with and without streams.

The accuracy of the bleed re-entry temperature prediction depends very much on the quality of the streams models for each engine component. In this study the combustion chamber was omitted. If a streams model of the combustion chamber were to be included then the re-entry temperature of the bleeds may well be different to that shown in figure 4.4.5.11. When bleed temperature updating was switched on and off, there were no perceptible differences in thrust predictions (although this may have been because of noise). However when switched on, there was a slight lowering of the whole engine temperature profile, which was repeated in spool speeds,

pressures, mass flows, etc. The author suspects that this was due to the cooling of the bleed to the HPT, lowering its inlet temperature and thus the engine cycle temperature.

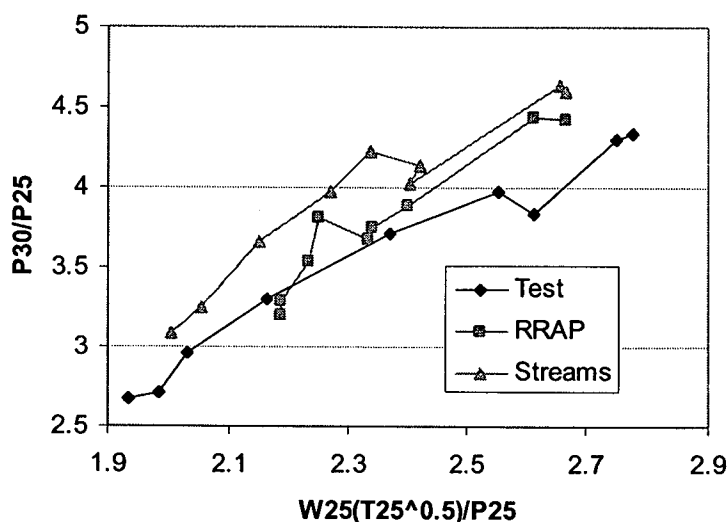


Figure 4.4.5.12 Transient working lines calculated from cold stab test measurements, and calculated by RRAP alone, and RRAP with the streams method subroutines incorporated into it.

Figure 4.4.5.12 shows the transient working lines calculated for RRAP with and without streams, and also from test measurements. The figure shows that there are significant differences between working line predictions made by RRAP with and without streams compared to that calculated from test data. There are a number of possible reasons why this is so. To start with, the BD31 version of RRAP used in this study is unlikely to correspond to the build of engine used to produce the test data. The component maps it uses may not correspond to the performance of components actually used, bleed mass flows may not be the same as those in the engine as it was finally built, and the BD31 internal handling bleed controls are unlikely to be the same as the corresponding algorithms in the sophisticated test bed engine control system. Start and end points for the cold stab acceleration working lines are all different to each other, and there is only a vague similarity between the shapes of line for each run (although that of streams RRAP appears to have better shape than that of RRAP alone). A further complication is that the HP3 handling bleed is taken from the middle of the high-pressure compressor. This means that the compressor aerodynamics and thermodynamics are different for each setting of the handling bleed. BD31 accounts for this by using a different high-pressure compressor map for each bleed setting. So figure 4.4.5.12 actually represents more than a single map, and streams RRAP and pure RRAP do not necessarily switch between the same maps at the same points in time during the manoeuvre, which again may or may not correspond to the switching on and off of the HP3 handling bleed in the test engine.

It would appear from this subsection that the inclusion of the streams method subroutines into RRAP brings the predictions it makes more in line with test results. However, all that can be concluded in reality is that the inclusion of streams method subroutines lowers the performance predictions produced by the BD31 version of RRAP. The fact that the test bed performance data used in this study is lower than that predicted by RRAP is likely to just be a coincidence. To properly ascertain whether or not the streams method does in fact improve engine performance predictions it would be necessary to ensure that the version of RRAP and the streams method models matched the build of engine used to produce the test data. In addition it would be necessary to conduct a thorough investigation using a much more full set of test data.

4.4.6 SECTION CONCLUSION

The purpose of section 4.4 was to demonstrate the effect that the author's heat soakage and tip clearance methods have on whole engine transient performance predictions.

In this section the author presented the results of incorporating the streams method subroutines into the BD31 version of RRAP. He has shown the relative sizes of influence of each of three transient phenomena as modelled by the streams method for the Trent 500 engine. The three transient phenomena were primary gas path heat soakage, bleed temperature updating, and instantaneous blade tip gap departures from their corresponding steady state values. Three transient manoeuvres were used in the study: a slam acceleration, a slam deceleration, and a hot reslam manoeuvre.

Subsections 4.4.2, 4.4.3, and 4.4.4 described how the transient tip gap and heat soakage subroutines, when switched on and off individually and in combination, affected spool acceleration, bleed temperatures, and HPC working line trajectories during slam acceleration, slam deceleration, and hot reslam manoeuvres. The manoeuvres were run to a fixed fuel flow versus time in order to isolate engine physics from the effect of the presence of a control system (see appendix B). It was shown that steady state heat leakage from the primary gas path via conduction into the secondary gas paths can significantly affect the steady state performance of the engine model. It was also shown that heat soakage, bleed temperature updates, and tip gap deltas all affect the transient performance of the engine and this is especially noticeable at the start of a hot reslam manoeuvre. Heat soakage had the largest effect in general.

Subsection 4.4.5 showed how RRAP and RRAP with streams compared against real engine test data. The streams method subroutines worsened pure RRAP performance predictions (mostly due to steady state heat leakage), bringing RRAP results more in line with test data. Given the uncertainty in the degree to which the BD31 version of RRAP represents the particular build of engine used to produce the test data used in this study, the apparent improved prediction compared to test data resulting from use the streams method subroutines is likely to be more to a matter of chance than design. A more thorough investigation would require the use of test data, hardware models, and an engine performance code that all correspond to the same engine build.

4.5 ESTIMATION OF STEADY STATE PERFORMANCE FROM TRANSIENT TEST DATA

4.5.1 SECTION INTRODUCTION

The purpose of section 4.5 is to discuss the relevant issues and suggests a possible approach regarding a method of extracting steady state engine performance information from transient test data.

4.5.2 MOTIVATION

The motivation behind trying to extract steady state performance information from transient test data is purely economic. As mentioned previously in chapter 2, test beds are expensive to run, and a large portion of the cost associated with an engine manufacturer's performance function is that of development testing. To establish the steady state performance of a new engine throughout its throttle range at a given set of environmental conditions, typically a series of small steps in throttle will be used from low to high power, each step being taken a nominal 3 to 5 minutes after the previous step (Curnock 2002). The period of waiting is there to allow the engine to thermally stabilise, and thus prevent transient/stabilisation phenomena from contaminating the test data. This waiting time is expensive. If it was possible to eliminate the waiting time by extracting steady state performance information from transient test data then significant cost savings could be achieved.

4.5.3 ISSUES

Kimzey and Wantland (1977) describe their attempt at doing this using transient test data from a two spool engine. The transient phenomena they included in their analysis were rotor energy, heat soakage, and component rematching. They reported that the largest effect was heat soakage, however in their analysis heat soakage received the simplest treatment – for each component they used a single time constant, a single metal temperature, and a gas temperature equal to the average of inlet and outlet temperatures. Their method was able to predict nett thrust and SFC to accuracies between 1 and 2%. They recommended that in future a transient mathematical model of the engine should be made that includes heat soakage and tip clearances. Yang et al (2002) used essentially the same approach as Kimzey and Wantland (1977) but on a single spool turbojet in an altitude test facility, however their predictions were slightly less accurate.

Both sets of researcher reported that the largest effect was heat soakage, and both sets of researchers used only simple heat soakage models in their analyses. The previous sections in this chapter have shown that large differences in predicted heat flows can exist between simple models (e.g BR081), and more involved models (e.g. multistage streams models).

It is the opinion of the author that inference of steady state performance information from transient test data to a useful degree of accuracy is unlikely to be realised in the near future. The main issue is that of uncertainty in the transient phenomena models.

To infer steady state performance information from transient test data necessarily involves the calibration of a steady state engine performance code on the strength of the accuracy of a transient model. Transient models are generally more uncertain than steady state models. In effect the process of inferring steady state performance information from transient test data is a little bit like trying to calibrate bathroom scales by asking someone of a known mass to jump up and down on them.

4.5.4 APPROACH – SINGLE COMPONENT

For the purpose of this discussion it is useful to break down the task and first consider the extraction of steady state performance information from transient test data of just a simply ventilated compressor performing a cold acceleration. It is assumed that tip deltas and heat flows during the transient are accurately known, and that a set of transient measurements for temperatures, pressures and mass flow exists.

From this starting point it may be possible to add the transient heat flow lost from the primary gas path back into the working fluid to find what the adiabatic delivery temperature should be, and then using the equation for non-adiabatic compression and expansion (equation 2.3.3.1) calculate what the adiabatic delivery pressure should be. Furthermore, knowing tip deltas and accurate efficiency and flow exchange rates, it may be possible to recalculate what adiabatic temperatures, pressures, mass flows, and efficiencies should be, and thus produce and estimate of a steady state compressor map from transient test data.

The first problem is that nobody in the compressor office would believe it. And the second problem is that in order to find out what tip deltas are it is necessary to know steady state tip closures, and for that it is necessary to know steady state inlet and outlet conditions, and for that it is necessary to have a steady state map, which is the very thing we are trying to produce. A straight through analytical approach, such as that described above, is unlikely therefore to be appropriate.

The alternative to a straight through analytical approach is an iterative approach. One way to use an iterative approach would be to start with a guessed steady state map. It could be used in combination with our accurate transient phenomena models to produce a prediction of transient performance. It would then be possible to compare this against the transient test data and use any differences to modify our guessed steady state map. The process could continue, combined with our transient model, until the guessed map produced transient performance data that exactly matched the transient test data. At which point it might be claimed that the guessed steady state map was an exact representation of the steady state performance of the compressor.

4.5.5 APPROACH – WHOLE ENGINE

The following assumes that a set of accurate steady state component performance maps exist, and that these are used by an engine code into which our accurate transient phenomena models have been incorporated.

In reality, after assembling an engine the individual components are generally in a significantly different environment to that which they experienced during rig tests. Rig ventilation flows are often not representative of real engine ventilation flows, temperature profiles are different, tip clearances are different, and inlet and outlet temperature and flow profiles are different. All of these effects make the engine on a test bed perform differently to the predictions of performance made by an engine code into which rig test component maps are incorporated.

The author suggests that it may be possible extend the iterative approach used to infer steady state performance for the simple compressor and apply it to the whole engine. It may be possible to take our steady state maps and our accurate transient phenomena models, and perform a series of transient performance code runs. Comparing the transient performance predictions thus produced against the transient test data, and identifying where there were differences it may be possible to put factors and deltas onto our steady state maps. This process could be repeated until we had modified all of the steady state maps to the extent that the transient performance predictions exactly matched the transient test data. At which point it may be possible to claim that our steady state engine model was an exact representation of the steady state performance of the engine.

4.5.6 CHOICE OF TRANSIENT MANOEUVRE

An important issue is what type of transients should be performed as part of the analysis. The initial idea associated with this project (Martin 2001) was to replace the traditional development test performance curve consisting of a series of 10 or 12 steps each with a 3 to 5 minute waiting time taking in total about an hour, with a single slow acceleration taking in total just a fraction of the time. Figures 4.5.1 and 4.5.2 show the results of running RRAP with streams (internal handling bleed control switched on) through a slow acceleration. For the first 10 seconds of the manoeuvre fuel flow was at 2,250 lbs/hr, following which fuel flow was increased linearly with time for a period of 2 minutes, after which a steady fuel flow of 22,000lbs/hr was held. Figure 4.5.1 shows streams method predictions for F values for the Trent 500 during the 2 minute acceleration. If we compare the figure 4.5.1 with figure 4.4.5.7 for the cold stab acceleration, we can see a stark difference in the size of transient F values experienced by the engine. For the cold stab acceleration F values peaked in some cases at many tens of percent, however the F values for the 2 minute acceleration barely differ from their steady state values.

Figure 4.5.2 shows streams method predictions for non-dimensional tip deltas during the manoeuvre. Comparing figure 4.5.2 with figure 4.4.5.9 for the cold stab acceleration it is clear that 2 minute acceleration tip delta values are much smaller than tip deltas predicted for the cold stab.

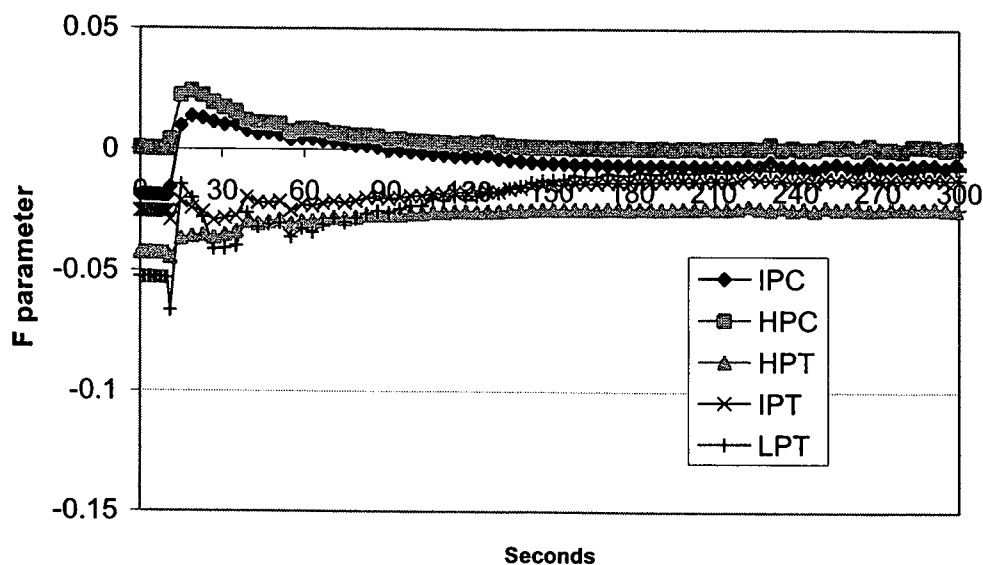


Figure 4.5.1 Streams method predictions for F parameter values during and following the 2 minute acceleration manoeuvre.

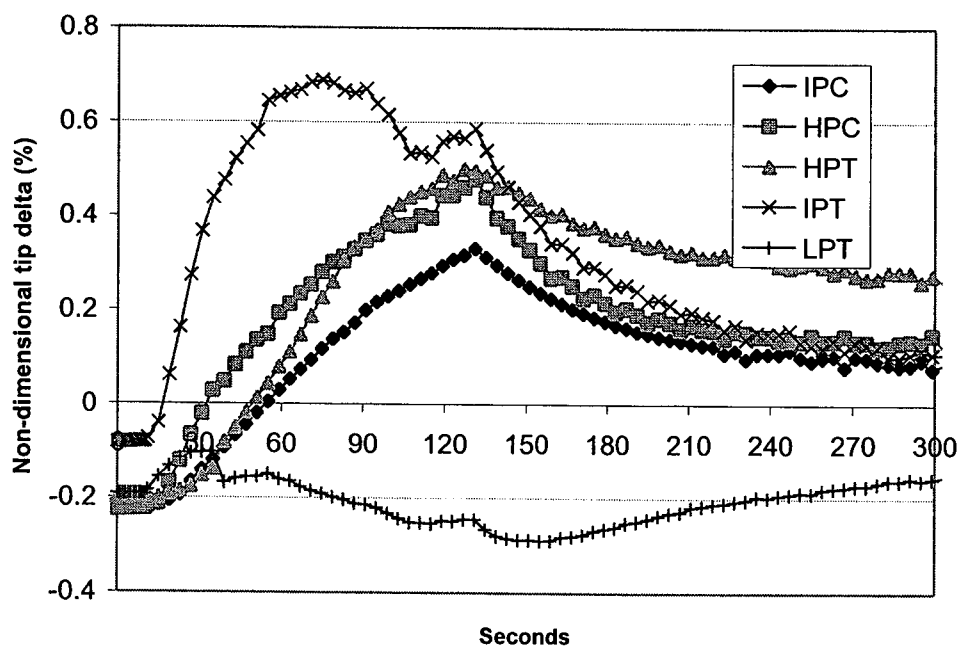


Figure 4.5.2 Streams method predictions for non-dimensional tip deltas during and following the 2 minute acceleration manoeuvre.

