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Rotating stall in variable geometry compressors.

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September 2016

Submitted in part fulfilment of the requirements for the degree of
Doctor of Philosophy in Mechanical Engineering of Imperial College
London
and the Diploma of Imperial College London

Abstract

The design and operation of gas-turbine engines is heavily influenced by the off-design stability of the compressor, which limits obtainable performance and may result in aeroelastic vibration issues. Whilst Variable Stator Vanes (VSVs) are widely used to mitigate this problem, research considering the mismatching effect of VSVs away from their optimal settings is limited. In this thesis, a high-speed variable geometry compressor is studied at part-speed conditions and VSV setting adjustments are made to deliberately trigger stable rotating stall and study its behaviour.

Examination of unsteady measurements reveals two “families” of rotating stall, each at different frequencies, where the dominant behaviour depends upon the VSV settings. Stall in the front stage is shown to consist of a spatially non-uniform and time-varying pattern of short lengthscale cells, which couple with rotor vibration and propagate as noise. Second stage stall is a longer lengthscale uniform disturbance consisting of fewer stall cells. The stalling pressure amplitudes are also found to correlate well to blade loading parameters from a one-dimensional meanline model.

Steady and unsteady CFD simulations at these stalled conditions confirm that the behaviour is due to regions of stall in the front stage tip region together with the hub of the second stage. These CFD calculations naturally result in the formation of stall cells and give a credible match to the experiment. Inviscid reasoning explains how this flowfield is due to spanwise static pressure gradients arising from part-speed closure of the VSVs.

Finally, the non-dimensional cell propagation speed (V_{stall}/U) for each family of stall is shown to be uniquely determined by the VSV settings. This appears to be linked to the axial flow velocity local to the cell and suggests that cell speed may be restated in a more universal non-dimensional form. Furthermore, simulations show the importance of flowfield coupling mechanisms in determining the number of stall cells, which are also driven largely by the VSV settings.

Declaration of Originality

The work presented in this thesis was conducted at the Mechanical Engineering Department, Imperial College and Rolls-Royce Derby between May 2012 and May 2016. This thesis is the original work of the author alone and all contributing sources have been cited within the text, acknowledgements and bibliography. No part of the work presented here has been submitted to any other University or Institution for any other qualification.

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This thesis contains approximately 50,000 words and 100 figures.

John Dodds

May 2016

Publications

Dodds, J. and Vahdati, M., 2015, “Rotating Stall Observations in a High Speed Compressor - Part I: Experimental Study” *ASME J. Turbomach.*, 137(5) p.051002

Dodds, J. and Vahdati, M., 2015, “Rotating Stall Observations in a High Speed Compressor - Part II: Numerical Study” *ASME J. Turbomach.*, 137(5) p.051003

Acknowledgements

The author has benefitted considerably from the continuous support of his supervisors Dr. Mehdi Vahdati and Professor Nick Cumpsty throughout this project. Both have provided an ongoing source of help and constructive advice and their patience is greatly appreciated. The CFD simulations in this thesis were enhanced significantly by the experience of Dr. Vahdati on the use of AU3D code. The many insightful discussions with Professor Cumpsty during the later stages of this PhD were also invaluable.

The author wishes to thank his employer Rolls-Royce PLC for their considerable support and for allowing the author the time and freedom to pursue this research, and also permitting the author the use of their test case, data and methods. Within the compressors department the author wishes to thank M. Adams, N. Baker, J. Bolger and C. Peng for their support. Within the vibration department the assistance of C. Morgan, J. Gallego-Garrido and S. Andrews was greatly appreciated during experimental testing and analysis. The author also wishes to extend special thanks to T. Rae and J. Green for enabling this project and their encouragement throughout.

Within Imperial College, the author would like to thank his colleagues S. Stapelfeldt, F. Zhao and P. Sureshkumar for their help both technically and for providing a stimulating work environment. The support of Z. Zachariadis is also appreciated in optimising the CPU cluster and promptly resolving many computational issues.

The author kindly acknowledges the financial support provided by the UK TSB “TUFT” programme. Several useful discussions with Ivor Day and Chris Freeman at the Whittle Laboratory were also of great benefit during the early phases of this project.

Finally, some things are more important than aerodynamics. My greatest thanks must go to my family. My parents Barbara and Eddie, my wife Kirsty and children Emily and Samuel have provided unwavering support, curiosity and kept a much needed sense of humour throughout this journey.

Nomenclature

Roman Letters

Symbol	Units	Description
A	m^2	Area
C	m	Chord Length
C_p	Jk^{-1}	Specific Heat Capacity
D	Pa	Dynamic Pressure
f	Hz	Frequency
H	J	Stagnation enthalpy
m	—	Spatial mode order
$\dot{m}$	kg/s	Massflow Rate
M	—	Mach Number
n_{stat}	—	Fraction of Engine Order (Static Frame)
n_{stat}	—	Fraction of Engine Order (Rotating Frame)
p'	Pa	Static Pressure (fluctuation)
p	Pa	Static Pressure (mean)
P_0	Pa	Total Pressure
r	m	Radius
s'	Pa	Unsteady Stress Component
s	$Jk^{-1}kg^{-1}$	Entropy
S	m	Tangential Spacing
T_0	k	Total Temperature
V	m/s	Velocity
U	m/s	Blade Speed
x	m	Axial Coordinate

Greek Letters

Symbol	Units	Description
γ	—	Ratio of specific heats
α	<i>Radians</i>	Swirl angle
θ	<i>Radians</i>	Angular Coordinate
ρ	kg/m^3	Density
ϵ	$^\circ$	Stagger angle
σ	—	Solidity
Ω, ω	<i>rad/s</i>	Rotational Speed
λ	—	Blockage Factor
η	—	Efficiency

Abbreviations

Abbreviation	Description
<i>BPF</i>	Blade Passing Frequency
<i>CFD</i>	Computational Fluid Dynamics
<i>DF</i>	Diffusion Factor
<i>DCA</i>	Double Circular Arc
<i>EO</i>	Engine Order
<i>FFT</i>	Fast Fourier Transform
<i>HPC</i>	High Pressure Compressor
<i>IGV</i>	Inlet Guide Vane
<i>IPC</i>	Intermediate Pressure Compressor
<i>NACA</i>	National Advisory Committee for Aeronautics
<i>RANS</i>	Reynolds Averaged Navier Stokes
<i>URANS</i>	Unsteady Reynolds Averaged Navier Stokes
<i>VSV</i>	Variable Stator Vane

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1 Introduction

The design and operation of modern gas turbine engines is heavily influenced by the problem of compressor stability. The onset of rotating stall and surge places a significant constraint on both the aerodynamic and mechanical designer and has considerable implications on the final cost, weight and performance of the engine. Whilst compressor stall and surge has been the subject of research for many years, there are a significant number of issues that remain poorly understood. Despite widespread use of variable stator vanes (VSVs) in gas turbine applications, their effect on rotating stall, particularly once moved away from their intended settings, has received little attention. This was recognised by Day[1] in a very recent literature review:

“In multistage compressors, stage matching is a problem which becomes more severe as pressure ratios increase. More work is needed on the scheduling of variable vanes and bleed valves - particularly when operating at part-speed. Are current scheduling practices optimal or are improvements possible?”

(Day 2016[1]).

In this thesis, attention focuses on the topic which Day[1] addressed in his Gas Turbine Lecture. The thesis presents a detailed investigation on the effect VSV settings have on part-span rotating stall in the front stages of an aeroengine style compressor. This chapter seeks to introduce the concept of compressor stall and outlines the key challenges posed by long term trends within the industry, both to the aerodynamic and mechanical designer. Having discussed these challenges, the problem statement for this research is then presented together with an overview of how this thesis is structured.

1.1 Aerodynamic challenges

The axial flow compressor has been in use for many decades in gas turbine engines, both for industrial and aviation applications. From a long term point of view, there is a constant desire to obtain the maximum possible pressure ratio and efficiency from the compressor, whilst minimising the cost and weight. To progress further there are a number of challenges facing the designer - most notably the problem of aerodynamic stability.

The most widely adopted way to visualise the design and operation of high-speed compressors is the “operating map”, such as that shown in Figure 1.1. Quasi non-dimensional characteristics of constant aerodynamic speed ($N/\sqrt{T_0}$) are shown on a map of inlet flow function ($\dot{m}\sqrt{T_0}/P_0$) against pressure ratio. The operating map would typically be obtained by rig testing, where the compressor shaft is driven directly by a motor to accurately control the speed, together with a variable area exit nozzle, which would be choked under normal operation. Gradual closure of the nozzle area would typically be performed at each speed to obtain characteristics from the working line (the locus of steady state operation when the compressor is installed in the gas turbine engine) up to the surge line¹. Through the use of dimensionless parameters for flow, pressure ratio and speed, the operating map of the compressor becomes (to a first approximation) independent of inlet conditions and will be equally valid both at sea-level and altitude.

Considering the behaviour at the design point, the blade aerofoil sections will be operating at incidence angles² close to the minimum loss condition and the levels of diffusion demanded by the blade passages will be quite tolerable. However, as the exit throttle is closed and the operating point moves up the characteristic, both the incidence and level of diffusion will rise to the extent that the blade and endwall surface boundary layers will experience adverse pressure gradients of increasing severity. Eventually, at the weakest point within the compressor (axially, circumferentially and radially) the boundary layer will separate leading to the formation of several localised passages of stalled flow.

¹The surge line connects the stall points at individual speeds, though technically surge or stall may occur at this boundary.

²The incidence angle refers to the difference between the inlet gas and blade angle.

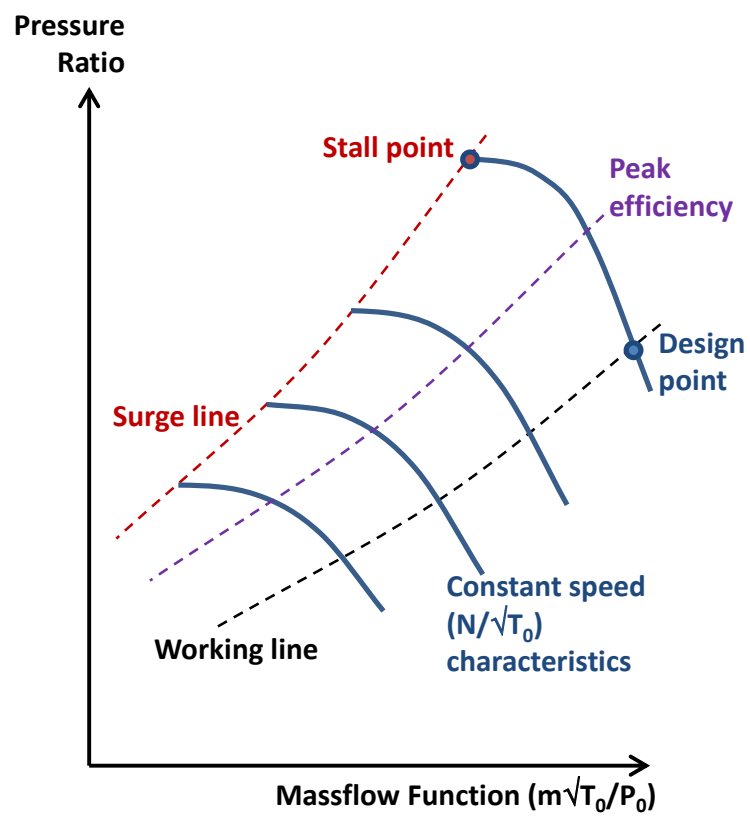


Figure 1.1: A typical operating map for a high-speed compressor.

The process which follows is termed “rotating stall” and has been known for several decades, with the “classic” explanation commonly attributed to Emmons et al.[2]. Figure 1.2 shows how rotating stall occurs based upon the Emmons model, which is essentially a two-dimensional simplification where blades stall uniformly across the span. Following the initial stall of one or several blades around the circumference, a region of blockage forms which naturally redistributes flow circumferentially, causing the incidence on one side of the cell (point “A”) to naturally rise, thus causing those blade passages to also stall. Simultaneously, the incidence on the other side (point “B”) naturally falls. This blockage based mechanism leads to a “stall cell” comprising of several stalled passages. This disturbance rotates around the annulus in the same sense as shaft rotation but at a fraction, typically 40-80%, of the shaft speed.

Rotating stall may consist of one large cell extending from the blade hub to tip (“full-span stall”), or may consist of multiple cells distributed around the circumference which occupy a region local to one of the blade endwalls (“part-span stall”). Rotating stall of either type may occur either as a persistent stable phenomena, or may exist only transiently as a pre-cursor to surge - where the compressed air stored within the exit volume discharges in a one-dimensional manner back through the compressor. Significant research undertaken in this field has enabled links to be established between the properties of the compression system and the phenomenon which is actually encountered (part-span stall/full-span stall/surge), which will be presented in chapter 2.

In the context of high-speed multistage compressors, it is typical for the compressor to become “mismatched” at part-speed conditions. This can be most easily understood by considering first the behaviour of the rear-most stage, which is forced to deliver a near constant exit flow function ($\dot{m}\sqrt{T_0}/P_0$) to the turbine, noting that the turbine nozzle guide vanes can effectively be modelled as a choked nozzle across much of the speed range[3]. This effectively fixes the operating point of the rear stage, such that at part-speed the “excursion” is tolerable in terms of incidence, diffusion and Mach number relative to the design point. However, moving further upstream and away from the choked nozzle, this part-speed excursion increases progressively to the extent that, without mitigation, the front

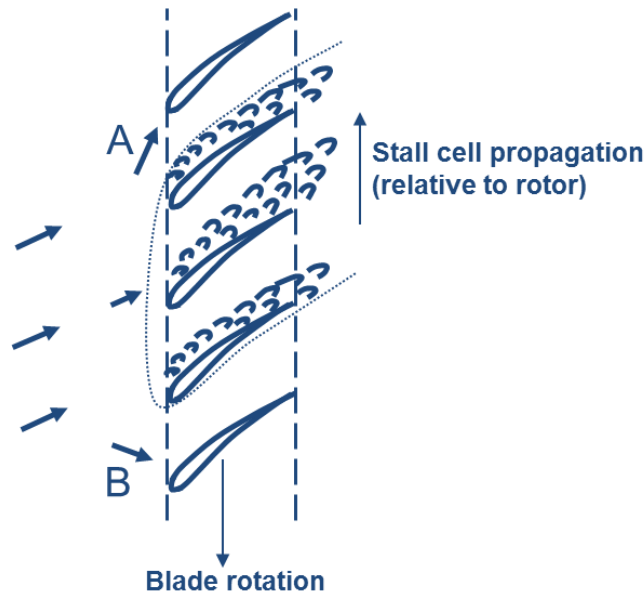


Figure 1.2: The Emmons model for rotating stall [2].

stages will tend to operate at very high incidence and diffusion levels relative to their design points. For this reason the front stages of any compressor will tend to become stalled at part-speed conditions, which subsequently drives the requirement for variable stator vanes (VSVs) - a design feature common to many gas turbine applications. By increasing the stagger of the variable stator vanes, the operating point excursion experienced by the front stages can be kept within reasonable limits and the stability of the compressor increased considerably relative to a fixed geometry machine.

However, whilst the required overall part-speed stability/surge margin requirements of the compressor can be achieved through the use of VSVs, significant challenges remain for the designer to resolve, not least the problem of how to optimise or “schedule” the VSVs across their speed range. Moreover it must also be ensured that the compressor behaviour is acceptable should the vanes become “mal-scheduled” away from their optimum settings due to manufacturing/assembly issues, deterioration mechanisms or errors within the VSV control system. In such situations it is quite feasible for the blades within the variable stages to approach their local stall

boundaries whilst the compressor as a whole still operates on its working line in a stable manner. This can lead to the occurrence of rotating stall confined to the front stages, with an unsteady flowfield which can potentially couple with blade vibration modes and cause resonant vibration issues. Similarly, the unsteadiness associated with rotating stall may also act as an acoustic source with the potential to cause elevated farfield noise levels.

In this thesis, a high-speed variable geometry compressor is tested with the VSVs deliberately adjusted away from their optimum settings by varying amounts. This causes rotating stall to form in the front stages in a stable manner, allowing the detailed behaviour to be investigated using both unsteady measurements (pressure transducers and strain gauges) and supporting numerical simulations.

A final point concerns the implications of compressor stall on the design and operation of the gas turbine engine. To achieve the overall operability requirements of the engine, the designer must always ensure sufficient margin is achieved between the working line and surge line to allow for rapid transients or deterioration effects without causing surge. However, this commonly prevents the compressor from being operated at its optimum condition for efficiency. As can be seen in Figure 1.1, it would be desirable from a performance point of view to design the engine such that the compressor operates on a higher working line, closer to the peak efficiency locus and delivering a higher pressure ratio. However, the closer proximity to the surge line would result in a lower stability margin, a scenario that would not be acceptable from an engine operability point of view. At this level it can be seen how the requirements for adequate stability prevent the designer from delivering the optimum fuel efficiency. This compromise can be found at many levels through the design process, with the stability requirements also having a very significant impact on the overall dimensions of the compressor, such as the number of stages, the number of blades per stage and also the requirements for any variable stages or interstage bleeds. For this reason compressor stability has a powerful effect on the overall cost, length and weight of the engine, in addition to the performance.

1.2 Mechanical challenges

The previous section briefly introduced the concept of compressor stall from an aerodynamic point of view. However, the challenges are equally applicable to the mechanical designer who must ensure that the blades and vanes of the compressor are free of any resonant vibration (or at least ensure that they are designed with sufficient strength to tolerate such vibration).

Considering a rotor blade within the compressor, the surrounding flow-field will be highly unsteady as the blade is subject to forcing as it passes through wakes shed from the upstream stators, together with the potential field arising from the downstream stators. These unsteady forcing mechanisms have the potential to cause resonant vibration of the rotor blade. For any given blade row, the blade will possess natural Eigenmodes of vibration which occur at discrete frequencies. Through laboratory testing or finite element modelling of such a blade (for example, clamping the blade in a rig and subjecting it to air-jet excitation across a range of frequencies), the natural modes can be identified. Typically the lowest frequency mode (Mode 1) will be referred to as “first flap” (which characterises the principal motion), whilst the second mode (Mode 2) will be referred to as “first torsion”. An infinite number of higher modes will also exist at progressively higher frequencies, each possessing a more complex mode shape.

The potential for resonant aeroelastic vibration of compressor blades is commonly visualised using a Campbell diagram [4]³. An example is shown in Figure 1.3, in terms of frequency against shaft speed. The frequencies of the eigenmodes are shown as near horizontal lines (for simplicity the variation in eigenfrequency with speed is ignored here). Also shown on the Campbell diagram are lines of constant “engine order” (frequency normalised by shaft speed). Typically low engine orders would be shown to represent sources of long wavelength inlet distortion (e.g. once per rev type excitation), whilst higher engine orders would be included specifically to identify the excitation frequencies of the surrounding rows. For example a rotor with 20 upstream stator vanes would experience excitation at 20 engine order (EO) and potentially also at 40EO due to any harmonic content in the stator wakes.

³The Campbell diagram is also commonly referred to as a Spoke diagram or Interference diagram.

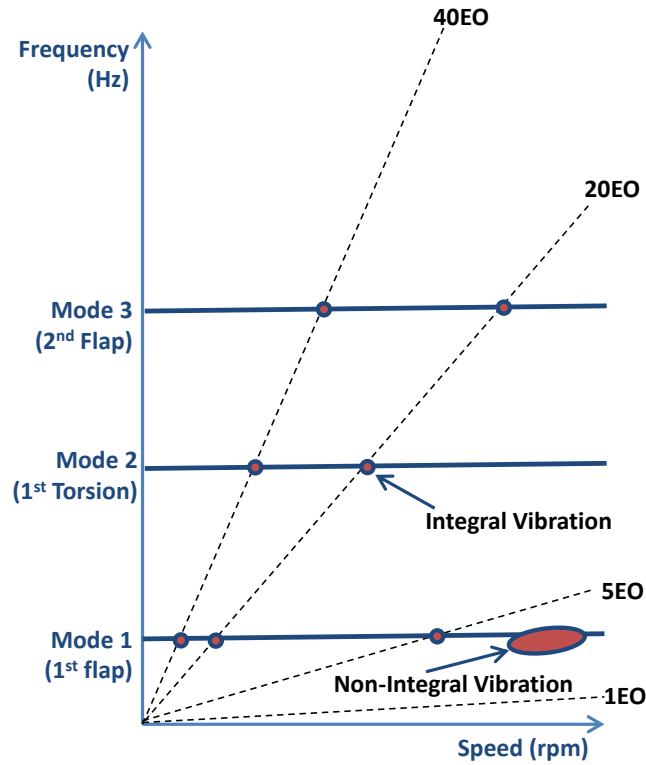


Figure 1.3: Typical Campbell diagram for a compressor rotor blade.

Potential resonances therefore arise at any point on the Campbell diagram where an engine order line intersects a vibration mode line. Careful design of the compressor blading is required (for example in terms of blade counts and frequencies) to ensure these so-called “mode-crossings” are minimised, avoided during normal engine running conditions or cleared by appropriate testing or modelling.

Any resonances occurring at the engine order/mode crossing points shown in Figure 1.3 are commonly referred to as “integral” vibration - this term implies that the forcing mechanism is perfectly synchronised to the speed of the shaft and is therefore always at a known frequency, with an exact integer engine order. However, considering now the rotating stall behaviour introduced in the previous section, it is quite possible for excitation of the

blade to occur at engine orders that are not integer in value and at frequencies that are not synchronised to the shaft speed. For example, the unsteady excitation arising from rotating stall need not occur at fixed engine order, and can occur anywhere between 40-80% of shaft speed. For this reason, aerodynamic phenomena such as rotating stall are commonly referred to as “non-integral” vibration mechanisms⁴, a term which also applies to other aeroelastic phenomena such as flutter, buffeting or acoustic resonance. Finally, the frequency range over which excitation may occur due to rotating stall is potentially very large and is inherently set by the number of stall cells which arise and their propagation speed. Full-span stall, for example, consisting of a single cell will occur below 1EO, whilst part-span stall may consist of over 30 cells (such cases are studied in this thesis) and occur at over 20EO. The precise nature of any rotating stall occurring within the compressor and thus the vibration modes potentially at risk, cannot in general be predicted in advance of the engine development programme, leading to costly delays if parts require redesign.

Finally, there is a long term desire to eliminate cost and weight from the compressor, together with a performance driven desire to increase pressure ratios and aerodynamic loadings. Both of these trends will increase the risk of stall induced non-integral vibration in the long term. One such trend is the gradual replacement of the conventional rotor blade/disc root fixing arrangement, which provides a source of mechanical damping due to friction at the interface, with “blisks”. A blisk consists a disc and blade machined from one solid forging, that whilst offering significant benefits in terms of weight (and potentially also performance), removes the damping mechanism present on the traditional arrangement. From a long term point of view the transition to blisks is also likely to exacerbate any vibratory responses.

1.3 Research objectives

The previous two sections of this chapter have therefore introduced significant challenges to both the industrial and academic community, which this thesis seeks to address as summarised in the following objectives:

⁴Non-integral may also be referred to as non-synchronous or asynchronous.

- Methods are required to predict during design the risk of rotating stall onset within the variable stages of the compressor. More specifically, it is desirable to predict how this risk may be reduced or eliminated with appropriately chosen VSV settings. Currently this process relies largely on a trial and error based process, involving costly high-speed rig testing.
- From a fundamental perspective, increased understanding is required to explain the fluid dynamic origins of stalled flow within the variable stages of high-speed compressors operating at speeds below their design point. The majority of published research on rotating stall concerns measurement or prediction of flow in low-speed laboratory compressors with fixed geometry.
- Tools are required to predict and understand the lengthscales, timescales and frequencies associated with rotating stall in variable geometry compressors such that the aeroelastic vibration risks can be better quantified without the reliance on extensive experimental testing. Similarly, any farfield noise problems associated with stable rotating stall require further understanding.
- Improvements are required in the interpretation of unsteady measurements taken from compressor rig and engine testing, to communicate information regarding the aerodynamic and aeroelastic health of the compressor to the designers. This would allow corrective action to be taken, both to prevent development engine failures and to improve and fix designs in the long term.
- Steady numerical CFD methods have become well established in the design and analysis of high-speed compressors over the last decade and the use of unsteady methods is becoming more common due to improvements in parallel computing methods. This thesis seeks to explore the suitability of URANS modelling techniques to predict rotating stall and partially address the concerns raised above.
- This thesis will be limited to studying cases of rotating stall in several stages of the compressor whilst the overall operation is held stable on the working line. This allows fundamental behaviour to be studied in a

stable and controlled manner. However, the understanding developed here is also likely to apply to the case of surge, where rotating stall is an unstable transient process before complete flow reversal through the compressor. This area is poorly understood both from an aerodynamic and aeroelastic point of view.

1.4 Thesis overview

Chapter 2 introduces some essential fundamental concepts of compressor operation including part-speed matching and the need for VSVs. This is accompanied by a literature survey of relevant research conducted in this field. A very comprehensive review of developments in the field of compressor stall and surge has recently been published by Day[1], which provides an in depth discussion on relevant research themes over the last 75 years. To avoid unnecessary duplication, therefore, chapter 2 presents only a brief overview of this field and seeks to place the research in this thesis into context. In particular it is noted that very little research has been published to date concerning the effect of variable stator vanes on rotating stall behaviour, despite the widespread use of variable geometry compressors across both the aviation and industrial power generation sectors.

The experimental and numerical methodologies used throughout this thesis are presented in chapter 3, which also introduces the test compressor. A significant element of this thesis concerns the use of unsteady instrumentation commonly used to monitor vibration levels (casing pressure transducers and blade mounted strain gauges), which here are used for aerodynamic measurement of unsteady rotating stall phenomena. Within chapter 3, therefore, a relatively simple example of full-span rotating stall is presented as a means to introduce the unsteady processing methods which are used more extensively later in the thesis.

In chapter 4, methods to trigger rotating stall by deliberate adjustment of the VSV settings are then presented from a simple one-dimensional viewpoint. This is followed by the results taken from the initial phase of the experiment, whereby the VSVs were adjusted by varying amounts and the trends observed on the unsteady instrumentation. It is shown how this

results in two “families” of rotating stall in the test compressor, each measured at different characteristic frequencies. Each of these can be linked to stalling of the first or second stage, depending upon the VSV adjustments applied.

The most extreme VSV settings are then investigated in detail in chapter 5, which reveals how the two different “families” of observed phenomena are associated with rotating stall with differing properties. In the second stage, part-span rotating stall with 4-6 cells is observed, with spontaneous transitions in the cell count as the compressor is accelerated. By contrast, deliberate stall of the front stage is shown to result in much shorter length-scale stall cells with over 10 present. Mechanisms for stall induced farfield noise propagation and stall induced vibration are also presented in chapter 5.

Chapter 6 proceeds to introduce the steady state CFD simulations. Initially the simulated stagewise characteristics and radial profiles are validated using experimental measurements to build confidence in the model. More detailed analysis is then conducted to explain the part-speed flowfield in the compressor, in particular to investigate two regions of separated flow which co-exist: stalled flow in the Rotor 1 tip region and a region of separation in the Stator 1/Rotor 2 hub region. Using a 2D streamline curvature model, it is shown how this complex flowfield can be explained by inviscid reasoning and is driven largely by radial equilibrium effects due to the swirl imparted by the Inlet Guide Vane (IGV) which closes down considerably at part-speed.

The steady single passage numerical model was then expanded, in chapter 7, to a full annulus unsteady domain and used to model the behaviour seen in the experiment, both at datum VSV settings and the extreme cases investigated in chapter 5. It is found that by adjusting the VSVs within the unsteady model, rotating stall naturally forms both in the Stator 1/Rotor 2 hub region and also in the Rotor 1 tip. A method is then used to decompose the numerical predictions into spatial Fourier modes allowing direct comparison between the unsteady (post-test) simulation and unsteady measurements taken from the pressure transducers in the experiment. This shows a credible line-up between CFD and experiment and suggests that

URANS methods are able to provide a capable tool to aid the designer in simulation of rotating stall with different VSV settings.

Chapter 8 addresses the more fundamental question of how the setting of the VSVs affects the resulting stall behaviour. An experiment is presented where the relationship between speed and VSV setting is adjusted through so-called “ganged” offsets to the VSV schedule. This shows clearly that the stall propagation speed is very heavily influenced by changes to the settings of the VSVs, whilst the impact of changes to aerodynamic speed alone is found to be weak. This observation is related back to the fundamental operation and shows a link between the local axial flow velocity and the stall propagation speed. Evidence is also shown in this chapter that the front stage stalling phenomenon is able to “lock-on” to a torsional vibration mode of the rotor blade when the stall frequency is sufficiently close to the modal frequency. In this case the stall pattern appears to naturally adjust its cell count to maintain this coupling at constant frequency.

The unsteady modelling process presented in chapter 7 is extended in chapter 9 to explore the effect of flowfield coupling mechanisms. This shows that the formation of rotating stall in the unstable regions of the compressor is heavily influenced by the surrounding stable regions, both axially and radially, which suppress the growth of stall cells beyond a certain size. This is shown to be consistent with published experimental work on low-speed compressors[5] and the effect in this test case is driven largely by the VSV settings. As this coupling mechanism can be considered to be predominantly inviscid, this chapter offers some insight into why URANS predictions are feasible for this phenomenon.

Finally, the conclusions are presented in chapter 10 together with recommendations for future research work in this field.

2 Background information and areas of research

This chapter introduces some key background concepts related to compressor stability, stall and surge and provides an overview of published research which is of relevance to this thesis. In particular, the off-design matching of high-speed compressors is considered in detail, an effect which dictates the need for Variable Stator Vanes (VSVs) and forms a key part of this investigation. Very recently Day [1] has published a very comprehensive literature survey of research conducted on this subject over the past 75 years and is an essential introduction to this topic.

Firstly, the basic concept of compressor stability is discussed by approaching the problem from the fundamental operation of a compressor stage. Subsequent to this, research within the field of rotating stall and surge is discussed which can be broadly grouped into several key areas, each addressed in the following sections. “Stall inception” relates to the transient behaviour at the stability boundary; “Fully developed stall and surge” relates to the longer term behaviour of the system and “Flowfield coupling” investigates how different stages of the compressor, or components of the compression system as a whole, can interact and have a stabilising or destabilising effect on each other. A discussion is then included on the operation of high-speed multistage compressors, where off-design axial matching becomes important and is of particular relevance to this thesis. Finally, recent progress in the field of CFD modelling of stall and surge is presented.

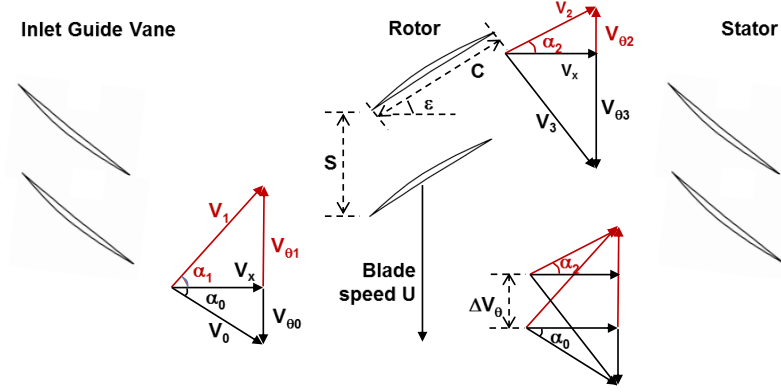


Figure 2.1: Velocity triangles for a compressor stage with key parameters defined. Black and red vectors indicate conditions in the stationary and rotating frames respectively.

2.1 Fundamentals concepts of compressor stability.

The fundamental operation of compressors is commonly described[6] using velocity triangles such as those shown in Figure 2.1, showing the typical mid-span behaviour of a compressor front stage with an upstream Inlet Guide Vane (IGV). The IGV delivers flow to the rotor at a swirl angle α_0 and velocity V_0 in the absolute frame of reference whilst the rotor delivers flow (to another downstream stator) a swirl angle α_2 and velocity V_2 in the relative frame of reference. It is straightforward to show[6] that the rise in stagnation enthalpy ΔH across the rotor is given by the change in the value of $\Delta(UV_\theta)$ across the rotor, i.e. the product of the blade speed U and swirl velocity V_θ . This immediately implies that work can be done by the rotor on the flow to raise its enthalpy (and therefore temperature) through a combination of blade speed and flow turning.

By simple trigonometric reasoning, this behaviour can be stated in a non-dimensional manner using the swirl angles α_0 and α_2 , which to a first approximation can be assumed to remain fixed as the compressor operating point is changed. This leads to Equation 2.1 which effectively defines the operating characteristics of an “ideal” compressor stage in terms of the *work coefficient* $\Delta H/U^2$ and *flow coefficient* V_x/U .

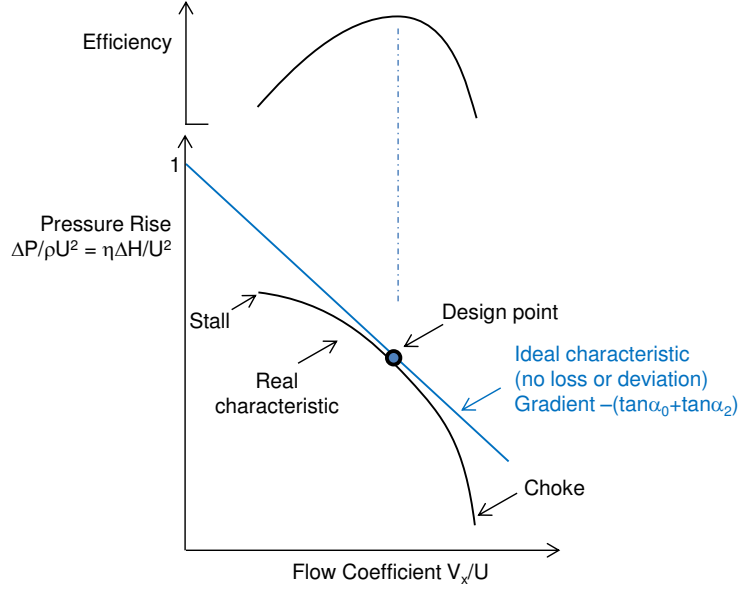


Figure 2.2: Example characteristic of pressure rise coefficient against flow coefficient for ‘ideal’ and ‘real’ compressor stage.

$$\frac{\Delta H}{U^2} = 1 - \frac{V_x}{U} (\tan \alpha_0 + \tan \alpha_2) \quad (2.1)$$

Importantly, the function of the compressor is to raise pressure at delivery to the combustor. Therefore it is important to relate the work done to the useful amount of compression achieved, by introducing the isentropic efficiency η_{is} [6]. It is straightforward to show that the pressure rise across the compressor (stated in non-dimensional form as $\Delta P/\rho U^2$) is equal to $\eta_{is} \Delta H/U^2$.

Considering an “ideal” process which is free of total pressure loss (entropy generation) in each blade row, Equation 2.1 is shown in Figure 2.2 on a map of pressure rise coefficient against flow coefficient as a straight line characteristic which is uniquely defined by the swirl angles α_0 and α_2 .

In reality, as the operating point is moved up the characteristic away from the design point, where the blades operate close to optimal incidence and maximum efficiency (as shown in Figure 2.2), the level of diffusion

experienced by the suction surface boundary layers on each blade row will increase, causing the blades to turn less effectively. This causes α_0 and α_2 to rise (higher deviation) causing a drop in work at a given flow. Furthermore, there will also be a substantial rise in the rotor and stator *inlet* swirl angles (α_1 and α_3) as the flow reduces which implies also a rise in incidence angle onto the blade above its design value. This, together with the increased level of boundary layer diffusion on both the blade and endwalls contribute to increasing losses (and falling efficiency) further up the characteristic, which reduces the pressure rise obtained from the work done by the rotor. This results in the characteristic for a real compressor stage as shown in Figure 2.2.

As the flow is reduced further, the rise in incidence angle and increased level of suction surface boundary layer diffusion will both approach their physical limits and one of these will eventually cause the blade to stall, typically from the endwall regions first. This may either occur from the leading edge, once an incidence limit is reached, or from the suction side towards the trailing edge if the static pressure rise is sufficiently high relative to the available dynamic pressure.

A key aspect of this thesis concerns the behaviour once this stall point is reached and rotating stall occurs. The following section explores relevant research published in the field of *stall inception*, the transient process occurring at this stability boundary.

2.2 Stall inception

As discussed in the previous section, once the operating point of a compressor stage moves along its characteristics to a sufficiently low flow-coefficient, stall will occur as soon as the angle of incidence onto the blade leading edge exceeds a critical limit, or once the level of boundary layer diffusion becomes excessive. It is clearly of value to the designer to understand where this stability boundary is to ensure that the compressor operates free of stall.

2.2.1 Empirical models

A key requirement for the aerodynamic designer is to ensure adequate “surge-margin” is available between the working line (the locus of steady in-engine operation) and surge line, as shown in Figure 1.1. This must be sufficient to allow for rapid transient operation of the engine, which may cause the compressor operating point to rise above its steady working line, and deterioration which may cause the surge line to fall during the life of the engine, for example due to increasing clearances between the rotor and casing¹. The widely used definition of surge margin is given in Equation 2.2, based on the pressure ratio on the working line and surge line (PR_{wl} and PR_{sl}) at a fixed inlet flow function.

$$Surge\ Margin = \frac{PR_{sl} - PR_{wl}}{PR_{wl}} \quad (2.2)$$

Based on Equation 2.2, the immediate question facing the designer is where the surge line is likely to be for a given design. Despite the wide array of available methods from simple 1D correlation based methods through to multistage CFD models, accurate prediction of the stall boundary across the operating range is still not possible. This subsequently means that high-speed rig testing is still required to map the surge line accurately.

Several attempts have been made to capture the essential physics which determine when a compressor stage becomes unstable, into simple empirical criteria which the designer can use for comparison with experimental data and other successful (or otherwise) designs. In simplest terms this could involve ensuring that overall design point parameters such as flow coefficient, work coefficient and reaction are maintained close to a trusted level based on experience and that blades are designed with negative incidence to ensure wide range to stall. However, the desire to maximise performance (with minimal development testing) has driven the need for criteria which more accurately represent the physics of the breakdown of flow into stall.

¹There is a well established link between high tip clearance and poor stability. This topic was recently studied by Young[7].

The work of Lieblein [8] considered how, approaching stall, the suction surface boundary layer limits the pressure rise capability of compressor blades. Using a series of NACA-65 cascade tests, he investigated the main parameters that influence the maximum obtainable loading of the blade. In doing so, Lieblein developed the famous Diffusion Factor, see Equation 2.3, where all the parameters are defined in Figure 2.1.

$$DF = \left[1 - \frac{V_2}{V_1}\right] + \frac{\Delta V_{\theta} s}{2V_1 c} \quad (2.3)$$

Essentially, the diffusion factor seeks to quantify the severity of the static pressure gradient on the aerofoil suction surface between its minimum value location (at the suction peak) and the trailing edge. Typically values up to 0.45 are considered acceptable for design, whilst anything over 0.6 would be considered stalled.

In general, diffusion factor is too simplistic to be generally applicable as an indicator of the point of stall inception. The diffusion factor is strictly applicable to standard aerofoil families (NACA, DCA) and incompressible flow at close to minimum loss incidence. The effects of endwall flows, which are also not captured in the diffusion factor, are also often the dominating influence at stall. These arise both at free-ends of blades in the form of over tip leakage, and the fixed-ended of blades as corner separations. However, despite these limitations, the parameter is useful during the preliminary design phase when more complex methods are not available or practical.

2.2.2 Modal and spike stall

An alternative approach to the subject of stall inception considers the unsteady behaviour of the compressor at the stability boundary. Significant experimental research into this field has been conducted, typically through the use of unsteady instrumentation in the form of hot-wire probes or pressure transducers spaced circumferentially around the compressor casing, which are monitored as the compressor is stalled. This has lead to two commonly accepted mechanisms by which rotating stall may form:

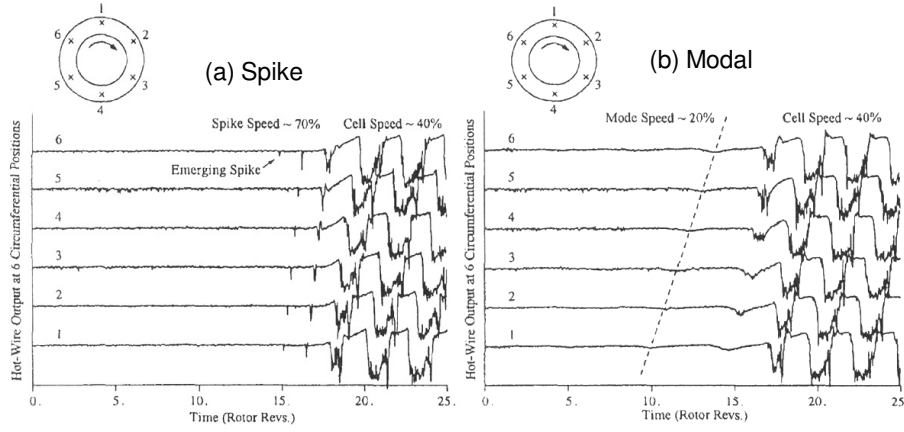


Figure 2.3: Occurrence of spike and modal pre-stall disturbances on circumferentially offset hot-wire probes (Camp and Day [14]).

Image included with permission from ASME.

- *Modal stall*[9], is a long wavelength two-dimensional disturbance that grows from infinitesimally small amplitudes across whole compressor, typically with a circumferential mode order of 1. This has been shown both analytically based on small perturbation theory[9] and experimentally[10] to occur when the compressor is operating on the peak of its total-to-static pressure rise characteristic and develops over timescales of the order of hundreds of shaft revolutions. Significant research into the field of active control has explored the possibility that such modal disturbances may be suppressed, which potentially may increase the stable operating range of the compressor [11].
- *Spike stall*[12], is a short lengthscale three-dimensional disturbance that grows from a finite amplitude in a localised blade passage. This has been shown experimentally to occur whilst the compressor is still on a negatively sloped characteristic and is not associated with the aforementioned modal waves. It is now well established that spike stall is common to the majority of compressors (spike stall is certainly the dominant mechanisms in high-speed machines) and typically develops over timescales of the order 1-2 shaft revolutions. This has implications for the use of active control[13] due to the rapid onset.

Some distinction between these two distinct stalling mechanisms has been made by Camp and Day [14], who identified that, whilst modal stall is a longer wavelength oscillation of the flow field occurring at the peak total-to-static pressure rise, short wavelength spike stall by contrast is a 3-dimensional phenomenon occurring when typically just one blade row exceeds a critical incidence, usually close to the tip. The occurrence of modal or spike stall in a particular situation was shown to be dictated by which of these two limiting conditions is reached first. For example, a compressor operating close to design speed would be well matched, and all stages may reach peak pressure rise simultaneously as the compressor is throttled towards stall. This situation would favour modal stall. Conversely a compressor operating off design at part-speed would tend to operate mismatched: with the front stages stalled and the rear stages choked (as will be shown later in this chapter). Therefore the high incidence onto the front stage rotors would promote spike stall before the overall peak static pressure rise is reached. Thus some compressors stall at peak (modal) whilst others stall on a negative slope (spike).

In the years since this work, significant effort has gone into understanding the fluid dynamic origins of spikes and their subsequent development into full rotating stall or surge. Inoue et al.[15] measured short wavelength transient disturbances and suggested that spike formation may be due to the blade reaching a limiting circulation at stall, shedding excess vorticity into a radial vortex which emanates from the suction surface, stretching up to the casing. Much more recently Young et al.[16] identified irregularities in the blade passing signal approaching stall. They were able to show, through a tactically placed array of pressure transducers, that the irregularity was due to the propagation of disturbances (which she referred to as “blue holes”), with length scale of order one blade pitch, at approximately half the rotor speed. Unsteady CFD modelling by Pullan et al.[17] concluded that these observed disturbances were indeed linked to the radial vortex of Inoue, proposing a propagation mechanism shown in Figure 2.4. The initial shed vortex propagates across the passage, raising the incidence onto the adjacent blade which also stalls. The disturbance then self-propagates in this manner and grows as each newly shed vortex is augmented by the vorticity of the previous. This sequence was proposed as the

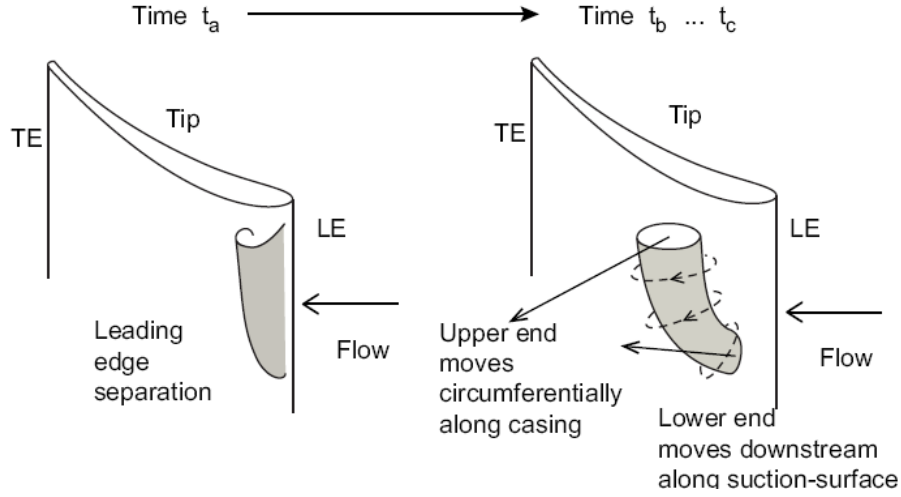


Figure 2.4: Radial vortex mechanism for spike inception (Pullan[17]).
Image included with permission from ASME.

mechanism for spike formation. Furthermore, Pullan[17] shows that the tip leakage flow is not a pre-requisite for spike formation, as both fixed and free blade tips were found, by CFD, to behave in a similar manner. Research into stalling behaviour in centrifugal compressors as show similar behaviour both experimentally[18] and numerically[19].

2.3 Fully developed stall and surge

Whilst the previous section discusses the transient stall inception process at the stability boundary, a separate subject concerns the long term stalling behaviour of the compressor and the system in which it operates.

The “post stall” behaviour of the compressor is known to commonly follow 3 paths[20][21], illustrated in Figure 2.5. For Figures 2.5(a) and 2.5(b), once the stability limit is reached the operating point transiently falls onto a stable “secondary characteristic” of lower pressure rise. In this state the compressor is operating in stable rotating stall and the flow structure is believed to follow the form described by Emmons, Pearson and Grant [2] who proposed the blockage based model for rotating stall depicted in the previous chapter in Figure 1.2.

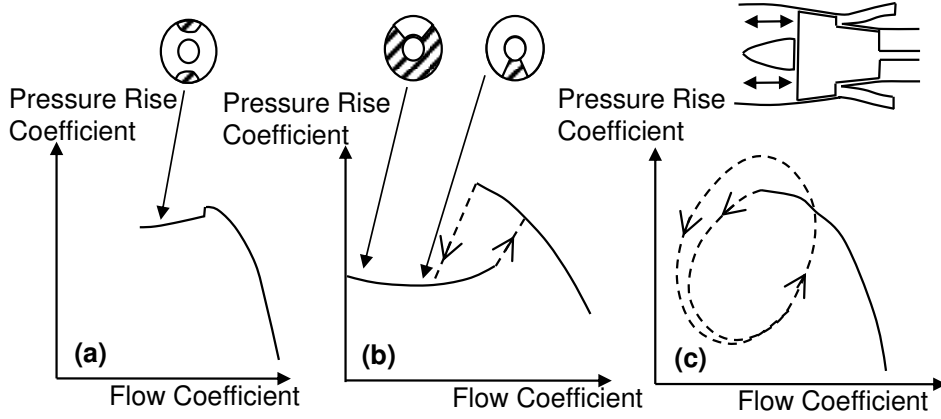


Figure 2.5: Typical characteristics for (a) part-span stall, (b) full-span stall and (c) surge [21]. *Image included with permission from ASME.*

The behaviour in rotating stall can be further split[20] as shown by contrasting Figure 2.5(a) and (b). Within Figure 2.5(a) the transition onto a secondary characteristic is relatively mild. This behaviour is typical of part-span “progressive” stall, whereby multiple stall cells are present local to the hub or, more commonly, the tip region, and can be confined to a limited number of blade rows. The remainder of the span operates normally. Figure 2.5(b), on the other hand, shows an immediate transition onto a secondary characteristic at much lower pressure rise. This is typical of full-span “abrupt” stall, whereby one single stall cell occupies the full radial and axial extent of the compressor. In this situation hysteresis occurs: opening the throttle does not immediately unstall the compressor, rather the operating point moves along the secondary characteristic until a second stability limit occurs at which point the compressor transitions back onto the unstalled primary characteristic. For compressors exhibiting part-span stall, hysteresis is small[20].

The progression into either part-span or full-span stall was studied extensively by Day et al.[20] who defined some simple criteria based on observed behaviour from a large number of low-speed test compressors. Firstly, the total-to-static pressure rise per stage within full-span stall was universal and insensitive to the pre-stall level at a value of approximately $0.11\rho U^2$. Secondly, Day et al. defined a blockage parameter based on the intersec-

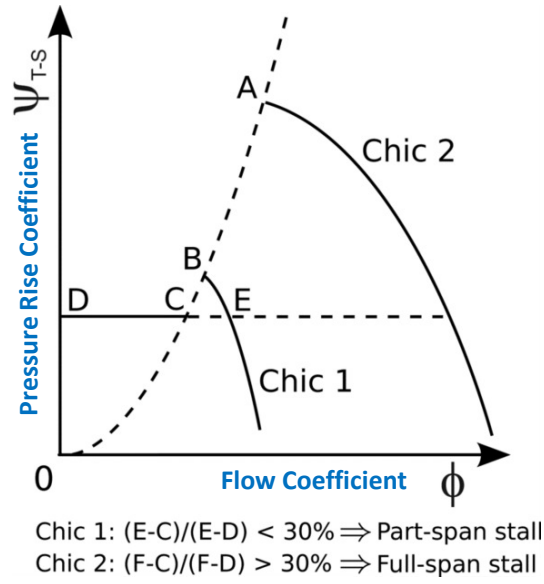


Figure 2.6: Blockage Model of Day et al.[20] for full-span and part-span stall. *Image included from [1] with permission from ASME.*

tion of this secondary characteristic with the stalling throttle line, shown in Figure 2.6. Physically the blockage parameter can be interpreted as the fraction of the annulus with zero flow. Based on the test data, a blockage of greater than 30% appeared to mark the boundary between full-span and part-span stall. Interestingly, the paper also shows that a well matched compressor with more stages and/or a higher design V_x/U will be more likely to encounter full-span stall and therefore a greater hysteresis effect.

Within the category of part-span stall, there are a range of physical behaviours reported in the literature. Day and Cumpsty [22] demonstrated how stable part-span stall was possible with 4 to 12 stall cells present, depending on how the compressor was configured in terms of design flow coefficient, reaction, number of stages and axial gaps. Kameier and Neise[23] and others[24][25] have all reported the occurrence of part-span rotating stall consisting of a non-uniform cell pattern with a high cell count and referred to the phenomenon as “rotating instability”. This was also linked with elevated farfield noise levels[23] and attributed to the effect of high tip clearances. It should also be noted that part-span stall is typically

higher risk from an aeroelastic vibration point of view, as the phenomenon generally occurs at higher frequencies than full-span stall and has greater potential to interact with the resonant modes of the compressor blades[26].

Figure 2.5(c)[21] shows the third post stall flow regime, known as surge. By contrast with rotating stall, which occurs within the compressor, surge is a result of instability of the whole compression system including the plenum volumes at inlet and exit and the exit throttle. During surge (after the initial inception phase), the high pressure gas stored within the exit plenum oscillates back and forth through the compressor and results in significant mechanical loads on the blades and other components[27]. In the case of an engine surge this may result in the combustion products reversing through the compressor, often with flames visible at the engine face. Once the plenum discharge is complete the conditions arise for normal flow to resume, re-pressurising the plenum until the unstable condition is once again reached. This leads to a cyclic oscillation of the gas within the whole system around the orbit on the compressor map in Figure 2.5(c) on a time scale of the order 1-10Hz (surge is a much lower frequency phenomenon than rotating stall), until corrective action is taken such as opening the throttle.

Greitzer [28] and [29] studied the problem of surge in significant further detail and proposed a simple one-dimensional model of the compression system based on the Helmholtz resonator concept. Greitzer found that the response of the system was heavily dependent on the non-dimensional B-parameter given in Equation 2.4, where L and A refer to the effective compressor duct length and area respectively, V_{plen} refers to the exit plenum volume, a is the speed of sound whilst U refers to the blade speed.

$$B = \frac{U}{2a} \sqrt{\frac{V_{plen}}{AL}} \quad (2.4)$$

At a low B-parameter (low-speed, low plenum volumes) the energy storage within the plenum was insufficient for surge to occur and rotating stall would occur instead. Conversely, at a high B-parameter, surge would occur, with the intensity and cycle period increasing with B. By testing a compressor with a variable exit plenum, Greitzer[28] showed both surge and rotating

stall could be initiated at similar compressor operating conditions, i.e. the post-stall phenomenon encountered depends not just on the compressor, but the installed “system level” environment.

2.4 Flowfield coupling

One significant aspect of multistage compressor stall is the unsteady interaction between blade rows and stages and even between compressors in the multi-spool environment. It is clearly of importance to understand how a compressor and its individual blade rows will behave in their installed environment relative to that in isolation. The empirical criteria discussed earlier do not capture any interaction of this type and any such coupling effects are commonly overlooked during compressor design.

Component interactions of this type have been studied by several authors, one such case is Greitzer and Grimsword[30] who showed that the presence of an exit diffuser can exacerbate the effect of circumferential total pressure distortion in the upstream compressor. This was shown to be because the function of the diffuser is to decelerate the flow and raise its static pressure, which it typically delivers to a uniform exit static pressure condition (a large volume). Assuming that the level of diffuser static pressure rise is always directly proportional to the inlet dynamic pressure D (i.e. $\Delta p_s/D = \text{constant}$), it follows that in any distorted sector of the compressor where D is low, the static pressure rise must fall leading to an *increase* in the local static pressure at diffuser inlet. This local rise in static pressure, as it was shown[30], causes this sector to operate further up its characteristic, reducing the flow further and “amplifying” the effect of the distortion. The same behaviour was demonstrated by Greitzer et al.[31], who further showed that an accelerating nozzle has the opposite effect and is able to suppress the effects of circumferential distortion.

The work of Longley and Hynes [5] on a three stage low-speed compressor also shows clearly the effect of flowfield coupling in a multistage compressor environment. Through deliberately mismatching their compressor, by staggering the blades of the rear two stages closed, the operating range of the front stage could be extended well beyond the stall point obtained as an

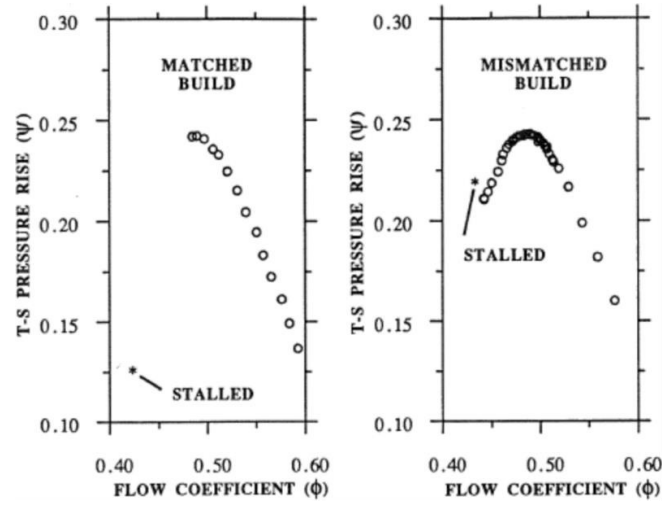


Figure 2.7: Characteristic of front stage in a matched build and a mismatched build(Longley and Hynes[5])
Image included with permission from ASME.

isolated stage or within a well-matched compressor, as show in Figure 2.7. Their explanation for this was that any stall cells forming in the front stage, would be attenuated by the stable downstream stages operating on negatively sloped characteristics (when staggered closed). This coupling between stages thus supported the stalled front stage and prevented the growth of stall disturbances beyond a short lengthscale such that the compressor overall remained stable.

2.5 High-speed compressor axial matching.

The vast majority of published experimental research in the field of stall and surge has been performed on low-speed compressors where the changes to operating speed have a negligible effect on the axial matching between stages. However, it is well known that in a high-speed multistage compressor the front stages become stalled at part-speed. The following section explains this fundamental behaviour and also explores the need for Variable Stator Vanes (VSVs) as a common method to ensure the compressor still has adequate stability margin.

2.5.1 Fundamental behaviour

The part-speed operation of high-speed multistage compressors can be most easily explained by considering the effect of operation with a choked exit nozzle. Such operation is a valid assumption aero-engine compressors, which typically operate with the turbine nozzle guide vanes choked across the majority of the operating range.

Based on well established compressible flow theory [32], the choked nozzle will impose a fixed non-dimensional exit flow function ($\dot{m}\sqrt{c_p T_0}/(AP_0)$) on the compressor, which is usually simplified to $\dot{m}\sqrt{T_0}/P_0$. As the speed is reduced from the design point, the compressor must still deliver a fixed exit flow function, however the lower blade speed (U) means that the compressor does less work on the flow (the enthalpy change along a streamline at fixed radius is given by $U\Delta V_\theta$) which must also result in a lower pressure ratio. Based on continuity of massflow and the assumption of isentropic flow (which is sufficient for explanation purposes), Equation 2.5 can be written which relates the change in flow function between compressor inlet and exit to the pressure ratio.

$$\left. \frac{\dot{m}\sqrt{T_0}}{P_0} \right|_{inlet} = \left. \frac{\dot{m}\sqrt{T_0}}{P_0} \right|_{exit} \times \left(\frac{P_{0\,exit}}{P_{0\,inlet}} \right)^{\frac{\gamma+1}{2\gamma}} \quad (2.5)$$

For the overall compressor this shows that, if the exit flow function is to remain constant, a fall in pressure ratio must be accompanied by a reduction in inlet flow function. Stated in more physical terms, as the speed changes the pressure ratio also changes, and the compressor must always accept the correct amount of inlet flow that it can compress into a fixed exit nozzle area.

Operation with a choked exit nozzle thus forces the compressor to operate along a fixed working line - as shown in Figure 2.8, which follows a line of constant exit flow function. Closing the exit throttle will lower the compressor exit flow function and progressively raise the working line (moving the compressor to higher pressure ratio and lower inlet flow function, con-

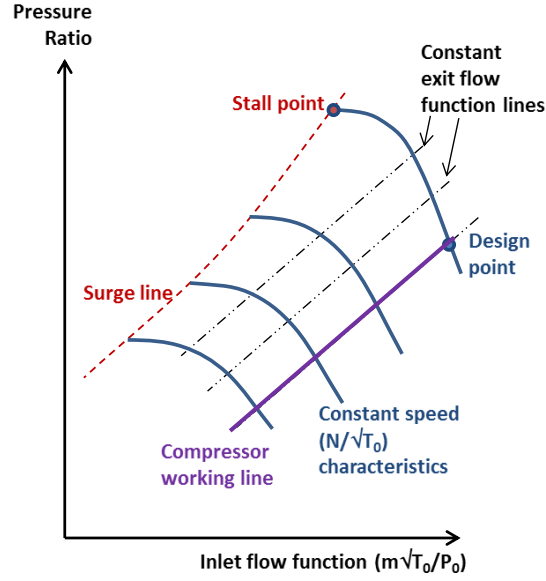


Figure 2.8: Typical high-speed compressor overall operating map with lines of constant exit flow function highlighted.

sistent with Equation 2.5) until eventually it intersects the surge line and the compressor stalls or surges.

The “stagewise” operation of the compressor is illustrated in Figure 2.9, which shows the operating map in the same manner now for both the front and the rear stage of a compressor, which for simplicity can be assumed here to have only 2 stages. An important point to emphasize again here is that the compressor exit flow function remains fixed across the speed range. It is therefore logical to explain the effect by starting at the rearmost stage and then move forwards (upstream).

As discussed, the rearmost stage of the compressor is forced to deliver a fixed exit flow function due to the choked exit nozzle, a condition which remains true across the speed range. As the compressor speed is reduced, the rearmost stage will do less work and deliver a lower pressure ratio.

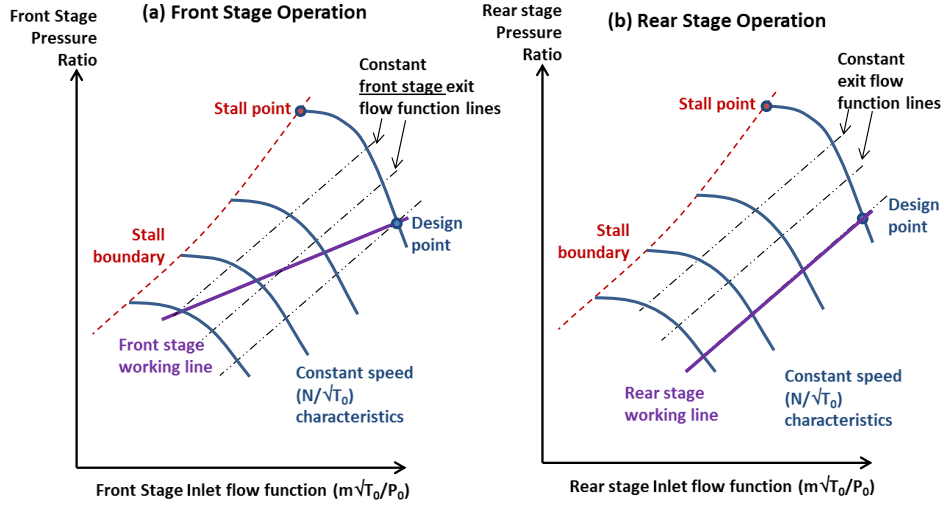


Figure 2.9: Typical operating maps for front and rear stages of a 2 stage high-speed compressor showing the problem of part-speed stall in the front stage.

As shown in Equation 2.5, applied now to just the rear stage alone, this will reduce the the flow function at entry to the rearmost stage. On the operating map in Figure 2.9(b), the rearmost stage will operate along a line of constant exit flow function which, as shown, keeps well away from its local stall boundary at part-speed.

As the flow function at entry to the rearmost stage falls with speed, this has immediate implications for the stage upstream. For the simple case of a 2 stage compressor, the front stage must always deliver a flow function at its own exit demanded by the downstream stage. As this falls with speed Equation 2.5 dictates that the front stage must move to higher pressure ratio and/or lower inlet flow. Stated another way, the front stage must rise progressively up its characteristic as the speed is reduced, to compress flow more as the rear stage “flow swallowing” capacity is reducing. In Figure 2.9(a) this can be seen visually, the front stage does not operate on a constant downstream flow function working line and actually moves progressively towards its stall boundary at lower speeds.

Based on this logic applied cumulatively across a multistage compressor, it becomes clear that the part-speed excursion in operating point of the front stages of any compressor will be considerably larger than that of the rear stages. The same is also true for a 3 shaft engine arrangement[33] where an intermediate pressure compressor (IPC) operates upstream of a high pressure compressor (HPC). In this case, it is the HPC flow reduction at lower speeds forces the IPC onto a flatter working line. Without mitigation, this effect therefore always results in the front stages of the compression system becoming stalled at part-speed and limits the obtainable surge margin.

This problem can be addressed in three ways. First, the front stages may be designed very conservatively to ensure adequate operating range to handle this part-speed excursion, however this is undesirable due to the severe compromises required in terms of design speed efficiency and obtainable pressure ratio. There is thus a limit to how far such an approach can be taken. Second, handling bleed may be used to raise the flow function through the front stages and lower their working line at part-speed (by discarding some “bleed” air at part-speed through a valve between the front and rear half of the compressor). This is also undesirable - whilst bleed valves can be opened only at part-speed, there is still an impact to cycle efficiency from discarding air from the compressor after several stages of compression. Moreover, the discharge of the bleedflow (typically into the engine bypass duct) is a significant noise source. Thirdly, the front stage stators may be of variable stagger design as shown in Figure 2.10[33]. These close down² at part-speed allowing the incidence and loading (i.e. diffusion) levels in the front stage blade rows to be kept within acceptable limits and thus increase the margin between the working line and surge. Whilst all three of these options are utilised in modern aeroengine compressors, it is the effect of variable stator vanes (VSVs) that are investigated here.

A major complexity with variable stator vanes is the problem of optimising their settings across the speed range to ensure adequate overall stability is achieved, whilst also ensuring efficiency is maximised. This is typically done on a compressor rig test similar to that presented in this thesis, where the optimum VSV setting are found at each speed and combined together

²In this thesis VSV closure means stagger increasing, where stagger ϵ is defined in Figure 2.1

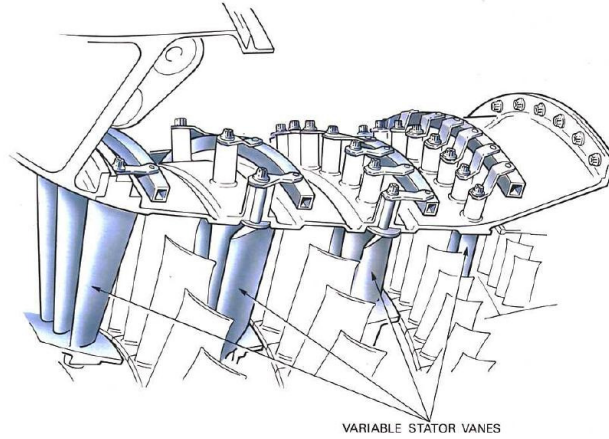


Figure 2.10: A typical VSV mechanism ([33]).

to define the “VSV schedule” - effectively how the vanes move across the speed range relative to their design speed settings. It is also common for the variable stator vanes on several stages to be driven from a single actuator, by coupling them together on a so-called “ganged” mechanical system such that the travel of each stage across the speed range is proportional to the other stages. Cumpsty [6] shows that for optimal performance an IGV closure in the region of 40° is required to ensure adequate part-speed stability, with the downstream vanes closing progressively less moving rearwards within the compressor. Such a typical VSV schedule is shown in Figure 2.11 which follows this idealised operation and maintains constant proportionality between each row of vanes.

The use of VSVs is therefore very effective in mitigating the part-speed mismatching effects discussed in this section. Considering Equation 2.1, closure of the IGV increases α_0 and effectively lowers there operating point to a lower work coefficient. This also reduces boundary layer loading levels and incidence angles on a given working line and gives rise to a significant improvement in overall range to stall. When the VSV schedule is optimised, it is therefore quite possible that the front stages no longer become unstable first at part-speed, and instead the stability is limited by stall onset in the first fixed stages, downstream of the variable stages.