The fact that slow acceleration predictions of transient phenomena are smaller in magnitude than those predicted for a fast acceleration would suggest that to improve the observability of transient phenomena performing fast accelerations rather than slow accelerations is the best way to do it. This is very obvious – the longer we take to perform a manoeuvre the more we approach steady state and the less visible become any transient phenomena.

In order to reverse calculate transient effects out of transient test data it would make sense to make them more visible. With this logic, fast transients would appear to be better for our purposes than slow transients. However fast transients present a fresh set of problems. When an engine performs a fast transient it is much more likely to encounter operability problems, and therefore cross control system limits. Whenever an engine hits control system limits the engine physics is complicated by the presence of the control system. This being the case, the process of extracting steady state information from transient test data would have to include accounting for the effect of the control system. Also, the faster the transient, the more the engine begins to feel the effect of phenomena not experienced during steady state, e.g. mass packing. Furthermore, when performing fast transients sensor lags start to become more significant, and therefore would also need to be accounted for in the reverse calculation process. All of these complications would appear to lead back to the idea that slow transients are probably better.

So the answer to the question of which types of transient manoeuvres it would be better to perform is likely to be both fast and slow transient manoeuvres. And this would be quite feasible given all the extra time that would be freed up by not doing the traditional stepped test bed manoeuvre.

4.5.7 SECTION CONCLUSION

The purpose of section 4.5 was to discuss the relevant issues and suggests a possible approach regarding a method of extracting steady state engine performance information from transient test data.

Subsection 4.5.2 outlined the economic motivation behind developing such a method, i.e. an order of magnitude test bed time and cost saving. Indeed, this was the business case for the sponsorship of this project. Subsection 4.5.3 mentioned previous work in this area and talked of the uncertainty problems inherent to a process of trying to use a transient model to calibrate a steady state model. Subsection 4.5.4 proposed a method of extracting steady state performance information from transient test data for a compressor in isolation. It is clear that an iterative approach is required. Subsection 4.5.5 discussed how the same approach might be extended to a whole engine performance code, i.e. iterative manipulation of component maps until transient runs performed with the code matched up with transient whole engine test data. Subsection 4.5.6 discussed how both fast transients and slow transients would probably be needed in order to calibrate the whole engine model.

The basic idea behind the iterative map optimising process described above is not new. In fact, a description of a similar procedure is given in Stamatis et al (1990), which is essentially the same process that the Rolls-Royce ANSYN computer

programme performs to match the engine deck to test data. The main difference is that both ANSYN and Stamatis et al (1990) are only used steady state.

If the goal of inferring steady state performance information from transient test data is to be successfully achieved then the author recommends two things. The first is that further work be done to improve the accuracy of transient phenomena models, and the second is to investigate the possibility of modifying a version of the ANSYN code or the method described by Stamatis et al (1990) such that it has the ability to calibrate engine models from transient test data as opposed to just steady state data. Clearly, if this can be done successfully significant economic benefits could be realised. However the author feels that whoever attempts this task may well be defeated not through an inability to create such a procedure but by the accuracy of the transient phenomena models needed for the job, and/or the set up times perhaps necessary to produce acceptably accurate predictions.

4.6 CHAPTER CONCLUSION

The main purpose of this chapter was to establish the extent to which the heat soakage and tip clearance methods developed by the author meet the project requirements laid down by the sponsor. Other relevant studies were also included.

In section 4.2 of this chapter, the author's heat transfer and thermomechanical tip clearance methods were used to construct a multistage model of the Trent 500 high-pressure compressor. The results it produced were compared with those of the Rolls-Royce in-house Trent 500 HPC transient tip clearance model (the Lewis method), and the Rolls-Royce whole engine performance code heat soakage model (BR081). In addition, a sensitivity study was performed to investigate the sensitivity of heat transfer and tip clearance methods to their discretionary variables (heat transfer coefficients and mass flow interaction fractions).

The results of the study showed that streams method tip gap predictions were outside of the accuracy requirements laid down by the sponsor (i.e. within 0.025mm of the Lewis method steady state, and 0.1mm transiently). This was mainly due to the simplifications made in the streams method model. Interestingly, the difference between the RMS of the Lewis method results and the mean of the Lewis method results was larger than 0.025mm at steady state, and almost larger than 0.1mm transiently. The results also showed that heat flows predicted by the streams method multistage model were much less than those predicted by BR081. This was mainly due to the fact that BR081 ignores conduction, whereas conduction is accounted for within the streams method and this impedes heat transfer. Also, in a multistage model forward stages preheat or precool air before it arrives at the later stages, thus reducing total heat flow. Absence of any authoritative test or finite element analysis data prevented assessment of the accuracy of any of the heat transfer calculations. The results of the sensitivity analysis showed that variation of the streams method's discretionary variables allowed a degree of scope for calibration of the model, should it be required to more accurately represent test data. Set up time for the Trent 500 high-pressure compressor model was approximately 2 days. Run times from within RRAP were less than one minute to converge for a steady state calculation, and faster than real time during transient runs (using a transient time interval of 0.05 seconds).

In section 4.3 of this chapter the heat transfer and thermomechanical tip clearance methods were used to construct a single averaged stage representation of the Trent 500 HPC in order to investigate the possibility of reducing set up and run times compared to the multistage model. Single averaged stage tip clearance predictions were compared with those of the multistage streams model and the Lewis method. The heat soakage predictions it produced were compared with those of the multistage streams model and BR081. In addition, sensitivity study was carried out and also an investigation into how easily the model could be calibrated to match Lewis method results.

The process of representing a multistage component with a single averaged stage proved to require a significant amount of engineering judgement. For example decisions had to be made on what the single stage material properties should be given the Trent 500 HPC is a six-stage component made of eight different materials. Moreover, deciding how to represent complex multiple bleed ventilation in just a single stage was also a challenge. The results of the study showed that tip gap change predictions for a single averaged stage were less accurate (by over 0.5 mm) than those produced by averaging individual stage values from a multistage model. The results also showed that the single averaged stage heat transfer predictions for the six-stage compressor, when multiplied by six, were much larger than predictions made by both the multistage streams model and BR081. The sensitivity study showed the single averaged stage model to be moderately sensitive to its discretionary variables, however after half a day of manipulating the discretionary variables and the model itself there still existed a significant difference (up to 0.25 mm) between its tip gap predictions and the average of stage values predicted by the Lewis method. Set up and run times for the single averaged stage model were much less than those for the multistage model however the extra speed came at the cost of accuracy, and the lengths of time needed for model calibration may be prohibitive.

In section 4.4 of this chapter, full multistage streams method models of the Trent 500 IPC, HPC, HPT, IPT, and LPT were incorporated into the BD31 version of RRAP. A series of sea-level transient manoeuvres were performed to establish the relative impact on whole engine transient performance of primary gas path heat soakage, bleed temperature updating, and blade tip deltas, in isolation and in combination. In addition, RRAP with streams and RRAP without streams were run to the same fuel flow versus time and day conditions as a set of test bed data for a Trent 500 cold stab acceleration.

The results of the first part of the study revealed that heat transfer to and from the primary gas path appeared to have the largest effect on engine performance, not only during transient manoeuvres but also under steady state conditions. It also showed that heat soakage, bleed temperature updates, and tip deltas all affect transient engine performance, especially during hot reslam manoeuvres. The results of the comparison against test bed data were less clear. The BD31 version of RRAP used in the study may or may not correspond to the particular build of Trent 500 used to produce the test bed data. The streams method appeared to bring RRAP predictions closer to the test data however this was more likely to be due to chance rather than design. The only real conclusion being that the streams method worsens RRAP's engine performance predictions.

In section 4.5 of this chapter, the author presented a brief discussion relating to the issues associated with attempting to infer steady state performance information from transient test data.

The author proposes one possible approach whereby whole engine code component maps are iteratively manipulated until transient runs performed with the code match up with whole engine transient test data. The author suggests that an existing steady state engine model manipulation method such as that described by Stamatis et al (1990) could be modified in order to do this. It is the opinion of the author that accuracy of the transient phenomena models is the biggest obstacle to achieving such a goal.

The requirements laid down by the sponsor for the engineering component of this project (see chapter 2) were that the methods developed by the author should:

- 1) Be able to handle both tip clearance and heat soakage
- 2) Be physics based and therefore have some ability to predict the response of untested hardware
- 3) Be accurate enough to adequately predict component performance effects, which for tip clearance calculations equates to predictions within 0.025 mm of Lewis method predictions steady state, and 0.1 mm transiently.
- 4) Be able to carry out the calculations in a time frame consistent with performance code calculations, i.e. at or near real time.
- 5) Be capable of being calibrated against engine test data.

The strategy adopted by the author in an attempt to meet these requirements was to further develop the GTTrans thermomechanical heat soakage and tip clearance methods within the context of the Rolls-Royce Trent 500 BD31 version of the RRAP performance code. From the studies presented in this chapter we can see how the methods developed by the author meet the project requirements, i.e.

- | | |
|--|-----------------------------|
| 1) Both tip clearance and heat soakage | - YES |
| 2) Physics based | - YES |
| 3) Accuracy | - tips NO, heat flows MAYBE |
| 4) Run at or close to real time | - YES |
| 5) Capable of being calibrated | - MAYBE |

In the opinion of the author, with enough refinement the accuracy of the streams method models could be improved to the extent that project accuracy requirements would be met. Also, if enough time is given to the calibration process, then streams method models could be adequately calibrated against more authoritative data from test or from finite element analyses.

Chapter 5: TECHNOLOGY TRANSFER

5.1 CHAPTER INTRODUCTION

The EngD programme requires that candidates address not only technical issues associated with their research project but managerial issues as well. The purpose of this chapter is to address the management topic associated to this research project.

Rolls-Royce PLC is a technology driven company. Its gross research and development for the first half of this year was approximately 10 percent of turnover (Rolls-Royce 2004). Part of this money is used to support more than 20 University Technology Centres (UTC) in the UK, Europe and the US (Rolls-Royce 2003). The UTC in Gas Turbine Performance Engineering at Cranfield University was established March of 1998, since then the UTC has produced a considerable number of high quality MSc, PhD, and EngD theses. Many of the methods produced by the UTC have already become part of the Rolls-Royce Performance Department's tool kit. In a typically proactive attempt to more fully leverage the research work coming out of the UTC, it was decided by Rolls-Royce (see Naylor 2002a) that the management issue to be addressed in this EngD project was to be the process of technology transfer from UTC to Rolls-Royce.

In this chapter the author presents a concise review of the relevant open literature, followed by an assessment of the issues associated with the process of technology transfer from UTC to Rolls-Royce, and finally suggests possible ways in which the process might be improved.

5.2 RELEVANT LITERATURE

The open literature contains many thousands of academic publications relating to the topic of technology transfer. A significant portion of this number relate to the transfer of technology developed within universities into the private sector. Good examples are Trot et al (1995), Gilbert & Cordey-Hayes (1996), Rogers et al (2001), and Siegel et al (2004). These authors between them propose a number of generic technology transfer models which all contain essentially similar elements. At the time of writing the author was only able to find a single paper that specifically related to the transfer of technology developed at a Rolls-Royce UTC into Rolls-Royce working practice. In this paper, Smith et al (1998) discuss the transfer of the ConCERT technique (Vickers et al 1996) developed at the Systems and Software Engineering UTC at York University to within Rolls-Royce. The ConCERT technique is a structured and disciplined approach to the specification of complex engineered software systems. The technology transfer model described by Smith et al (1998) is not unlike those presented by other authors. It contains five stages (c.f. figure 5.2.1): Observation, where industrial practice is observed and analysed to identify opportunities for improvement; Innovation, where new technology is developed; Acceptance, gaining agreement to apply the technology within the industrial environment; Implementation, where academic theory is applied within the industry; Maintenance, where industry

incorporates it into working practices, this is seen as the point at which technology transfer has been achieved.

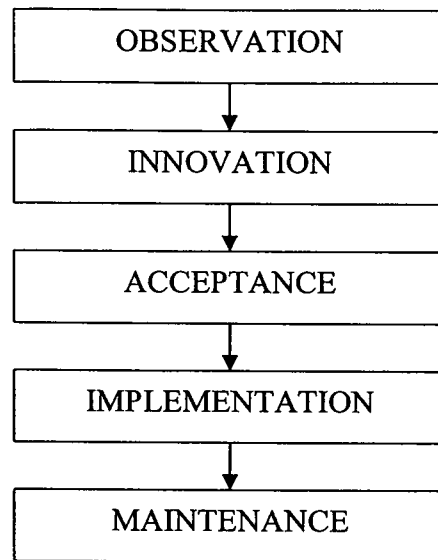


Figure 5.2.1 Technology Transfer Process Model (Smith et al 1998)

In their paper Smith et al (1998) identify a number of different barriers that may affect or prevent the technology transfer process, these are: knowledge barriers, social barriers, and cultural barriers.

Knowledge barriers can exist between the academic and the industrialist. The academic will possess broad theoretical knowledge about the problem domain they are working within. The industrialist on the other hand will possess detailed and specific knowledge about the particular problem area being investigated. Both parties therefore will possess a different understanding of the problem domain, and even a different vocabulary to describe that domain. This difference can result in the two parties effectively speaking different languages.

Social barriers arise from the way in which people interact and develop the relationships necessary to enable the technology transfer process to occur. Key to this relationship is trust and respect, which is built through shared knowledge and understanding, and the establishment of common ground. An industrialist who delivers a presentation to other industrialists achieves a greater degree of acceptance than when the same thing is presented by an academic, and vice versa. Further differences can derive from personality conflicts and the differences in working style between academics and industrialists.

Cultural barriers relate to the inherent differences between academic culture and the industrial culture, in terms of the different objectives, motivations, mechanisms for work and internal pressures that people are subject to. A common barrier to

academic-industrial relationships is the academic need to publish in order to progress their career, in contrast to the industrial requirement for commercial confidentiality. Industrialists are subject to the demands on constraints of time, cost, and resource. Academia, on the other hand, is interested in understanding and the furthering of knowledge, and by virtue of their environment, academics are not subject to the same time constraints and pressures. When working together, this can lead to very different objectives and expectations from the technology transfer process.

Although classified by slightly different terms, other researchers (e.g. Rogers et al 2001, Polak 2002, Boulter & Bendell 2002), identify essentially the same issues as being the major obstacles to the technology transfer process.

One particularly significant barrier to the university to industry technology transfer process is the means by which it most often takes place, i.e. academic publications. Rogers et al (2001) conducted a study of 55 university based industrially sponsored research centres. They report that academic publications are mainly written for other academics rather than for potential users, and are not an effective means of technology transfer. In echo of this sentiment, it was stated by Rolls-Royce at the 4th EngD steering group meeting for this project that significant cost, in terms of both money and time, can be incurred translating new methods documented in a UTC thesis into a tool that is usable in the Rolls-Royce working environment (Naylor 2002a).