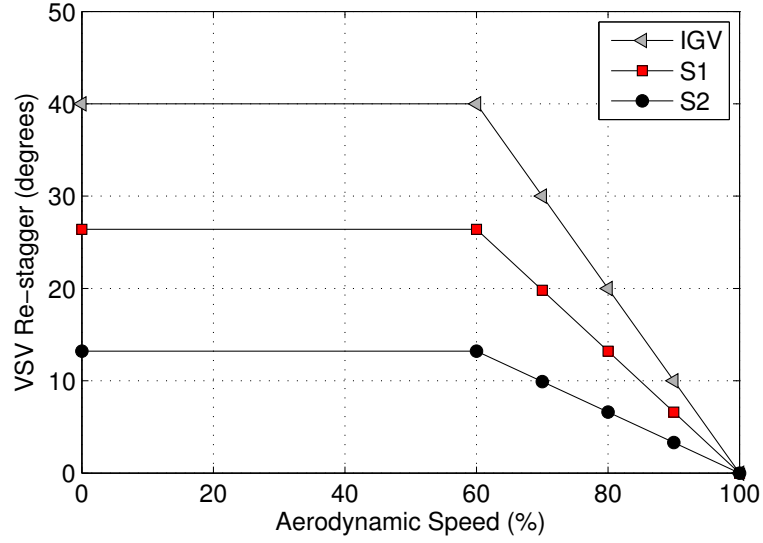


Figure 2.11: A typical variable stator vane (VSV) schedule.

2.5.2 High-speed compressor research

The majority of published research on stall and surge is based upon low-speed compressor testing where the phenomena can be studied in a controlled manner, although the matching effects discussed in the previous section are not present as the flow remains incompressible. Such testing on high-speed machines is considerably more complex.

Day and Freeman[34] presented a systematic comparison between behaviour at low and high-speeds, using a three stage compressor at the Whittle Laboratory and a Rolls-Royce Viper engine, with an 8 stage 5:1 compressor. This comparison demonstrated similarities between both machines and confirmed that rotating stall always precedes surge even at high-speed conditions. By testing the engine at different speeds, Day and Freeman were also able to confirm that stable full-span rotating stall occurred at lower speeds, whilst surge occurred at high-speed, consistent with Greitzer[28]. It was also reported[34] that the engine always exhibited spike stall inception, with no evidence of modal waves at any speeds. Finally, low-speed operation of the engine resulted in multi-cell part-span stall local to the front stages (referred to as “front-end stall” [34]). This was shown to arise due to the matching effects discussed in the previous section, which cause the front

to become stalled at part-speed whilst the compressor as a whole remains stable. For this reason the presence of rotating stall in the front stages did not impair the operation of the engine during starting.

Escuret and Garnier[35] also presented the stalling behaviour of a high-speed 4 stage compressor with a variable IGV. At all conditions the compressor stall inception route was always via short lengthscale spikes, though adjustment of the IGV setting caused the spikes to form in different stages. This was consistent with the IGV adjustment redistributing loading within the compressor and causing different stages to become unstable first.

2.6 CFD methods

Although unsteady 3D CFD methods have been in use for some time, the significant computational effort required to model an unsteady process such as rotating stall has until recently been out of reach. Some progress has been made using “reduced order” models where some simplification is made.

An early CFD application to the rotating stall problem was made by He[36], who performed unsteady computations to capture rotating stall in a stage with differing rotor-stator counts on a two-dimensional stream surface. For 10 and 8 rotors and stators respectively the model shows the formation initially of 2 stall cells. He therefore suggested that the stall cell count was due to the difference-order resulting from the blade counts. This was supported by a second calculation with 9 stators which stalled directly via a single cell. A similar stream surface calculation was performed by Gourdain et al.[37] closer to the tip region of a single stage compressor, who also reported the presence of unstable disturbances present in the solution for a significant period prior to the appearance of cells, which were attributed to modal waves. However, it is likely in such a calculation setup that three-dimensional spike stall may have not been possible. Subsequent calculations by the same authors [38] with a fully three-dimensional analysis have shown a credible match to experimental data in terms stalling massflow, although the detailed rotating stall behaviour was not correctly predicted. In this case, the model initially exhibited part-span stall for an extended duration whilst the experiment rapidly stalled in a full-span manner.

In the previous cases the analysis was restricted to low-speed single stage compressors, however Vahdati et al.[39] have demonstrated that part-span rotating stall can be simulated in the front stages a high-speed variable geometry compressor using a RANS solver³. Using a single passage model, they were also able to resolve the axisymmetric cyclic flow process occurring during a surge. This work clearly demonstrated that capability exists to simulate such flow features using a RANS based method. Using the same method, Choi et al.[40] presented simulations of full-span stall for a high-speed single stage fan, in this case capturing the inception, stabilisation on a secondary characteristic and the recovery process as the throttle was opened. The throttle hysteresis typical of full-span stall was also demonstrated in this model. A second publication by Choi et al.[41] showed that this predictions agree well with experimental data.

The use of CFD methods has also provided some further physical understanding of rotating stall. Using a partial sector model, Vo et al.[42] proposed criteria for spike stall inception, related to the over-tip leakage. The more recent numerical analysis by Pullan et al.[17] (already discussed), has developed this further, showing that spike stall inception is quite possible at fixed-end blades, without over-tip leakage. Similar behaviour has also been simulated in centrifugal compressors [19].

³Reynolds-Averaged Navier-Stokes solver, referring to CFD techniques which compute/estimate the turbulent Reynolds Stress term based upon the averaged flow quantities.

3 Methods

In this chapter the test compressor is introduced, together with the instrumentation and data processing methods. Analysis of unsteady aerodynamic phenomena using fast response instrumentation forms a significant element of this thesis, and this chapter therefore provides an overview of how such analysis is performed, both in the time domain and using frequency domain (i.e. Fourier) methods. For this purpose example cases are presented of rotating stall and surge, seen on this test compressor whilst operated with the VSVs set to their optimal angles, which is shown to behave in a manner consistent with other cases reported in the open literature [28] [34]. These examples provide useful opportunities to introduce the analysis methods, in preparation for more complex cases studied in later chapters, where the VSVs are deliberately offset to trigger stall locally in the front stages.

Following this, an overview is provided of the numerical methods used in this thesis to complement the experiment. Lower fidelity analysis was performed using a one-dimensional meanline model, used to rapidly assess different Variable Stator Vane (VSV) settings. A two-dimensional throughflow model was also used to aid understanding of powerful inviscid radial equilibrium effects, which arise due to part-speed operation with the VSVs closed down¹. For several selected conditions tested in the experiment, steady and unsteady CFD simulations were also performed, which contribute a significant part of this thesis. The later sections of this chapter therefore introduce these methods.

¹In this thesis VSV closure means stagger increasing, where stagger ϵ is defined in Figure 2.1

3.1 Experimental methods

3.1.1 Description of test compressor

A schematic view of the test rig is shown in Figure 3.1. The high-speed research compressor was designed by Rolls Royce and is representative of the type used in a modern aircraft engine application. The compressor has 8 stages with an Inlet Guide Vane (IGV) upstream of the first rotor. The IGV and the first two stator rows (Stators 1 and 2) are of variable stagger type, with each row controlled by an independent actuator which drives a unison ring connected to each vane by a set of individual levers. A typical Variable Stator Vane (VSV) mechanism was shown in Figure 2.10[33]. The independent actuation of the vanes of each stage on this rig offers additional flexibility when compared to a typical engine application, where in the latter case it is common for a single actuator to drive several stages together through a “ganged” mechanism, forcing the stators from one stage to always move in linear proportion to the stators in the other stages. In this experimental setup the use of independent “unganged” actuation on each stage allows different VSV configurations to be tested with relative ease.

Tip clearances and stator shroud clearances are representative of a modern high-speed compressor and the Rotor 1 relative inlet flow is supersonic at design conditions. The compressor was driven by a 10MW rated motor and discharges into an exit plenum, downstream of which is a throttle valve mechanism allowing the compressor to be operated on a wide range of working lines. Adjustment of the throttle allows constant speed characteristics to be mapped between choke and stall/surge. Detailed examples of this will be covered later in this section.

The compressor operates with ambient inlet temperature, which remained approximately constant throughout the test. Therefore the absolute shaft speed (N) and aerodynamic speed ($N/\sqrt{T_0}$) are within 1% of each other when normalized by their value at design speed with standard daytime atmospheric conditions. For simplicity, the difference between these speed definitions has been ignored as it does not significantly affect this investigation.

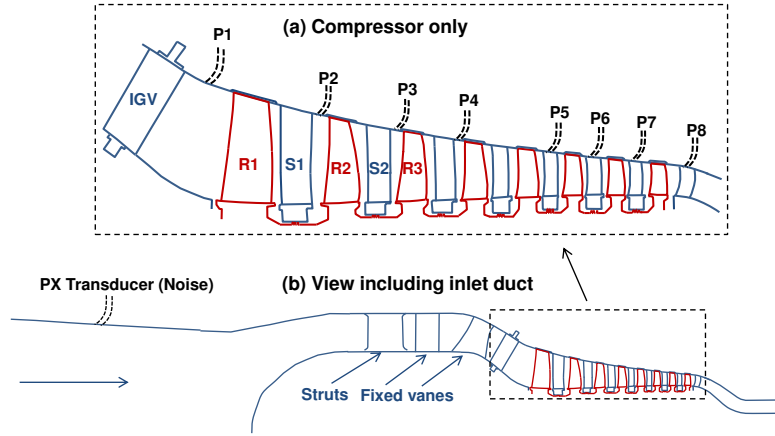


Figure 3.1: Schematic view of the compressor rig.

Unsteady pressure measurements were recorded on 8 fast response pressure transducers (Kulite XTE-190-50A) distributed axially as shown in Figure 3.1(a), denoted P1-P8. To provide an indication of any possible farfield noise sources an additional transducer was located far upstream in the inlet duct as also shown in Figure 3.1(b), denoted PX. All transducers were placed circumferentially within 45° of each other, as space allowed. These probes are industry standard for close coupled pressure measurement in turbomachinery environments and each consist of a piezoresistive semiconductor element with an embedded pair of strain gauges coupled to form a Wheatstone Bridge. The signal for each probe is fed into a data acquisition system for live monitoring and stored to a drive for subsequent analysis.

Vibration measurement was possible using strain gauges mounted with adhesive to the rotor and stator blades of stages 1 to 4, which were included primarily to identify any high stresses. However, as is shown later in this chapter, the strain gauges also provide valuable insight into the unsteady flow on the blade surface, even when the blades are away from resonance and the vibration levels are small. To accurately capture the blade stresses, the gauge positions were chosen carefully based upon the vibration modeshapes (from either Finite Element modelling or laboratory testing) to ensure they were located in active regions of the blade where the effective strain was large. For rotating instrumentation, a telemetry unit within the drum was used to transmit data back into the stationary frame of reference.

All unsteady data (both from the pressure transducers and strain gauges) was acquired at 96 kHz, which is sufficient to capture all blade passing and rotating stall phenomena, which are typically in the 0-10kHz range for this compressor. This recorded data was subsequently low pass filtered (down-sampled) to a bandwidth suitable for detailed analysis. Through this process the Nyquist anti-aliasing criteria is met as all phenomena of interest occupy a bandwidth significantly smaller than half the sampling rate. All Fourier analysis was performed using HGL Tornado[43] and subsequently analysed in MATLAB[44].

Steady state instrumentation was also present on the test rig in the form of kiel shrouded total pressure and temperature probes. These were mounted on inlet and exit rakes (for overall performance) and upon the leading edges of a small number of stator vanes on each stage (for stage-wise performance). In chapter 6 this instrumentation is used to calibrate the steady CFD model and explore the underlying flowfield, although during the transient manoeuvres presented in subsequent chapters, this instrumentation is not reliable (due to the slow time response of the pressure lines between the measurement location and instrumentation) and is therefore ignored. Accurate flow measurement was provided by an airmeter at inlet to the test facility, downstream of which the flow passes through a large settling tank before entering the compressor section.

3.1.2 Frequency domain analysis of unsteady phenomena.

It is instructive at this point to discuss the methodology used in later chapters for the analysis of unsteady phenomena, using the fast response instrumentation. Figure 3.2 shows the operating map of the compressor across a range from 50 to 100% speed, as measured during the test using instrumentation at compressor inlet and exit. The operating map is presented in the common format, with Pressure Ratio shown against Inlet Flow Function, with characteristics of constant aerodynamic speed shown. These points were measured from below the engine representative working line (also shown in Figure 3.2) to the last stable point as the exit throttle was closed in sequential steps.

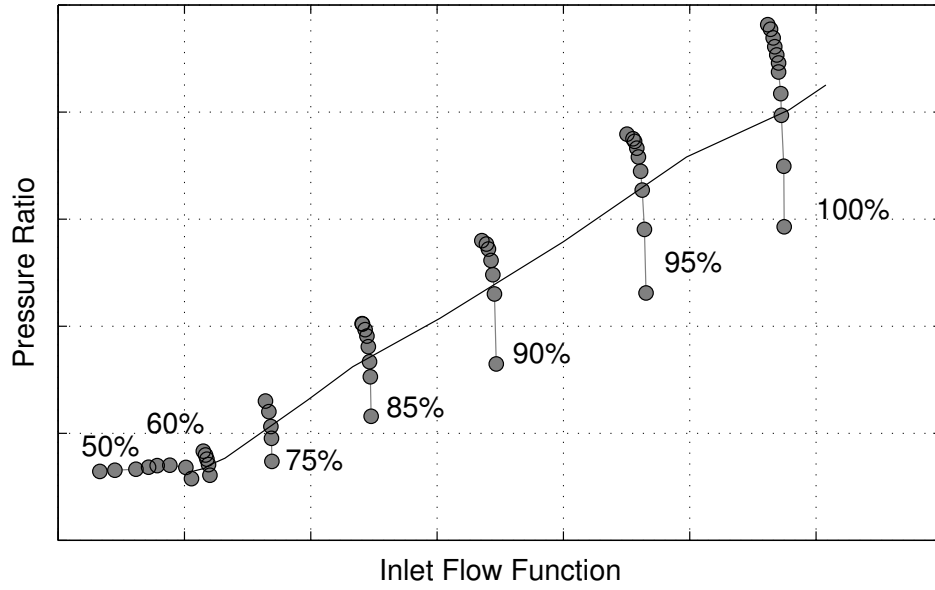


Figure 3.2: Operating map of the test compressor, with VSVs in their nominal settings.

Considering now the behaviour at a fixed speed, a stationary pressure transducer, such as that at location “P1” (at Rotor 1 leading edge - see Figure 3.1), will see a time varying signal due to the passing pressure field of Rotor 1 (and to a lesser degree the downstream rotors). With the shaft speed held constant the Fast Fourier Transform (FFT) spectrum for this signal will be dominated by the blade-passing frequency (BPF) of the Rotor 1 blade row which is equal to the product of the number of blades and the shaft rotational speed (in revolutions per second). This spectrum, as measured, is shown in Figure 3.3 where a clear single peak can be seen when the compressor was stabilised close to 85% speed (for simplicity the condition chosen here, in terms of speed and variable stator vane settings, was free of rotating stall). Figure 3.3 was obtained by computing a 1024 point fast Fourier transform (FFT), across a frequency range sufficient to capture the Rotor 1 blade passing frequency. The frequency shown in Figure 3.3, is a non-dimensional “reduced frequency”, calculated based upon a characteristic velocity and length scale, which in this case has been estimated from the mean inlet axial velocity at this condition, together with the Rotor 1 blade chord length at mid-span. However, frequency may also be normalised

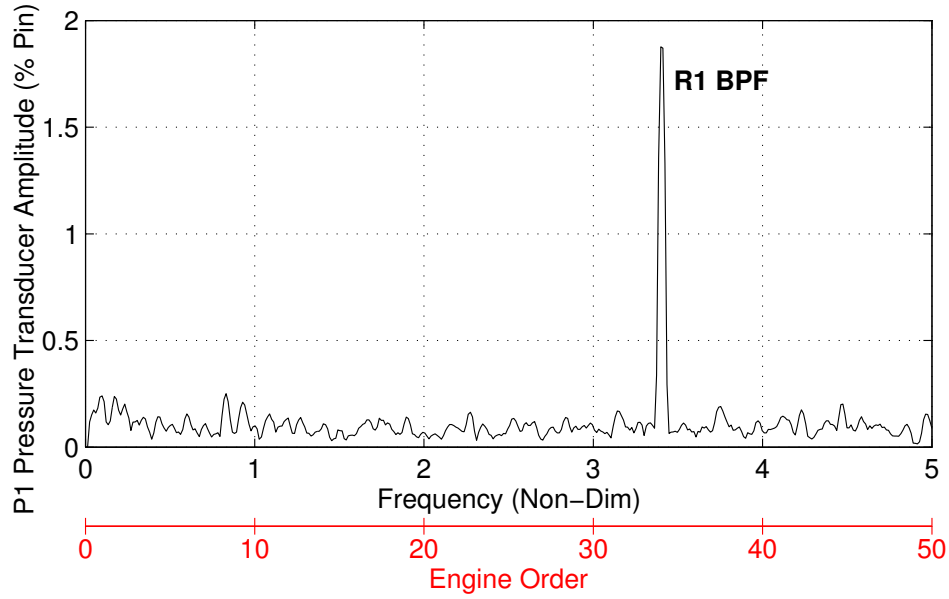


Figure 3.3: Spectral analysis of P1 pressure transducer (R1 inlet) whilst stabilised at part-speed condition.

based upon the shaft speed (revolutions per second), which is commonly referred to as the Engine Order (EO), as also shown in Figure 3.3. When interpreted this way, the Rotor 1 blade passing response aligns exactly with 34EO, which corresponds to the number of individual Rotor 1 blades.

The behaviour as the compressor is accelerated through this condition, from 70% to 95% speed, is shown in Figure 3.4, at the same location. This was obtained by decomposing the raw time signal through this transient manoeuvre into 360 discrete intervals and calculating, for each interval, the FFT in the same manner as shown in Figure 3.3. Figure 3.4 was produced by arranging these spectra side by side into a Campbell Diagram[4], in which the amplitude at each frequency and speed is represented by a shade of colour and presented using a logarithmic scale. In this case the reduced frequency is used as the ordinate, and lines of constant engine order are highlighted, for example an EO5 line has been added manually, and EO34 is seen to be present naturally in the signal due to the Rotor 1 blade passing, which clearly rises linearly with increasing speed. For the remainder of this thesis this format will be retained for visualising data and for simplicity all discussion within the text will use the Engine Order notation.

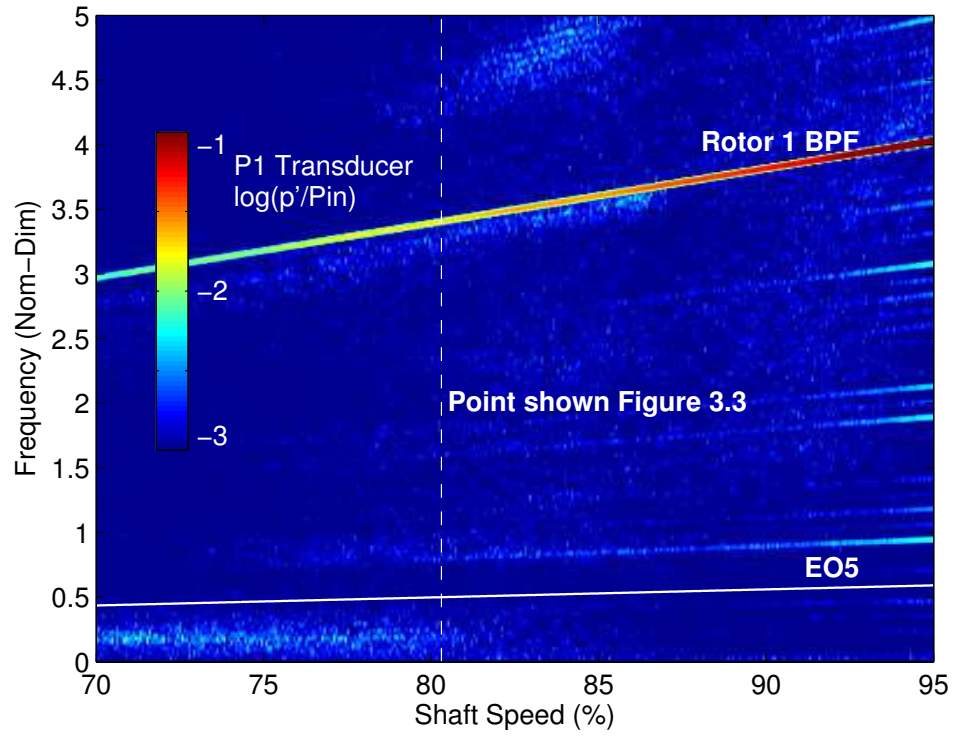


Figure 3.4: Spectral analysis of P1 pressure transducer (R1 inlet) whilst accelerating from 70 to 95% speed. Dashed line corresponds to spectrum shown in Figure 3.3.

3.1.3 Methods for analysis of rotating stall and surge

The previous section introduced the spectral processing techniques used throughout this thesis on a simple example, where non-integral phenomena such as rotating stall and flutter were not present. This showed the blade passing signal from the front stage rotor at steady state and transient conditions. This thesis will progressively build upon these techniques to explore more complex phenomena. In this section, therefore, the methodology is extended to examples whereby the compressor exhibited rotating stall and surge.

Whilst the later chapters will focus on the unsteady behaviour in the front stages at the working line condition as the VSVs are adjusted to provoke stall, two key points of interest in this section are those at the stability boundary of the compression system at 95% speed and 50% speed. These serve as useful “events” to introduce the processing techniques used in the later chapters. It can be seen in Figure 3.2 that at 50% speed the characteristic takes on a different shape, in comparison to those at higher speeds. In the case of the 50% speed line, the pressure ratio quickly reaches a maximum and then flattens and moves progressively to lower inlet mass flow as the throttle is closed in gradual steps. At 60% speed and above, by contrast, this behaviour is not seen. The physical reason for this is that the 50% speed case is exhibiting rotating stall to the left of the stability boundary, whilst at speeds of 60% and above, surge is encountered. This behaviour is consistent with the compression system (i.e. compressor, duct and plenum) operating at sufficiently low-speed that the exit plenum has insufficient energy to trigger a surge. In terms of the well established research of Greitzer [28] [29], discussed in chapter 2, the “B-Parameter” is sufficiently low at 50% speed that instability leads to rotating stall, rather than surge.

The behaviour seen in Figure 3.2 at the overall level presents a useful opportunity to explore the unsteady data for these cases and present the methodology that will be used in later chapters. Figure 3.5 shows the time signals recorded from the P2 pressure transducer (location shown earlier in Figure 3.1) for both the 50% and 95% speed characteristics as the compressor was throttled beyond its stability limit. At 50% speed the time signal is shown across a period of 16 minutes as the compressor was progressively

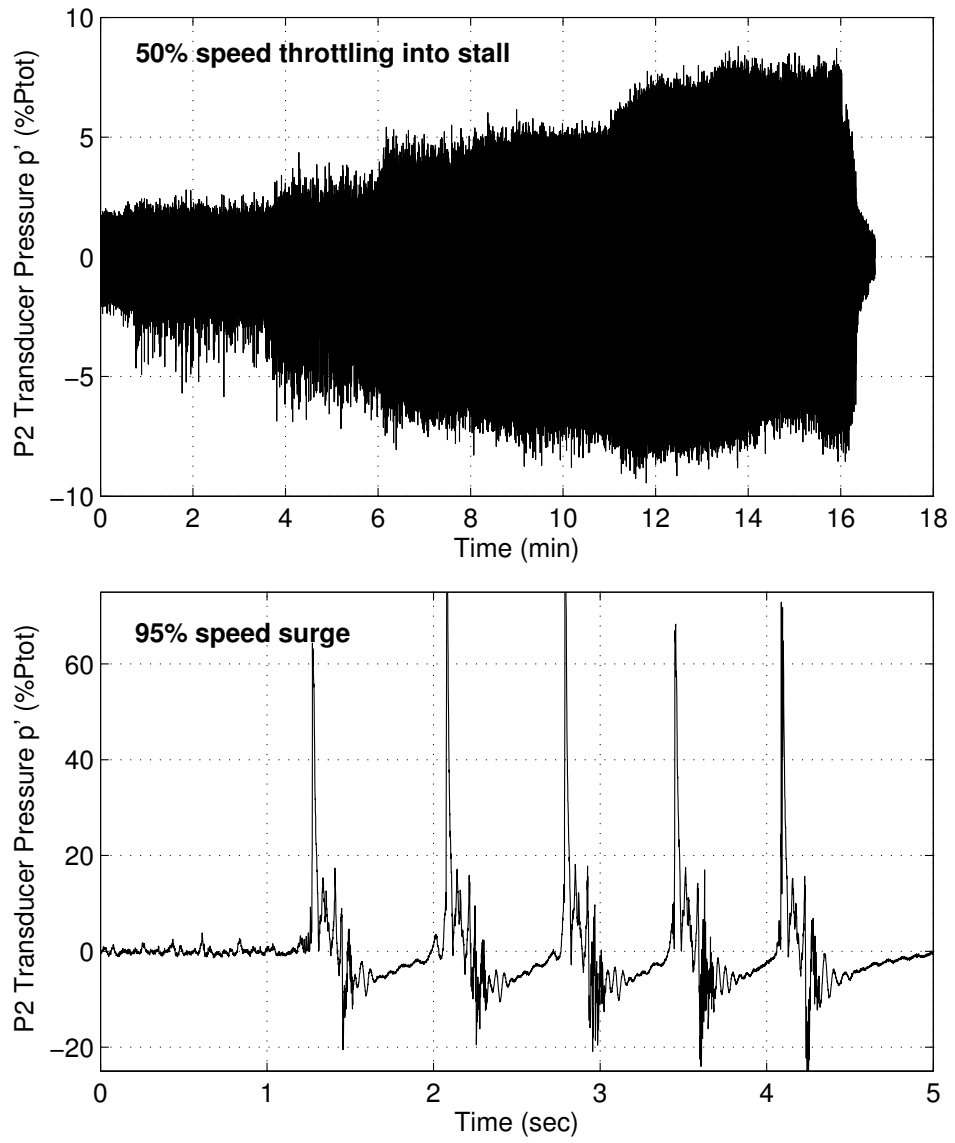


Figure 3.5: P2 (Rotor 2 leading edge) pressure transducer behaviour during 50% speed rotating stall (top) and 95% speed surge (bottom). VSV in optimal settings.

throttled and stabilised in steps along the flat part of the characteristic in Figure 3.2. A stable phenomenon forms and becomes progressively more intense as the throttle is closed further. The behaviour at 95% speed (also seen in Figure 3.5), by contrast, consists of 5 rapid pulses before the exit throttle was opened and the compressor allowed to recover, with the whole process complete within 5 seconds. In this case the speed is sufficiently high that surge occurs, again consistent with the findings of Greitzer [28] [29], whereby the air within the compressor exit volume discharges rapidly through the compressor and refills in a cyclic process which repeats until the exit throttle is opened sufficiently to allow the compressor to recover.

Whilst the time domain analysis of the type shown in Figure 3.5 provides useful information regarding the transient behaviour during a stall/surge event, further insight can be obtained through the Fourier analysis techniques introduced earlier. Focusing attention on the 50% speed event, a spectral analysis of the P2 (Rotor 2 inlet) and P8 (Stator 8 mid chord) transducers is shown in Figure 3.6. The processing techniques are identical to those discussed earlier, although the data is shown over a narrower frequency range (the bandwidth being one tenth of that used to analyse the acceleration manoeuvre in Figure 3.4). During 16 minutes of progressive throttle closure along the flat 50% speed characteristic in Figure 3.2, a series of approximately horizontal lines are seen to emerge and become stronger at both axial locations (front and rear) which, as highlighted, appear with a fundamental frequency of 0.44EO and a series of higher harmonics.

The behaviour in Figure 3.6 can be understood as follows. With stall cells rotating circumferentially, the static pressure field induced by the cells will rotate past the fixed pressure transducer location. This gives rise to a time signal of a particular waveform, with crests and troughs corresponding to the passing of successive stalled and unstalled flow regions. Figure 3.7 shows how this waveform actually appears for transducer locations across all 8 stages, during a time interval consisting of 30 shaft revolutions (close to the final point on the characteristic before the exit throttle was opened). A clear periodic signal is visible on all stages particularly from P2 (Rotor 2) rearwards, with the fundamental waveform having a period of approximately 2.3 shaft revolutions at all locations, consistent with the 0.44EO fundamental frequency in Figure 3.6.

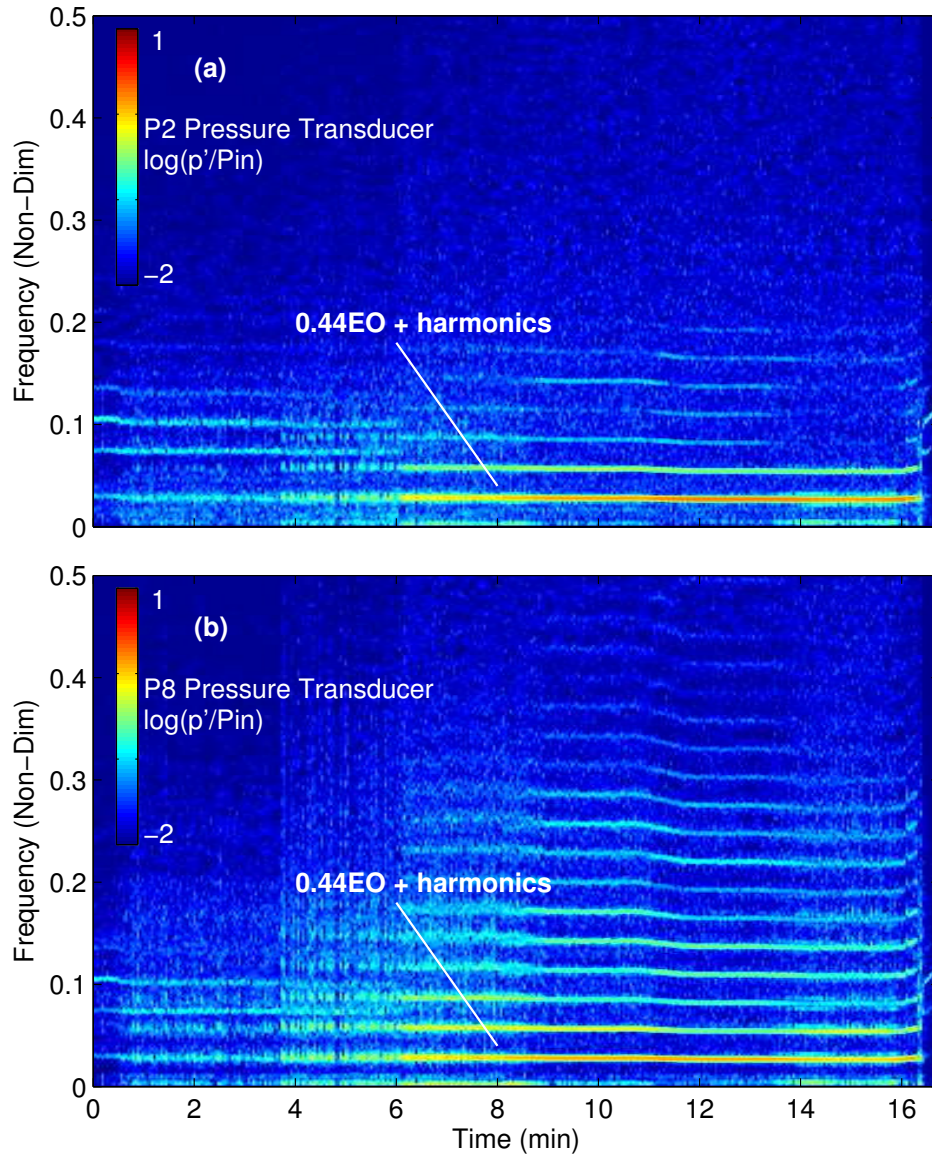


Figure 3.6: P2 (R2 inlet) and P8 (S8 mid chord) Pressure transducer spectral behaviour during progression into stall at 50% speed. Stall signal grows at 0.44EO with harmonics.

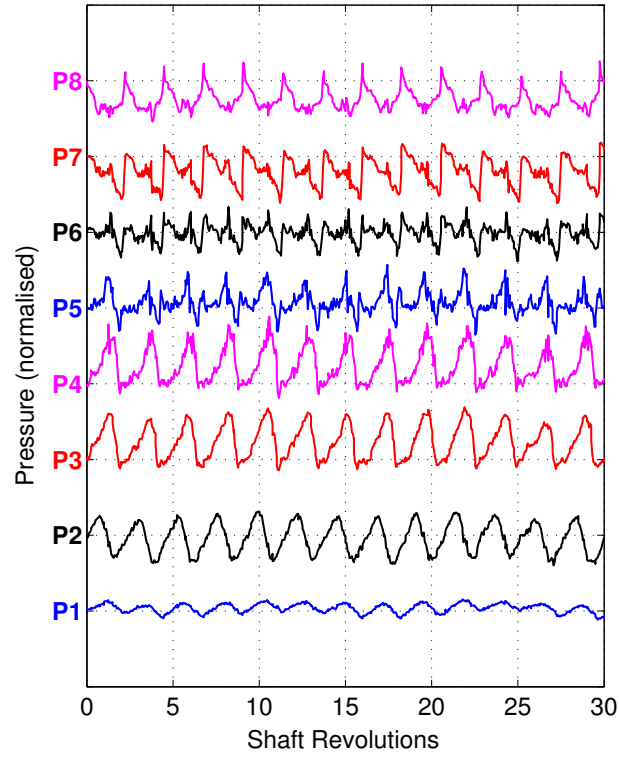


Figure 3.7: Time domain response of pressure transducers on all stages shown for 30 shaft revolutions during stall at 50% speed.

Interestingly, at location P2 (Rotor 2), the waveform more closely resembles a pure sinusoid than on the downstream stages, which are more complex with considerably stronger temporal pressure gradients. Considering the fundamental principles of Fourier decomposition, a pure sinusoidal waveform will consist of a single fundamental tone in the frequency domain. As the waveform becomes more complex, however, the corresponding spectrum will exhibit more and more high frequency harmonics. This can indeed be observed in the frequency domain analysis for P2 and P8 locations in Figure 3.6, where approximately 6 harmonics can be seen at location P2, whilst 15 can be seen at location P8.

Digressing briefly, it is of interest to consider why the waveform should be more complex (in the sense of more harmonics) in the middle and rear stages than it is in the front. As was discussed in chapter 2, at low non-dimensional speeds multistage compressors will become mismatched with the front stages moving towards stall. However, in a variable geometry compressor such as this one, the variable stator vanes (VSVs) are closed down by a considerable amount, to the extent that the front stages are offloaded and protected from stall. For this reason, it becomes the mid stages of the compressor, i.e. the front of fixed geometry part from stage 3 rearwards, that will often tend to be most susceptible to stall and thus the least stable at this speed. Such behaviour can be confirmed by breaking the overall characteristic at 50% speed (from Figure 3.2) into 3 separate blocks, as is shown in Figure 3.8, with blocks comprising of the variable stages, the mid stages and the rear stages.

It is clear from Figure 3.8 that the mid stages become unstable first as the compressor is throttled, followed by the rear stages also collapsing onto stalled characteristics. The front variable stages are more robust and remain on a negatively sloped characteristic. This is consistent with Figure 3.7, which shows the P4 (stage 4) location exhibits the highest amplitude activity and also the strongest pressure gradients, which in turn give rise to the presence of higher harmonics. Moving forward to the front stage (P1), which is “shielded” by the closed-down VSVs, it is suspected that the higher harmonics decay more rapidly, leaving only the lower frequency components visible.

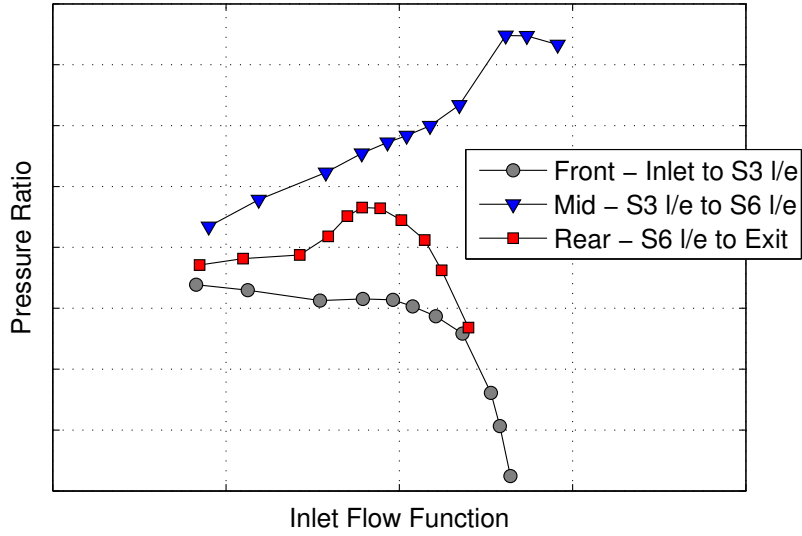


Figure 3.8: Block characteristics for compressor at 50% speed as compressor is throttled into full-span rotating stall.

Recalling the behaviour in Figure 3.6 and 3.7, it is therefore observed that as the compressor is throttled into rotating stall a signal with a fundamental $0.44EO$ frequency develops with a series of harmonics. Figure 3.9 shows the equivalent behaviour on the stage 1 and 8 rotor strain gauges, which are processed in an identical manner to the pressure transducers. Whilst the primary purpose of the strain gauges is to capture any high stresses resulting from forced vibration or flutter phenomena, analysis of their spectral behaviour alongside the pressure transducers provides further insight into the behaviour in stall. This is because any unsteadiness in the flow surrounding the blade (such as rotating stall) will still cause very light off-resonant vibration in the blade, which the strain gauge is still sufficiently sensitive to detect.

The behaviour in Figure 3.9 is more complex than on the pressure transducer, as several other features are visible in addition to the “stall” phenomenon which are discussed shortly. The activity of most interest, however, is the behaviour similar to that seen on the pressure transducer shown in Figure 3.6. In this case the corresponding stall signal consists of a fundamental frequency at $0.56EO$, noting this is $(1-0.44)EO$, again with a sequence of harmonics suggesting that the rotating stall phenomenon appears

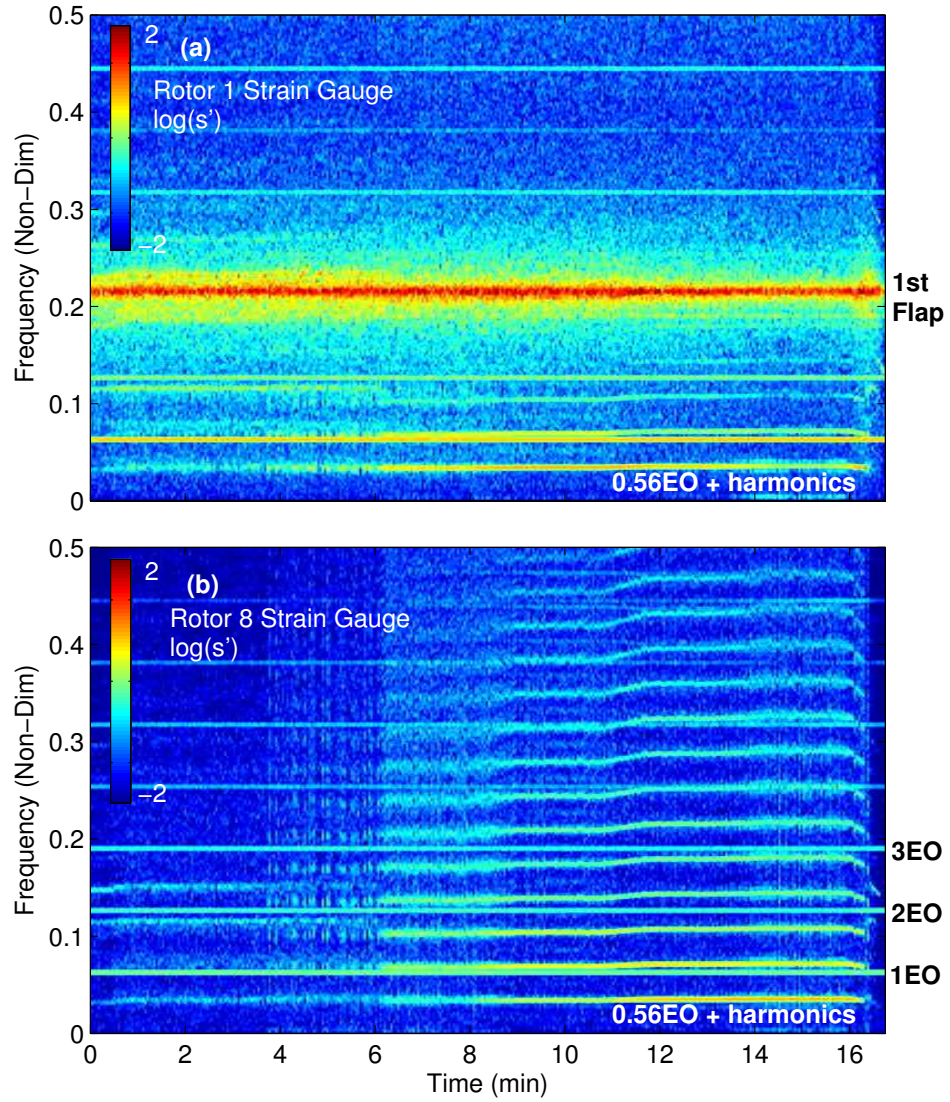


Figure 3.9: Rotor 1 and 8 strain gauge spectral behaviour during progression into stall at 50% speed. Rotating stall signal grows at 0.56EO with harmonics. Constant integer engine order tones also visible in addition to Rotor 1 1st Flap mode.

with similar spectral behaviour on both stationary and rotating instrumentation but at different frequencies, with the higher harmonics also more clearly visible on the rear stage.

In addition to the stall activity in Figure 3.9, a region of vibration activity is seen on the Rotor 1 strain gauge at a constant frequency. This is the first flap “Eigenmode” of the rotor blade which will always be visible at very small amplitudes (of the order 1MPa when away from resonance). Similar vibration mode activity cannot be seen on the Rotor 8 strain gauge because the blade is smaller and subsequently has a first flap frequency which is high enough to be out of the range plotted. Moreover, it can also be concluded from Figure 3.9 that this stall phenomenon is also unlikely to pose a concern from a blade vibration point of view, as the activity is at considerably lower frequency than the blade eigenmodes. Only when a stall phenomenon has a forcing frequency close to that of the vibration modes will resonance occur. The only excitation due to rotating stall in this case will come from the higher harmonics of the stall cell pattern which, as Figure 3.9 shows, become considerably weaker at higher frequencies and do not have sufficient energy to excite significant vibration.

Activity can also be seen in Figure 3.9 at constant integer engine orders at both locations (highlighted “1EO” to “3EO”). This results from general asymmetry in the compressor, such as manufacturing/build variations and rotordynamic effects. For example, if the stator vanes are not uniform around the circumference (particularly likely to be true for the VSV stages) then the local flowfield will show a more complex circumferential variation than simply the stator wakes themselves. In terms of Fourier analysis (in this case spatial rather than temporal), this waveform can be decomposed into a series of spatial modes (or nodal diameters), with only whole wavenumbers possible around the circumference - as non-integer wavenumbers would destructively interfere. This means that any non-rotating asymmetry will naturally appear with a 1 lobed component, a 2 lobed component and so on, which in turn will manifest itself on the rotating instrumentation at 1EO, 2EO respectively, consistent with Figure 3.9.

3.1.4 Rotating stall diagnosis in two frames of reference.

An observation which will be exploited in the subsequent chapters concerns the different frequency at which stall occurs on the stationary pressure transducer and the rotating strain gauge. The rotating stall examples presented here can be considered a subset of a wider range of rotating disturbances, for example a similar shift in frequency arises due to other phenomenon such as acoustic resonance or flutter. Kurkov[45] and Mengle[46] both show how flutter of a rotating blade assembly in a given mode is also visible on a stationary pressure transducer spectrum, due to the static pressure perturbation induced by the vibrating blade. This is offset from the vibration frequency by an integer number of engine orders equal to the nodal-diameter pattern in which the vibration occurs (i.e. the interblade phase difference).

In the 50% speed rotating stall example in Figure 3.6 and Figure 3.9 the fundamental stall frequencies occurred at 0.44EO and 0.56EO respectively. This difference is because the rotating stall phenomenon is being measured/observed in two different frames of reference. For example, consider a pattern of m stall cells propagating around the circumference at a speed V_{stall} , which (typical of rotating stall) is a fraction of rotor blade speed U but in the same direction. It follows that stationary and rotating instrumentation will detect this spinning pattern as a waveform at a different fundamental frequencies given by Equations 3.1 and 3.2 respectively, where in this case $n_{stat} = 0.44$ and $n_{rot} = 0.56$.

$$n_{stat} = \frac{mV_{stall}}{U} \quad (3.1)$$

$$n_{rot} = m(1 - \frac{V_{stall}}{U}) \quad (3.2)$$

Simple rearrangement leads to Equation 3.3 and Equation 3.4 which gives the number of lobes and the propagation speed in terms of the two measurements.

$$m = n_{stat} + n_{rot} \quad (3.3)$$

$$\frac{V_{stall}}{U} = \frac{n_{stat}}{n_{stat} + n_{rot}} \quad (3.4)$$

Therefore, provided reliable measurements of rotating stall can be obtained in both static and rotating frames of reference (capable of resolving the fundamental frequency), the number of cells is then simply the summation of the non-integer rotating and static engine orders. This technique therefore confirms that this 50% speed stall example (with the VSVs in their nominal settings) must consist of a single cell passing through the entire length of the compressor. From Equation 3.4 it can also be established that the single cell is rotating at 44% of shaft speed. Whilst this example is relatively simple, the technique is used extensively in subsequent chapters to investigate stall with multiple cells and gives reliable results (i.e. the number of cells m being integer to within 2 decimal places) even with over 30 stall cells.

It is also important to consider the possibility of other phenomena in which spinning modes rotate in the opposite sense ($V_{stall}/U < 0$) to the shaft, or faster than the shaft ($V_{stall}/U > 1$). Acoustic resonance and blade flutter are practical examples in which this may occur. In this situation it is straightforward to show that Equation 3.3 must be modified such that the circumferential mode order becomes the difference between n_{stat} and n_{rot} . This situation does not arise in this thesis, with the exception of a special case presented in chapter 5, where a rotating stall pattern and the blade pressure field (each of different wavelengths) interact to create a third waveform which *does* travel faster than the shaft speed.

It should be noted that this experimental technique is of limited use for investigating rapid transients such as spike stall inception, where very short lengthscale disturbances initially appear in a localised circumferential sector of one stage and rapidly grow into mature stall cells. For such analysis the traditional techniques [12] consider the phase difference between pressure transducers distributed around the circumference. This method has not been utilised here as the circumferential coverage of pressure transducers is not sufficient. Rather, this thesis focuses on investigation of stable, fully developed rotating stall in both rotating and static frames of reference making extensive use of the techniques outlined here. It also should be emphasized that fluid measurements in this experiment are limited to static pressure perturbations. In reality a phenomenon such as rotating stall is likely to also have significant total pressure variation.

3.2 Numerical methods - low fidelity

The experiment presented in the subsequent chapters of this thesis makes use of the 3 rows of variable stator vanes in this compressor as a method for deliberately stalling the front stages. This allows the subsequent physical behaviour to be studied whilst the compressor as a whole system remains stable on its working line. However, it is not immediately clear which blade rows will be most susceptible to stall from consideration of the VSV setting angles alone. More specifically, the IGV/Stator 1/Stator 2 settings will be opened and closed by varying amounts relative to a chosen baseline configuration, in multiple permutations - each of these which will have a differing aerodynamic impact. Consideration of the geometric vane angles alone gives little insight into the physical implications for the flow. For this reason a number of supporting numerical analyses were performed consisting of rapid 1D meanline predictions and 2D throughflow/streamline curvature models (3D RANS/URANS simulations are covered in the next section). These lower fidelity methods were developed by Rolls-Royce although similar methods are well established in the open literature, as will be discussed below.

3.2.1 One-dimensional meanline method

As discussed above, the experiment presented in this thesis involved testing with a range of different VSV settings to provoke rotating stall locally in the compressor front stages, by operating the compressor along its working line with the downstream stages maintaining stable compressor operation overall. A 1D meanline model was therefore used to allow these VSV settings to be systematically compared and “ranked” relative to each other in terms of their aerodynamic impact on each blade row of the compressor.

In this model, the geometric boundary conditions consist of the annulus, i.e. the area and meanline radius at each axial location throughout the machine, together with the meanline blading information (inlet/outlet angles, space-chord ratios, thickness-chord ratios etc). This information, together with the operating boundary conditions of aerodynamic speed, inlet pressure and inlet temperature, form the model inputs. The model then solves iteratively the continuity and Euler work equations successively through the compressor using a stage stacking technique. For each blade row the total pressure loss, deviation and blockage growth (i.e. the viscous effects) are represented by empirical correlations of the type already documented in the open literature [47],[48],[49],[50].

Using this method, the model can be run at any point across the operating range in terms of flow, speed, pressure ratio and also any VSV settings (at least within the limits of the correlations). The primary use of the model is to assess the susceptibility of the front stages to stall, whilst still on the working line, and importantly how adjustment of the VSVs redistributes loading between blade rows. Once a solution is obtained from the 1D model for each point on the operating map, in a matter of seconds, the predicted velocity triangles are then defined throughout the compressor. This allows the susceptibility of each row to stalling to be assessed across a range of VSV settings using well established criteria[8]. The overall and stage characteristics predicted by the 1D model are also compared to experimental measurements in chapter 6.

3.2.2 Two-dimensional throughflow method

Whilst the 1D meanline method is effective in rapidly assessing different VSV schedules and understanding which blade rows will be most susceptible to stall, it quite clearly does not take into account any spanwise redistribution of flow that occurs naturally between the hub and casing due to the constraint of radial equilibrium. From a purely inviscid point of view, this 2D behaviour becomes important in this compressor and is explored in detail in chapter 6.

To address this, a 2D streamline curvature (throughflow) method (of the type similar to that described by Denton[51]) is used to model the compressor inlet duct and front stage to aid understanding. For modelling simplicity the downstream stages are removed as the fundamental behaviour, due to radial equilibrium effects arising from closing the IGV at part-speed, can be adequately captured with only the front stage. In this model the radial equilibrium equation is solved along 21 axisymmetric streamlines between the hub and casing. An example of the 2D domain and solution, in terms of streamlines (or lines of fixed cumulative massflow) is shown in Figure 3.10.

The model does not account for any variations around the circumference, such as the presence of blades and vanes, with the relative flow angles at the blade trailing edge locations provided as model inputs based on the blade outlet angle, with some user assumed level of deviation. Similarly, the total pressure loss at the exit of each blade row and the axial distribution of blockage are specified as model boundary conditions. This effectively means that, for the purposes of understanding, the *viscous* effects of loss, deviation and blockage can be held constant, whilst changes in the compressor speed, flow coefficient and VSV angle settings can be explored in terms of their *inviscid* radial equilibrium effect. Chapter 6 shows that this behaviour is particularly powerful, as closure of the IGV at part-speed imparts considerable swirl into the compressor which, as the 2D code is used to show, is accompanied by significant spanwise gradients of both static pressure and axial velocity. This has implications for how the blade rows stall in a non uniform manner across their span.

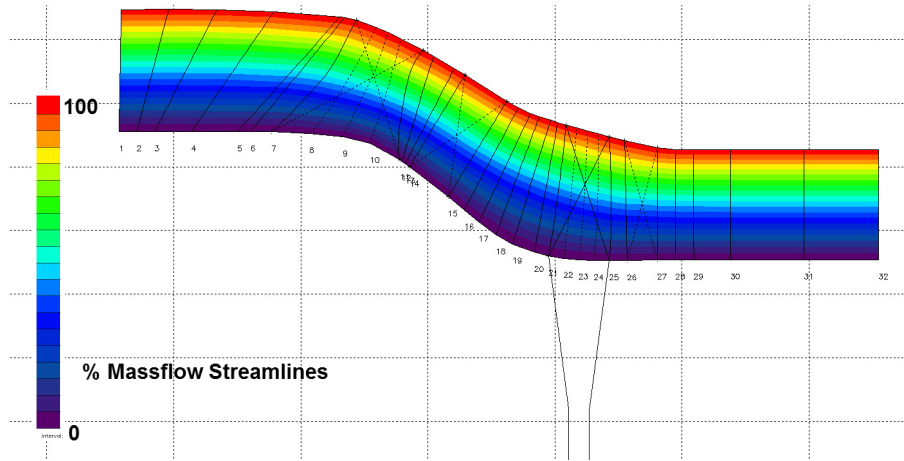


Figure 3.10: 2D Streamline Curvature model for compressor front stage - predicted streamlines.

3.3 Numerical methods - CFD

A significant objective of this thesis is to understand the capabilities of URANS CFD methods for simulation of rotating stall, by performing full annulus unsteady computations at stalled conditions identified during the experiment. This firstly aims to establish whether credible rotating stall predictions can be obtained (potentially to aid the engineer during design or testing phases of compressor development) and secondly to offer deeper insights into the physical behaviour in stall than can be obtained through experimentation alone. It is emphasised here that investigation of detailed changes to the code, for example turbulence modelling parameters or numerical schemes, is beyond the scope of this thesis and the same settings were used throughout this investigation.

3.3.1 Numerical code setup

The 3D CFD analyses presented in this thesis were performed using computational fluid dynamics (CFD) techniques that are standard across both the industrial and academic communities, utilising a code that has been developed at Imperial College over 20 years. This code has been developed extensively for simulation of both pure aerodynamic and aeroelastic phe-

nomena, the latter using a mesh motion technique to represent vibration of the blades. Throughout this study the calculations are purely aerodynamic in nature and no prescribed vibration is applied to the blade. A significant number of publications have utilised this method which describe the numerical techniques in detail[52]. More recently this method has been used to demonstrate credible simulations of rotating stall on both high-speed compressors [39] and fans [40], in the latter case showing results consistent with experimental data [41].

The CFD computations were performed using an implicit, time-accurate 3D compressible Reynolds-averaged Navier-Stokes (RANS/URANS) solver which is based on a cell-vertex, finite volume technique. The solver uses the well established RANS/URANS based turbulence model of Spalart and Allmaras [53], for which the parameters have been fine tuned to give reliable predictions for turbomachinery applications over a number of years and are held fixed throughout this investigation. The solver utilises a wall function and the near-wall first cell height is chosen to achieve a Y^+ value between 20 and 100.

In the modelling approach used in this thesis, the calculation mesh is represented using an edge-based data structure and is presented to the solver as a set of node pairs connected by edges. A pre-processing stage, prior to the main calculation, is used to compute edge weights at the inter-cell boundaries allowing the solver to have a unified data structure for which the nature of the hybrid mesh is concealed from the main calculation loops. An implicit second-order time integration is used to advance the system of equations in time, which for steady flows uses a point relaxation procedure consisting of Jacobi iterations. Techniques such as residual smoothing and local time stepping are also employed to accelerate the solution. For the unsteady computations presented in chapter 6 a dual time stepping procedure is used with external Newton iterations in addition to the internal Jacobi iterations. Further information regarding the solver is given by Sayma[52].



Figure 3.11: Domain for steady single passage mixing plane analysis.

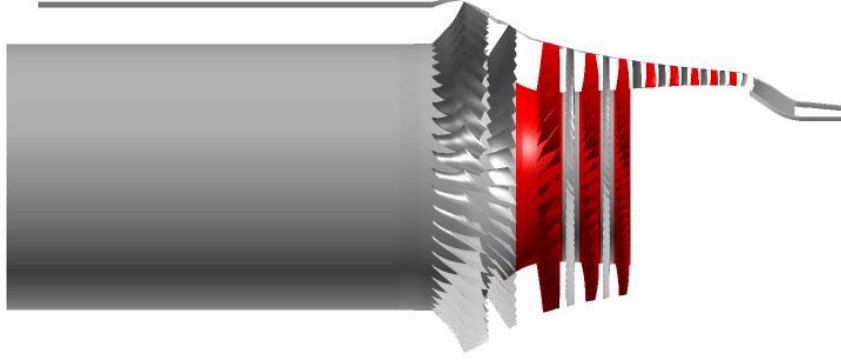


Figure 3.12: Domain for unsteady full annulus sliding plane analysis (note blade rows shown in red indicate rotors).

3.3.2 Calculation strategy

Initially, steady state computations were performed for the whole compressor domain shown in Figure 3.11. The steady model assumes periodic flow between adjacent passages and mixing planes are used to circumferentially mix out the flow quantities at the inlet and exit boundaries of each blade row, in a similar manner to that described by Denton[54]. A fully gridded exit throttle was placed downstream of the domain and its physical area adjusted to position the compressor on the correct experimental working line and map out constant speed characteristics. Using this modelling approach, ambient atmospheric boundary conditions can be used at inlet and exit as the exit throttle will operate choked.

The corresponding unsteady domain shown in Figure 3.12 was generated by expanding the single passage mesh and solution circumferentially to cover the full annulus across the front 7 blade rows (to Rotor 3 exit) with sliding plane interfaces[52] between blade rows. Downstream of this location

the domain transitions back to a single passage mixing plane (i.e. steady) computation for the rear stages, by making the modelling assumption that the unsteady rotating stall behaviour is confined to the region upstream of Rotor 3. This assumption is validated firstly using experimental data in chapter 5 and is later confirmed by additional calculations in which the transition to mixing planes is moved further downstream.

The geometry was meshed using techniques described by Sbardella[55] and comprises of an unstructured grid within the blade to blade plane, with a structured grid in the radial direction consisting of 45 radial mesh points. A typical blade to blade mesh is shown in Figure 3.13, on a mid-span surface through Rotor 1 and Stator 1. Triangular prismatic elements were used away from the blade surface whilst hexahedral cells were used in the region around the blade itself to form an O-Mesh. In total the computational domain shown in Figure 3.12 consisted of approximately 70 million nodes, with typically 0.3 million nodes per passage. A steady single passage mesh refinement study is presented in chapter 6, demonstrating that the simulation results are insensitive to an increase in mesh density by a factor of two in the bladerows of interest. The mesh density is assumed to also remain adequate for the unsteady calculation, providing the spatial lengthscale of any stall cells forming in the simulation remains sufficiently large, i.e. greater than one blade passage.

The clearances between the rotor blades and the casing were set at a level consistent with the test rig, with 5 radial mesh points across the clearance gap. Fillets at the blade ends and leakage flows through the shroud cavities beneath the stator vanes were not included. However, steady analyses with these features included showed their relative importance to be vastly outweighed by the effect of the variable stator vane settings, in determining the susceptibility to stall.

A temporal discretization level of 1200 time steps per shaft revolution was used, with this choice being based upon observations from the experiment, in terms of the typical frequencies (or engine orders) at which rotating stall was observed, thus ensuring sufficient temporal resolution in the model to resolve the measured behaviour. This is explored further in chapter 7 which presents a timestep sensitivity study.

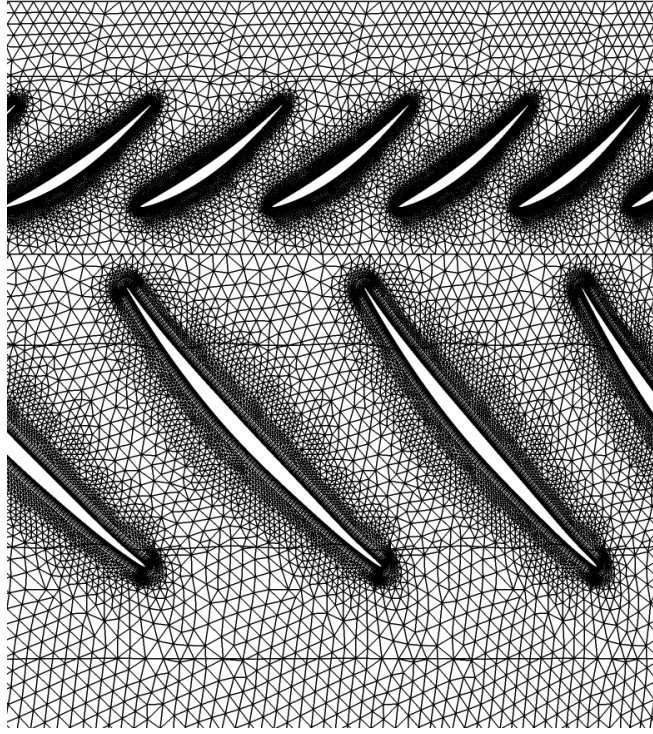


Figure 3.13: Computational mesh shown on mid-span surface through Rotor 1 (bottom) and Stator 1 (top).

With the computational domain shown in Figure 3.12 and timestepping discussed earlier the resulting simulations progressed at a rate of approximately 2 days per shaft revolution using 64 CPUs (Intel® Xeon® CPU E5-26900, 2.90GHz utilising an Infiniband 40Gb/s interconnect). The chapters that follow show that using this modelling strategy, rotating stall can be captured and stabilised by running the model for 10-15 shaft revolutions from a suitably chosen steady solution as a starting point. The maximum runtime was therefore approximately 1 month, although further adjustment of the VSVs from this stabilised unsteady condition allowed additional stalled conditions to be achieved more quickly.

4 Experimental study 1: VSV schedule modifications

Before this thesis progresses to explore detailed unsteady phenomena seen in the test compressor, this chapter shows how the variable stators provide a valuable experimental opportunity to deliberately mismatch the compressor in a flexible and controlled manner. This allows stall to be provoked in the front stages whilst the compressor as a whole remains stable, providing an ideal environment to study the nature of rotating stall and explore also how it behaves as the compressor operating condition is changed. An experiment is then presented in which a number of different VSV schedules were tested, by adjusting the stagger of each variable stator row by different amounts and in various combinations. For each case the measured unsteady flow behaviour is presented, making use of the 1D aerodynamic model to diagnose which stages of the compressor are stalling.

4.1 Triggering rotating stall using the VSV schedule.

In this thesis the variable stagger vanes are used to deliberately mismatch the compressor and trigger stall in the front stages, allowing its behaviour to be studied. This can be easily achieved by forcing the VSVs to follow schedules which deviate from the idealised case presented in Figure 2.11. The next section therefore proceeds to explore how “over closure” of Stator 2 at part-speed may be used to provoke stall in the front stages.

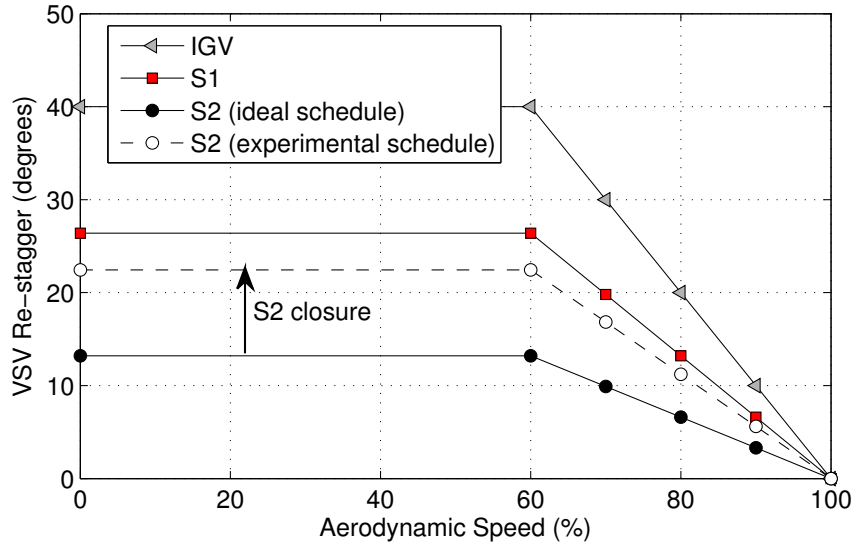


Figure 4.1: Example of an “experimental” VSV schedule in which S2 is deliberately closed more at part-speed, to trigger rotating stall in the front stages.

Chapter 2 showed, in Figure 2.11, the “idealised” VSV schedule, in which the vanes close progressively from their high-speed settings, with IGV closure of 40° typical at low-speeds[6]. The corresponding movement of Stators 1 and 2 are progressively less, by approximately two thirds and one third of the IGV re-stagger respectively. A VSV schedule configured this way is likely to offer the optimal efficiency, as aerodynamic loading and incidence angle changes will be evenly distributed between the front stages.

Consider, therefore, the effect of changing the VSV schedule as shown in Figure 4.1. Compared to the idealised VSV schedule, Stator 2 has been closed down further as the compressor speed is reduced from its design point. At any given operating point, the compressor exit flow function can be assumed to be fixed by the downstream component. For the stand-alone compressor rig considered here this would be the exit nozzle, which remains choked and determines the flow function the compressor is required to deliver at its exit. As Stator 2 is closed down, this requirement leads to a process of axial rematching within the compressor, which has the effect of pushing the blade rows upstream of Stator 2 towards stall.

This effect occurs because the closure of Stator 2 causes a reduction in work and therefore pressure ratio delivered by the downstream part of the compressor, with this reduction occurring primarily on Rotor 3. As the exit flow function must remain constant, this causes the flow function at entry to Stator 2 to fall (this follows directly from application of Equation 2.5 to the compressor stages from Stator 2 rearwards all grouped together). Consider now the effect on the front stages. In response to this fall in downstream flow capacity, which is analogous to a throttle of reducing flow area, the front stages (upstream of Stator 2) must move up their characteristics to compress the flow more into this smaller area. Stated in an alternative way, the closure of Stator 2 will cause a reduction in the static pressure rise delivered by the downstream part of the compressor. Assuming that the compressor delivers to a plenum at a constant static pressure, this will result in the static pressure rising at inlet to Stator 2, which again implies that the front stages must move up their characteristics towards stall. It follows that with sufficient Stator 2 over-closure, the front stages of the compressor can be pushed into stall.

This behaviour can be seen clearly by running the 1D model on two different VSV schedules, to give the characteristics shown in Figure 4.2, from 70 to 90% speed. The characteristics for a “block” containing the IGV to Rotor 2 (i.e. rows upstream of Stator 2 leading edge) are shown, together with another block for Stator 2/Rotor 3, and finally an additional block composed of all blade rows from Stator 3 leading edge rearwards. The main observation here is that the deliberate over-closure of Stator 2 causes a pronounced fall in flow at inlet to the Stator 2/Rotor 3 block, as the speed is reduced. Consequently, the upstream rows simply rise up their characteristics and will eventually reach their local stall boundary, despite the compressor remaining stable overall. Through such a process, sufficient closure of Stator 2 can trigger rotating stall locally in the front stages of the compressor even when the compressor overall remains at stable working line conditions, as the rear stages remain on stable characteristics and prevent the compressor from surging.