5.3 ANALYSIS OF THE CURRENT SITUATION

5.3.1 ROLLS-ROYCE PERFORMANCE FUNCTION

The Rolls-Royce performance function is responsible for ensuring that engine performance (i.e. thrust, fuel consumption, stability) meets the requirements laid down by airworthiness authorities and meets guarantees made by Rolls-Royce to the customer.

The Rolls-Royce performance function is involved with an engine throughout its whole life cycle from the preliminary calculations made at its first conception all the way through to support of engines in service. The engineering process is shown in figure 5.3.1.

The Rolls-Royce performance function can be divided into three groups: the advanced propulsion design team, the core performance team, and the engine project performance teams.

The advanced propulsion design team produce engine propulsion concepts to meet medium and long term market requirements. Their main activities are associated with the early stages of new engine design.

The project teams are responsible for all of the performance function related activities (modelling, testing, etc) that correspond to particular engine project following its release by the advanced projects team. Project teams grow and shrink depending on

what stage of figure 5.3.1 the particular engine project is at. The project team for an engine preparing for certification by airworthiness authorities may have tens of engineers in it, whereas a single smaller project team may be sufficient to oversee performance activities for more than just one type of in service engine. The project teams are responsible for the performance of engine types following the modifications and upgrades that inevitably occur throughout its in service life.

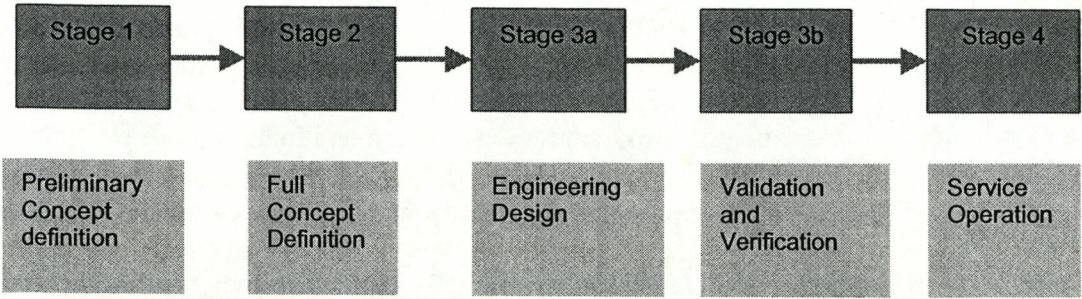


Figure 5.3.1 The Engineering Process (Nayer 2004)

The core team is responsible for producing and maintaining the methods and tools used by the various performance function teams.

The project teams work to strict budgets and deadlines and are under no obligation to use methods produced by the core team if such a method does not deliver benefits that exceed the cost of learning and using it. Because of this, the core team generally focuses on producing methods that are specifically requested by the project teams since producing unrequested, or more specifically, unused methods represents a waste of core team time and money.

The performance function leaders are responsible for ensuring the continual improvement of performance function capabilities. In recent years they have worked with academics from the Cranfield University School of Engineering to set up at Cranfield the Rolls-Royce UTC in Gas Turbine Performance Engineering. From a Rolls-Royce perspective the purpose of the UTC is to assist the performance function in developing new modelling and analysis techniques, to help attract external funding to such research activities, to access Cranfield's internationally respected academic capability base, and to provide a source of well qualified future performance engineers.

5.3.2 UTC IN GAS TURBINE PERFORMANCE ENGINEERING

The Cranfield UTC was established in 1998, and started with just one PhD student. Since then it has grown quite significantly. As the UTC has grown and matured there has been a shift toward greater reliance on business processes based on lessons learned along the way. The UTC continues to change and improve. The paragraphs that follow contain a description of its basic structure as it stands at the time of writing.

ROLLS-ROYCE ROLES:

The UTC Owner is ultimately responsible within Rolls-Royce for the Cranfield UTC. He must justify all UTC activities, and competes for funding from the pool of money set aside to pay for the entire network of Rolls-Royce UTCs around the globe. He is responsible for UTC strategy and direction.

The UTC Coordinator is responsible for ensuring that each work package stays on track, is properly supported by Rolls-Royce, and delivers what the performance function needs. He actively supports the UTC Owner with strategy decisions, however, the UTC Coordinator is much more involved operationally.

Theme Owners are technical experts who are responsible for supporting a particular research theme, which is typically comprised of more than one work package. At the time of writing there were ten Theme Owners. Theme Owners are involved operationally rather than strategically.

CRANFIELD UNIVERSITY ROLES:

The UTC Director is ultimately responsible within Cranfield University for the Cranfield UTC. He is responsible for working with the UTC Owner in defining strategy and direction.

The UTC Deputy Director's role is similar to that of the UTC Director in that it supports strategy, however the UTC Deputy is much more involved operationally with each project. The UTC Deputy is responsible for ensuring each work package meets academic requirements.

The UTC Administrator is responsible for organising meetings, keeping records, and ensuring that all parties are made aware of relevant issues at appropriate times. The UTC Administrator is the first point of contact between the two organisations.

Research Officers and Research Assistants carry out the research associated with each of the UTC work packages. They are permanent members of staff who may be associated with more than one work package at any one time.

Research students are the PhD and EngD students. Each research student carries out the research associated with a single work package. Research students are generally part of the UTC for only the duration of their degree programme, i.e. 3 or 4 years.

MSc students are students studying for an MSc in Thermal Power at Cranfield University. The MSc is marked half by written examinations and half by thesis. MSc students carry out work associated with a single work package, and are part of the UTC for just one year.

The latest agreed structure (as of August 2004) consists of 2 Research Officers/Assistants, 4 PhD/EngD research students, and 8-10 MSc students, in addition to the UTC Director, Deputy Director, and Administrator.

UTC MEETINGS:

The UTC Owner and UTC Coordinator meet regularly within Rolls-Royce to discuss UTC strategy. The UTC Director and UTC Deputy Director do the same at Cranfield.

Quarterly management meetings are held where the UTC management group (i.e. UTC Owner, UTC Coordinator, UTC Director, and UTC Deputy Director), and others by invitation, review UTC progress and revise UTC strategy.

Quarterly technical review meetings are held where the UTC Coordinator and UTC Deputy Director meet with research staff and students to assess the technical progress of each work package.

Meetings between research staff, students, and theme owners take place as and when they are felt necessary. Meetings between students and their academic supervisors (either research staff or the UTC Director or Deputy Director) take place at least monthly but usually much more frequently.

Every year there is also a UTC annual review consisting of a series of management meetings and technical presentations. Individuals from within Rolls-Royce (non-Cranfield UTC) and also outside of Rolls-Royce (e.g. representatives from the Department of Trade and Industry) attend this event.

5.3.3 TECHNOLOGY TRANSFER ISSUES

As part of his investigation into technology transfer issues specific to the Cranfield UTC, the author carried out a number of interviews. He met with the UTC Director and UTC Deputy Director at Cranfield, and spoke with the UTC Owner and also the head of the core performance team at Rolls-Royce. In addition, the author scrutinised the Cranfield UTC quality plan, and also the Rolls-Royce generic quality plan for the development of new or enhanced performance methods software, i.e. Widdowson & Nayer (2003).

During the investigations the author accumulated a body of information relating to the issues and challenges associated with Cranfield UTC to Rolls-Royce technology transfer. The details of that information are very interesting but are not meant for an audience as broad as that of this thesis, however the following paragraphs describe the more general relevant issues.

From the interviews it was found that the academics viewed the activities of the UTC slightly differently to the industrialists. The academics focused much more on creativity, new concepts, and exploration, whereas the industrialist's focus was on the delivery of more near term benefits. This is hardly surprising given the nature of their two environments. Indeed, the exact same observation can be seen in the open literature (e.g. Smith et al 1998, Polak 2002).

From a study of the quality plans it became apparent that even though work carried out within the scope of each plan could often be related, neither plan made any mention of the existence of the other plan.

From the interviews with Rolls-Royce it was discovered that not all of the methods that had already been successfully incorporated into the Rolls-Royce performance methods toolbox were actually being used by the engine project performance teams. An example was given of a novel method produced by the UTC that the core performance team tried to incorporate into the Rolls-Royce environment. The engineers found it difficult and time consuming. One problem was that documentation associated with the method was not always helpful. This was further complicated by the fact that the method was developed in complete isolation from the Rolls-Royce working environment. Another example was given where a method produced by the UTC was successfully incorporated into the Rolls-Royce environment but an engine hardware design change resulted in the method becoming no longer of any use. Another example was given where a UTC method was technically very good and was successfully incorporated into the Rolls-Royce environment but the amount of work involved with preparing input data for the new method was found to be prohibitive, and the quicker to use existing method continues to be used by Rolls-Royce engineers.

Even though the UTC methods in these examples each met their technical objectives and all became available for use in the Rolls-Royce environment, it appears that the methods were not used in practice because of reasons outside of their technical project objectives.

From the author's investigations, the main issues relating to Cranfield UTC technology transfer appear to be requirements specification, method development environment, and method documentation.

The problem of method documentation basically reinforces what was reported by Rogers et al (2001), i.e. academic publications are mainly written for other academics rather than for potential users, and are not an effective means of technology transfer. This problem is quite a difficult one because the nature and format of academic publications have been fixed by many years of historical precedent, and are unlikely to change in the foreseeable future. When a university delivers say a PhD thesis to a company, the university may argue that technology transfer has taken place. At this point however, the company still has a mammoth task to perform before the technology described in the thesis can be used to deliver economic benefit. Only then is the company likely to agree that technology transfer has finally taken place.

Requirements specification is really the key issue. The problem of method development environment versus method usage environment is directly related to it. If UTC projects are specified incompletely, or without the specification of a development platform (e.g. computer programme), or without the involvement of the end users, then any gaps in the specification will be filled in, by default, by the research and MSc students, whose main objectives are to write a thesis and gain a qualification. This situation will produce UTC methods that meet their project specification, but fail to meet the needs of end users.

The issue of requirement specification is currently receiving attention from the Cranfield UTC strategy makers. One of the reasons why the author was set to task assessing and making recommendations on the issue of technology transfer was to

provide a degree of input to this process. As the UTC has grown and developed, lessons have been learned and changes have been made. Some of the changes currently taking place are the shift away from such UTC research topics as engine diagnostics and heat soakage (alas!), and a shift toward novel engine cycles in support of the advanced projects team, and greater focus on engine test bed analysis, windmilling, and in flight relight.

5.4 THE AUTHOR'S PROPOSAL

This section outlines the author's proposal for an addition to the Cranfield UTC quality plan. If adopted the proposed addition would serve to link the Cranfield UTC quality plan to the RR quality plan (see Widdowson & Nayer 2003), and also address the issues of requirements specification and documentation.

The brief for the management topic assigned to this EngD project was to address the process of technology transfer. It was not to address university creativity. Creative thinking is one of the things that make universities like Cranfield such an attractive partner for industrial research. Imposing too many controls and constraints can easily stifle creativity. It is not in the interest of Cranfield or Rolls-Royce to do anything which would devalue their relationship. The constraints and controls proposed in this section are intended to only to inform the discussion on current UTC reforms to the technology transfer process, rather than prescribe the exact course of action.

The basic idea behind the author's proposal is a stage/gate process. The stages in the process are not completely dissimilar to what already happens within the UTC, the main difference is in the gates. The early gates are intended to ensure that proposals entering the UTC are properly specified. Later gates are intended to ensure that the documentation associated with a particular UTC work package is prepared in such a way as to facilitate the passage of the new method through the Rolls-Royce generic quality plan for the development of new or enhanced performance methods software (see Widdowson & Nayer 2003). Working according to the proposed process should result in UTC-ready project proposals, and Rolls-Royce-ready thesis packs. A schematic diagram of the proposed process is shown in figure 5.4.1. Descriptions of each stage and gate included in the process are given below.

The roles of people involved in the process are the same as those that exist at present, however there is one addition, i.e. the Rolls-Royce end user representative.

Abbreviations used hereafter are:

RR	Rolls-Royce
RRO	UTC Owner
RRC	UTC Coordinator
RRT	UTC Theme Owner
RRE	Rolls-Royce end user
CU	Cranfield University
CUD	UTC Director
CUV	UTC Deputy Director
CUR	UTC research officers & assistants, research students & MSc students

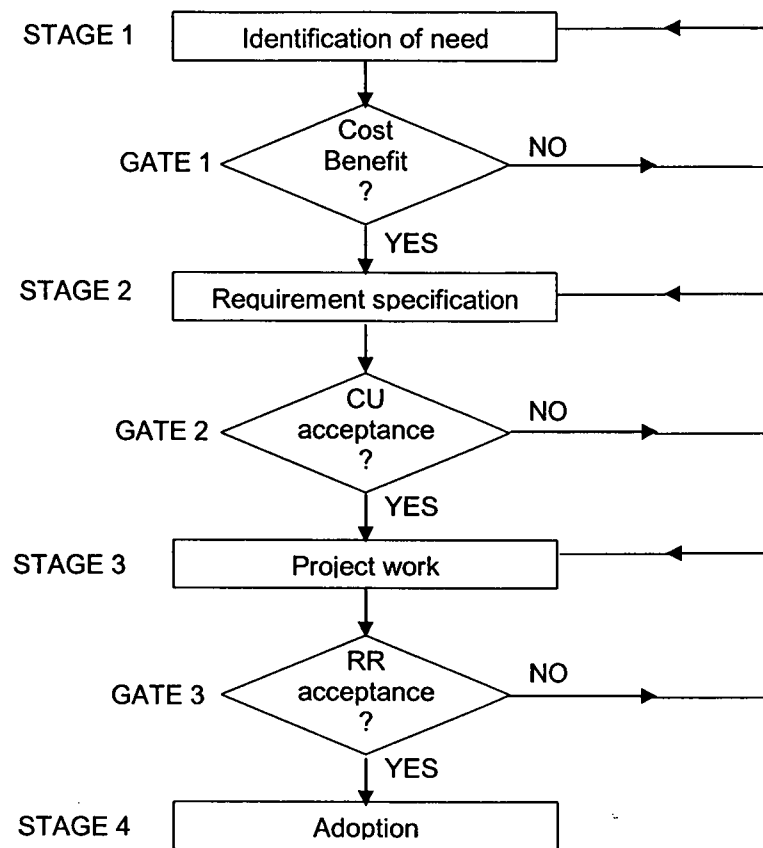


Figure 5.4.1 The proposed technology transfer stage/gate process

STAGE 1a: IDENTIFICATION OF CURRENT/FUTURE NEEDS

People involved: RRO, RRC, RRT, RRE, CUD, CUV

This stage is essentially about strategy. At this stage RR and CU work together to identify where existing RR methods might not meet current/future needs. Discussions might take place regarding the expected future operational activities of RR engine project performance engineers, given macroeconomics, market trends, and competitor behaviours; emphasis would be on the long-view. Outside thinking from CUD and CUV is likely to be especially valuable at this stage. Already, the UTC management team hold regular (quarterly) meetings to discuss this topic, and RR technology strategists have their own six-monthly internal meetings to discuss these issues.

The output from this stage would be lists of possible UTC projects that both RR and CU agree would be of real long-term benefit to RR and require sufficient creative contribution to be worthy of an academic degree. This list would be prioritised and projects would be passed on to stage 1b starting from the top of the list.

STAGE 1b: PREPARATION OF GATE 1 DOCUMENTS

People involved: RRO, RRC, RRT, RRE, CUD, CUV

Gate 1 is the go/no-go decision for the project based upon a cost/benefit analysis. Information needed to assist with the decision is contained in the gate 1 documents. Preparation of gate 1 documents is very much a RR activity because it requires detailed knowledge of RR performance department operations and methods, however input from CU might be needed to ensure from as early as possible that project proposals meet academic degree requirements. Gate 1 documents are as follows:

Operational expectation statement

Brief statement of the expected current/future operational activities of project team engineers in the RR performance department.