It is also important to note that as Stator 2 is closed down, the Stator 2/Rotor 3 block also moves progressively onto a steeper, more vertical characteristic. This implies that Stator 2/Rotor 3 is progressively offloaded and

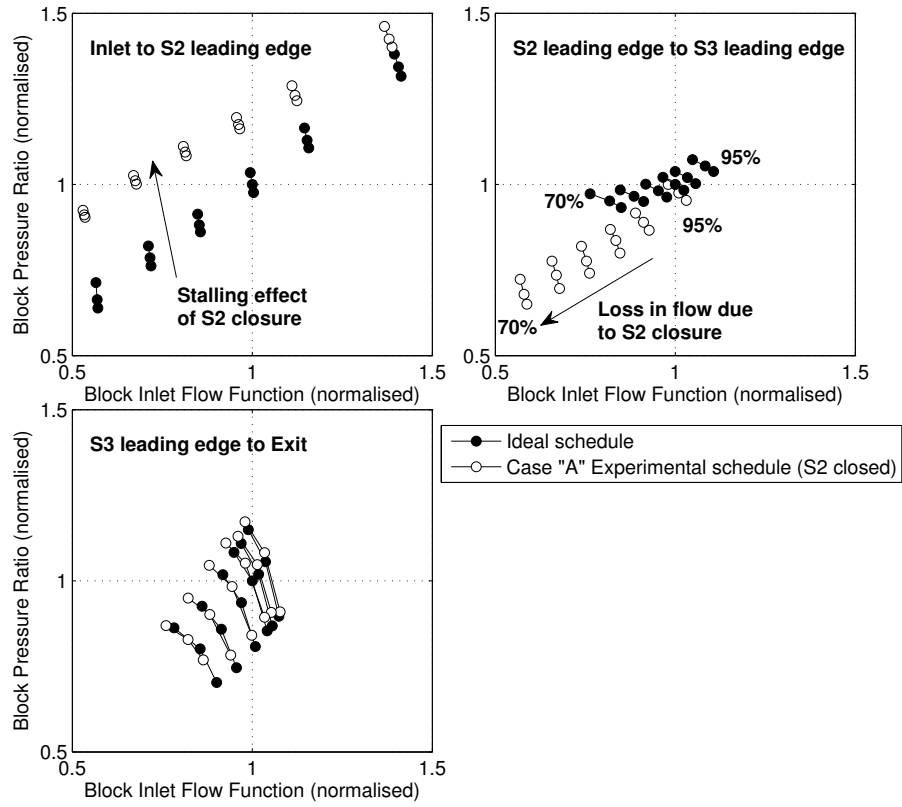


Figure 4.2: 1D meanline model - predicted operating maps for front stages, Stator 2/Rotor 3 and rear stages. Shown on idealised VSV schedule and experimental schedule with S2 overclosure.

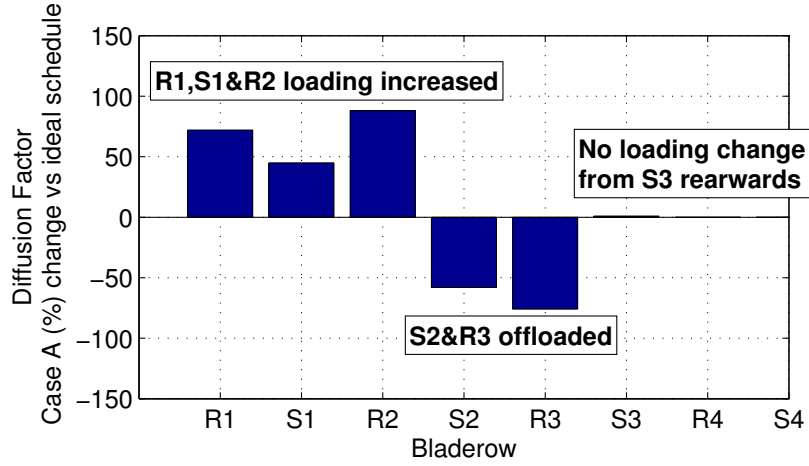


Figure 4.3: 1D meanline model - predicted diffusion factor % change for Rotor 1 to Stator 4 at 85% speed, moving from idealised VSV schedule to Stator 2 closed VSV schedule.

driven towards choke, whilst the upstream rows are simultaneously driven towards stall. The fixed geometry rows from Stator 3 rearwards remain unaffected (at least within the context of this simplified 1 dimensional analysis).

It follows that for this investigation, VSV schedules with Stator 2 deliberately closed down more, as the speed drops from the design point, will tend to be the most active in terms of front stage rotating stall behaviour. Therefore, such a case was chosen for this experiment, noting that this is clearly not an optimal VSV schedule from an overall performance point of view but is ideal for allowing stall behaviour to be explored. This experimental VSV schedule is referred to herein as Case “A”.

Using the 1D meanline model, which calculates the velocity triangles across the whole operating map of the compressor, it is also straightforward to derive the loading levels on each blade row. For example, Figure 4.3 shows how the Lieblein Diffusion Factor[8] changes between the idealised and Case A VSV schedules, the latter having Stator 2 overclosed as discussed above. The calculations for both schedules were performed at a fixed point on the operating map which corresponds to 85% speed on the Case A VSV schedule. Figure 4.3 shows how the trends discussed above, as

Stator 2 is closed down, apply at an individual blade row level. By closing Stator 2, both Stator 2 and Rotor 3 move to reduced loading whilst the upstream rows consisting of Rotor 1, Stator 1 and Rotor 2 move to higher loading and become susceptible to stall. In the next section, this method is extended to explore the behaviour as all 3 VSV rows are perturbed relative to Case A and this loading is “redistributed” between different blade rows.

4.2 Description of experiment

In the remainder of this chapter, data is presented for acceleration manoeuvres from 50% to 105% of design speed, unless otherwise stated. During each of these accelerations, the variable stator vanes (VSVs) were all progressively opened (turning towards the axial direction) in a smooth manner, similar to the typical VSV schedule shown in Figure 4.1. These accelerations were then repeated, with deliberate fixed offsets applied to the stagger of the VSVs, in addition to their normal movement across the speed range. During each of these manoeuvres the data from the casing mounted pressure transducers and rotor mounted strain gauges was recorded. It should also be stated that the acceleration rate was sufficiently slow to be considered quasi-steady at each operating point, with each 50-105% manoeuvre completed in approximately 4 minutes. All manoeuvres were performed with the exit throttle set to match a fixed compressor working line shown in the chapter 3 in Figure 3.2, and were repeated to ensure consistency of results.

In total 26 different VSV configurations were tested, covering several permutations of IGV/Stator 1/Stator 2 opening and/or closing relative to Case A. A sample of 8 configurations was chosen for more detailed analysis, as shown in Table 4.1. It is the relative effect of each of these configurations that is of prime interest, which the next section seeks to quantify from a 1D aerodynamic point of view.

Configuration	IGV	S1	S2	Predicted effect (relative to datum)
A	0	0	0	Datum
B	-1.9	+0.8	-2.0	S1&R2 offloaded
C	+1.9	-0.8	+2.0	Mild S1&R2 stall
D	+4.5	-3.5	+4.5	Severe S1&R2 stall
E	+1.9	-0.8	-0.9	R1 offloaded
F	-1.1	+1.8	+2.0	Mild R1 stall
G	-4.5	+3.5	+3.5	Severe R1 stall
H	-4.5	-4.5	+3.5	Severe stall R1,S1,R2 Combined

Table 4.1: Selection of VSV angle offsets (degrees) applied relative to Datum schedule “Case A”.

4.3 One-dimensional analysis - assessment of VSV schedule adjustment.

The previous section of this chapter showed how closure of Stator 2 causes the compressor to rematch and push all blade rows upstream towards stall, whilst unloading Rotor 3. It follows that Rotor 1, Stator 1 and Rotor 2 are most susceptible to stalling in this experiment. Adjustment of the IGV and Stator 1 will redistribute the additional loading between these blade rows. Following the same logic presented earlier, for a given rotor the loading will be increased by closure of the downstream VSVs combined with opening of the upstream VSV. For example, the *worst* VSV setting for Rotor 1 will always occur when the IGV is opened, combined with closure of Stators 1 and 2.

In order to systematically rank the effect of each configuration, in terms of the susceptibility of each of these blade rows to stall, the 1D model was again utilised to derive diffusion factor changes similar to those shown earlier in Figure 4.3. Models were generated for each of the 26 different VSV configurations at a fixed operating point, corresponding to the inlet flow and pressure ratio matching the Case “A” operating point at 85% speed (this condition was chosen as typical of where rotating stall is observed, as will be presented in the subsequent analysis). Each configuration was then ranked using the well-known Lieblein Diffusion Factor [8] to determine the subsequent aerodynamic loading effect upon Rotor 1, Stator 1 and Rotor 2.

The results of this process are shown in Figure 4.4(a), which provides a “map” of the Rotor 1 versus Rotor 2 diffusion factor changes on each VSV configuration relative to Case A. The 8 settings listed in Table 4.1 are shown in red with the remaining 18 points shown in grey. It can be seen in Figure 4.4(a) that the sequence of configurations (B, A, C, D) will push Rotor 2 towards stall with minimal effect on Rotor 1. This occurs when the IGV is closed, Stator 1 is opened and Stator 2 is closed (relative to Case A), as was shown in Table 4.1. A corresponding group of configurations (E, A, F, G) can also be identified in Figure 4.4(a) which will push Rotor 1 towards stall with minimal effect on the Rotor 2 loading, in this case with the IGV opened combined with Stators 1 and 2 closed.

Figure 4.4(b) also shows the corresponding predictions of the Stator 1 versus Rotor 2 loading, the good correlation implying that for all configurations tested, Stator 1 and Rotor 2 are expected to approach stall as a pair. The question of how this physically occurs (in terms of which row stalls first) is studied in the later chapters.

A further configuration referred to as Case “H” can be seen in Figure 4.4 which simultaneously raises the loading of all three bladerows together towards stall. This is achieved by opening the IGV and Stator 1 together in combination with closure of Stator 2 relative to Case A and will also be investigated later in this chapter.

It should be emphasised at this stage that the 1D model is simply used to ‘rank’ these VSV schedule changes and identify trends. In reality the subsequent chapters will show how the actual flow in the compressor front stages exhibits significant spanwise variations with the Rotor 1 tip liable to stall at high incidence, whilst the Stator 1/Rotor 2 hub region is also susceptible to separation due to excessive diffusion in the Stator 1 hub. The use of a 1D analysis is therefore clearly of limited scope and cannot be used to offer fundamental insights into the fluid dynamics of part-span stall or provide a physically valid criteria for stall onset. However, it will be demonstrated in the following sections that, for estimating how changes to the VSV schedule are likely to distribute loading between each blade row, particularly when used in combination with observed unsteady behaviour, the model is still of benefit.

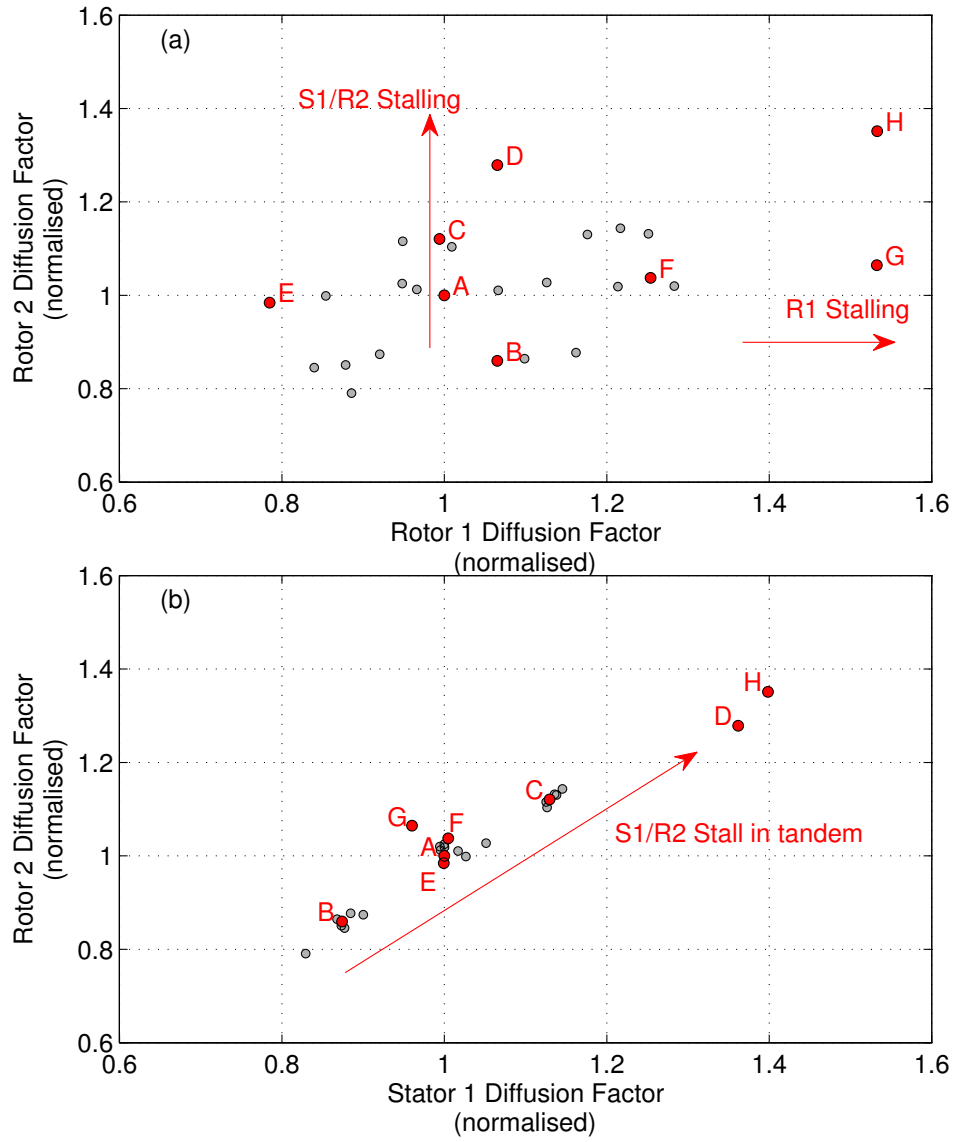


Figure 4.4: 1D predicted diffusion factor for R1, S1 and R2 (normalised relative to Case 'A').

Having used the 1D model to identify these trends - in terms of grouping together different configuration which isolate Rotor 1 stall (E,A,F,G) and Stator 1/Rotor 2 stall (B,A,C,D), attention now turns to the observed unsteady behaviour observed on the pressure transducers, progressing through each of these sequences.

4.4 Effect of Stator 1/Rotor 2 stalling

The progression through Cases B, A, C and D was predicted, in Figure 4.4, to push the Stator 1/Rotor 2 region towards stall whilst having minimal effect on Rotor 1. This occurs when the IGV, Stator 1 and Stator 2 are closed, opened and closed respectively relative to the datum schedule.

As discussed in the previous section, the experiment consisted of shaft acceleration at a constant rate over approximately 4 minutes, from 50% to 105% shaft speed on each VSV configuration. The pressure transducer and strain gauge time signals were recorded for further study. The most interesting behavior occurred in the 70-95% speed region hence attention will focus here, although the worst VSV configuration, Case D, was terminated at 91% speed.

Figure 4.5 shows spectral data from the ‘P2’ transducer (at Stator 1 exit) for each of these 4 configurations, using the frequency domain analysis methods presented in chapter 3. For discussion purposes, 5EO lines have been added manually to Figure 4.5. For reasons that become apparent over the subsequent analysis, the 5EO line turns out to be a useful reference feature by which to differentiate different phenomena, though it should be emphasised that the 5EO value does not carry any special physical significance. On the right hand side of each case, the peak amplitudes observed through the manoeuvre are also shown.

The main feature observed in Figure 4.5, other than Rotor 1 blade-passing (other BPF tones are out of the range shown), is unsteadiness growing within the frequency range below 5EO, as highlighted. This is not visible at all for the unstalled Case B and reaches amplitudes of over 5% of inlet total pressure for the most stalled Case D. This implies that the deliberate

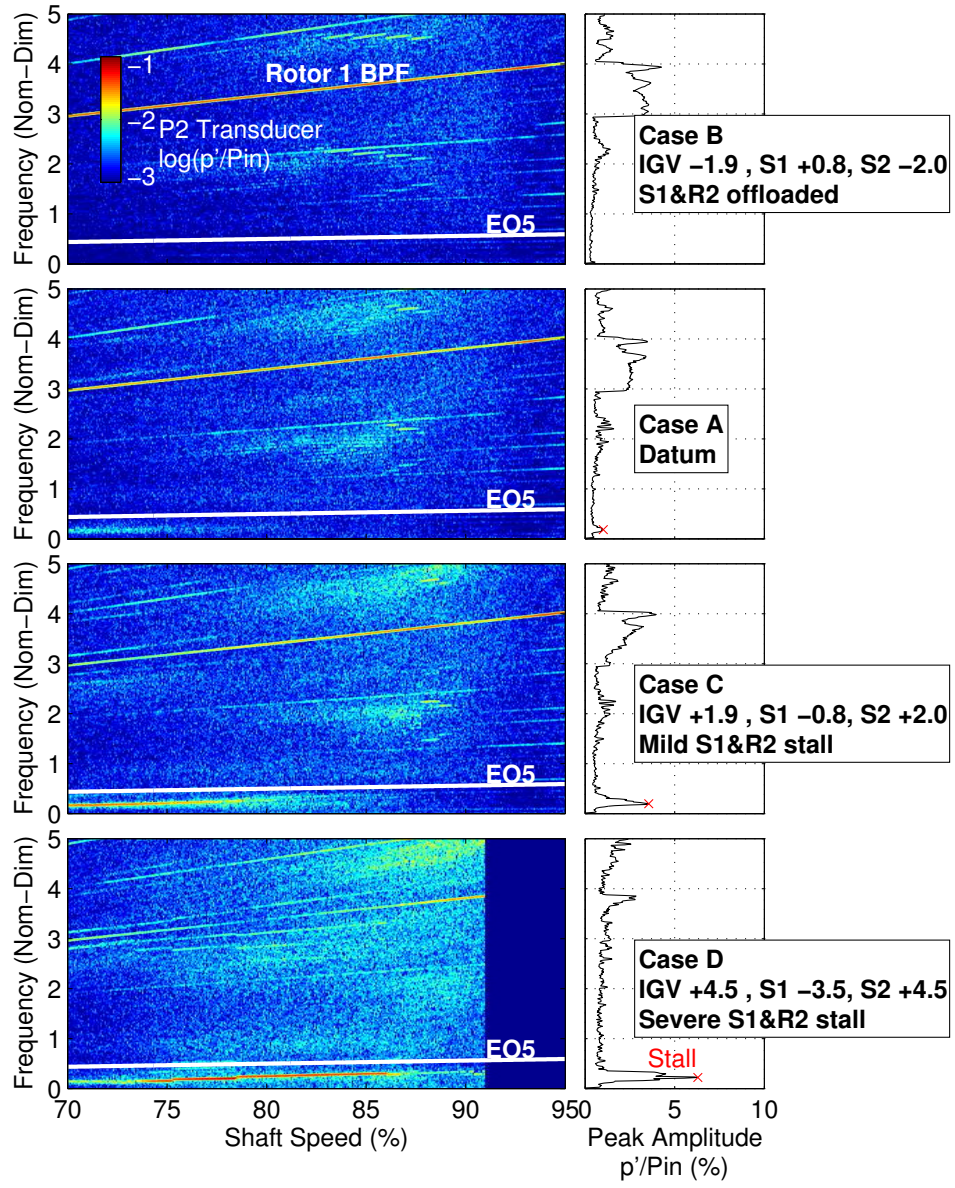


Figure 4.5: Spectral analysis of transducer at location P2 for Case B, A, C and D VSV settings, as S1/R2 loading is increased. Stall activity is observed in 0-5EO range.

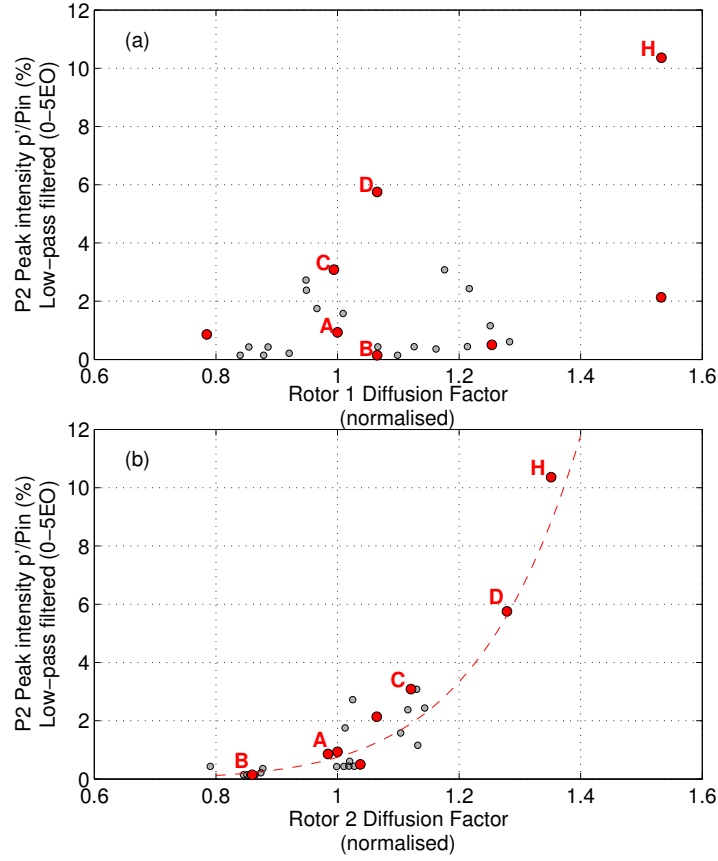


Figure 4.6: Correlation of low-pass filtered peak unsteady pressure to diffusion factor for (a) Rotor 1 and (b) Rotor 2, P2 location.

stall of Stator 1/Rotor 2 results in significant unsteady flow at a frequency below 5EO. At this stage no attempt is made to diagnose the physical cause of the unsteadiness (e.g. to confirm it is due to rotating stall or otherwise). Here, it is simply noted that activity below 5EO is only present when the Stator 1/Rotor 2 aerodynamic loading is high and is therefore believed to be associated with stalling in this region.

This observed behavior can be generalised to all 26 VSV configurations in addition to the 4 presented so far, by correlating the peak “stall” intensity observed against the diffusion factors derived earlier from the 1D model. This corresponds to applying a low pass filter to the pressure transducer signal and taking only the peak amplitude in the 0-5EO frequency range. The results are shown in Figure 4.6, where reasonable correlation

can be seen in Figure 4.6(b) between the intensity and the Rotor 2 diffusion factor (the Stator 1 correlation is of similar quality but not included for brevity). By contrast, the corresponding Rotor 1 correlation in Figure 4.6(a) is poor. This implies that this phenomenon (unsteadiness below 5EO) is indeed present for all VSV configurations which push Stator 1/Rotor 2 towards stall.

4.5 Effect of Rotor 1 stalling

Considering again each of the configurations assessed and presented in Figure 4.4, the pressure transducer traces for the group of VSV configurations denoted E, A, F and G are now examined. These progressively push Rotor 1 into stall with minimal effect on Rotor 2. This occurs when the IGV, Stator 1 and Stator 2 are opened, closed and closed respectively relative to the datum VSV settings.

Following identical logic to that presented in the previous section, the spectra, taken this time from the P1 transducer (upstream of Rotor 1 leading edge), are presented in Figure 4.7, with the 5EO line added in the same manner as shown earlier. In this sequence, unsteadiness can be observed to grow at frequencies *above* 5EO as Rotor 1 becomes more stalled, in contrast to the lower frequency phenomenon discussed in the previous section.

For Case G, the signal appears initially within a narrow band labeled “G1”. At higher speeds this appears to break up into broadband like activity labeled “G2”, which was still present when this manoeuvre was terminated at 91% speed. A similar sequence can be observed at lower speeds on the less severe Case F, with narrowband activity also giving way to broadband like activity, with the sequence occurring at lower speeds than observed on Case G. However for Case F, a further region of narrowband behavior can be seen at a higher frequency, beyond the cessation of the broadband phase, this is labeled “F1”. Finally, some faint activity can also be observed even on the datum Case A and is labelled “A1”. The progression through Cases E, A, F and G suggests that deliberate stall of Rotor 1 results in unsteady flow across a wide range of frequencies above 5EO.

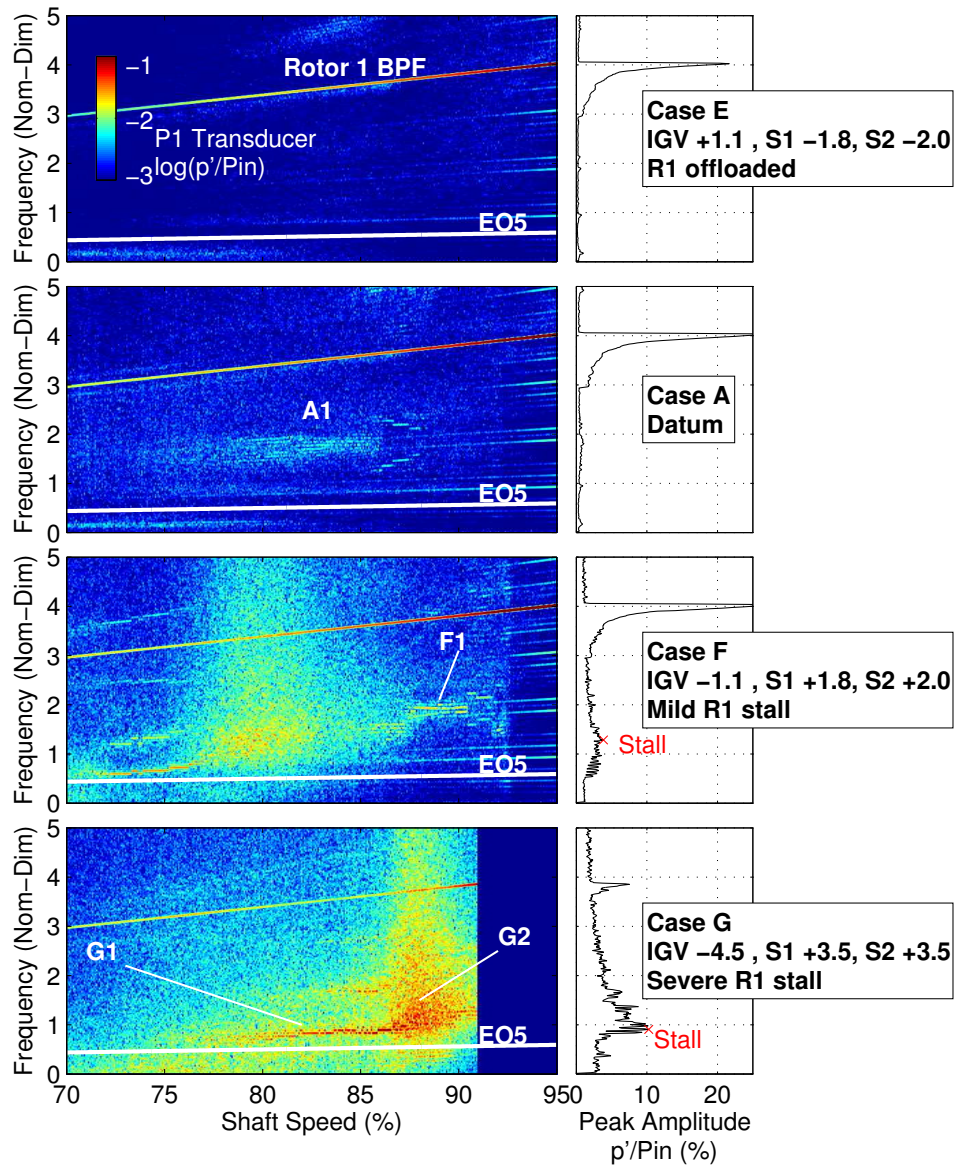


Figure 4.7: Spectral analysis of transducer at location P1 for Case E, A, F and G VSV settings, as R1 loading is increased. Stall activity is observed in >5EO range.

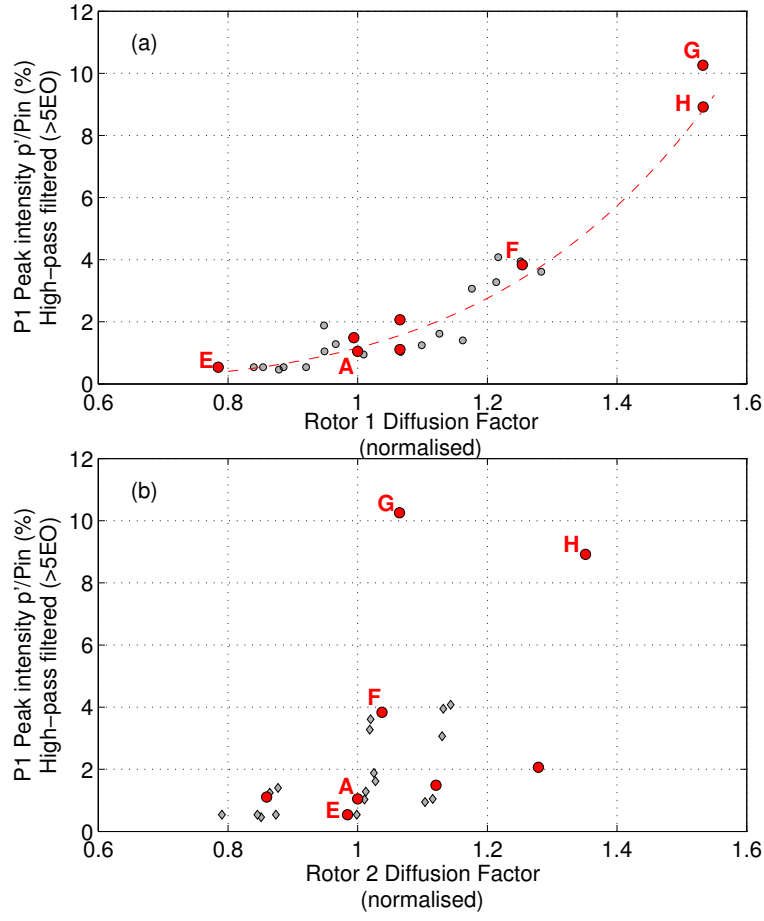


Figure 4.8: Correlation of high-pass filtered peak unsteady pressure to diffusion factor for (a) Rotor 1 and (b) Rotor 2, P1 location.

Figure 4.8 shows the intensity of this signal, taken as the peak above 5EO (high-pass filtered with blade passing tones also removed) when correlated against the meanline diffusion factors derived earlier. A strong correlation can now be seen in Figure 4.8(a) between the measured intensity and the Rotor 1 diffusion factor. By contrast, the correlation with the corresponding Rotor 2 (and also Stator 1) diffusion factor in Figure 4.8(b) is poor. This implies that this higher frequency pressure fluctuation observed close to the tip of Rotor 1, is present for all VSV configurations which push Rotor 1 towards stall.

4.6 Noise and vibration during Rotor 1 stall

The previous section revealed how changes to the VSV schedule which tend to stall Rotor 1 (opening the IGV combined with closure of Stator 1 and Stator 2) resulted in the appearance of unsteady flow across a broad frequency range above 5EO. This was based on observations made on a stationary pressure transducer immediately upstream of Rotor 1 leading edge in location P1 (refer to Figure 3.1). During this part of the experiment, simultaneous activity was also observed on the transducer at location PX (also shown on Figure 3.1). This location provides an indication of potential farfield noise sources and is approximately 2 tip diameters upstream of Rotor 1. Furthermore, strain gauges positioned on Rotor 1 showed vibration activity implying that the unsteady flow was coupling with the blade vibration modes and resulting in resonant responses.

Figure 4.9 shows the progression through Cases E, A, F and G presented using the spectral analysis methods already described, using the upstream PX pressure transducer. This can be contrasted directly with Figure 4.7 which effectively showed the equivalent plot for the “nearfield” location P1. For Case E (Rotor 1 offloaded) the level of upstream noise is relatively low, with Rotor 1 blade passing frequency still visible. However, progression to Cases A and F, with Rotor 1 more highly loaded, shows periods of significant noise propagation at approximately 82% and 90% speed respectively. Comparing these features with Figure 4.7, similar features were also observed at this speed and frequency (labelled “A1” and “F1”). This behaviour implies that this Rotor 1 “stall” phenomenon has a further interesting property - at some conditions upstream noise propagation is observed.