Appraisal of existing RR methods

Brief description of relevant existing RR methods (with references) and their capabilities, and how each does/will under-perform in light of the operational expectation statement (i.e. the premise for the summary of requirements)

Summary of requirements

Prioritised list of new method requirements/deliverables, quantified where possible, including expected levels of difficulty and/or likelihood of achievement.

Cost benefit breakdown

Worked calculations of the productivity and cost improvements that could be realised if project engineers had access to a method that fulfilled the general summary of requirements. In addition, worked calculations of the learning times, set up times, run times, and roll out cost targets that would have to be met for the project to deliver a positive net present value.

These documents would not have to total more than two pages per project idea. The preparation of these documents would probably require a little more time than is spent on current project ideas at this stage in the UTC process, however the work does not necessarily have to be done by a single individual. Extra hours spent here would ensure that MSc projects are properly aligned to Rolls-Royce interests.

GATE 1: COST BENEFIT DECISION

People involved: RRO

Based upon an inspection of the cost/benefit analysis and other gate 1 documents, the UTC Owner would decide whether or not to fund UTC development of the proposed new project, writing the reason for the decision on the gate 1 documents. If the

decision were to be negative then the proposal would be archived for future reference, if positive, it would then pass to stage 2.

STAGE 2: REQUIREMENT SPECIFICATION (GATE 2 DOCUMENTS)

People involved: RRO, RRC, RRT, RRE, CUD, CUV

Without the information contained in the gate 2 documents, the burden of detailed definition of the project defaults to the CUR. CUR are generally unfamiliar with RR operations. If the details of project definition are left to the CUR then the end results will probably emphasise the criteria required for an academic publication (e.g. thesis), rather than those required for effective technology transfer.

It is essential that RRO and RRC, or their representatives, prepare gate 2 documents in consultation with one or more RRE who will have access to the kind of detailed information that is needed at this stage. Anything left undefined by RR will by default be defined by the CUR. In addition CU academics should be consulted to ensure that projects proposed have sufficient academic content to meet the requirements of a degree. Gate 2 documents are:

Gate 1 documents, updated as necessary
See above

Team list

This is a list of names, contact details, roles, and responsibilities of those assigned to assist the CUR. Such people might include the RRT, the academic supervisor, the associated RRE, a relevant RR computing platform expert, etc.

Usage details

This document supports the summary of requirements from the gate 1 documents. This document should include the following items

Which phases of the engine development process the method will be used (aftermarket, preliminary design, etc), by who, and to do what

A list of data that will be available to the end-user as input to the method at the expected time of use, i.e. stage in the engine development process (e.g. station vector data, geometry, engine test data, etc.), and also its format (e.g. spreadsheet, text file, ALICE, etc.)

A list of output data the method will be required to produce, and the required accuracy or certainty levels (for comparison against existing RR methods), and also the required format (e.g. text file, spreadsheet)

Target learning, set up, and run times (for comparison against existing RR methods)

The required computing platform the method is to be developed upon by the CUR at the UTC, e.g. spreadsheet, stand alone C++/FORTRAN, RRAP customer deck (which version), Matlab, etc.

NOTE: Although the usage details document clearly prescribes the project deliverables it places no restraint whatsoever on how the deliverables should be achieved. In this way creative input from the CUR would be promoted.

The gate 2 documents are unlikely to require more than a couple of pages of writing, but are likely to result in a step positive change in CUR productivity, who at present typically spend many months either formally or informally trying to figure out what project deliverables should be rather than creatively exploring how they might be delivered. Every hour spent by the RRO, RRC and RRE adding detail at this stage is likely to save many months of CUR time trying to achieve the same thing without any of the relevant knowledge or information.

GATE 2: UTC ACCEPTANCE DECISION

People involved: RRO

Gate 2 is the go/no-go decision for proposals going into the UTC. For a proposal to pass this gate it must meet RR performance function needs, meet the academic requirements of a higher degree, and contain sufficient information to be given to a UTC researcher to productively start work upon. Under-defined project proposals are unlikely to deliver the required results.

The reason why the RRO should be responsible for this gate is because he is responsible for the RR money spent on the UTC. Each UTC proposal would be scored against some agreed standard. Letting only high scoring (i.e. properly defined) projects into the UTC will increase the probability of useful project outcomes and will promote long-term UTC profitability.

The scheduling of gate 2 activities must take into account the timing of the academic year.

STAGE 3a: PROJECT WORK

People involved: CUR, RRT, RRE, Academic Supervisor, RRC, CUV

The first activity at this stage would be an appraisal of open literature methods in light of the operational expectation statement (gate 1 document), to complement the appraisal of RR existing methods (gate 1 document). Next, the CUR would work on concept design, consulting with the RRT, academic supervisor, and RRE as necessary. Each design to be investigated would require a verification plan, again created in consultation with the RRT and the academic supervisor. Once the postgraduate's concept designs have been approved at the regular quarterly UTC technical review meetings, he would then start the development, programming, verification testing, etc. Some projects may just involve concept design and verification, others might also include some validation, other projects might be revisits of previous UTC projects, in which case the project's gate 1 and 2 documentation will have been updated on their way to stage 3.

STAGE 3b: PREPARE GATE 3 DOCUMENTS

People involved: CUR, RRT, RRE, Academic Supervisor

At this stage the CUR writes up his work. It may be useful for the UTC to lay down specific formats for each type of report. For UTC research students submitting project theses it is likely that a number of non-thesis documents would also be required. These extra documents would contain information for the specific purpose of assisting the preparation of the RR quality plan gate documents. These documents would not have to be very long or have to take long to prepare, but nevertheless would be valuable for technology transfer purposes. Gate 3 documents would include:

Gate 1 documents – updated as necessary

See above

Gate 2 documents – updated as necessary

See above

Academic thesis if part of a higher degree programme

To the required academic format (hardcopy or electronic copy)

UTC report if not part of a higher degree programme

To the required format (hardcopy or electronic copy)

Source code

Perhaps written to UTC software standards (i.e. rules on commenting and variable names)

Verification data – if not fully presented in thesis

RR quality plan gate document (Widdowson & Nayer 2003)

Verification plan – if insufficient verification testing was done in thesis

RR quality plan gate document (Widdowson & Nayer 2003)

Validation data – if not fully presented in thesis

RR quality plan gate document (Widdowson & Nayer 2003)

Validation plan – if insufficient validation testing was done in thesis

RR quality plan gate document (Widdowson & Nayer 2003)

Training documentation plan

RR quality plan gate document (Widdowson & Nayer 2003)

These documents together would become the UTC project thesis/report pack. The extra documentation would not necessarily have to involve lots of work, or all be written by the CUR. Involvement of the RRT and RRE would ensure that a much better job would be done due to their greater familiarity with Rolls-Royce quality procedures.

GATE 3: RR ACCEPTANCE DECISION

People involved: RRO, RRC, CUD, CUV

At this gate the decision is made as to whether to pass the method (i.e. thesis pack) to stage 4, to shelf it, or to put it back through the UTC another time. The thesis pack could be scored against some agreed standard to help make the decision. A low scoring thesis pack may be the result of a low scoring proposal (process feedback).

STAGE 4: ADOPTION

People involved: RRE

The RR end user representative drives this process (Widdowson & Nayer 2003). His task will have been made much easier due to the presence of Rolls-Royce-ready thesis packs. The information included in the thesis packs will provide much of what is required by the RR quality plan gates. A high scoring thesis pack may even make it possible to pass through more than one RR quality plan gate at a single sitting.

ADDITIONAL COMMENTS

The UTC is not a software house. The proposed process recognises that in the majority of cases a UTC project will not produce a plug and play tool or method. The role of the UTC is the development and proof of concepts. Indeed, whatever comes out of the UTC still has to pass through the RR internal quality control process before the core performance team can release it into the user community. The proposal outlined above is intended to facilitate that process while at the same time respecting the roles of the parties involved.

The proposed process excludes UTC researchers from the project deliverables definition stage. This is left to the UTC management in collaboration with the Rolls-Royce end user representative (i.e. the one who will have to use it when it is finished). In the proposed process, the role of the UTC researcher in developing a new method is restricted to that of taking whatever inputs will be available to the end user at time of use, and applying fresh thinking and creativity to the problem of how to use these inputs to deliver the outputs required by the end user.

Admittedly the proposed process involves more paperwork than is involved in the current UTC arrangement. In stages 1 and 2 of the process the additional paperwork is there to ensure that the projects proposed actually deliver benefit to Rolls-Royce, and give the researcher clear instruction regarding project deliverables. In stages 3 and 4 of the process the purpose of the paperwork is to facilitate the preparation of documents required by the Rolls-Royce quality control process.

The author has identified where the proposed process would most suitably be inserted into Cranfield UTC quality plan, i.e. Chapter 9 Configuration Management, which deals with the procedures controlling project definition, deliverables, documentation, source code, etc.

One suggestion has been that the documentation involved with the proposal is too much for an MSc project, and that the current system of single paragraph MSc project proposals written without involvement of an end-user is sufficient. Also, that the documentation outlined in the proposal should only be put together for PhD/EngD projects, and that the PhD/EngD candidates should do the putting together. The argument is that this would help PhD/EngD candidates build contacts within Rolls-Royce. Of course, the UTC management group will have to decide for itself whether or not it is happy with the status quo.

5.5 HEAT SOAKAGE AND TIP CLEARANCE METHODS

The author has tried during the development of his heat soakage and tip clearance methods to address some of the technology transfer issues associated with this project. The author used the Rolls-Royce RRAP performance code as the platform upon which to develop the methods as opposed to other non-Rolls-Royce options, and made detailed studies of the methods currently used in the Rolls-Royce environment, as well as investigating the open literature. Furthermore, from the start he made a special effort to make lots of visits to Rolls-Royce and to maintain a dialogue with the relevant company experts. Indeed, a trend has now started in that other Rolls-Royce sponsored EngD researchers are following suit and are spending more time at Rolls-Royce discussing project related issues.

Despite these technology transfer efforts, it has been decided by Rolls-Royce that the heat soakage and tip clearance methods developed by the author will not at this time be taken further and implemented into the Rolls-Royce performance function toolkit. Rolls-Royce has decided that the Lewis method will be taken forward and developed further (at the time of writing, a Lewis method model for the Trent 500 high-pressure turbine was being constructed).

The heat soakage and tip clearance methods developed by the author either fully met or partially met all of the project requirements laid down by the sponsor (see chapter 4), nonetheless, the decision was made that the author's methods would not be taken any further by Rolls-Royce at this time. The first reason for this was that input file set up times for the methods were excessively long. The second reason was to do with how the various departments within Rolls-Royce are organised and interact with each other. The performance department often need to know compressor tip clearances for different flight manoeuvres. They get this information from the compressor thermals department who are the ultimate authority on compressor tip clearance values within the company. If the performance department use tip clearance values that do not come from or are exactly the same as those issued by the compressor thermals department, then information quality controls will have been compromised, and performance department calculations that make use of tip clearance values will be without guarantee. Interestingly neither of these two issues (i.e. set up time and authority of information) appeared on the original list of project requirements. Even if the authors methods had fully met all of the requirements specified at the start of the project then these two issues would likely have led to an identical conclusion, i.e. no further development of the author's methods at this time. This reiterates the importance of detailed project specifications written with end user involvement. Nevertheless, the author is fully confident that the changes currently

taking place within the UTC will lead to the prevention of similar circumstances in the future, and it is hoped that the proposal and recommendations presented in this chapter will usefully inform this current process of change.

5.6 RECOMMENDATIONS

As mentioned previously the Cranfield UTC is currently undergoing a number of changes. The author's recommendations in relation to this process of change are as follows:

The UTC technology transfer control process outlined in section 5.4 should be incorporated into chapter 9 of the Cranfield UTC quality plan. Benefits would include greater clarity of project requirements primarily through involvement of end users, and greater coordination between the Cranfield UTC and Rolls-Royce quality control systems through improved new method documentation. Although some new forms would need to be created, wherever possible existing forms (i.e. from the Cranfield UTC quality plan, or from the Rolls-Royce quality control system) should be used in the preparation of such documentation.

UTC software standards should be implemented to reduce the often lengthy amounts of time UTC researchers have to spend deciphering code written by previous researchers. Standards governing commenting, explanations of variables, national language used, units, etc. need not be overly complicated, however many man-hours could be saved if good programming practices were implemented.

In addition, the author makes recommendations to echo those made by Rolls-Royce in the interviews carried out as part of this technology transfer investigation, i.e. the problems associated with development of methods in isolation from the Rolls-Royce environment could be mitigated if UTC researchers spent more time at Rolls-Royce learning about usability requirements, software compatibilities, etc. In addition, technology transfer would also be helped if UTC personnel were more familiar with the Rolls-Royce quality plan.

5.7 CHAPTER CONCLUSION

The purpose of this chapter was to address the issues associated with the process of technology transfer from Cranfield UTC to Rolls-Royce. Section 5.2 gave a brief review of some of the relevant open literature. Section 5.3 followed with a description of the current Cranfield UTC arrangement. Section 5.4 described some specific technology transfer issues that are faced by the Cranfield UTC, i.e. project specification, method development environment, and method documentation. Section 5.5 outlined the author's proposal for an addition to the Cranfield UTC quality plan, which if adopted would help address those specific issues. Section 5.6 discussed technology transfer issues relating to the heat soakage and tip clearance project. And finally, the chapter ended with a number of recommendations that it is hoped will be of assistance to the UTC management group as they go about making changes to the Cranfield UTC.

Chapter 6: CONCLUSION

6.1 CHAPTER SUMMARY

Chapter 1 of this thesis outlined the Cranfield University Engineering Doctorate degree programme and introduced the Rolls-Royce University Technology Centre concept. It then went on to identify the category of academic contribution made by the work described in this thesis. The chapter ended with an outline of the remaining chapters of the thesis.

Chapter 2 of this thesis described the business case that led to sponsorship of this EngD project. It also gave a review of the relevant open literature, i.e. whole engine performance modelling; heat soakage phenomena; and existing heat soakage phenomena models. The chapter also listed the project requirements laid down by the sponsor, identified the research gap to be filled by this piece of research, and defined project scope. The purpose of the chapter was to justify the project scope and objectives.

Chapter 3 of this thesis described the approach taken by the author to model convection, conduction, radiation, and thermomechanical engine geometry changes that occur transiently within a gas turbine engine. Sample calculations were compared against studies using the finite element analysis package ANSYS®. The chapter also described how the author integrated the new methods into the whole engine transient performance code provided by the sponsor. The purpose of the chapter was to present the heat soakage and tip clearance models developed by the author.

Chapter 4 of this thesis described the results of a series of studies making use of the methods developed by the author. The first study involved use of the new heat soakage and tip clearance methods to construct a multistage model of a Rolls-Royce Trent 500 high-pressure compressor. The results it produced were compared to those produced by BR081 and the Lewis method. The second study involved constructing a simplified single stage model of the same compressor and comparing the results it produced to those produced in the first study. The third study involved the new methods being used to construct multistage models of all of the turbomachinery (except fan) in the Trent 500 engine. These models and methods were then incorporated into the Trent 500 transient performance code provided by the sponsor - the BD31 version of RRAP. Comparative transient test runs were made between the original engine code, the code updated with the new models and methods, and a set of test data provided by the sponsor. The chapter also presented a discussion of a method whereby it might be possible to extract steady state engine performance information from transient test data. The main purpose of chapter 4 was to establish the extent to which the heat soakage and tip clearance methods developed by the author meet the project requirements laid down by the sponsor. The chapter showed that the author's heat soakage and tip clearance methods met 3 of the 5 requirements laid down by the sponsor at the start of the project. In the opinion of the author, further refinement of the methods and models would probably lead to all 5 requirements being met.

Chapter 5 of this thesis addressed the management topic associated with this EngD project. It gave a brief review of open literature relevant to the topic of technology transfer from university to sponsor. It then described the Cranfield UTC arrangement as it currently stands. The chapter then went on to describe the technology transfer issues faced by the UTC, i.e. project specification, method development environment, and method documentation. It then described the author's proposal for a possible addition to the Cranfield UTC quality plan that would help address those specific issues. The chapter also discussed technology transfer issues relating to the heat soakage and tip clearance methods developed by the author. The chapter ends with some recommendations, namely, the adoption of the proposed addition to the UTC quality plan, the adoption of UTC software standards, and more time spent by UTC researchers at Rolls-Royce gaining familiarity with the end user environment and Rolls-Royce quality plan.

6.2 CONTRIBUTION

The academic contribution made by the work described in this thesis falls into the category of 'improvement of an existing design'. This thesis describes significant improvements that have been made to the GTTrans thermomechanical heat soakage and tip clearance methods. These improvements have been made by the author in the following ways:

The original methods use a lumped capacitance approach to model transient convective heat transfer within the components of a gas turbine engine. Within a gas turbine the effect of conduction on transient heat transfer can be significant, especially where engine metal is thick (e.g. disc cobs). Also, under steady state conditions conduction leads to heat leakage out of the primary gas path into the bleed air paths, and this can affect the steady state performance of the engine. In the new methods developed by the author, conduction is accounted for using a two dimensional axisymmetric finite volumes representation of component hardware, and convection from non-trivial component hardware geometries is managed by what the author has called the streams method. These two developments have significantly improved upon the existing GTTrans transient heat transfer calculation methods.

In the original methods the thermomechanical disc growth model divides each disc into three sections: rim, diaphragm, and cob, each section having its own temperature. The temperature of the whole disc is assumed to be the mass weighted average of the three sections and is used to calculate thermal disc growth. Mechanical growth is estimated assuming that the disc has only a single thickness equal to that of the diaphragm. The sum of thermal and mechanical growth is factored by a value found from a separate more detailed disc study to give total disc growth. In the new methods developed by the author, a greater degree of sophistication is used. The disc is divided into a number of sub-discs/rings of hyperbolic section such that the geometry of the disc is more accurately represented. The temperature of each ring is calculated using the finite volumes method (above). Ring interface pressures are iteratively guessed and ring thermomechanical growths are calculated until the gaps between the rings are within a given tolerance. As has been shown in this thesis, the

rings method gives more representative results than the original GTTrans thermomechanical disc growth method.

In the original methods multistage components are represented by a single averaged stage. The geometry of this single stage is defined by taking the average of each of the component's individual stage geometries. A study described in this thesis has shown that the tip clearance calculated using a single averaged stage can be quite different from the average of stage tip clearances calculated using a multistage model. Also, the fact that forward stages preheat/precool air before it interacts with rearward stages means that heat flow predictions from single averaged stage model are much larger than those produced by multistage models.

In the original methods, bleed air was assumed to behave as a temperature reservoir, maintaining the same temperature from the point at which it was bled all the way through until it remixed with the primary gas path. In the new methods developed by the author, the streams method is used to update bleed temperatures as a result of convective heat transfer and windage heating in the bleed cavities. The effect of these two sources of bleed heating can be particularly significant in multistage disc cavities, where the cumulative effect of windage heating affects the temperatures of particularly the downstream discs thus having a considerable effect on component tip clearance calculations.

The author has also made a contribution toward the management issue dealt with in this EngD thesis. Following investigations in to the technology transfer issues facing the Cranfield UTC, the author has proposed a process which can be conveniently added to the Cranfield UTC quality plan. If adopted the stages and gates which constitute the proposed process would result in better defined UTC project proposals and Rolls-Royce-ready UTC methods and tools.

6.3 RECOMMENDATIONS

Recommendations regarding the process of technology transfer from Cranfield UTC to Rolls-Royce working practice can be found in section 5.6. The recommendations described below relate more to heat soakage and transient blade tip clearance modelling.

The author recommends that heat soakage and tip clearance methods to be used within whole engine performance codes should follow the way things are usually done in the rest of gas turbine performance engineering, i.e. via performance characteristic maps, or alternatively make use of characteristic equations such as those of the Lewis method or Khalid and Hearne (1980).

Performance maps are created during the process of new engine development that characterise the aerodynamic and thermodynamic performance of individual engine components. These maps are derived first from heavyweight numerical tools, such as computational fluid dynamics, and are later derived from test data. The data held on these maps is the most accurate available and is easy and quick to access via a variety of look up routines. In the same way the aerodynamic and thermodynamic behaviour of engine components is summarised on look up maps, it should be possible to create

maps of the thermal behaviour of individual engine components (Fiola 2002). Indeed, if such maps could be created and incorporated into whole engine performance codes then the result would be extremely fast and accurate heat soakage and tip clearance representation within such codes. The speed would arise from the fact that information regarding thermal behaviour could simply be looked up on a map, and the accuracy would come from the fact that the data captured on the map is likely to be the most accurate that is available, being derived in the first instance from heavy weight numerical analyses, and then later possibly from test measurements. Such a system would be far quicker and easier to update than would full physics models (e.g. the streams method). In addition, the authority of Rolls-Royce departments responsible for guaranteeing the quality of information such as tip clearances would be maintained. Characteristic equations would serve as equally well as performance maps.

Once the relevant Rolls-Royce departments have created computational fluid dynamics and finite element models of a component (which in time would be validated against test data), then these should be the source of all data used by the performance department in its whole engine performance codes. If the performance department incorporated its own physics based methods into its engine codes then prior to the existence of finite element models it would be able to make its own estimations of the thermal behaviour of a new engine. However, once finite element models came into existence a conflict in authority might arise. It is clear to see that a physics based method (e.g. the streams method) would only be of use to performance engineers for a very short period at the start of a new engine development programme prior to the existence of engine component finite element models. Throughout its life typically an engine type will continue to be upgraded. When upgrades involve the change of a piece of hardware finite element and other detailed analyses usually precede the physical manufacture of the new piece of hardware. This means that throughout its life either test data or heavyweight numerical analyses will be available to provide input to component thermal behaviour maps.

In light of what has been said regarding the physics based nature of heat soakage and tip clearance methods the author recommends that models for each phenomena ought to be developed separately, rather than developing a single method that can handle both heat soakage and tip clearance.

The author recommends that heat soakage in the whole engine performance codes be represented by a method such as Rielger (1999) or perhaps BR081 in both primary and secondary gas paths with the inclusion of windage so as to account for bleed temperatures variations during transients. For tip clearances methods such as that of Lewis, Khalid & Hearne (1980), or Moll et al (2003) should be used. Incorporation of time constant methods or look up maps into whole engine performance codes should probably be a task given to UTC Research Officers and Research Assistants rather than to UTC postgraduates. The reason for this is that such a task is likely to be too big for an MSc student to do but at the same time require insufficient creative input to meet the requirements of either a PhD or EngD degree.

Alternatively, the UTC could pursue an NPSS type approach. Modern commercially available physical analysis computer codes such as ANSYS[®] are able to perform both computation fluid dynamics and finite element analysis all on the same model. Such

models could be iteratively called as batch processes from within the whole engine performance code. Perhaps a more useful approach in terms of technology transfer would be not to use ANSYS® but rather to call existing Rolls-Royce software package models (CFD, FEA, air system networks, etc.) from within a RRAP code. However, this is as much a task for software engineers as it is a task for gas turbine performance engineers and this should be allowed for in the staffing plans for any such project.

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APPENDIX A

STREAMS METHOD MODELS OF TRENT 500 TURBOMACHINERY

Appendix A contains sections of Trent 500 airflow diagrams and streams method representations for the IPC, HPT, IPT, and LPT. No model for the fan was constructed and the model for the HPC has been discussed at length in Chapter 4. Appendix A also contains charts of stage averaged transient tip closures (from cold static) during the acceleration, deceleration and hot reslam manoeuvres of chapter 4.

INTERMEDIATE-PRESSURE COMPRESSOR (IPC)

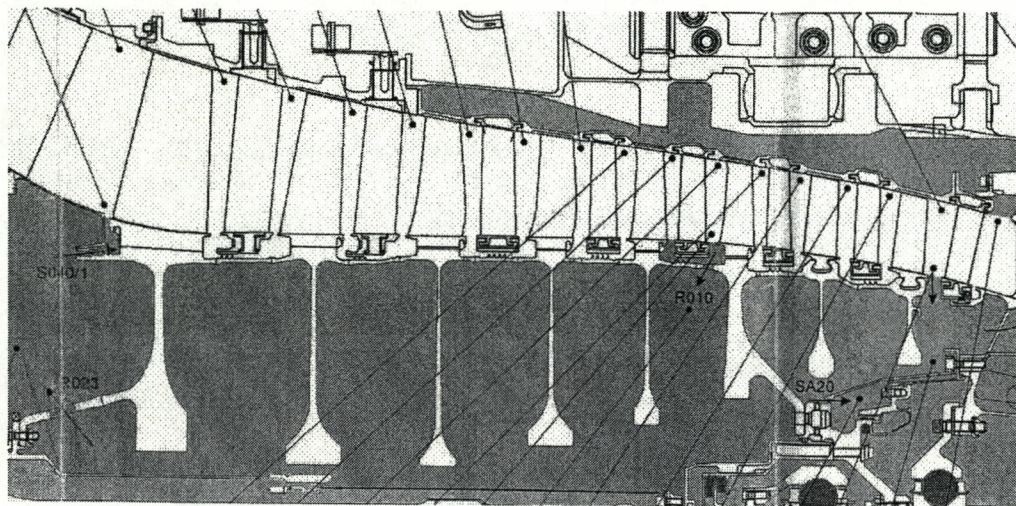


Figure A.1

Cut out from an airflow diagram showing the basic geometry and bleed paths for the Trent 500 IPC.

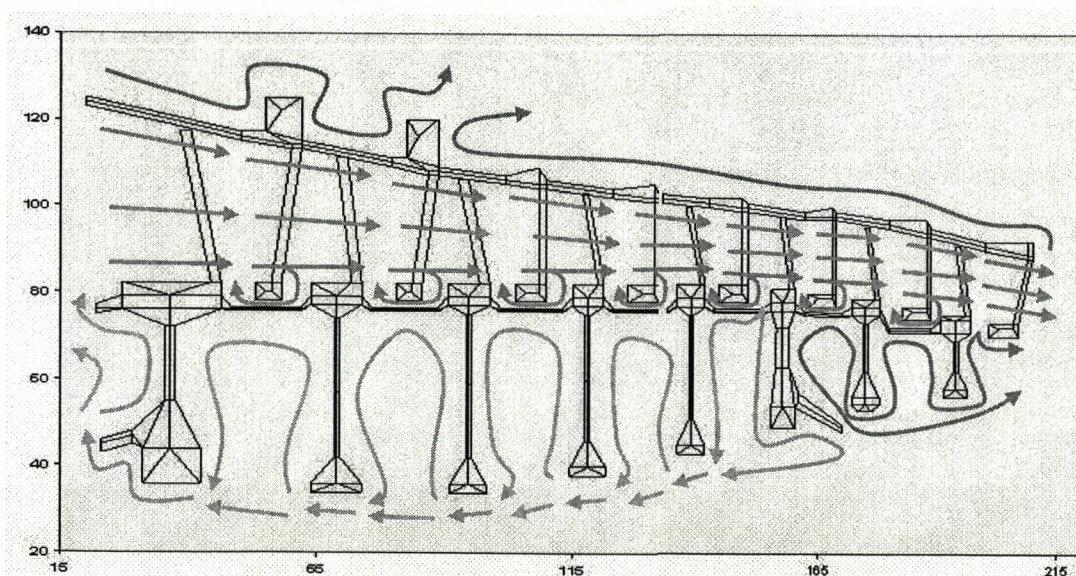


Figure A.2

Streams model representation of the Trent 500 IPC.

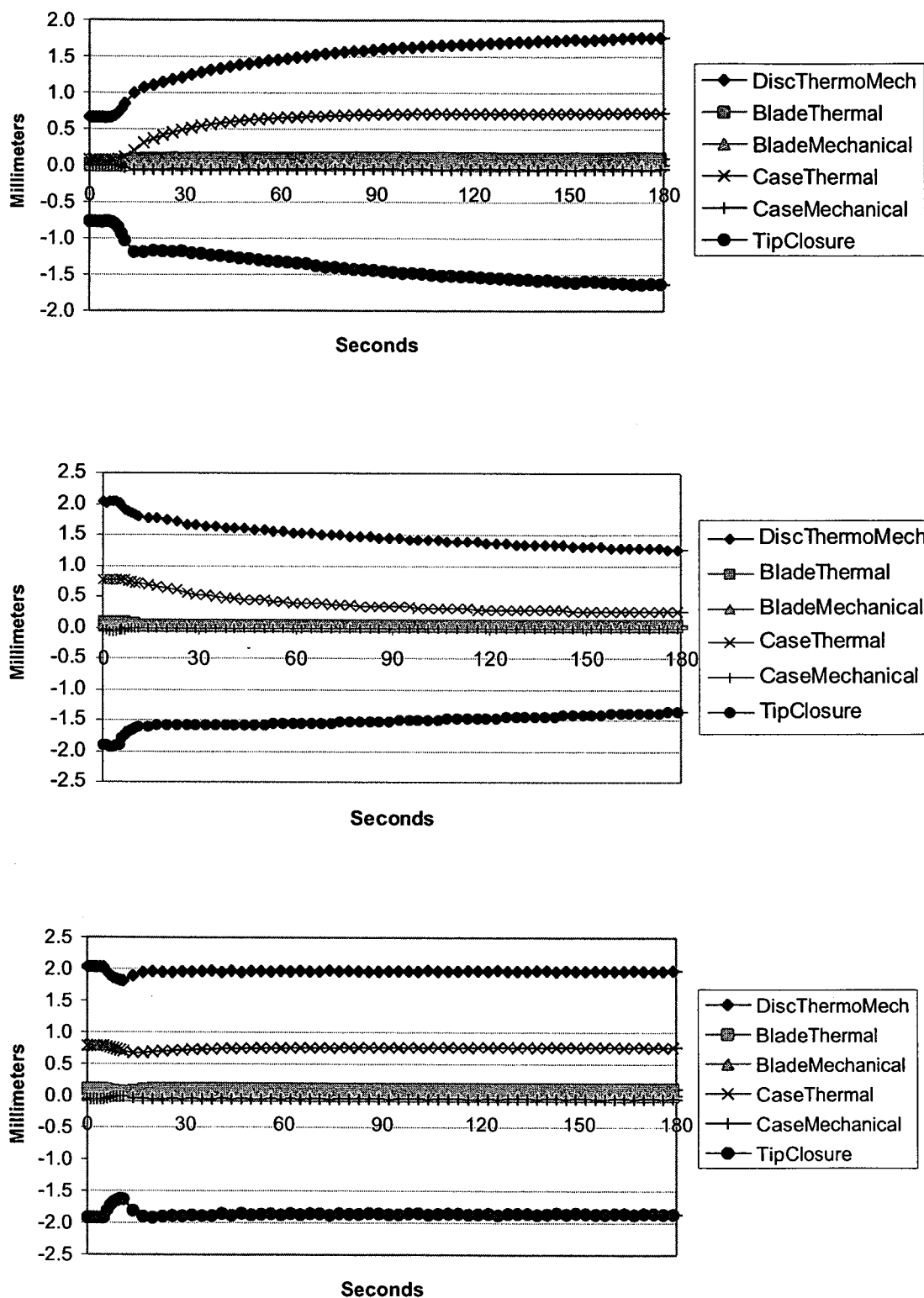


Figure A.3

Stage averaged tip closure and components of tip closure calculated using the streams model of the Trent 500 IPC (top: slam acceleration, middle: slam deceleration, bottom: hot reslam)