It can also be observed that the most extreme VSV schedule for Rotor 1 stall, referred to as Case G, does not appear to show the same level of upstream noise as Cases A and F, although weaker activity can be seen at approximately 90% speed immediately before the manoeuvre was terminated. It is likely that due to this case being stopped at 91% speed the noise phenomenon was missed. This is supported by noting that for Cases A and F, the noise phenomenon appeared initially at 82% speed and then at 90% speed respectively, implying that the phenomenon does shift to higher

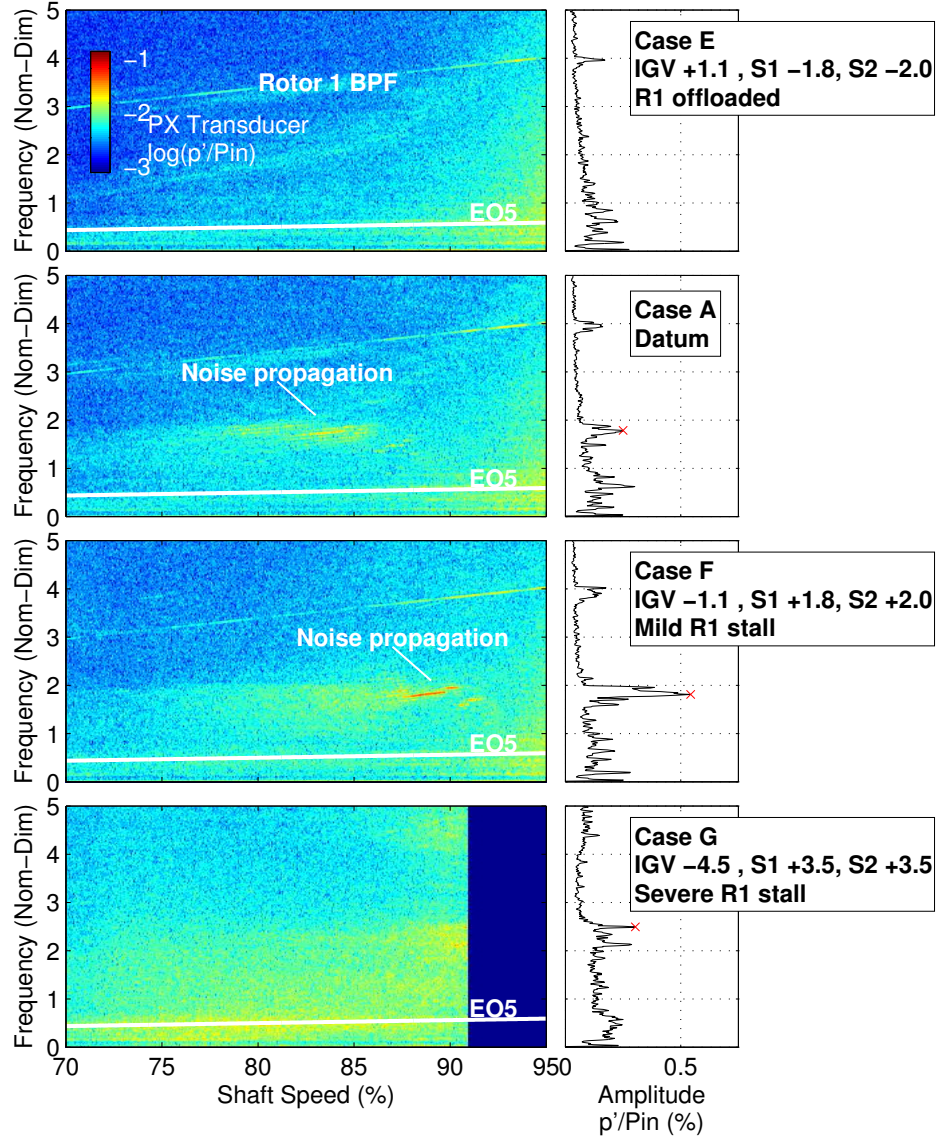


Figure 4.9: Spectral analysis of transducer at location PX for Cases E, A, F and G as Rotor 1 is pushed in to stall.

speed for the more highly loaded cases. Therefore the potential for the noise propagation moving to speeds above 90% on the most highly stalled Case G is logical. The fundamental mechanism by which noise propagation may occur, from a stalled rotor, is presented in the next chapter.

Analysis corresponding to that with the pressure transducers shown in Figure 4.7 has been performed using data acquired from a strain gauge mounted upon the Rotor 1 blade, to identify whether this phenomenon results in any significant vibration. The results of this are shown in Figure 4.10, where another trend can be observed as the VSVs are adjusted to push Rotor 1 further into stall, through Case E, A, F, G. For the unstalled Case E, the peak vibration amplitudes observed are all relatively low. However, as Rotor 1 becomes stalled on Case A and F, activity appears to couple with a vibration mode at a constant non-dimensional frequency. This corresponds to second torsion (2T) mode of Rotor 1, which shows non-integral excitation at approximately 86% and 91% speed for Cases A and F respectively, not dissimilar to the noise phenomena discussed earlier which also showed a drift to higher speeds as the loading was increased. The most heavily stalled Case G shows lower levels of torsional vibration, potentially for similar reasons given for the noise phenomenon (i.e. the acceleration was terminated at 91% speed and the torsional vibration may have been missed).

It is of interest to question whether the correlation between the unsteady measurements and the 1D diffusion factor, such as that presented in Figure 4.8 for the P1 pressure transducer local to Rotor 1, also applies to the noise and vibration phenomena discussed here. This is presented in Figure 4.11, where a strong correlation is observed between both measurements (noise and vibration) and the predicted Rotor 1 loading for all points, with the exception of Cases G and H which are both extreme VSV schedules which, as already discussed, were terminated at 91% speed, prior to the expected onset.

The potential benefit of the correlations in Figures 4.8 and 4.11 should also not be overlooked. These are based on a purely 1D prediction of the velocity triangles within the variable stages, using a model typical of the type which would be used during the preliminary design of the engine and

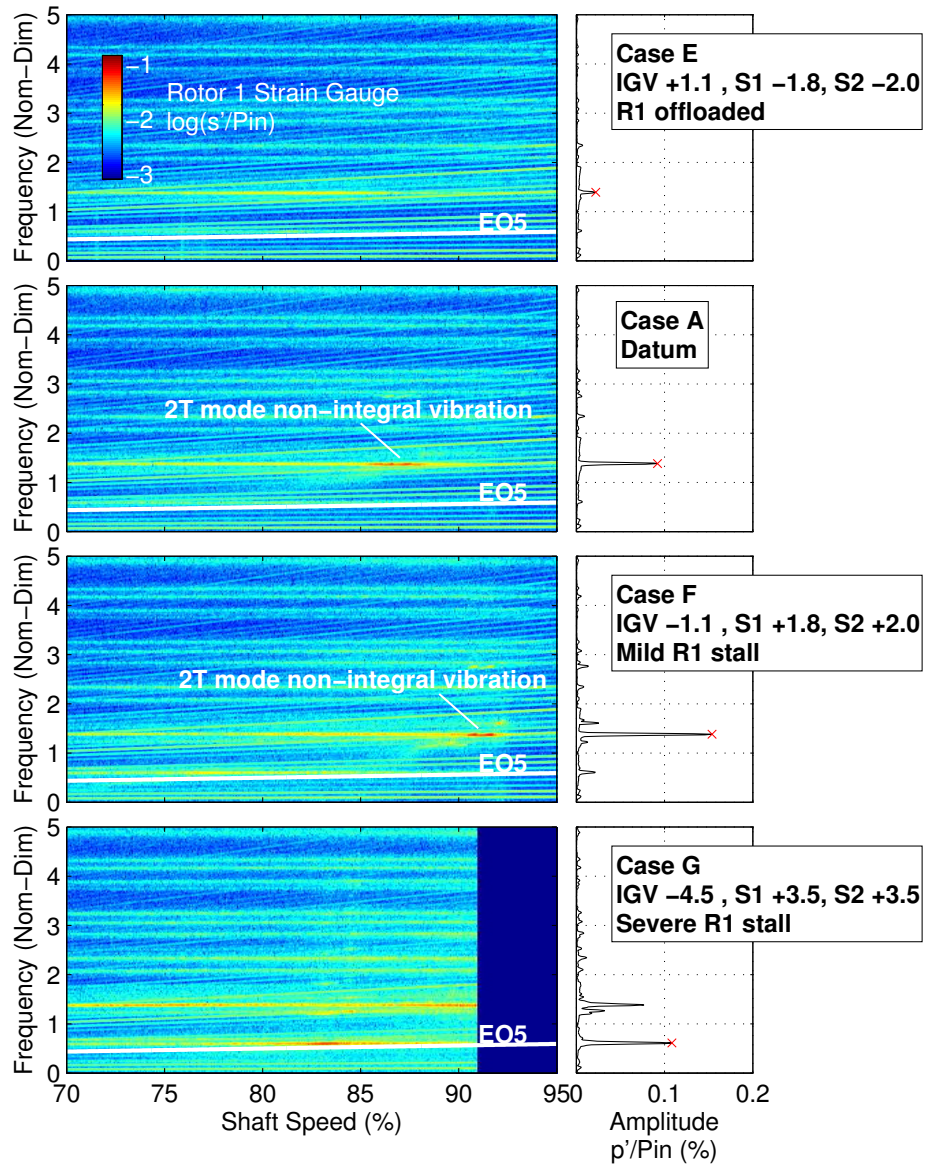


Figure 4.10: Spectral analysis of strain gauge on Rotor 1 for Cases E, A, F and G as Rotor 1 is pushed in to stall.

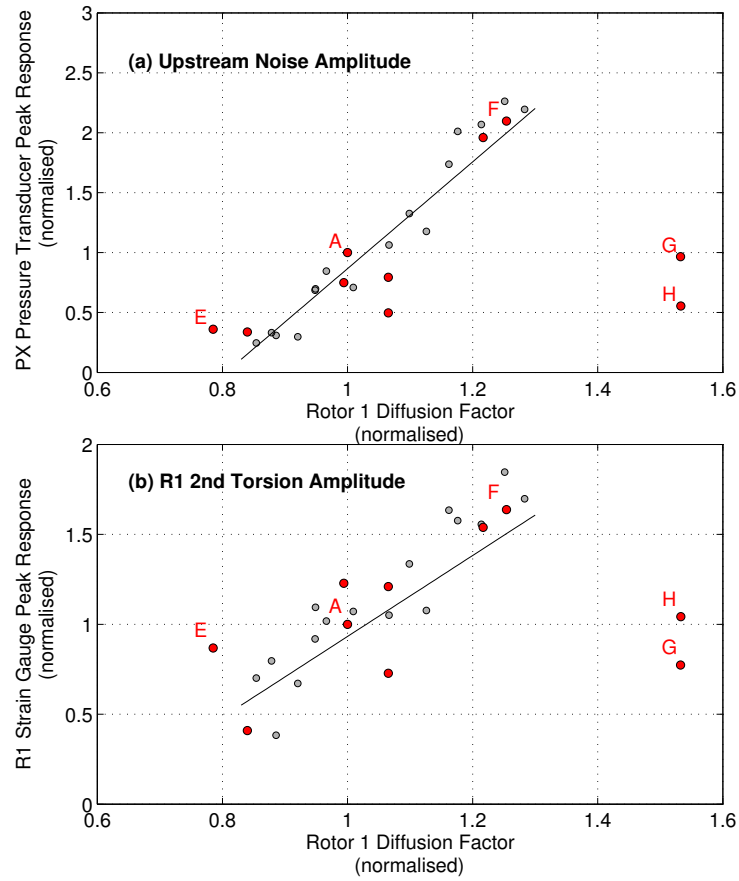


Figure 4.11: Correlation to Rotor 1 diffusion factor for (a) “Upstream noise” PX pressure transducer unsteady pressure and (b) 2T mode amplitude on Rotor 1 strain gauge.

before the detailed design of any blading. This section shows that a strong correlation can be observed between these 1D predicted diffusion factors and the complex unsteady stall phenomenon responsible for forced vibration and upstream noise propagation. Both of these are design attributes that would be desirable to identify and eliminate as early as possible in the compressor development programme. Such detailed information is therefore of significant practical benefit to both the aerodynamic and mechanical designer.

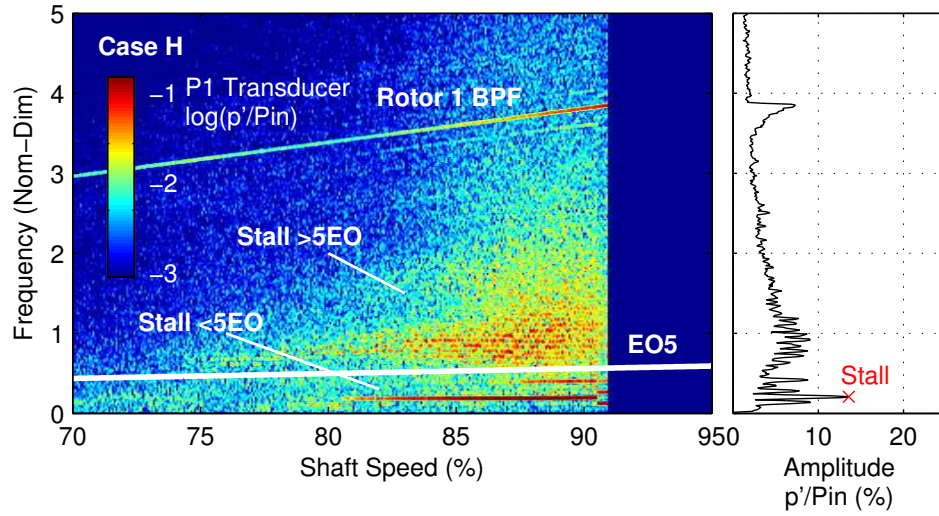


Figure 4.12: Spectral analysis of transducer at location P1, Case H - high loading for R1, S1 and R2 combined.

4.7 Concurrent stalling

The previous sections showed that as Rotor 1 approaches stall (through deliberate adjustment of the variable stator vanes), activity appears at frequencies *above* 5EO on the casing static pressure transducers with the amplitude showing a good correlation to meanline predicted aerodynamic loading parameters. By contrast, stall of the Stator 1/Rotor 2 region gives rise to activity *below* 5EO. A situation also arises in this experiment where Rotor 1, Stator 1 and Rotor 2 all approach stall simultaneously, referred to as Case “H”. Figure 4.4 shows that for this configuration both the Rotor 1 and Rotor 2 loadings are as high as they were for the most extreme cases in isolation. This case also fits the meanline correlations in Figure 4.6 and Figure 4.8 for both families. It is therefore expected that simultaneous activity above and below 5EO must be possible.

Figure 4.12 shows the spectral behaviour for Case H from the P1 pressure transducer, with the 5EO line also added for ease of interpretation. Intense activity both above and below 5EO can indeed be seen during this manoeuvre, with the activity occurring simultaneously above approximately 80% speed. This implies that the unsteady behaviour observed in isolation, as the

Rotor 1 and Stator 1/Rotor 2 regions are pushed into stall independently, also persist as separate phenomena which can occur simultaneously. The interpretation of this at a fluid dynamic level will be explored in the subsequent chapters.

4.8 Concluding remarks

This chapter began by considering how rotating stall may be triggered deliberately by adjusting the VSV setting angles away from an idealised schedule. In particular, “over-closure” of the rearmost variable vane will cause a reduction in the flow swallowed by the downstream stage, which in turn will move the upstream rows towards stall - making schedules configured this way ideal for this experiment to trigger rotating stall and study its behaviour.

The experimental procedure was discussed, consisting of slow acceleration manoeuvres with the compressor held on a fixed working line, which were repeated for 26 different variable stator vane schedules (defined in terms of their offset relative to a datum case). The 1D meanline method was used to assess the aerodynamic loading effect of each of these schedules in terms of the mid-span diffusion factor on Rotor 1, Stator 1 and Rotor 2. Using this analysis method, a group of the schedules tested were identified that progressively push the Stator 1/Rotor 2 region simultaneously into stall (i.e. higher diffusion factor) with minimal effect on the Rotor 1 loading. A similar group stalled Rotor 1 whilst Stator 1/Rotor 2 loadings remained almost constant.

Trends within the unsteady experimental data were then explored by comparing spectral data from fast response pressure transducers. It was observed that the Stator 1/Rotor 2 stalling behaviour was associated with unsteady activity at relatively low frequency below the 5th engine order (5EO). Rotor 1 stalling behaviour was associated with activity at higher frequency, always above 5EO, with this phenomenon showing the additional interesting observations of upstream noise propagation and coupling to a torsional vibration mode of the Rotor 1 blade. Moreover, a strong correlation was observed between the peak amplitudes within these frequency

ranges and the diffusion factors predicted by the meanline model. Simultaneous activity both above and below 5EO was also observed for the extreme case where both regions of the compressor were pushed into stall simultaneously, suggesting that these two separate stall “families” remain distinct from each other.

It should be emphasised that, at this stage, no attempt has yet been made to attribute the unsteady activity to a particular phenomenon such as rotating stall. In the chapters that follow, each of these observed families of stalling activity will be explored further, both through more detailed signal processing and the use of unsteady CFD simulation.

5 Experimental study 2: Detailed analysis of extreme VSV settings.

In this chapter attention now focuses in detail on the physical behaviour giving rise to the signals reported in chapter 4. It is recalled that testing with multiple VSV schedules revealed two key “families” of unsteady activity observed on the pressure transducers, each occurring at different frequencies. By adjusting the VSVs to deliberately stall the Stator 1/Rotor 2 block, unsteady activity was seen at frequencies *below* 5EO, whilst deliberately stalling Rotor 1 resulted in activity across a higher frequency range *above* 5EO. The two most extreme VSV schedules associated with each of these phenomena were named Case “D” and “G” for the Stator 1/Rotor 2 and Rotor 1 families respectively.

5.1 The Stator 1/Rotor 2 stall family (low frequency)

In this section the most extreme Stator 1/Rotor 2 stall configuration is studied in detail in an attempt to understand the fundamental cause of the activity seen below 5EO. This was referred to as Case D and was generated by closure of the IGV, opening of Stator 1 and closure of Stator 2 by 4.5° , 3.5° and 4.5° respectively, relative to the datum Case A VSV schedule.

5.1.1 Frequency shifting technique

Figure 5.1(a) shows the Campbell diagram for Case D, for which the 0-5EO signal was strongest. A narrower frequency bandwidth has now been chosen compared to the previous chapter allowing more detailed features to be identified. Corresponding data is also shown in Figure 5.1(b) for a rotor strain gauge.

From Figure 5.1(a), as the shaft speed is increased, the unsteadiness below 5EO can be observed in more detail. Of particular interest are the spontaneous changes in frequency, occurring at speeds of approximately 75, 79 and 86%. Selecting a speed of 78% (where the stall signal was at its maximum amplitude), the stalling frequency corresponds to 2.27EO, as shown in Figure 5.1(a).

Turning attention now to the corresponding rotor strain gauge spectra in Figure 5.1(b), similar features can be observed, with spontaneous frequency changes visible at the same speeds. Note that the strain gauge here is simply responding to small off-resonant vibration due to unsteadiness in the surrounding flow - i.e. the “stall” phenomenon is not resulting in high vibration levels as its frequency is separated from the nearest vibration mode, but it is still detectable by the strain gauge. Also visible in Figure 5.1(b) are the engine orders naturally present in the signal and a blade vibration mode, at a frequency slightly higher than that of the stall. These features were also identified in the 50% speed rotating stall example presented in chapter 3 and are not discussed further here. Selecting a speed of 78%, the frequency of the stall signal in the rotating frame corresponds to 2.73EO, as shown in Figure 5.1(b).

At this stage it can therefore be confidently stated that observations in two different frames of reference (stationary pressure transducer and rotating strain gauge) are both showing evidence of the same phenomenon, but at different frequencies. Recapping the analysis methodology presented in chapter 3, Equations 3.3 and 3.4 were derived, to show how a disturbance rotating relative to both probes will give rise to a shift in the resulting frequencies. In the case presented in chapter 3, summation of rotating static engine orders gave the number of stall cells.

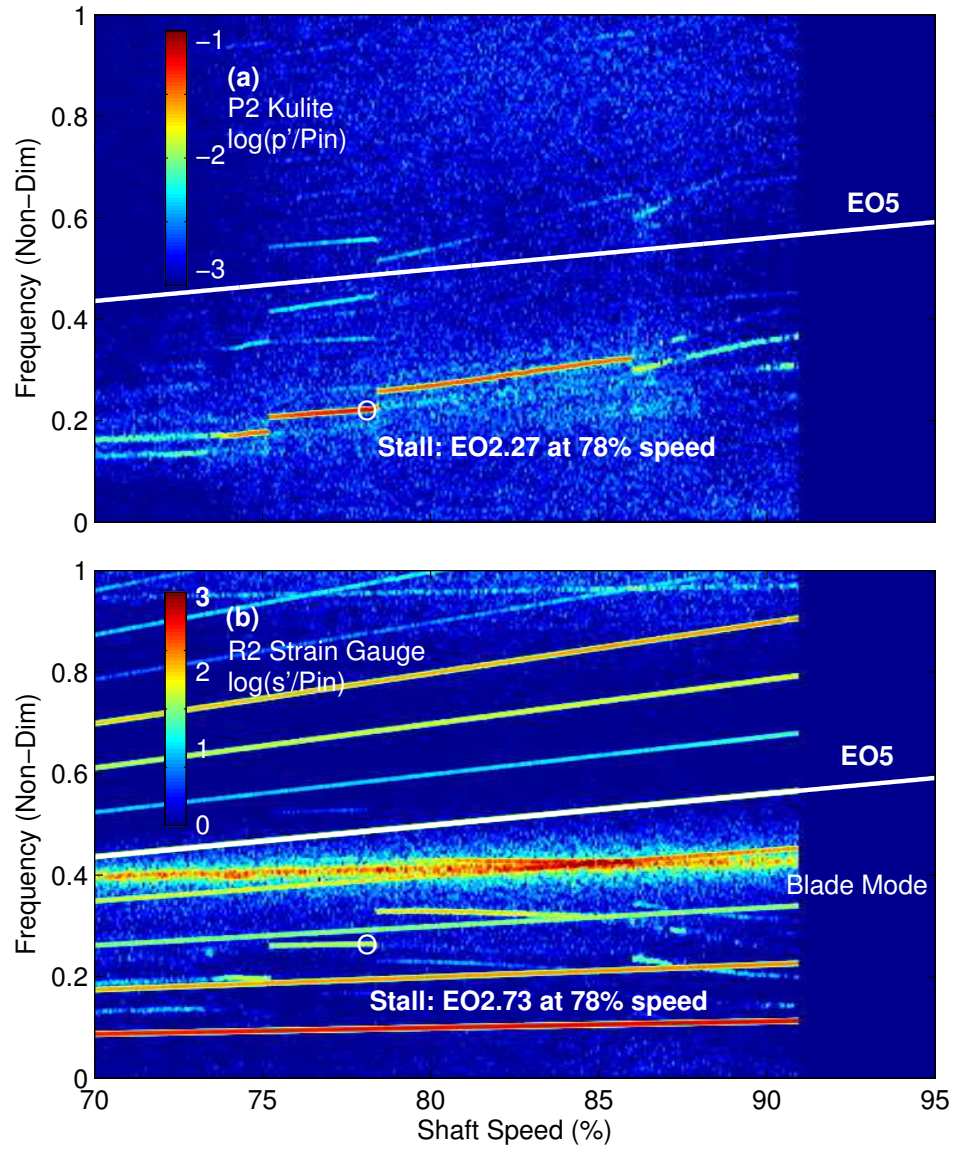


Figure 5.1: Case D - spectral analysis of (a) P2 stationary transducer and (b) Rotor 2 strain gauge.

Application of the same process is possible to the behaviour in Figure 5.1. At 78% speed the rotating and static orders sum exactly to 5, implying the presence of 5 cells which must be rotating at 45% of blade speed (to satisfy Equation 3.4). This is typical of part-span rotating stall[22].

Further confidence in the calculated stall cell count can be gained by repeating the process at other speeds - where it becomes apparent that combination of the static and rotating engine orders always results in an integer to within 2 decimal places. Figure 5.2 shows the result of tracking the stall signal across the whole manoeuvre, applying Equations 3.3 and 3.4 at each speed. The sharp transitions in frequency correspond to apparently spontaneous changes in the number of stall cells (4,5,6,5). It should be emphasized that changes in the shaft speed and VSV setting took place in a smooth manner across the manoeuvre. The transitions are therefore not in any way deliberately provoked by the way the compressor is operated and appear to occur naturally.

At each of these points the propagation speed V_{stall}/U also changes, the trend being that the stall cells travel faster whenever the number of cells reduces (i.e. when the cells grow). This appears to be in conflict with past experience - during stall inception, for example, it is commonly reported [12] that short length scale spikes initially propagate at high-speed, decelerating as they grow into a mature cell. A detailed explanation for this anomaly is given later in this thesis, when considered alongside numerical simulations.

Figure 5.2(c) also shows the intensity of the stall signal for transducers P1-P4 (see Figure 3.1). At lower speeds the signal is strongest at location P2, consistent with Stator 1/Rotor 2 stalling. However, the downstream location P3 dominates at higher speeds. Whilst it might be inferred from this that the blade rows downstream have also become stalled, this is considered unlikely. For the sequence of Cases B, A, C and D, Stator 2 is progressively closed (Table 4.1), each time further *unloading* Stator 2/Rotor 3, as follows from the logic presented in the previous chapter. This block will therefore tend to suppress, rather than trigger the formation of stall cells. Therefore, conventional stalling in these rows is unlikely to be the reason for the higher amplitude at the P3 location. This anomaly will be also be explained later in this thesis when considered alongside the numerical simulations.

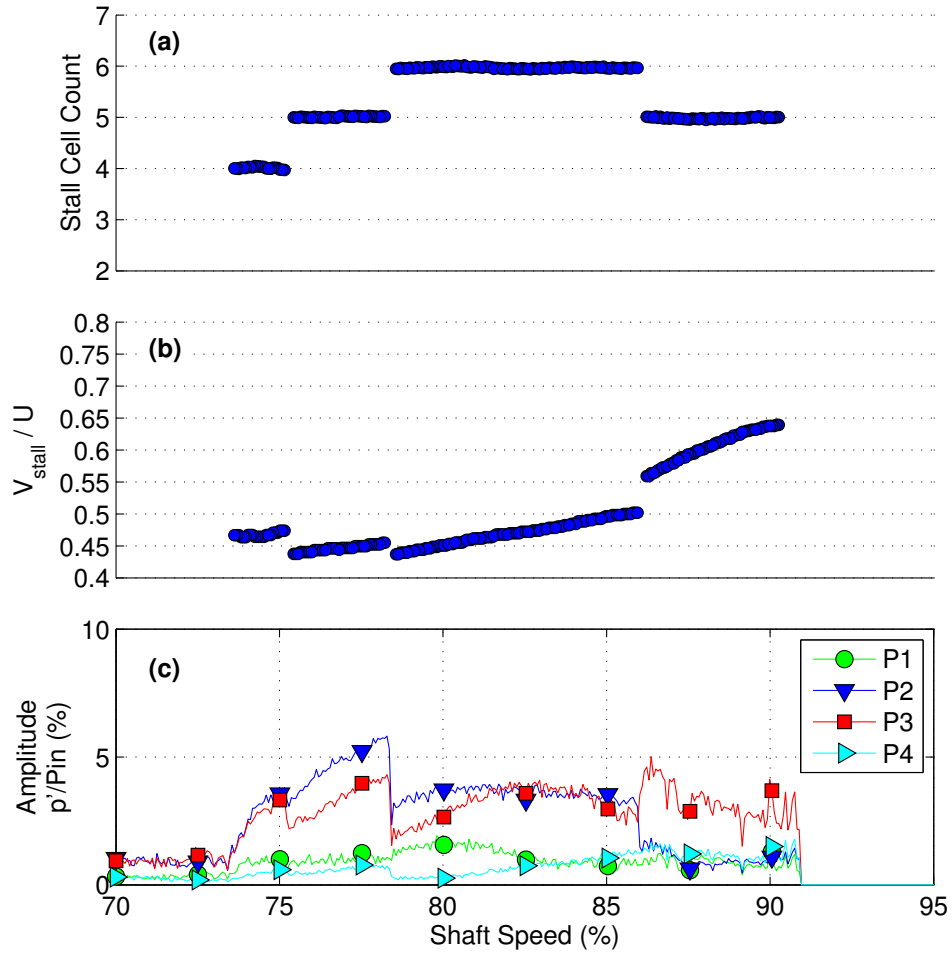


Figure 5.2: Case D - behaviour of rotating stall through manoeuvre in terms of (a) cell count, (b) propagation speed (c) pressure amplitude distribution across 4 axial measurement locations.

5.1.2 Comparison of acceleration/deceleration.

An additional useful deduction can be made from Figure 5.3, which shows the pressure transducer spectral response against time for the duration of the whole acceleration and deceleration parts of the Case D manoeuvre. The left-most part of the manoeuvre, up to a time of 200 seconds, corresponds exactly to that already shown and analysed in Figure 5.1, which consisted of only the acceleration part. The rightmost part, however, corresponds to the deceleration of the shaft back to 50% speed across the subsequent 200 seconds. It is evident from the symmetry of Figure 5.3 that during the deceleration phase the transitions in cell count occur in the same manner. This implies that the stall behaviour is unique to one operating condition and does not exhibit hysteresis (i.e. the observed stall behaviour at a given condition does not depend on how that condition was approached). Based on other experimental studies [20] this does appear unusual - typically full-span stall occurs with a single cell and exhibits considerable throttle hysteresis (i.e. a small throttle closure can trigger rotating stall which requires a much larger subsequent opening of the throttle to allow the compressor to recover). By contrast, part-span stall, consisting of multiple cells confined to the blade endwalls, exhibits much smaller throttle hysteresis. Evidently, the presence of multiple cells in this test case fits more consistently with the latter case of part-span stall, with little hysteresis. The spanwise location of the stall cells is therefore addressed in the next section using a purely experimental technique, which will subsequently be supported by numerical analysis in the later chapters.

5.1.3 Radial location of stall cells

The evidence presented so far indicates that deliberate stalling of the Stator 1/Rotor 2 region of the compressor, through VSV adjustments, results in the formation of 4 to 6 rotating stall cells, with the most extreme case showing spontaneous switching in the number of cells. Based on previous reported cases of rotating stall, these observations are far more consistent with part-span stall, where multiple cells are commonly observed, than with full-span stall - which typically consists of a single cell[22]. This section

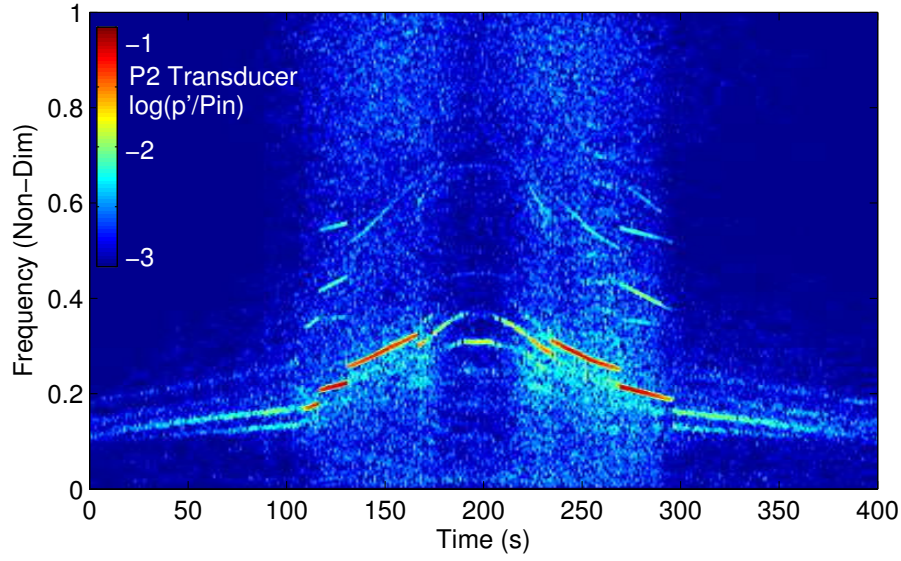


Figure 5.3: Case D - spectral behaviour on acceleration and deceleration.

therefore seeks to determine the spanwise location of these stall cells using the available unsteady experimental data. It is also helpful at this point to state upfront some findings from the numerical investigation in chapters 6 and 7, which show that the hub region of Stator 1/Rotor 2 becomes stalled at these part-speed conditions.

Therefore, a question arises regarding whether rotating stall can indeed form in the hub region and persist there as a stable phenomenon. In virtually all reported cases of part-span rotating stall, the cells exist in the rotor tip region. However, this experience is commonly obtained from low-speed testing which does not accurately represent the aerodynamics of this test case, where significant part-speed closure is applied to the variable stator vanes. Chapter 6 explores this effect in more detail and shows that Stator 1 hub naturally stalls as a result of radial equilibrium effects as the VSVs close down, whilst the outer span remains at quite tolerable loadings. Furthermore, in the majority of reported cases, including that presented here, pressure transducers (or hot-wire probes) are positioned only along the outer casing wall. It is quite unusual to fit fast response instrumentation to the inner hub region, given the accessibility constraints, making rotating stall disturbances local to the inner wall considerably more difficult to detect.

A means is therefore sought here to differentiate between rotating stall occurring at the casing or hub. Whilst the static pressure perturbations were only measured locally at the casing during this experiment, using pressure transducers, strain gauges mounted upon the stator vanes offer an alternative unsteady measurement in the stationary frame of reference which can be considered as the integrated effect of flow unsteadiness across the whole vane. It is therefore of interest to seek occurrences of this stall phenomena during the experiment where stall-like activity is observable on only the strain gauge, which responds to the effect of stall along the whole span, whilst not observed on the pressure transducer, which is dominated by the pressure amplitude on the casing. In such instances the implication would be that rotating stall cells are confined to the hub.

Therefore, a method was devised whereby the signal from the P2 pressure transducer was compared directly with the corresponding signal from a strain gauge mounted immediately upstream on the Stator 1 vane, close to the hub. This is shown in Figure 5.4, for Case C. The stall signal can be seen on both the casing transducer and stator strain gauge at identical frequencies (as both measurements are in the same frame of reference). Note also that in this case the signal also occurs at similar frequencies to the more extreme Case D VSV schedule (below 5EO), but does not exhibit the spontaneous changes in cell count. At higher speeds, as the stall signal decays, the point of stall cessation occurs significantly earlier on the casing transducer (at approximately 86% speed) than it does on the strain gauge (at approximately 89% speed). As the acceleration rate was slow, this corresponds to over 1000 shaft revolutions during which stall can only be observed on the stator strain gauge. The sustained presence of unsteadiness in the Stator 1 region, which cannot also be detected at the casing, implies that this phenomenon is indeed local to the inner span region and rotating stall may occur in the Stator 1/Rotor 2 hub region as a stable phenomenon. This observation will be explained and supported further by the numerical simulations in chapters 6 and 7.