HIGH-PRESSURE TURBINE (HPT)

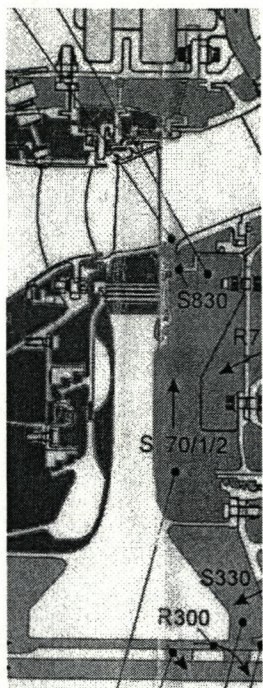


Figure A.4 Cut out from an airflow diagram showing the basic geometry and bleed paths for the Trent 500 HPT

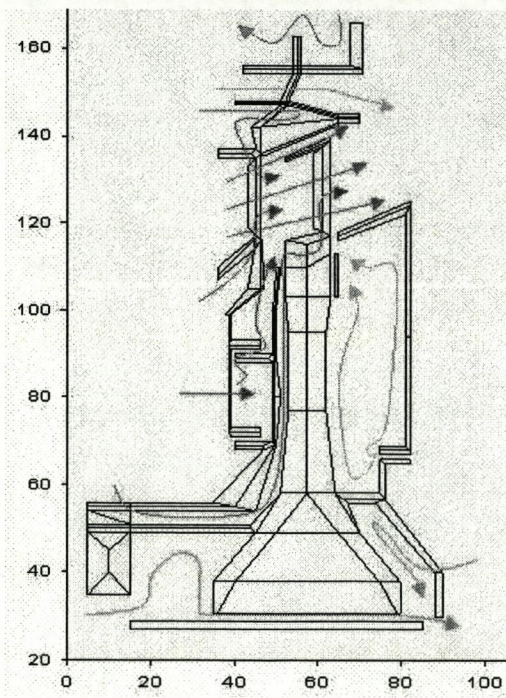


Figure A.5 Streams model representation of the Trent 500 HPT

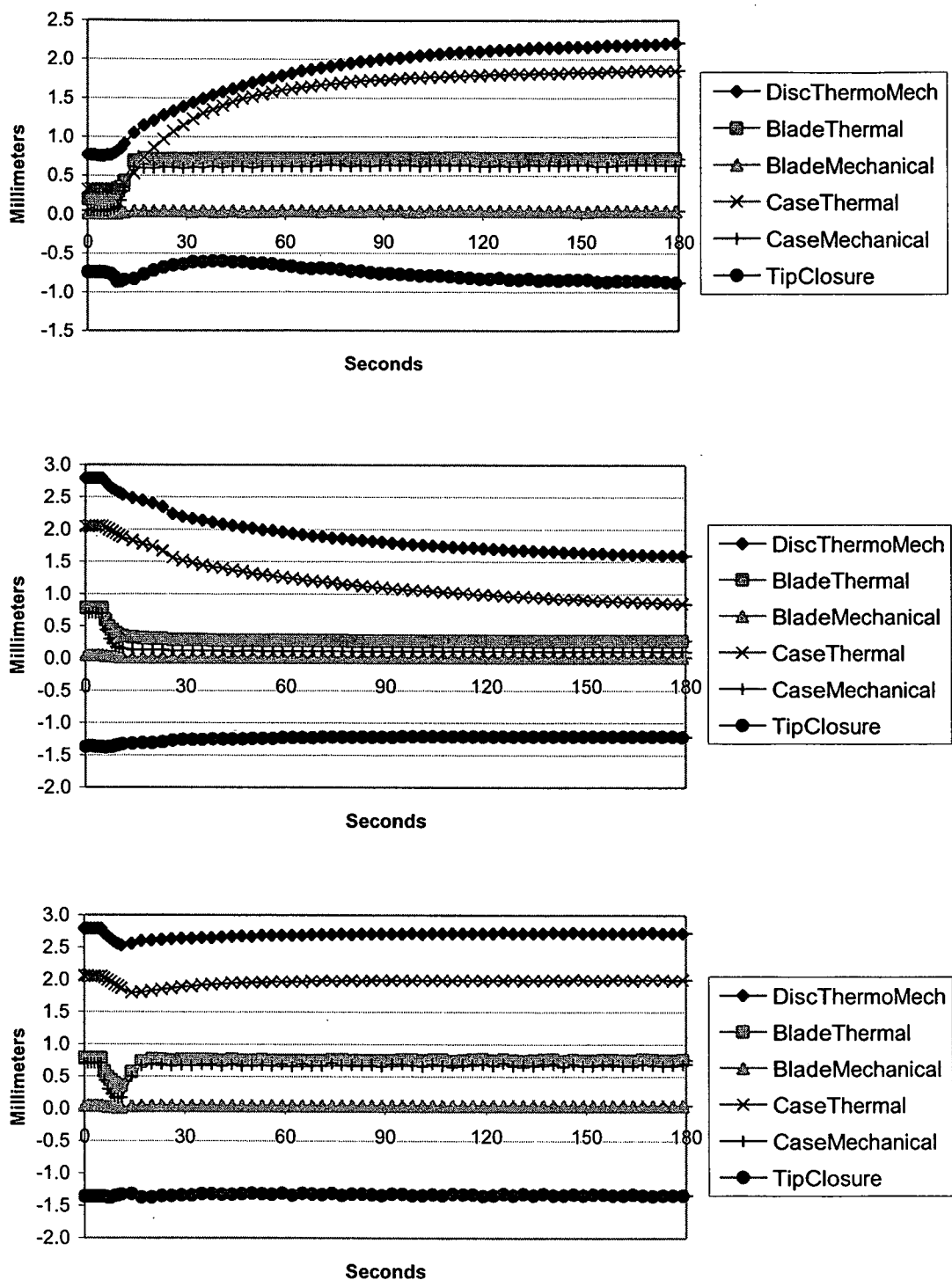


Figure A.6

Tip closure and components of tip closure calculated using the streams model of the Trent 500 HPT (top: slam acceleration, middle: slam deceleration, bottom: hot reslam)

INTERMEDIATE-PRESSURE TURBINE (IPT)

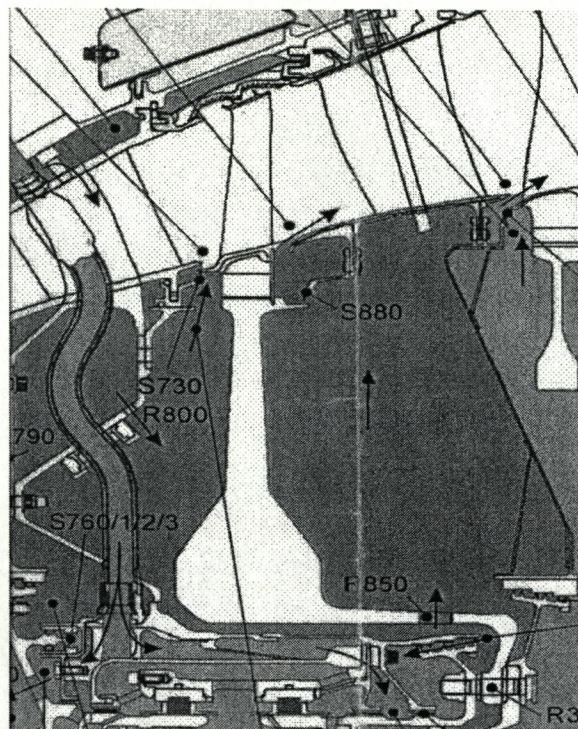


Figure A.7

Cut out from an airflow diagram showing the basic geometry and bleed paths for the Trent 500 IPT

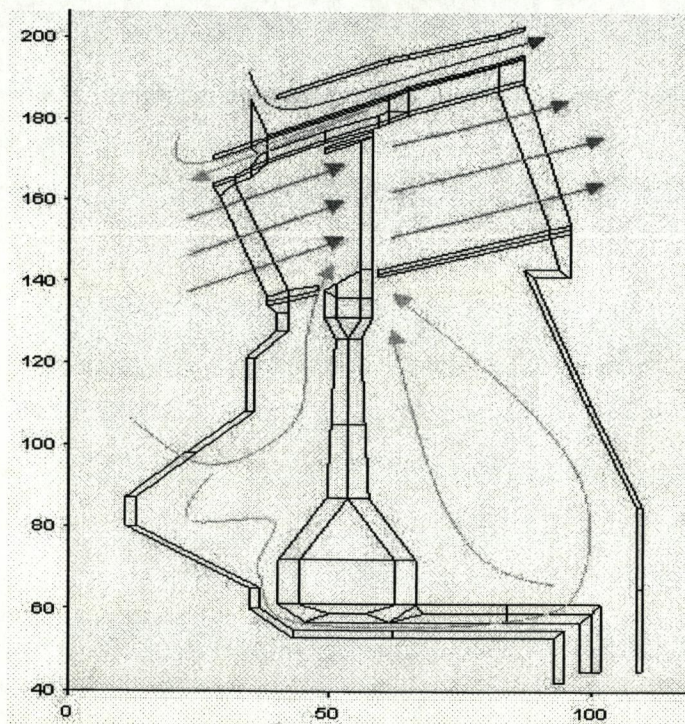


Figure A.8

Streams model representation of the Trent 500 IPT

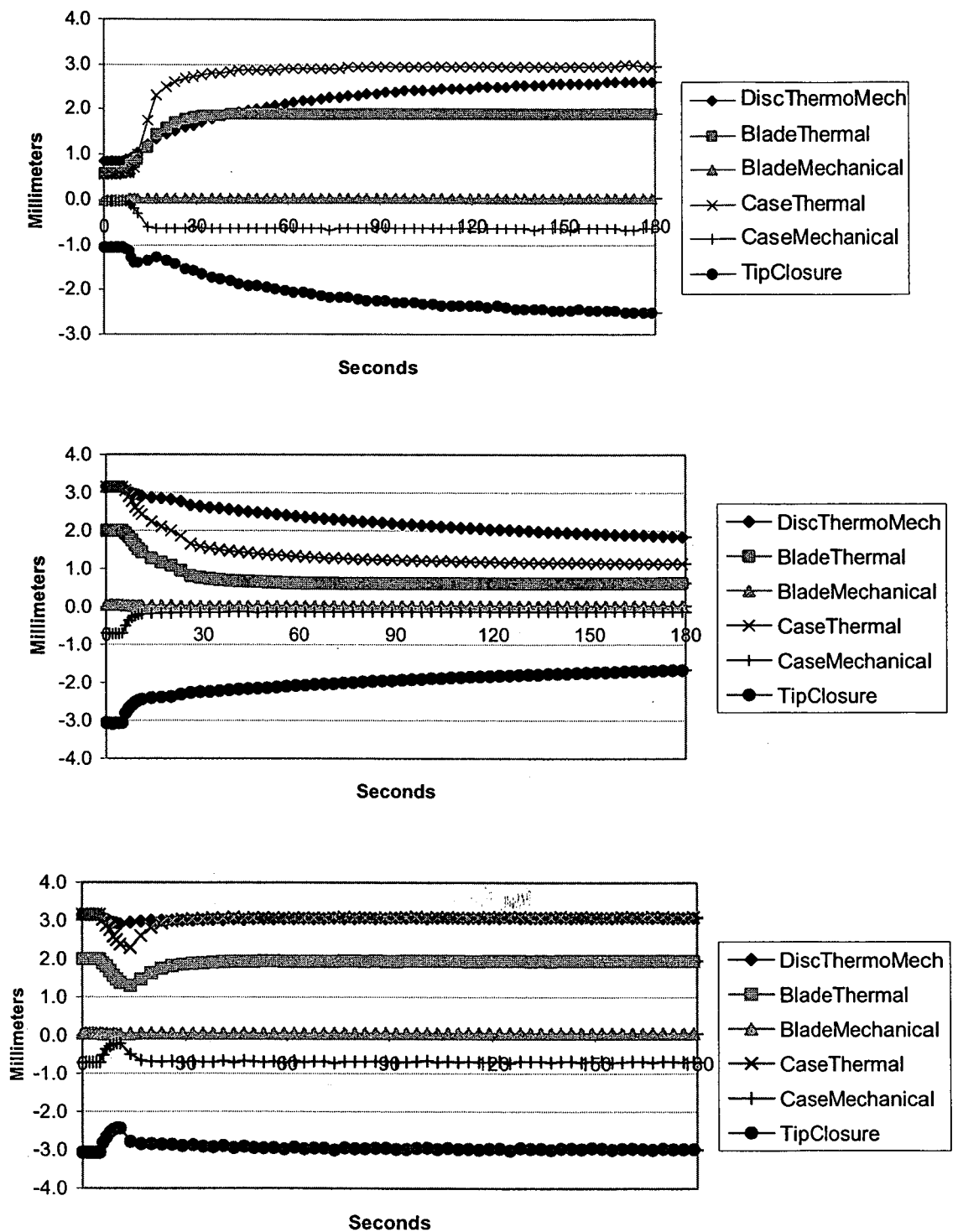


Figure A.9

Tip closure and components of tip closure calculated using the streams model of the Trent 500 IPT (top: slam acceleration, middle: slam deceleration, bottom: hot reslam)

LOW-PRESSURE TURBINE (LPT)

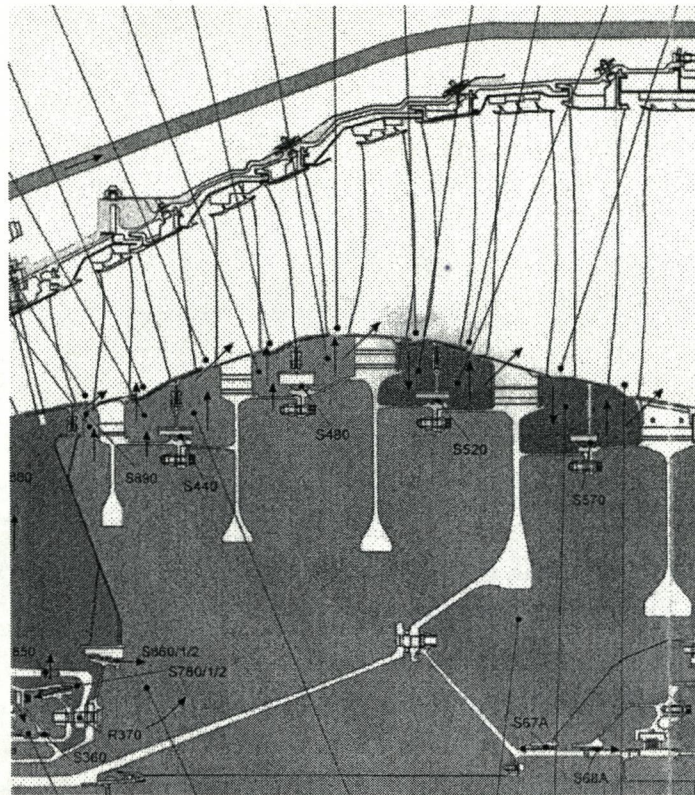


Figure A.10

Cut out from an airflow diagram showing the basic geometry and bleed paths for the Trent 500 LPT

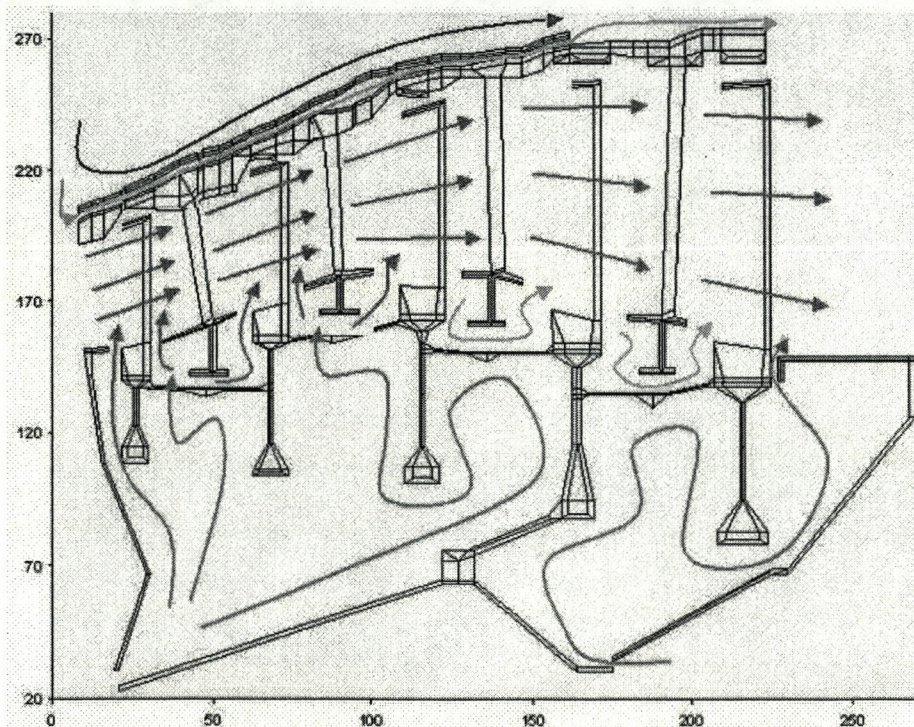


Figure A.11

Streams model representation of the Trent 500 LPT

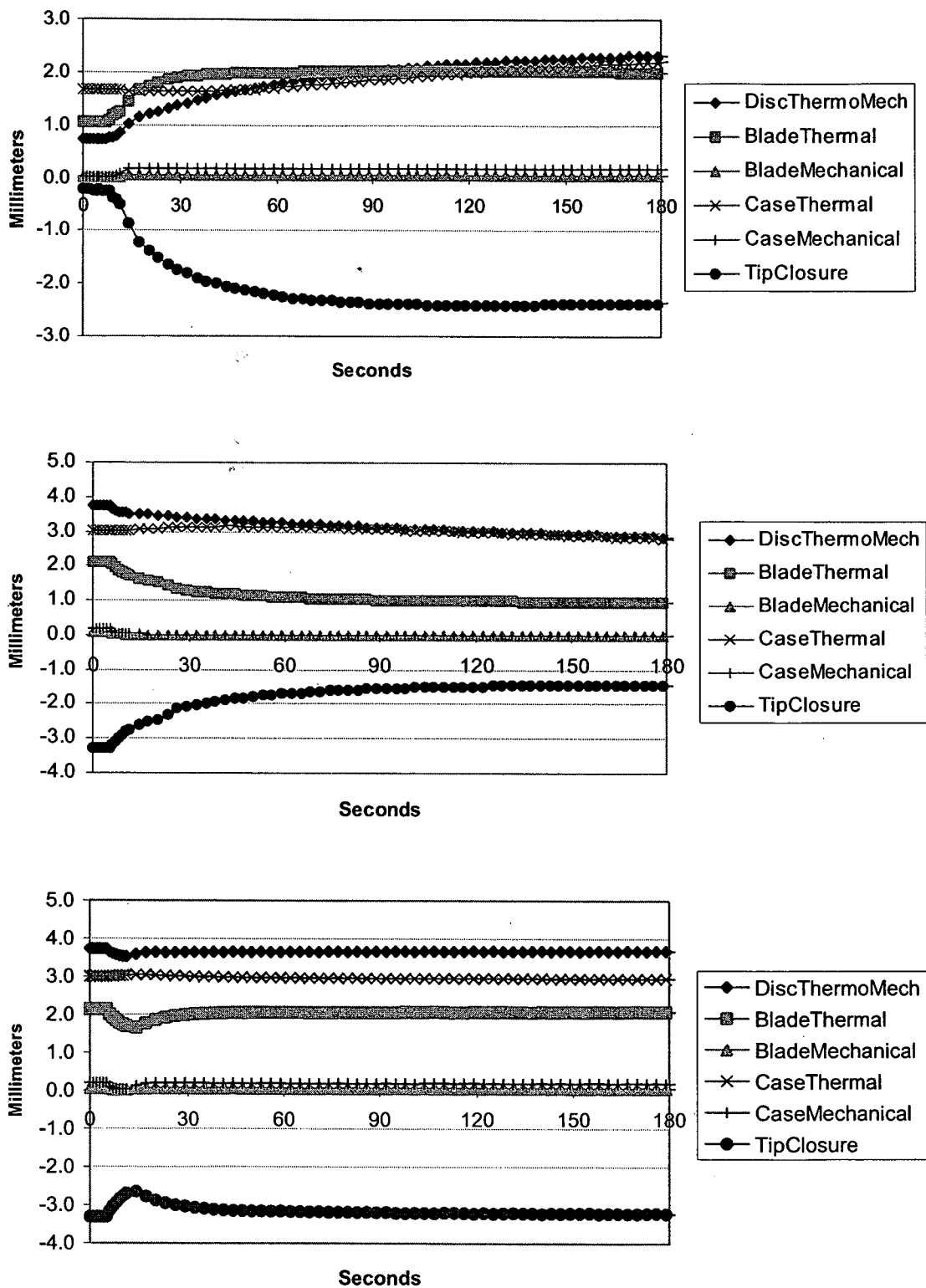


Figure A.12

Stage averaged tip closure and components of tip closure calculated using the streams model of the Trent 500 LPT (top: slam acceleration, middle: slam deceleration, bottom: hot reslam)

APPENDIX B

HOW CONTROL SYSTEMS AFFECT ENGINE TRANSIENT PERFORMANCE

Civil turbofan digital control systems are highly sophisticated pieces of equipment. They are essentially computers mounted onto the engine casing and packaged robustly enough to withstand such things as lightening strikes. The control algorithms they contain are extremely complex, especially those governing engine transient manoeuvres. The control system takes measurements of key engine parameters such as speeds, pressures, and temperatures, passes them through the algorithms, and then sends output signals to the fuel control valve and variable geometry actuators. Typically this sequence is repeated every 10–30 ms (Walsh and Fletcher 1998).

The purpose of the engine control system is to control thrust. This is generally done using an easy to measure proxy for thrust, such as spool speed or exhaust pressure ratio, because measuring thrust directly is very difficult (due to uncertain static pressure distributions surrounding the engine). In operation the controller has to cope with the effect of changing intake air pressure and temperature, due to changes in altitude and flight Mach number, in order to maintain air to fuel ratios within specific limits so as to give correct combustion. In addition, the controller has to ensure that the engine does not cross limits for compressor stability, spool speed, turbine blade temperature, and combustor flame out. Furthermore, the controller must ensure that the engine meets transient response time requirements laid down by airframe manufacturers and airworthiness authorities.

The pilot controls the engine using a power lever. Engine control systems generally employ a look up schedule of power lever angle versus engine speed demanded. The controller uses the difference between the speed demanded and the actual measured speed to either increase or decrease the fuel flow delivered to the engine. The controller uses a set of jump and rate limits to ensure that fuel flow changes and fuel flow rates of change do not result in the engine crossing its limits for compressor stability, spool speed, temperature, etc.

The two most common control philosophies are the referred fuel flow method, traditionally used in hydro-mechanical control systems, and the NDOT technique, which has become practical with the advent of digital controllers and which has now become adopted almost exclusively for aircraft engines (Walsh and Fletcher 1998).

The referred fuel flow method is based on the fact that jet engines behave non-dimensionally. Control strategies during transient operation are invariably based upon nondimensional or quasi-nondimensional parameter group relationships. Transiently, if nozzles and nozzle guide vanes are choked for a given engine geometry, and referred fuel flow is scheduled against referred speed (c.f. figure B1), then to a first order of accuracy all other referred parameter groups will follow a unique trajectory during transient manoeuvres. Because parameters such as referred torque imbalance and referred spool acceleration rate are sensitive to intake air pressure, then actual spool acceleration rates will be different at different altitudes and Mach numbers. Furthermore, using the referred fuel flow method deteriorated

engines or engines with customer off-takes will accelerate and decelerate at different rates (compared to new engines) due the fact that such engines require higher steady state fuel flows (c.f. figure B1). However, when using the referred fuel flow method customer off-takes and deterioration have little adverse impact on the high-pressure compressor transient working line, since transient referred fuel flow and hence working lines are unchanged (Walsh & Fletcher 1998).

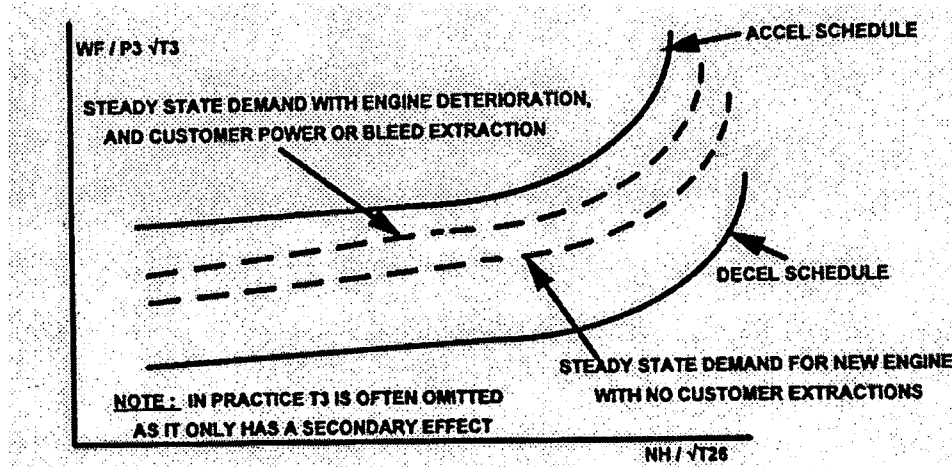


Figure B1 Referred fuel flow method acceleration and deceleration fuel schedules (Walsh and Fletcher 1998).

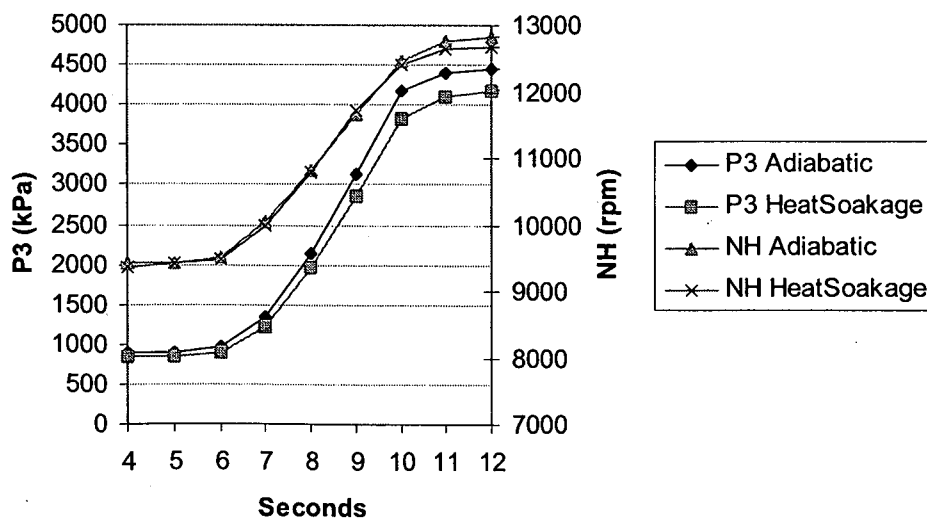


Figure B2 High spool speed, NH, and high-pressure compressor delivery pressure, P3, as calculated using the BD31 version of RRAP during the speed transient of a sea level acceleration manoeuvre run to a fixed schedule of fuel flow versus time with streams method models switched on (Heat Soakage), and switched off (Adiabatic).

Figure B2 shows that for a fixed schedule of fuel flow versus time, the physical response of a heat soaking engine lags behind that of an adiabatic engine. This has important implications for engines governed by a referred fuel flow method

controller. For a heat soaking engine at a given speed during acceleration the measured P3 will be lower and thus the fuel metered to the engine will be lower, and so the lag in response occurring due to heat soakage in the engine will be exaggerated by the control system.

As mentioned above, modern digital control systems for aero gas turbine engines generally use the NDOT method rather than the referred fuel flow method. Full authority digital engine control (FADEC) is widely used on large aero-engines but less so on smaller engines of lower cost. Although the cost of the cost of the actual computer is small, the need for certification of the software leads to very high software development costs (Cohen et al 1996). The NDOT method controls the rate of change of the high-pressure spool speed. The advantage of this technique is that it provides repeatability of engine acceleration and deceleration times, independent of engine deterioration, customer off-takes, etc. This is favoured particularly for twin engines, which under an NDOT controller will always respond at the same rates so that the pilot does not have to address yaw along the runway. However, the transient compressor working line excursions are greater under an NDOT controller with engine deterioration, etc, than they are with a referred fuel flow controller. The NDOT controller is based on a schedule of predicted acceleration fuel flow against high-pressure spool speed with a closed loop trim on the NHDot error to correct for external effects such as power off-take (Thompson 1992).

Another important transient effect that occurs due to the influence of the control system can be seen in figure B3. Heat soakage effects such as transient tip clearances temporarily affect component efficiencies until the engine reaches thermal stabilisation. If exhaust pressure ratio (EPR) is used as a proxy for thrust and a slam acceleration is performed, the control system is likely to overfuel in an attempt to compensate for lower component efficiencies in an attempt to meet the required EPR. This can result in a turbine entry temperature (TET) overshoot at the end of the speed transient that can persist for some minutes. This can have a very detrimental effect on engine life and time between overhauls, and thus the cost of ownership of the engine.

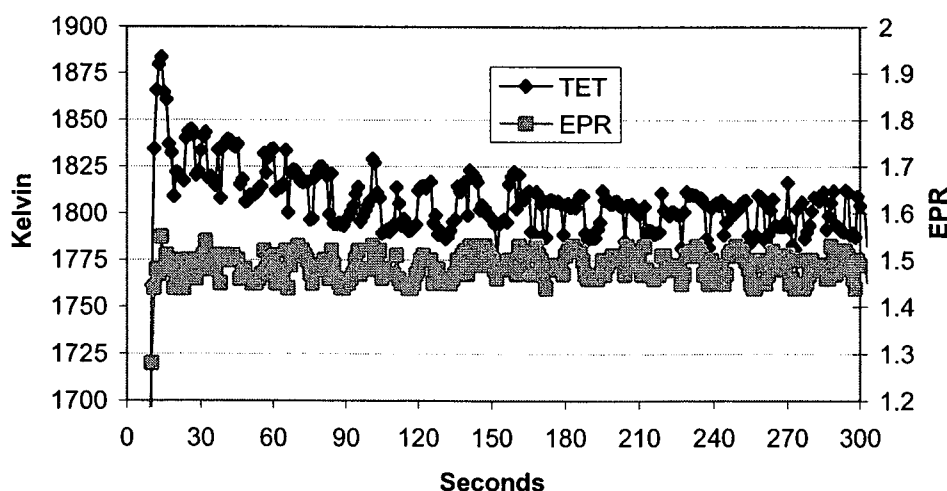


Figure B1 Turbine entry temperature (TET) overshoot following a slam acceleration manoeuvre as calculated by the BD31 version of RRAP, including streams method subroutines, run to a fixed exhaust pressure ratio (EPR).