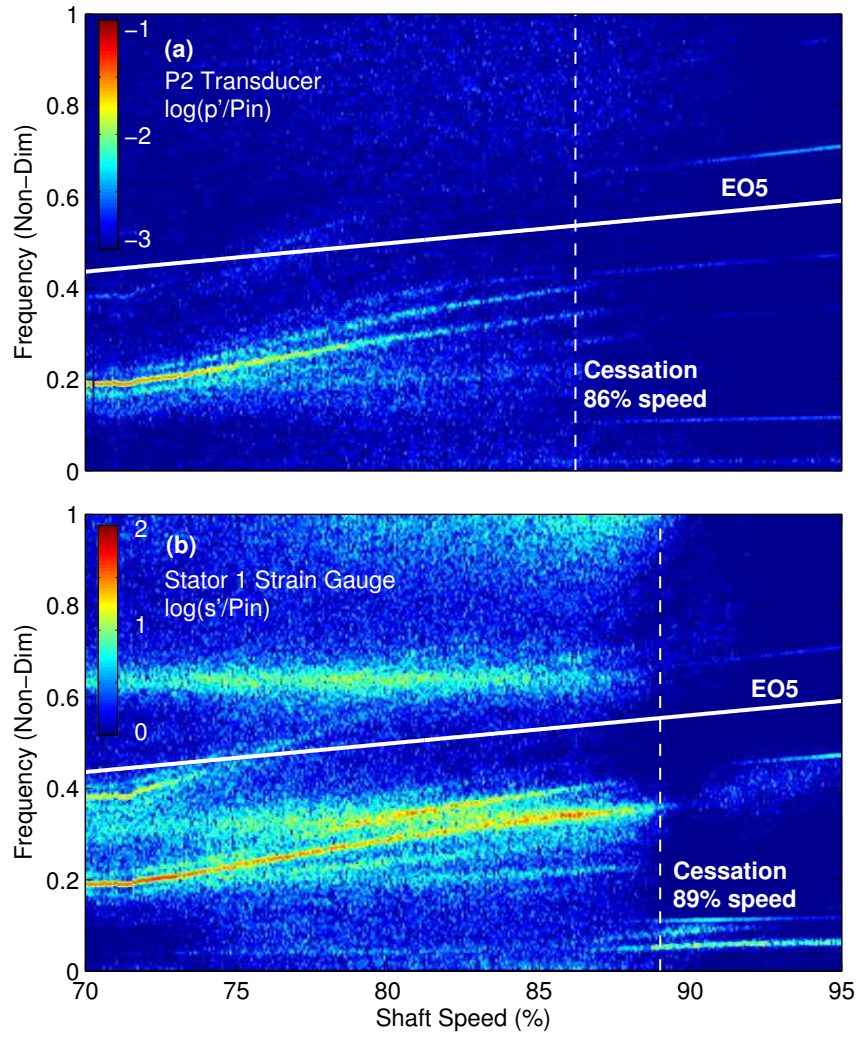


Figure 5.4: Case C acceleration showing stall cessation points on (a) P2 Pressure Transducer and (b) Stator 1 strain gauge.

5.2 The Rotor 1 stall family (high frequency)

Recalling each of the configurations assessed and presented in chapter 4, it was observed that progressively pushing Rotor 1 into stall (with minimal effect on Stator 1/Rotor 2) gave rise to unsteady flow at frequencies above 5EO. This was shown in Figure 4.7, for which the most extreme case was observed with the variable vanes in the Case G setting. The signal appeared initially within a narrow band labeled “G1”. At higher speeds this appeared to break up into broadband like activity labeled “G2”, which was still present when this manoeuvre was terminated at 91% speed.

5.2.1 Frequency shifting technique

Following a similar logic to the analysis of the lower frequency rotating stall phenomenon linked to Stator 1/Rotor 2, the physical behavior giving rise to this higher frequency activity is now investigated. The Case G manoeuvre is shown over a narrower bandwidth in Figure 5.5. This is sufficient to give better clarity of the G1 feature, however the broadband activity at higher speed is not investigated further in this thesis and falls outside the frequency bandwidth shown. At this scale the narrowband signal seen earlier in Figure 4.7 (labeled G1) is actually composed of several discrete frequency “bands”. The significance of multiple bands in the spectrum, in contrast to the isolated band behaviour associated with Stator 1/Rotor 2 (e.g. Figure 5.1) will be discussed in the next section.

Selecting a speed of 86%, where the amplitude peaks, the frequency of the signal (choosing the band with the highest intensity) corresponds to 8.43EO, as highlighted in Figure 5.5(a). The rotor strain gauge spectrum is shown below this in Figure 5.5(b), also showing multiple bands within the same speed range. The corresponding peak at 86% speed occurs at 6.57EO. It follows from exactly the same logic given earlier (application of Equations 3.3 and 3.4) that summation of rotating and static engine orders implies the presence of a 15 lobed pattern travelling at 56% of shaft speed. This is also indicative of rotating stall, albeit at a shorter length scale than that in the previous section.

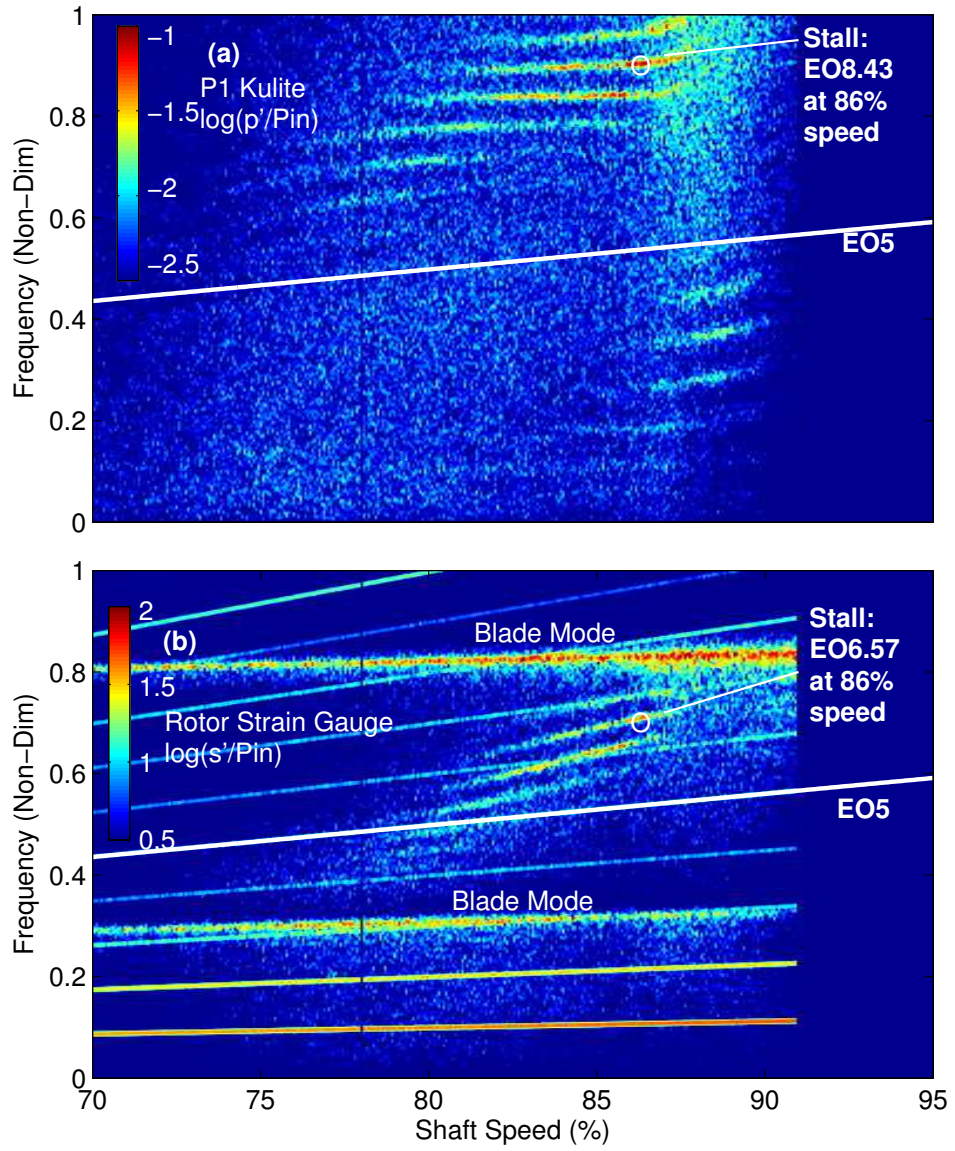


Figure 5.5: Case G - spectral analysis of (a) P1 transducer and (b) Rotor strain gauge.

Figure 5.6 shows the result of applying of Equations 3.3 and 3.4 across the whole manoeuvre. Each frequency band, observed in the static and rotating frames, can be attributed to a particular circumferential mode, with modes 13 to 16 easily identifiable to within 2 decimal places of accuracy, although some trial and error is required to match the appropriate feature on both the static and rotating probes to give an integer cell count value. All modes appear to rotate at identical speeds of up to 60% of shaft speed, with the propagation speed (V_{stall}/U) falling very slightly as the shaft accelerates. The trend in stall propagation speed V_{stall}/U with shaft speed is reflected in the slope which the stall bands possess when viewed in the spectra at each location - on the stationary transducer the stall bands are nearly horizontal (their engine order falling), whilst on the rotating gauge the stall bands are rising steeply in frequency (engine order rising). Application of Equations 3.3 and 3.4 confirms these trends are associated with the change in V_{stall}/U during the acceleration (further investigated in chapter 8).

The intensity of the tracked stall signal across the speed range is also shown in Figure 5.6(c) for the transducers at locations P1-P4 (see Figure 3.1). At all speeds the signal is attenuated very rapidly downstream and is barely visible by location P2 (Rotor 2 leading edge). This is consistent with a higher number of cells than the Stator 1/Rotor 2 stalling phenomenon, which Figure 5.2 showed was clearly visible across a whole stage. A higher number of cells would tend to decay more rapidly in the axial direction due to both the potential field effect (i.e. the lengthscale of axial decay can be shown to be approximately proportional to the circumferential spacing between disturbances) and also will tend to mix more rapidly in a viscous sense with smaller cells.

5.2.2 Reasons for a multiple mode response.

The behaviour linked to Rotor 1 rotating stall shows an important difference when compared with the Stator 1/Rotor 2 family of stall, with the former showing the simultaneous presence of multiple frequency bands whilst the latter shows a single well-defined spectral peak. Two physical interpretations of this observation are given in the following discussion.

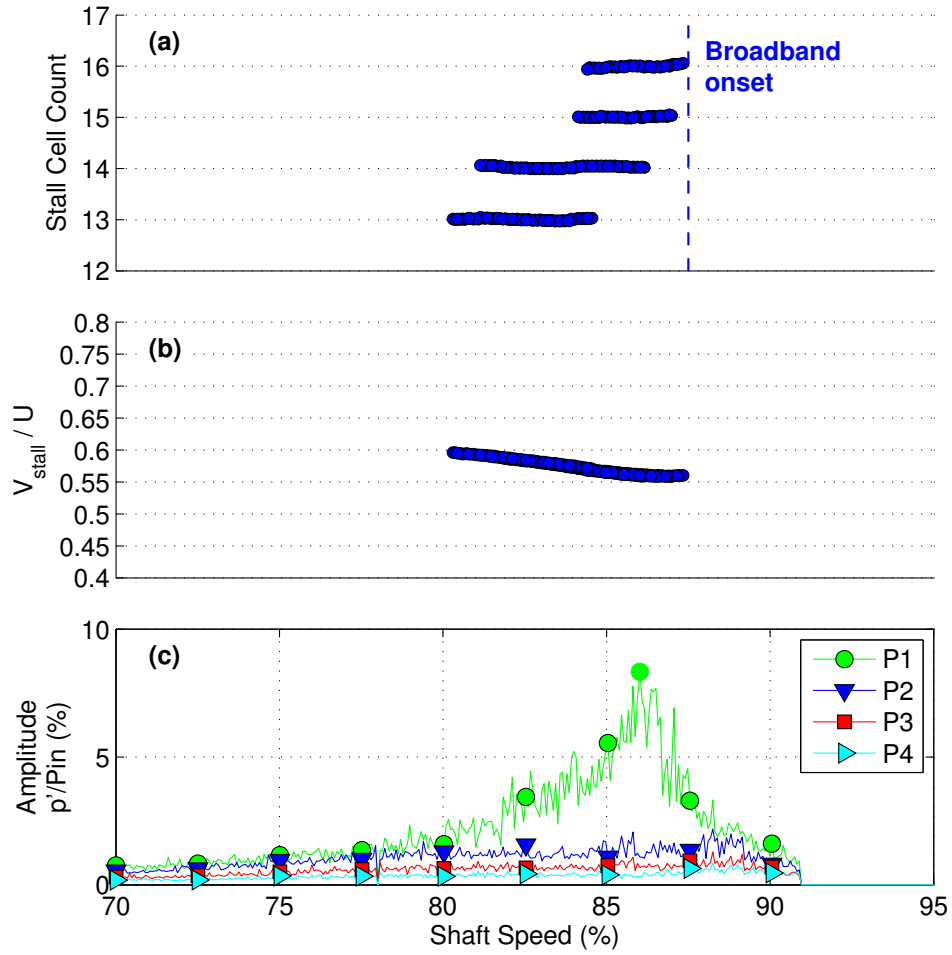


Figure 5.6: Case G - behaviour of rotating stall through manoeuvre in terms of (a) cell count, (b) propagation speed (c) pressure amplitude distribution across 4 axial measurement locations.

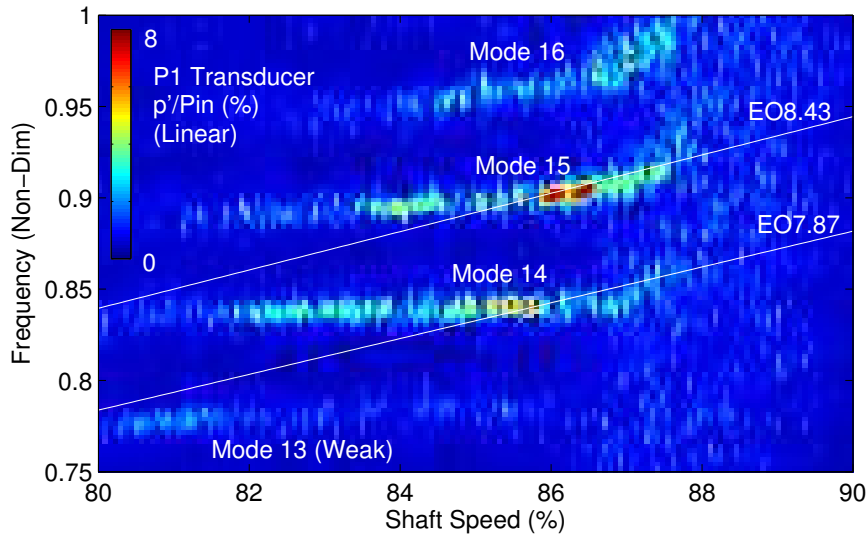


Figure 5.7: Case G - close up view of spectrum in Figure 5.5 showing multiple mode response.

Firstly, the intensity of each band (or circumferential mode) appears to oscillate, suggesting a pattern of stall cells that is non-uniform in time; the cells periodically break up and re-form, but fluctuate within the range between 13 to 16 cells. This can be seen in Figure 5.7, showing a close-up view of the spectrum shown earlier in Figure 5.5, now shown on a linear scale, with the corresponding mode orders added. The dominant mode order does not remain constant and oscillates between 14, 15 and 16.

A second reason for the multiple modes relates to the spatial distribution of the cells. Consider the situation whereby multiple stall cells, which are perfectly uniform in size and shape, are distributed in an exact equispaced arrangement around the circumference. The resulting spectral analysis would reveal a single well-defined peak (most likely also with the presence of harmonics, due to the fact that it is highly improbable that stall cells will also be purely sinusoidal in shape). This is also analogous to an rotor blade assembly which is manufactured with zero variability around the circumference - a pressure transducer spectrum for this would also possess a single well-defined blade-passing frequency, which for N blades would clearly occur at N engine order.

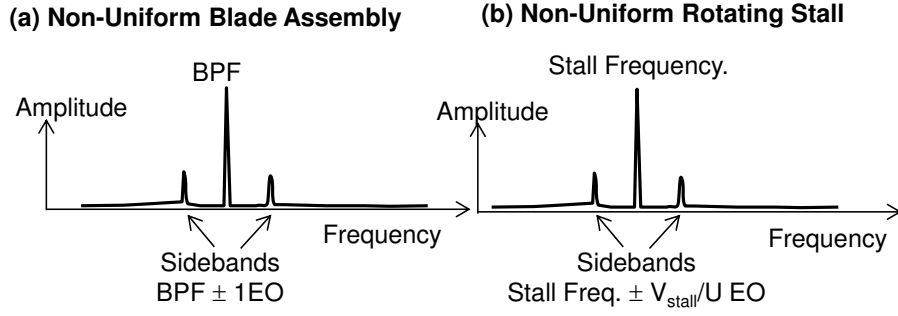


Figure 5.8: Analogy between (a) non-uniform rotor assembly and (b) non-uniform stall cell pattern showing sideband spacing relative to dominant frequency.

Keeping with the rotor blade assembly analogy, it is useful to then consider the equivalent spectral behaviour when variability in blade geometry is included (particularly if the blades were to not be uniformly spaced). In this case the spatial mode N would no longer be “pure” and would consist of a superposition of a very large number of additional modes ($N-1$, $N+1$, $N+2$ etc), noting of course that only integer mode numbers would be valid (non-integer wave-numbers would destructively interfere). The resulting temporal spectrum would therefore still be composed of a dominant peak at N engine order, however, the spectrum would also possess “side-bands” spaced by *whole integer engine orders* to represent the spatial non-uniformity of the geometry.

Returning to the case of rotating stall, therefore, Figure 5.8 shows how this analogy can be used to explain the multiple mode response. For a rotor, the side-bands are spaced by frequency intervals *equal to the rotor shaft speed (i.e. 1EO)*. By this logic, a pattern of spatially non-uniform stall cells must exhibit side-bands spaced by frequency intervals equal to the *stall propagation speed*. In terms of engine order notation (which by definition is frequency normalised by rotor speed), the expected side-band spacing must be given by an engine order equal to V_{stall}/U i.e. 0.55-0.6EO, as depicted in Figure 5.8. This spacing is precisely that seen in the close up in Figure 5.7; the frequency bands are separated by fractional engine orders equal to the stall propagation speed V_{stall}/U .

This implies that the Rotor 1 stall phenomenon is considerably less uniform both spatially and temporally in comparison to the lower frequency phenomenon related to Stator 1/Rotor 2. Contrasting this with other examples, this aligns most closely with the disturbances reported by Kameier and Neise[23] who categorised the phenomenon they observed as “rotating-instability”, though in their case the phenomenon was attributed to operation with high rotor tip clearances.

The preceding explanation also reveals a further point of experimental interest. It is commonly assumed that the “minimum” experimental arrangement required to derive the number of stall cells must consist of multiple transducers around the circumference (to determine phasing between difference locations) or alternatively, a single transducer combined with a rotating measurement (to perform analysis of the type shown in this thesis). However, for this case of the Rotor 1 stall, which exhibits multiple modes in its spectral response due to non-uniformity, the spacing between the frequency bands provides a key additional piece of information which gives the stall propagation speed. Once this is known (and can be obtained from a single transducer only), the number of cells can easily be determined from Equation 3.1. This implies that the details of rotating stall cell count and propagation speed can potentially be obtained from a single transducer alone. This is potentially of benefit during development testing where instrumentation (particularly in the rotating frame) is limited. For example, the spacing between the stall bands on a stationary measurement could be used to estimate the corresponding forcing frequency in the rotating frame and therefore the risk of potential mode interactions.

5.2.3 Stall induced noise

In chapter 4, it was shown how the Rotor 1 stall family exhibited upstream noise propagation and non-integral vibration. For these, the most extreme VSV configuration was Case F (noting that Case G was more severely stalled, but was terminated before these disturbances were encountered). Recalling Figure 4.7, activity was identified on the P1 transducer, local to Rotor 1 leading edge, after the cessation of the broadband phase and was labelled “F1”. At very similar speeds Figures 4.9 and 4.10 showed that

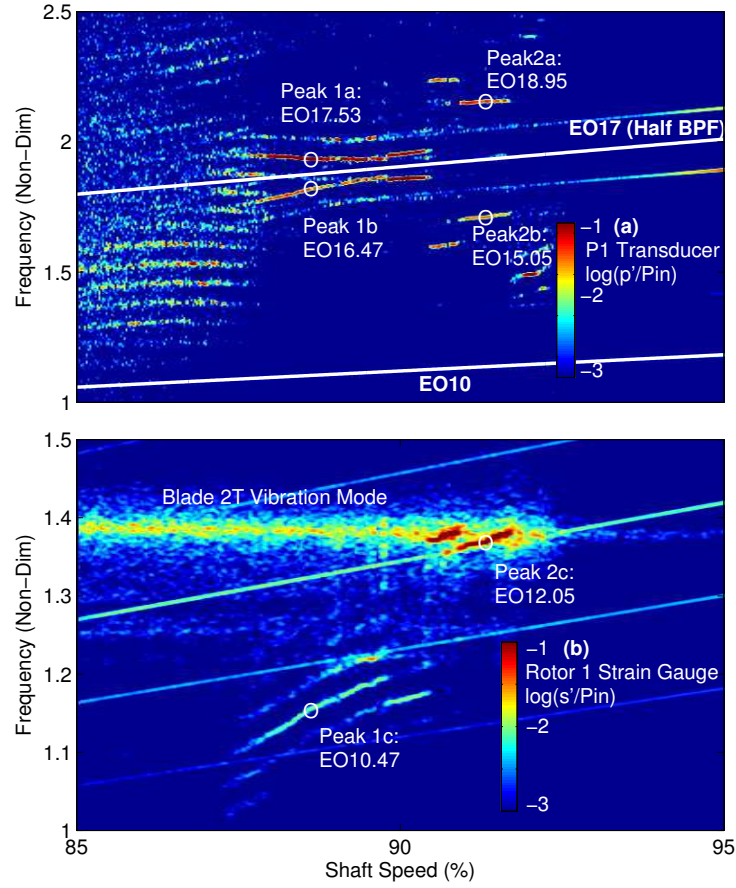


Figure 5.9: Case F - analysis of spectra from (a)P1 transducer and (b)Rotor 1 strain gauge, showing detail of feature “F1” (from Figure 4.7).

upstream noise and Rotor 1 vibration on the PX transducer and the strain gauge respectively. In this section, the “F1” feature is investigated further using the frequency shifting method already described.

Figure 5.9 shows the spectral data for Case F at a narrower bandwidth allowing the key details of feature “F1” to be identified, in the familiar format with the stationary P1 transducer and Rotor 1 strain gauge shown. As the broadband activity subsides above 85% speed, multiple bands can again be observed, not dissimilar to those discussed in the previous section, albeit at higher frequency. Two distinct regions of activity have been highlighted, aligning to the maximum upstream noise and vibration occurrences.

Starting with the lower speed activity, the feature labeled ‘Peak 1a’ is shown at a speed of 88.5% with a frequency of 17.53EO, on the P1 transducer. A weaker, second peak labeled ‘1b’ also occurs at 16.47EO. It should be noted at this stage that both peaks sum to 34, which is the number of Rotor 1 blades. Physically this occurs due to a spatial aliasing effect that will be described shortly. At the same speed the corresponding measurement in the rotating frame of reference shows a single peak at 10.47EO labeled ‘peak 1c’.

Ignoring, for a moment, the second peak at 16.47EO, application of Equations 3.3 and 3.4 shows that this can be interpreted as a 28 lobed pattern propagating at 63% of shaft speed in the direction of shaft rotation. This implies disturbances of very short lengthscale, comparable to that of a blade passage (Rotor 1 has 34 blades). The propagation speed V_{stall}/U is slightly higher than the case in the previous section (see Figures 5.5 and 5.6). This observation shows some similarity with the findings of Young et al.[16], who also reported stable disturbances of lengthscale comparable to that of a blade passage, which later research by Pullan et al.[17] demonstrated were due to radial vortex like structures (Figure 2.4) moving relative to the blade.

It is therefore necessary to develop an explanation of how a pattern of 28 stall cells may result in propagation of noise upstream and in particular why this only occurs for certain stall conditions. This requires a digression into duct acoustics. Firstly to very briefly explain the concept of acoustic “cut-on” and secondly to extend this to the concept of rotor-stator interaction tone noise. It follows that the noise mechanism presented here arises from a “rotor-stall” interaction which can be analysed in a similar manner to a conventional rotor-stator interaction.

Tyler and Sofrin[56] performed pioneering work in the field of duct acoustics and provide well established analytical methods for the analysis of noise generation/transmission in turbomachines. Their analysis starts from the Wave Equation (Equation 5.1), where c is the local speed of sound and p relates to a generalised static pressure perturbation. For simplicity the annulus is assumed narrow (very high hub-tip ratio) and the flow stationary - as these assumptions allow the general concept to be explained in a concise manner.

$$\frac{\partial^2 p}{\partial t^2} = c^2 \left[\frac{\partial^2 p}{\partial x^2} + \frac{1}{r^2} \frac{\partial^2 p}{\partial \theta^2} \right] \quad (5.1)$$

A complex solution is assumed of the form given by Equation 5.2, consisting of a spinning pressure perturbation which at a fixed moment in time consists of m circumferential lobes with an axial wavenumber given by k . At a fixed point in space the angular frequency is given by ω .

$$p(x, \theta, t) = A e^{i(\omega t - kx - m\theta)} \quad (5.2)$$

Substitution of Equation 5.2 into the wave equation and re-arranging leads to a relationship for k given by Equation 5.3, where M_{wave} is the Mach number at which the pressure wave travels circumferentially. This applies equally to any static pressure disturbance such as a rotating blade row, a rotating pattern of stall cells or an interaction pattern created between two independent perturbations.

$$k = \pm \frac{m}{r} \sqrt{M_{wave}^2 - 1} \quad (5.3)$$

Tyler and Sofrin[56] showed that a key property of Equations 5.2 and 5.3 is that when M_{wave} is below 1 (e.g. a blade row spinning with subsonic blade speed), the axial wavenumber is imaginary, which in combination with Equation 5.2 must imply a solution of the form e^{-kx} , whereby the pressure wave decays exponentially in the axial direction. This is known as the cut-off condition and results in zero transmission of noise to the farfield. By contrast, as the wave speed increases to the point at which it becomes supersonic, the behaviour changes radically and k becomes real. This results in a solution of the form $e^{\pm ikx}$ which implies propagation to the farfield with pressure waves spiraling along the duct. This is known as cut-on and is the means by which noise from turbomachinery can be transmitted along the annulus to the farfield.

Progressing further, it is implied from Equation 5.3 that cut-on will only occur when the wave speed is supersonic. This would imply that lower speed subsonic compressors would not transmit noise at all, as the resulting waveform would propagate circumferentially at a speed below Mach 1 and thus be cut-off. However, this is known to not be the case in many subsonic compressor and fans which still exhibit farfield noise effects. This was also explained by Tyler and Sofrin[56], who also showed how cut-on may still occur within lower speed compressors, due to interaction between the rotor and the stator (or other stationary disturbance). For a rotor with B blades and a stator with V vanes, an interference pattern will manifest itself due to the interaction of wakes and potential fields of each row. This will create additional spatial waveforms with mode orders given by the sum and difference of the blade/vane counts. For example the primary difference order would be $m = |B - V|$. By purely kinematic arguments, this interaction pattern must rotate circumferentially at a Mach number M_i quite distinct from the blade speed M_{blade} , given by Equation 5.4:

$$M_i = M_{blade} \frac{B}{|B - V|} \quad (5.4)$$

This implies that the rotor-stator interaction wave will rotate at a different speed from the shaft, to the extent that the interaction mode may become cut-on if M_i becomes sonic, whilst the spinning pressure field associated with the rotor alone remains cut-off (for $M_{blade} < 1$). For this reason certain rotor/stator blade count combinations are undesirable from a noise transmission point of view even in a relatively low-speed compressor. In this situation the interaction mode will be detected on a stationary pressure transducer at the rotor blade-passing frequency.

Returning now to the case of stall induced noise, a similar interaction pattern may be generated for a case of B blades and V stall cells, which may also potentially be cut-on should the interaction wave have sufficiently high Mach number. However, in the case of rotating stall, Equation 5.4 must be modified because the stall cells also rotate relative to the stationary observer. This can easily be accommodated by writing Equation 5.4 for the

stall-rotor interaction in a frame of reference moving with the stall cell, such that the situation becomes kinematically identical to a rotor/stator interaction. The equation for the circumferential Mach number of the *stall-rotor* interaction pattern M_{istall} , in the stationary frame is then:

$$M_{istall} = M_{blade} \frac{B - \frac{V_{stall}}{U} V}{|B - V|} \quad (5.5)$$

The experimental data presented earlier in this section shows the presence of a pattern of 28 stall cells rotating at 63% of shaft speed. The corresponding number of rotor blade is 34, giving rise to a series of interaction modes of which the primary difference order will be 6. Using these numbers in Equation 5.5 gives a propagation speed M_{istall} for the 6-lobe stall-rotor interaction of $2.73M_{blade}$. Moreover, returning to Equation 3.3, it can be confirmed that this stall-rotor interaction mode must be the reason for the additional 16.47EO signal also seen in Figure 5.9. This implies that whilst rotor-stator interactions always appear to a stationary observer at the rotor blade passing frequency, rotor-stall interactions always appear at a frequency given by the difference between blade passing and stall passing frequency.

This analysis therefore confirms that the signals seen in Figure 5.9 are due to a pattern of 28 stall cells moving at $0.63U$ (or $0.63M_{blade}$), which periodically “beat” with the rotor to create a third waveform of 6 lobes which must rotate at $2.73M_{blade}$. As the rotor tip operates close to sonic at design speed, the stall-rotor interaction pattern must be cut-on and will propagate upstream, potentially as a farfield noise source. The stall pattern itself, however, rotates below sonic velocity and must be cut-off, decaying along the duct.

This behaviour can be confirmed by visualising the behaviour on the PX pressure transducer. This is shown in Figure 5.10 across exactly the same range shown on the nearfield P1 location (Figure 5.9). The 16.47EO peak, due to the interaction tone can still be seen very clearly in the signal some 2 tip diameters upstream of the source, confirming that this interaction is cut-on. By contrast the 17.53EO peak (which is stronger in the nearfield

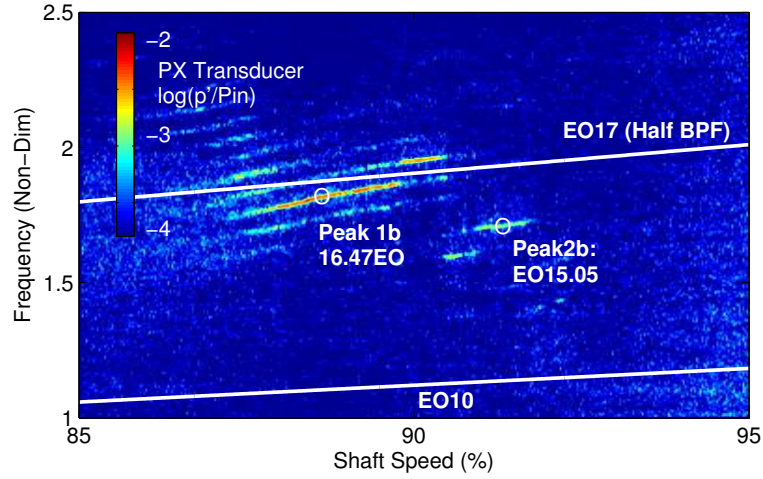


Figure 5.10: Case F - analysis of PX kulite spectra showing which tones propagate to the farfield, for contrast with “nearfield” location P1 in Figure 5.9

location) due to the stall pattern alone is no longer visible, which confirms that this disturbance is cut-off.

One further point concerns the research of Kameier and Neise[23], who reported similar noise propagation from a rotor operating with large tip clearances. It is emphasised that for the test case presented in this thesis, this stall induced noise propagation is driven largely by the VSV settings (offloading Rotor 1 very clearly suppresses the upstream noise level as evident in Figure 4.9). Chapter 6 will show that inviscid radial equilibrium effects are the primary reason for Rotor 1 becoming stalled, which implies that stall induced noise is possible even at small tip clearance levels.

5.2.4 Stall induced vibration

Turning attention back to Figure 5.9, the features labeled ‘Peak 2a and Peak 2b’ are now considered which occur at higher speeds, just above 90%. These may be aligned with equivalent behaviour observed on the rotating strain gauge in Figure 5.9(b) to calculate that this signal is due to a pattern of 31 stall cells rotating at 61% of shaft speed. Following the same logic presented in the previous section, this must interact with the 34 blade rotor

to create a third waveform of 3 lobes which must rotate at $5.02M_{blade}$ and will also be cut-on and propagate to the farfield, which can be confirmed by checking the farfield pressure transducer in Figure 5.10. This shows the presence of ‘peak 2b’ due to the cut-on interaction wave.

In this situation a further interesting observation can be made in Figure 5.9(b) on the Rotor 1 strain gauge. Below 90% speed the 28 cell stall pattern forces the rotor at a frequency of $10.47EO$ and has significant frequency separation relative to the second torsion (2T) mode, which is also highlighted. Physically this results in low stresses at this speed. As the compressor is accelerated beyond 90% speed, however, the behaviour changes considerably and the stall activity shifts spontaneously from this off-resonant condition, to the point that the stall bands coincide exactly with the vibration mode. Two stall bands can be seen to intersect the vibration mode corresponding to 32 and 31 cell stall patterns.

The physical interpretation of this behaviour, which is shown more extensively in chapter 8, is that below 90% speed the 28 stall cell pattern is a purely aerodynamic phenomenon which occurs at a frequency well away from the vibration mode. Moving to higher speed, however, causes the stall phenomenon to suddenly “lock-on” to the vibration mode once the stall and vibration frequencies become sufficiently close. In this case the stall pattern naturally adjusts the number of cells to maintain this aeroelastic coupling at a constant frequency. In chapter 8, this behaviour is shown to exist by combining the responses for different VSV configurations together, which allows considerable further insight to be obtained.