APPENDIX C

FORTRAN SOURCE CODE FOR RINGS METHOD ALGORITHM

```

SUBROUTINE SimpleHyperbolicRingMatchNewton
+ (numRings,xRo,xRi,xTho,xThi,xTemp,xAlpha,xE,xPois,xrho
+ ,xOmega,xDovetailPull,dDiscRad,xSigRo,Loops,COMP,STAGE)

IMPLICIT NONE

C   PARAMETERS

INTEGER,INTENT(IN)::NumRings           !number of rings (10 max)
REAL(4),INTENT(IN)::xRo(1:10)         !ring outer radius Ro
REAL(4),INTENT(IN)::xRi(1:10)         !ring inner radius Ri
REAL(4),INTENT(IN)::xTho(1:10)        !ring thickness at Ro
REAL(4),INTENT(IN)::xThi(1:10)        !ring thickness at Ri
REAL(4),INTENT(IN)::xTemp(1:10)       !ring temperature
REAL(4),INTENT(IN)::xAlpha(1:10)      !ring coeff therm expansion
REAL(4),INTENT(IN)::xE(1:10)          !ring young's modulus
REAL(4),INTENT(IN)::xPois(1:10)       !ring poisson's ratio
REAL(4),INTENT(IN)::xRho(1:10)        !ring density
REAL(4),INTENT(IN)::xOmega             !angular velocity
REAL(4),INTENT(IN)::xDovetailPull      !force at Ro of ring1
REAL(4),INTENT(OUT)::dDiscRad           !delta Ro of ring1 (from cold static)
REAL(4),INTENT(INOUT)::xSigRo(1:10)    !radial stress at Ro
INTEGER,INTENT(OUT)::Loops              !number of iterations
INTEGER,INTENT(IN)::COMP                !component number
INTEGER,INTENT(IN)::STAGE               !stage number

C   LOCAL VERSIONS OF PARAMETERS (DOUBLE PRECISION)

REAL(8) Ro(1:10),Ri(1:10),Tho(1:10),Thi(1:10)
REAL(8) Temp(1:10),Alpha(1:10),E(1:10)
REAL(8) Pois(1:10),Rho(1:10),Omega,DovetailPull

C   LOCAL VARIABLES

REAL(8) dRo(1:10)      !ring delta Ro (from cold static)
REAL(8) dRi(1:10)      !ring delta Ri (from cold static)
REAL(8) SigRo(1:10)     !ring radial stress at Ro
REAL(8) SigRi(1:10)     !ring radial stress at Ri
REAL(8) SigHo(1:10)     !ring hoop stress at Ro
REAL(8) SigHi(1:10)     !ring hoop stress at Ri
REAL(8) WRo(1:10)       !wobble radial stress at Ro (for Newton method)
REAL(8) WRi(1:10)       !wobble radial stress at Ri (for Newton method)
REAL(8) dWo(1:10)       !wobble ring delta Ro (for Newton method)
REAL(8) dWi(1:10)       !wobble ring delta Ri (for Newton method)
REAL(8) dSigdR           !partial derivative
REAL(8) Gap(1:10)        !gaps between rings
REAL(8) Tcold            !cold static reference temperature
REAL(8) GapTolerance     !ring gap tolerance
INTEGER IterationLimit   !iteration limit
INTEGER BigGaps          !number of gaps out of tolerance
INTEGER j                !counter

C   CONSTANTS

REAL(8) a(1:10),b(1:10),F(1:10),ConstA,ConstB
REAL(8) m1(1:10),m2(1:10),Ko1(1:10),Ko2(1:10),Ko3(1:10)
REAL(8) Ki1(1:10),Ki2(1:10),Ki3(1:10)
REAL(8) X1(1:10),X2(1:10),X3(1:10),X4(1:10),X5(1:10)
REAL(8) X6(1:10),X7(1:10),X8(1:10),X9(1:10),X10(1:10)
REAL(8) X11(1:10),X12(1:10),X13(1:10),X14(1:10),X15(1:10)
REAL(8), PARAMETER :: PIE = 3.14159265358979

C   COPY INPUT PARAMETERS TO LOCAL VARIABLES

Tcold = 288.0D0

```

```

Ro                = DBLE(xRo)
Ri                = DBLE(xRi)
Tho               = DBLE(xTho)
Thi               = DBLE(xThi)
Temp              = DBLE(xTemp)
Alpha             = DBLE(xAlpha)
E                 = DBLE(xE)
Pois              = DBLE(xPois)
rho               = DBLE(xrho)
Omega             = DBLE(xOmega)
DovetailPull     = DBLE(xDovetailPull)
SigRo             = DBLE(xSigRo)

C    INITIAL GUESSES FOR RADIAL STRESSES

IF (xSigRo(1) == 0.0) THEN
    DO j=1, NumRings
        SigRo(j) = DovetailPull / (PIE * 2.0 * Ro(j) * Tho(j))
        SigRi(j) = SigRo(j)
    END DO
ELSE
    DO j=1, NumRings-1
        SigRi(j) = SigRo(j+1)
    END DO
ENDIF
SigRi(NumRings) = 0.0D0

C    CONSTANTS USED IN MAIN ITERATIVE LOOP

DO j=1, NumRings

a(j) = DLOG(Tho(j)/Thi(j))/DLOG(Ro(j)/Ri(j))
b(j) = (1.0-Pois(j)**2.0)
F(j) = (-b(j)*Rho(j)*Omega**2.0)/(E(j)*(8.0+(3.0+Pois(j))*a(j)))
m1(j) = -(0.5*a(j))-((0.25*a(j)**2.0)-a(j)*Pois(j)+1)**0.5
m2(j) = -(0.5*a(j))+((0.25*a(j)**2.0)-a(j)*Pois(j)+1)**0.5
Ko1(j) = (E(j)/b(j))*(3.0+Pois(j))*F(j)*Ro(j)**2.0
Ko2(j) = (E(j)/b(j))*(m1(j)+Pois(j))*Ro(j)**(m1(j)-1.0)
Ko3(j) = (E(j)/b(j))*(m2(j)+Pois(j))*Ro(j)**(m2(j)-1.0)
Ki1(j) = (E(j)/b(j))*(3.0+Pois(j))*F(j)*Ri(j)**2.0
Ki2(j) = (E(j)/b(j))*(m1(j)+Pois(j))*Ri(j)**(m1(j)-1.0)
Ki3(j) = (E(j)/b(j))*(m2(j)+Pois(j))*Ri(j)**(m2(j)-1.0)

C    ConstB constants
X1(j) = 1.0 / (Ko3(j)-Ki3(j)*Ko2(j)/Ki2(j))
X2(j) = (Ko2(j)/Ki2(j))*X1(j)
X3(j) = (Ki1(j)*(Ko2(j)/Ki2(j)) - Ko1(j))*X1(j)

C    SigHo constants
X4(j) = ((1.0+3.0*Pois(j))*F(j)*Ro(j)**2.0)*(E(j)/b(j))
X5(j) = ((1.0+m1(j)*Pois(j))*Ro(j)**(m1(j)-1.0))*(E(j)/b(j))
X6(j) = ((1.0+m2(j)*Pois(j))*Ro(j)**(m2(j)-1.0))*(E(j)/b(j))

C    SigHi constants
X7(j) = ((1.0+3.0*Pois(j))*F(j)*Ri(j)**2.0)*(E(j)/b(j))
X8(j) = ((1.0+m1(j)*Pois(j))*Ri(j)**(m1(j)-1.0))*(E(j)/b(j))
X9(j) = ((1.0+m2(j)*Pois(j))*Ri(j)**(m2(j)-1.0))*(E(j)/b(j))

C    dRo constants
X10(j) = Ro(j)*Alpha(j)*(Temp(j)-Tcold)
X11(j) = Ro(j)/E(j)
X12(j) = Pois(j)*Ro(j)/E(j)

C    dRi constants
X13(j) = Ri(j)*Alpha(j)*(Temp(j)-Tcold)
X14(j) = Ri(j)/E(j)
X15(j) = Pois(j)*Ri(j)/E(j)

END DO

GapTolerance = Ro(1) * 0.000001
IterationLimit = 100000

DO Loops = 1,IterationLimit

C    GET DRO AND DRI FOR EACH RING (Roark p706)

```

```

DO j=1, NumRings
  ConstB = SigRo(j)*X1(j) - SigRi(j)*X2(j) + X3(j)
  ConstA = (SigRi(j)-Ki1(j)-Ki3(j)*ConstB)/Ki2(j)
  SigHo(j) = X4(j) + ConstA*X5(j) + ConstB*X6(j)
  SigHi(j) = X7(j) + ConstA*X8(j) + ConstB*X9(j)
  dRo(j) = X10(j) + X11(j)*SigHo(j) - X12(j)*SigRo(j)
  dRi(j) = X13(j) + X14(j)*SigHi(j) - X15(j)*SigRi(j)
END DO

C      CHECK IF GAPS ARE WITHIN TOLERANCE

BigGaps = 0
DO j=1,NumRings-1
  IF (ABS(dRi(j)-dRo(j+1))>=GapTolerance) BigGaps=BigGaps+1
END DO

IF (BigGaps == 0) EXIT

C      REGRESS RADIAL STRESSES USING NEWTON TYPE METHOD

DO j=2,NumRings
  WRo(j) = SigRo(j) + ((dRi(j-1)-dRo(j))/Ro(j))*E(j)/100
  WRi(j-1) = WRo(j)
END DO

DO j=1, NumRings
  ConstB = WRo(j)*X1(j) - WRi(j)*X2(j) + X3(j)
  ConstA = (WRi(j)-Ki1(j)-Ki3(j)*ConstB)/Ki2(j)
  SigHo(j) = X4(j) + ConstA*X5(j) + ConstB*X6(j)
  SigHi(j) = X7(j) + ConstA*X8(j) + ConstB*X9(j)
  dWo(j) = X10(j) + X11(j)*SigHo(j) - X12(j)*WRo(j)
  dWi(j) = X13(j) + X14(j)*SigHi(j) - X15(j)*WRi(j)
END DO

DO j=2,NumRings
  dSigdR = (WRo(j) - SigRo(j)) / (dWo(j) - dRo(j))
  SigRo(j) = SigRo(j) + dSigdR * (dRi(j-1) - dRo(j))/25
  SigRi(j-1) = SigRo(j)
END DO

END DO

C      CONVERT LOCALS INTO OUTPUT PARAMETERS

dDiscRad      = SNGL(dRo(1))
xSigRo        = SNGL(SigRo)

C      ZERO STRESS GUESSES TO PREVENT CONTAMINATION NEXT CALL

SigRi  = 0.0D0
SigRo  = 0.0D0
WRi    = 0.0D0
WRo    = 0.0D0

RETURN
END SUBROUTINE

```


APPENDIX D

ROLLS-ROYCE EQUATIONS FOR HEAT TRANSFER COEFFICIENT CORRELATIONS AND WINDAGE HEATING (SI units)

Heat transfer coefficient correlation for forced convection of flow through a duct:

$$D = 4.A/P$$
$$Re = W.D/(A.Visc)$$
$$Pr = Cp.Visc/K$$

For $Re \leq 2100$

$$h = 1.860.Re^{1/3}.Pr^{1/3}.(D/L)^{1/3}.(Visc/Viscw)^{0.14}.K/D$$

For $Re > 2100$

$$h = 0.036.Re^{0.8}.Pr^{1/3}.(D/L)^{1/18}.(Visc/Viscw)^{0.14}.K/D$$

D	Hydraulic diameter of the duct
A	Flow cross sectional area of the duct
P	Wetted perimeter of the duct
Re	Reynolds number
W	Mass flow rate of the fluid
Visc	Bulk viscosity of the fluid
Pr	Prandtl number
Cp	Specific heat at constant pressure of the fluid
K	Thermal diffusivity of the fluid
L	Starting length of the duct
Viscw	Viscosity of the fluid at the wall temperature
H	heat transfer coefficient

Heat transfer coefficient correlation for forced convection from a rotating disc:

$$Re = (Rho.Omega.R^2)/Visc$$
$$Pr = Cp.Visc/K$$

For $Re \leq 240000$

$$h = 0.616.Pr^{1/3}.Re^{1/2}.K/R$$

For $Re > 240000$

$$h = 0.0267.Pr^{0.6}.Re^{0.8}.K/R$$

Re	Reynolds number
Rho	Density of the fluid
Omega	Angular velocity of the disc
R	Radius of interest on the disc
Visc	Viscosity of the fluid
Pr	Prandtl number
Cp	Specific heat at constant pressure of the fluid
K	Thermal diffusivity of the fluid
h	Heat transfer coefficient

Heat transfer coefficient correlation for natural convection from a vertical surface:

$$X = \text{abs}(T_f - T_s) / (\text{Visc}^2 \cdot T_f)$$

$$\text{Gr} = L^3 \cdot \text{Rho}^2 \cdot g \cdot X$$

$$\text{Pr} = \text{Cp} \cdot \text{Visc} / K$$

$$\text{Ra} = \text{Gr} \cdot \text{Pr}$$

$$\text{For } 0 < \text{Ra} < 10^4$$

$$b = \log(\text{Ra})$$

$$m = 0.005106 \cdot b^2 + 0.1298 \cdot b + 0.17$$

$$h = 10^m \cdot K/L$$

$$\text{For } 10^4 \leq \text{Ra} \leq 10^9$$

$$h = 0.59 \cdot \text{Ra}^{0.25} \cdot K/L$$

$$\text{For } \text{Ra} > 10^9$$

$$h = 0.129 \cdot \text{Ra}^{1/3} \cdot K/L$$

Gr	Grashof number
L	Wall height
Rho	Density of the fluid
g	acceleration due to gravity
abs()	Function: absolute value
Visc	Viscosity of the fluid
Tf	Temperature of the fluid
Ts	Temperature of the solid
Pr	Prandtl number
Cp	Specific heat at constant pressure of the fluid
K	Thermal diffusivity of the fluid
Ra	Rayleigh number
log()	Function: logarithm to base 10
b	Intermediate variable
m	Intermediate variable
X	Intermediate variable
h	heat transfer coefficient

Windage heating generated by rotating discs and cones:

$$\text{Theta} = \max(H, 22) \cdot \pi/180$$

$$A1 = \text{Rho} \cdot 10^{-12}$$

$$A2 = \text{Omega}^3$$

$$A3 = (\text{Rad} \cdot 10^3)^4$$

$$A4 = (\sin(\text{Theta}))^{0.2}$$

$$A5 = 2\pi \text{Rad} \cdot 10^3$$

$$b = (A1 \cdot A2 \cdot A3 \cdot A4) / A5$$

$$A6 = \text{Rho} \cdot 10^{-12}$$

$$A7 = \text{Omega}$$

$$A8 = (\text{Rad} \cdot 10^3)^2$$

$$A9 = \text{Visc} \cdot 10^{-6}$$

$$\text{Re} = (A6 \cdot A7 \cdot A8) / A9$$

$$X1 = 1.2275 \cdot \log(\text{Re})^{-2.58}$$

$$X2 = 0.5502 \cdot \log(\text{Re})^{-3.58}$$

$$X3 = (X1 - X2) \cdot b \cdot \text{Area} \cdot 10^6$$

$$Q = -dt \cdot X3 \cdot 10^3$$

A1...A9	Intermediate variables
X1...X3	Intermediate variables
b	Intermediate variable
H	Hade angle of surface [degrees]
Theta	Hade angle of surface [radians]
Rho	Density of gas
Omega	Angular velocity
Rad	Radius of interest
Visc	Viscosity of gas
Area	Wetted area of surface
dt	Time interval
Q	Quantity of heat generated during time interval (heat flow from gas to metal is positive)