5.3 Summary of experimental observations

This chapter aimed to build on the observations from chapter 4 - which showed that this variable geometry compressor exhibits two distinct families of stalling behaviour. Deliberate stall of the Stator 1/Rotor 2 block resulted in unsteady activity observed at frequencies below $5EO$ on the stationary pressure transducer, whilst deliberate stall of Rotor 1 resulted in activity across a wider range of frequencies above $5EO$.

By studying the most extreme cases of each of these stall families, the physical behaviour giving rise to these signals is better understood. The most extreme Stator 1/Rotor 2 stall behaviour was found to be due to 4 to 6 stall cells, with the precise cell count changing spontaneously as the compressor was accelerated. Comparison of spectral data taken from a stator strain gauge alongside a nearby casing mounted pressure transducer showed clearly that activity persists on the Stator 1 vane surface even when the casing kulite becomes silent - this implies that this family of rotating stall is confined to the hub, which will be investigated in much more detail in the numerical study in chapters 6 and 7.

The higher frequency stall associated with Rotor 1 appeared considerably more complex and consisted of multiple frequency bands in the pressure transducer spectrum, before the signal broke down into broadband activity. This was attributed to a spatially and temporally non-uniform pattern of 13-16 cells that periodically breaks up and reforms.

A further point of interest concerned the point at which upstream noise propagation and non-integral vibration were observed, when Rotor 1 becomes stalled. A mechanism was described whereby the stall cell disturbance itself was acoustically cut-off and did not propagate upstream. However, a rotor-stall interaction mode (similar to a rotor-stator interaction) was found to propagate circumferentially at sufficiently high Mach number to act as a cut-on acoustic source which transmitted noise far upstream. This mechanism was confirmed by the use of pressure transducers mounted several tip diameters upstream of the front stage rotor.

In the next chapter, the steady state RANS CFD predictions are presented and compared to steady state experimental data. These models are then used to provide further insight into the stall phenomena observed in this experiment.

6 Steady state CFD analysis

Having introduced the test vehicle and experimental results, this chapter proceeds to explore the steady state behaviour obtained from simulations using the single passage CFD model, which were performed after the experiment. Firstly, the configurations of interest chosen for simulation are discussed, followed by a comparison of the numerical results to steady state experimental data taken from the datum VSV schedule (Case A). This reveals flow separations in the front stages of the compressor which can be best understood and explained using the 2D streamline curvature model.

6.1 Test configurations chosen for simulation

The experimental results presented in chapter 4 showed how, by operating the compressor with the VSVs adjusted in a manner to deliberately provoke stalled flow in the front stages, unsteady behaviour was observed on the pressure transducers associated with Rotor 1 stall and Stator 1/Rotor 2 stall. This behaviour appeared at frequencies above and below 5EO for the Rotor 1 and Stator 1/Rotor 2 stall families respectively. Chapter 5 then presented a more detailed analysis of the most extreme cases (D and G) to explore the physical behaviour giving rise to these different signals.

For the purposes of numerical simulation, the configurations of interest have been listed in Table 6.1. Analysis of Case A has been undertaken and both the experimental data and numerical results in this chapter show that this configuration has regions of stalled flow in the endwall regions of the front stages. A further configuration referred to as Case “Z” was also analysed, for which the Stator 2 vane was opened by 5° relative to Case A. The 1D analysis in chapter 4 showed how the Stator 2 angle could be

adjusted to axially rematch the compressor, which moves operating point of the upstream stages along their characteristics. This makes Case Z ideal for comparison with other cases as a completely stall free solution, as will be confirmed in a later section. Finally, the most extreme VSV settings giving rise to the Stator 1/Rotor 2 and Rotor 1 rotating stall families were analysed (Cases D and G). However, due to problems with obtaining a stable solution for Case D, the VSV offsets applied were only 80% of those applied during the experiment. To avoid confusion, therefore, this configuration will be referred to herein as Case “D*”.

Configuration	IGV	S1	S2	Predicted/observed effect (vs. datum)
A	0	0	0	Datum
D*	+3.6	-2.8	+3.6	Severe S1&R2 stall (<5EO)
G	-4.5	+3.5	+3.5	Severe R1 stall (>5EO)
Z	0	0	-5.0	Off-load R1 and S1/R2

Table 6.1: Selection of VSV angle offsets applied relative to Datum schedule Case A.

6.2 Comparison to experiment - stage matching

The Case A VSV schedule was first selected for simulation using the steady state single passage domain (shown in chapter 3, Figure 3.11). Solutions were obtained at 85% speed at several points on the overall compressor map by progressively closing the exit throttle. The unsteady experimental results presented in the previous two chapters show this speed to be an appropriate condition to warrant further investigation. Figure 6.1 shows the overall characteristics calculated with this model compared to experimental data also obtained for the Case A VSV schedule at 85% speed. To confirm the mesh resolution was adequate, a further simulation on the working line is shown with the mesh density for Rotor 1, Stator 1 and Rotor 2 increased by a factor of 2 in the spanwise direction, as these bladerows exhibited stalled flow in the endwall regions. The corresponding data obtained from the 1D meanline model is also included for completeness. A further CFD simulation is also shown for Case Z in which the Stator 2 vane opened by 5° , this has the effect in Figure 6.1 of pulling more flow through the compressor inlet.

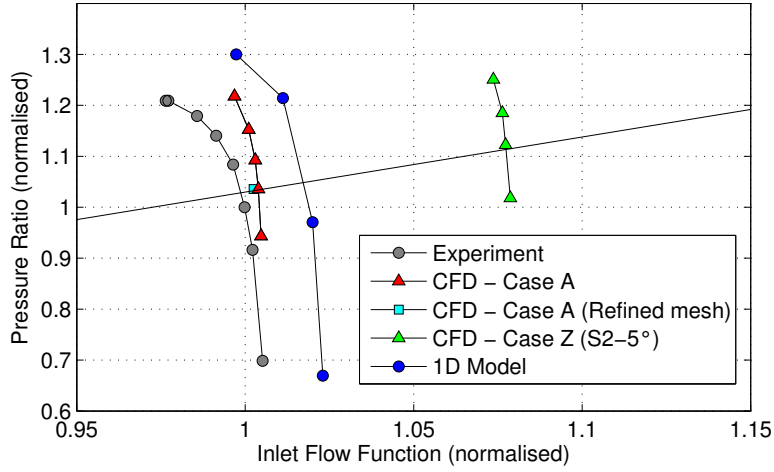


Figure 6.1: Overall performance map showing characteristics at 85% speed obtained from the experiment, steady CFD model and 1D mean-line model.

The overall level of agreement on inlet flow function at this speed, between the experiment and both models, is satisfactory and suggests that both the 1D and 3D models offer credible predictions. The inlet mass flow function predicted by the CFD model is within 1% of the experiment and this is insensitive to the mesh density. The overall characteristic shape and surge point obtained from the steady CFD simulation also show reasonable agreement with the experiment, although the CFD is clearly more upright approaching surge, suggesting that further modelling improvements may be required to predict accurately the absolute surge boundary. However, attention is restricted in this thesis to simulation of behaviour at the working line condition, as the VSVs are adjusted to trigger *local* instability confined to the front stages.

Figure 6.2 shows the corresponding block characteristics for the front 4 stages of the compressor, each block comprising of a rotor (1 to 4) and its *upstream* stator vane. This is consistent with the steady measurements which were always taken at stator leading edge, where the simulations have also been pitch-wise averaged at the same axial locations. From Figure 6.2 it is again noted that both the 1D and 3D models for the Case A VSV schedule show a good match to the corresponding experimental data. The effect of deliberate Stator 2 closure, discussed in chapter 4 (section 4.1),

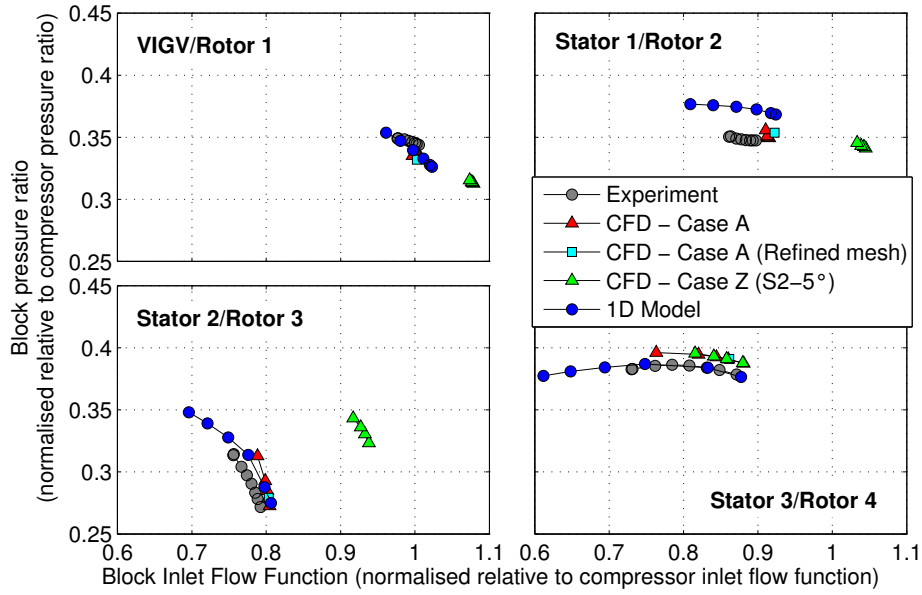


Figure 6.2: Block characteristics obtained from the experiment, steady CFD model and 1D meanline model.

can also be seen in the Stator 2/Rotor 3 block in both the simulations and experiment, which operates on a near vertical characteristic. This has significant implications for the upstream stages for which the operating point effectively becomes insensitive to the exit throttle as their exit flow function becomes fixed by the near constant Stator 2/Rotor 3 inlet flow function. Consequently, the front two blocks experience virtually no change in their inlet flow function as the whole compressor is throttled from the working line up to its surge point.

With the Stator 2/Rotor 3 inlet flow function controlling the operating point of the front two stages, it follows that the setting angle of Stator 2 has a powerful effect, which can clearly be seen from the CFD simulation performed with this vane opened by 5° (Case Z). This has the effect of significantly increasing the Rotor 3 pressure ratio. The mass flow function at entry to Stator 2/Rotor 3 also increases to achieve a constant massflow function demanded by the downstream block, for which the Cases A and Z characteristics overlay almost exactly. Stated another way, with Stator 2 opened the Stator 2/Rotor 3 block swallows more flow at its inlet, and compresses it more to deliver the same flow function at entry to the down-

stream block (based on Equation 2.5). This increase in Stator 2/Rotor 3 inlet flow function thus has a very significant effect on the front stages which are pulled down their characteristics and will tend to move away from stall. The next section shows the measured and computed flowfield at these conditions and confirms that the datum Case A setting has significant regions of separated flow, whilst the corresponding Case Z simulation is stall free.

6.3 Comparison of models to experiment - radial profiles

Figure 6.3 shows the radial profiles obtained from the experiment using total pressure probes mounted on the stator 1-3 leading edges. These are shown with the equivalent operating point from the CFD simulation, and are all at working line conditions for the datum Case A VSV schedule at 85% speed, for both the standard and refined meshes. At all locations there is a good agreement between the model and the measured profiles, confirming that the key features of the flow in the front stages of this compressor are adequately captured and remain mesh independent. A distinct dip at Stator 2 leading edge at around 20% span is clearly visible in both the experimental and CFD profiles. This is indicative of a region of separation, which can be visualised more clearly using contours of entropy on the periodic surface at mid blade pitch shown in Figure 6.4(a). The contours show that this feature is consistent with a region of separated flow in the Stator 1/Rotor 2 hub region. Figure 6.4(a) also shows clearly the presence of another region of separated flow in the Rotor 1 tip region.

The separations at Rotor 1 tip and Stator 1/Rotor 2 hub seen in Figure 6.4(a) can firstly be understood from a 1D point of view by considering the mechanism described in chapter 4. For “Case A” the Stator 2 vane has been deliberately closed down to provoke stalling behaviour in the front stages. The stagewise characteristics in Figure 6.2 confirm that, relative to Case A, opening Stator 2 will offload the front 2 blocks and move both away from stall. The equivalent entropy contours for this case are therefore shown in Figure 6.4(b) where it can be seen that these regions of separation are completely eliminated by a 5° open offset to the Stator 2 angle.

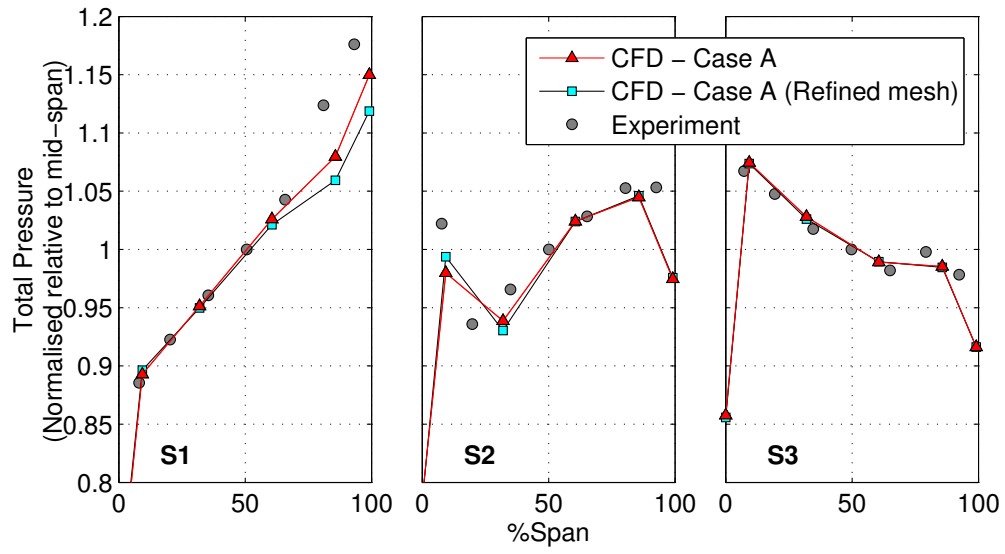


Figure 6.3: Stator 1-3 leading edge radial total pressure profile comparison between steady single passage simulation and experiment for Case 'A' at 85% speed at working line conditions. A refined mesh calculation is also shown.

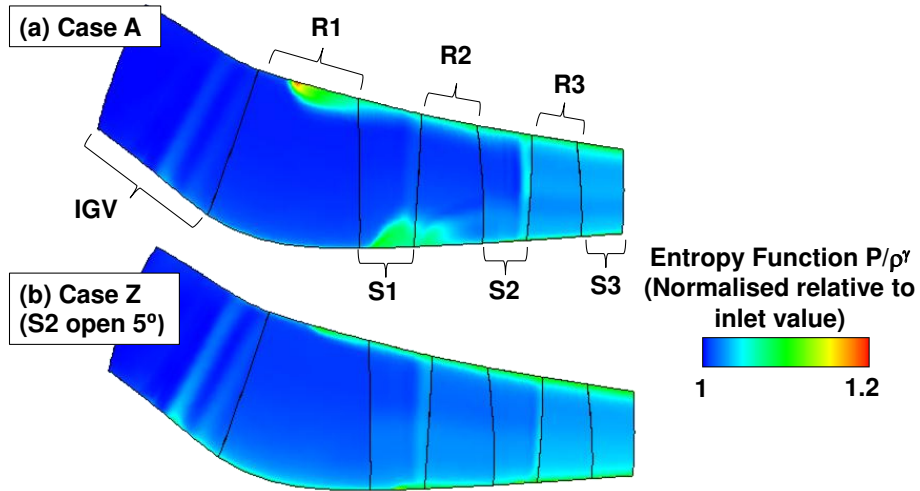


Figure 6.4: Contours of constant entropy on a mid-passage surface for the steady single passage solutions (a)Case A and (b) Case Z. Both at 85% speed, working line condition.

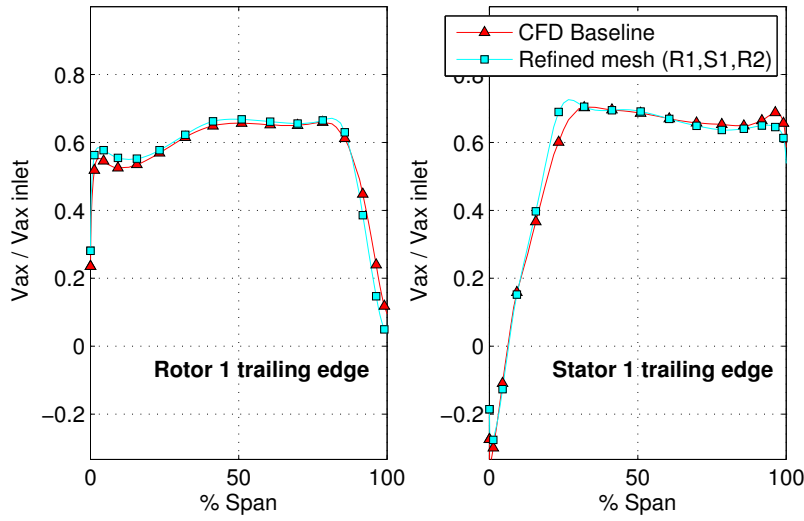


Figure 6.5: Comparison of datum and refined meshes - radial profiles of axial velocity at Rotor 1 and Stator 1 trailing edge

Whilst the one-dimensional effect of Stator 2 adjustment can be understood by considering operation on the block characteristics in Figure 6.2, this is not sufficient to explain the complex radial behaviour seen in Figure 6.4(a), with both the Rotor 1 tip and Stator 1/Rotor 2 hub region appearing to stall simultaneously. Comparison to the experimental data in Figure 6.3 also confirms that this simulation is credible and not likely to be a modelling anomaly. The next section explains how this can arise from a fundamental viewpoint using inviscid radial equilibrium considerations.

To further confirm that the simulations were not sensitive to mesh resolution, Figure 6.5 shows an additional comparison of the datum and refined meshes, where the refinement was made to Rotor 1, Stator 1 and Rotor 2 to twice the datum number of radial grid points. This shows spanwise profiles of axial velocity at Rotor 1 and Stator 1 trailing edge. At each location the simulations for both meshes are closely aligned in terms of profile shape, suggesting that the chosen mesh settings are robust. In chapter 7 this is assumed to also remain true for the full annulus unsteady cases, with the caveat that the rotating stall structures forming in the simulation remain of greater lengthscale than a single blade passage.

6.4 Inviscid radial equilibrium effects

Building upon the steady state numerical simulations presented in the previous section, it is logical to question how a flowfield can exist in which both Rotor 1 tip and Stator 1 hub can become stalled. It has already been shown, both through 1D analysis in chapter 4 and steady 3D analysis earlier in this chapter how stall in the front stages may be provoked by closure of Stator 2 at part-speed. This section therefore demonstrates how the behaviour in Figure 6.4 can be understood by considering 2D radial equilibrium effects.

It is important to recognise that the environment within which this flowfield occurs is not at all typical of a standard low-speed laboratory rig arrangement commonly presented in the academic literature. In a high-speed variable geometry compressor the Variable Stator Vanes (VSVs) close considerably at part-speed conditions, with IGV closures of 40° quite typical[6], as already discussed in chapter 4. At these conditions the IGV will impart a significant level of bulk swirl into the flow at Rotor 1 inlet, which will cause an inviscid spanwise redistribution of flow to maintain radial equilibrium. The swirl imparted by this IGV closure results in a strong hub-low spanwise static pressure gradient at Rotor 1 inlet, causing flow to be redistributed radially with the axial velocity becoming considerably higher at the hub than at the tip. Stated another way, the closure of the IGV is very effective at protecting the hub of Rotor 1, but leaves the tip exposed with very little change in incidence. When this two-dimensional effect is combined with the throttling effect of Stator 2 closure (as discussed earlier), the tip incidence naturally rises to cause the tip to stall.

The Stator 1 hub stalling behaviour also results from the IGV closure. The strong hub-low spanwise static pressure gradient imparted at Rotor 1 inlet, due to the increased swirl, remains at Rotor 1 exit and in fact becomes more pronounced as the absolute swirl naturally increases further across the rotor. It is left for the Stator 1 vane, which reduces swirl, to flatten this static pressure profile. This naturally loads up the hub to the point that the diffusion reaches levels that the suction surface boundary layer flow cannot tolerate. This causes the Stator 1 hub to also stall.

This concept can be explored in more detail using the two-dimensional throughflow method introduced briefly in chapter 3. Using this modelling technique can offer significant insight, as whilst viscous effects (total pressure loss/entropy generation, deviation and blockage) are represented in the model, they can be held constant irrespective of the compressor operating point or VSV settings. For this reason the physical mechanisms discussed below are considered to be largely inviscid and driven by radial equilibrium effects. The model thus provides a useful insight into the fundamental physics governing the behaviour of the front stages of a variable geometry compressor.

A model was therefore created for the front of the compressor up to and including Stator 1 (recall Figure 3.10), which is sufficient to explore this radial equilibrium effect and isolate it from 1D matching changes due to the downstream stages. Streamline curvature simulations were performed at both design speed and 85% speed, with the 85% speed model repeated several times with the imposed IGV exit gas angle adjusted to represent different IGV settings - from zero closure (i.e. no change relative to design speed) to 43° closure. By the same process, the Stator 1 exit gas angle was simultaneously adjusted to represent Stator 1 closure equal to two thirds of the physical IGV restagger (i.e. to mimic a typical ganged VSV schedule).

6.4.1 One-dimensional ideas

The behaviour of the compressor front stage (Rotor 1 and Stator 1) throughflow model is presented in the traditional work coefficient ($\Delta H/U^2$) against inlet flow coefficient (V_x/U) view in Figure 6.6. When viewed in this manner, it is seen how the operating point with a fixed IGV is effectively speed independent as both the 100% and 85% speed points, with the same flow function at stage 1 exit, have very similar operating points.¹

As the IGV closes, however, the operating point moves down and to the left, consistent with consideration of simple velocity triangles. Equation 6.1 (based on velocity triangles[6]) shows how, neglecting viscous effects, the

¹ It is important to recall that for reasons described in Section 2.5.1, operation with a fixed IGV is not practical in a multistage context as the effect of the downstream stages would cause the front stage to rise up its characteristic and stall at part-speed.

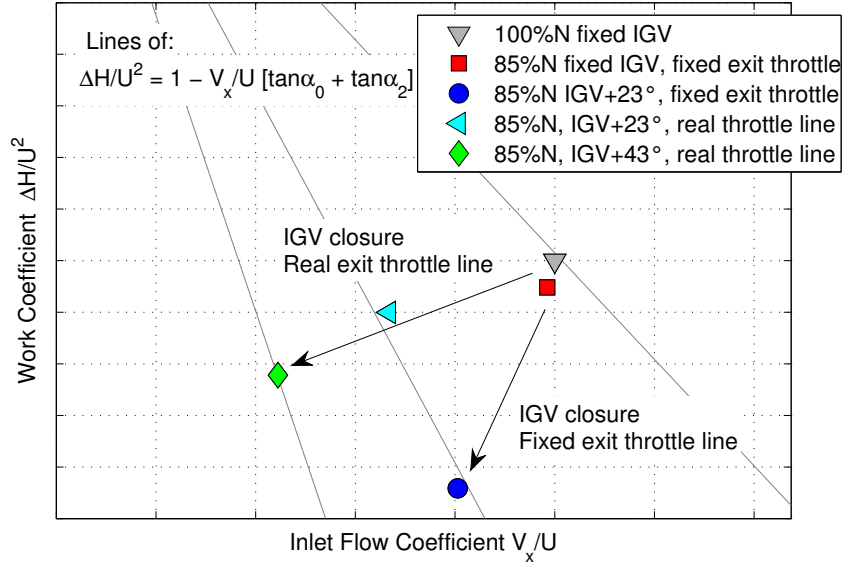


Figure 6.6: Throughflow model of front stage only, with operating points at 100% speed and 85% . IGV setting angles from 0-43° modelled at 85% speed.

relationship between Rotor 1 inlet flow coefficient and work coefficient is uniquely defined by the absolute flow angle at IGV exit (α_0) and the relative flow angle at Rotor 1 exit (α_2), as were defined in Figure 2.1.

$$\frac{\Delta H}{U^2} = 1 - \frac{V_x}{U} (\tan\alpha_0 + \tan\alpha_2) \quad (6.1)$$

Equation 6.1 gives rise to ‘idealised characteristics’ on Figure 6.6 which are straight lines all converging to a work coefficient of 1 at zero flow. Again, neglecting viscous effects of deviation, it can be assumed that the Rotor 1 exit relative gas angle (α_2) remains fixed, whilst the IGV absolute exit gas angle (α_0) will rise in line with the closure of the IGV. This results in a movement of the operating point to lower flow coefficient and onto a more upright characteristic - which subsequently also helps stabilise the compressor at part-speed conditions and is fundamental to the benefit offered by variable geometry from an overall compressor stability point of view.

It is also important to isolate the effect of the IGV closure from the effect of the downstream stages at part-speed, which, as discussed earlier is driven strongly by the closure of Stator 2. This is also visualised in Figure 6.6 by showing how the operating point moves as the IGV is closed down with two different exit flow conditions. One assumes that the downstream stages have no throttling effect, as if a choked nozzle was applied at the exit to the front stage. The other accounts for the exit flow change along the real throttle line for the front stage (which can be derived from experimental data from the Case A schedule). For the case with the IGV set 23° closed, it can be confirmed that the operating points both lie approximately on the same characteristic - where the compressor really operates at the higher operating point when on the Case A VSV schedule.

Therefore, from a 1D point of view, the IGV closure is able to cause a significant reduction in the compressor inlet axial velocity, quite independent of the compressor speed and moves the operating point onto a progressively more vertical characteristic at lower flow coefficient. Downstream throttling effects, such as closing the exit nozzle, or more significantly, closing Stator 2, will move the operating point *up* this characteristic to a higher work coefficient.

6.4.2 Axisymmetric throughflow ideas

Focusing now on the 2D implications of operating with the IGV (and downstream VSVs) closed down significant amounts, the basic analysis that follows is based on the concept of radial equilibrium for turbomachines that can be traced back to Wu and Wolfenstein [57]. It is straightforward to show that the presence of a high swirl velocity implies also a high spanwise static pressure gradient (given by $\rho V_\theta^2/r$), which inherently links all spanwise sections. Following well establish theory ([57][6]), the assumption of radially uniform entropy and stagnation enthalpy leads to Equation 6.2, which also neglects, for now, the meridional duct curvature upstream of Rotor 1. This shows how changes in swirl velocity are inherently coupled with spanwise changes in axial velocity:

$$V_x \frac{\partial V_x}{\partial r} + V_\theta \frac{\partial V_\theta}{\partial r} + \frac{V_\theta^2}{r} = 0 \quad (6.2)$$

Assuming that closure of the IGV at a fixed speed results in a bulk increase in V_θ across the span, together with fall in the mean value of V_x (shown earlier in Figure 6.6 based on 1D reasoning), it follows from Equation 6.2 that a significant negative spanwise axial velocity gradient must also exist in order to satisfy these simple inviscid flow requirements². Therefore inviscid reasoning (with a number of simplifying assumptions) implies that closure of the IGV will re-distribute flow radially, with additional flow passing through the hub whilst flow reduces at the tip (i.e. in relative terms V_x falls at the tip and rises the hub). It is this inviscid distortion of the inlet flow (in terms of velocity and static pressure) that causes the pronounced regions of endwall separation seen in the CFD simulations, which the remainder of this section will use the 2D model to quantify.

Figure 6.7 shows the radial profiles at Rotor 1 inlet for the 5 through-flow simulations shown in Figure 6.6, where the model also accounts for the annulus curvature upstream of Rotor 1. This allows the radial equilibrium effect to be studied on the test case presented in this thesis. Within Figure 6.7, radial profiles for 4 different parameters are shown - tangential velocity ($\frac{V_\theta}{U}$, which is also normalised relative to its mid-span value at 100% speed), static pressure ($\frac{p}{p_{mid}}$ normalised by the mid-span value), flow coefficient ($\frac{V_{ax}}{U}$, also normalised relative to its mid-span value at 100% speed), together with the Rotor 1 incidence angle change relative to 100% speed.

From Figure 6.7(a), it is noted that as the IGV is closed the mean swirl velocity imparted to the flow not only rises but the shape of the profile also alters as discussed in the preceding analysis. Recalling this analysis, the requirement for radial equilibrium implies that the raised level of swirl must be accompanied by a significant change in the static pressure gradient, which is seen in Figure 6.7(b). At design speed, with the IGV in its open setting, the static pressure is lower at the casing than at the hub, which

²In reality the term $\partial V_\theta / \partial r$ also changes with IGV closure and iteration is required to obtain the correct axial velocity profile, though this is captured using the 2D model.

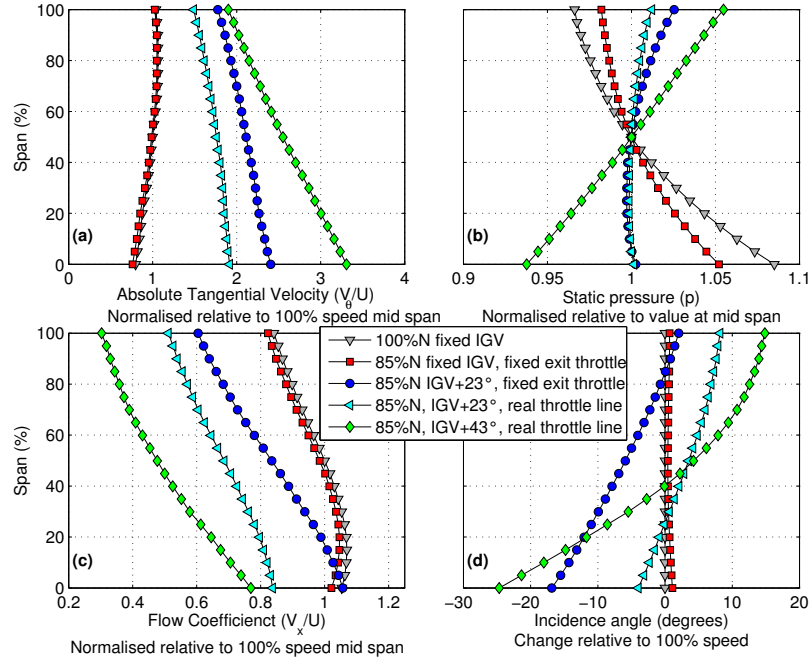


Figure 6.7: Spanwise profile from 2D simulations in Figure 6.6. Rotor 1 inlet V_θ/U , P_{static} , V_x/U (all normalised) and incidence change.

is consistent with the meridional curvature of the duct at Rotor 1 inlet. However, as the IGV is closed down, the static pressure profile shape is reversed and becomes higher at the casing due to the high level of swirl. This is true irrespective of whether the front stage is operating to a constant exit flow or on the real throttle line (the throttle line predominantly has only a one-dimensional effect).

The Rotor 1 inlet spanwise profile of axial velocity (or flow coefficient) is also shown in Figure 6.7(c) and is in line with simple radial equilibrium considerations presented earlier. The effect of dropping speed is minimal. Closure of the IGV causes, in addition to a significant drop in the mean axial velocity, a significant spanwise redistribution of flow, with more flow passing through the hub than through the casing. This has immediate implications for the stability of Rotor 1 tip which, as Figure 6.7(d) shows, now has a large incidence angle variation across the span relative to the 100% speed, fixed IGV condition. At a fixed exit flow function the effect of closing the IGV by 23° (red squares to dark blue circles in Figure 6.7(d)), is to drive the

incidence at the hub down considerably with little change at the tip. Stated another way, closure of the IGV protects the hub of Rotor 1 considerably more than it protects the tip.

Considering therefore the Rotor 1 incidence angle moving up to the real throttle line in Figure 6.7(d), at 23° IGV closure, the effect is largely one-dimensional - i.e. the incidence rises uniformly across the span. This is quite tolerable at the hub which has been “protected” by the IGV closure and remains at a negative incidence relative to the design speed level. However, the tip incidence is now increased further to levels considerably higher than at design speed. This mechanism ultimately drives the tip region to stall (the over tip leakage is likely to have an additional contribution here, but the rise in incidence being the more fundamental cause).

To summarise, therefore, the Rotor 1 tip stall is driven by two factors combining: First, the closure of the IGV provides excessive incidence protection to the hub region whilst providing very little protection to the tip - which is essentially a 2D effect. Second, the throttling effect of the downstream stages (Stator 2 over-closure in particular), causes a 1D rise in incidence across the entire span which drives the tip towards stall.

Focusing attention now on the stalling behaviour at the hub of Stator 1, Figure 6.8 shows the spanwise profile of static pressure at (a) Rotor 1 inlet, (b) Rotor 1 exit and (c) Stator 1 exit obtained from the throughflow model. It is seen that the distorted hub-low static pressure profile already present at Rotor 1 inlet when the IGV is closed down (due to the high inlet swirl) is even more pronounced at Rotor 1 exit. This occurs because the absolute swirl velocity rises across the rotor to a higher level than at inlet. Noting that Stator 1 naturally diffuses the flow back towards the axial direction (removing swirl) it is logical to expect the static pressure profile to flatten again across Stator 1, as indeed is seen in Figure 6.8(c). This implies that the static pressure rise at Stator 1 hub must be considerably larger than that at the casing.

A further observation can be made at Rotor 1 inlet in Figure 6.7(c), which showed that the flow coefficient is higher at the hub than at the tip. This implies that the Rotor 1 hub will deliver less work than the tip, by consideration of simple velocity triangles as outlined earlier (e.g. Equation 6.1), so

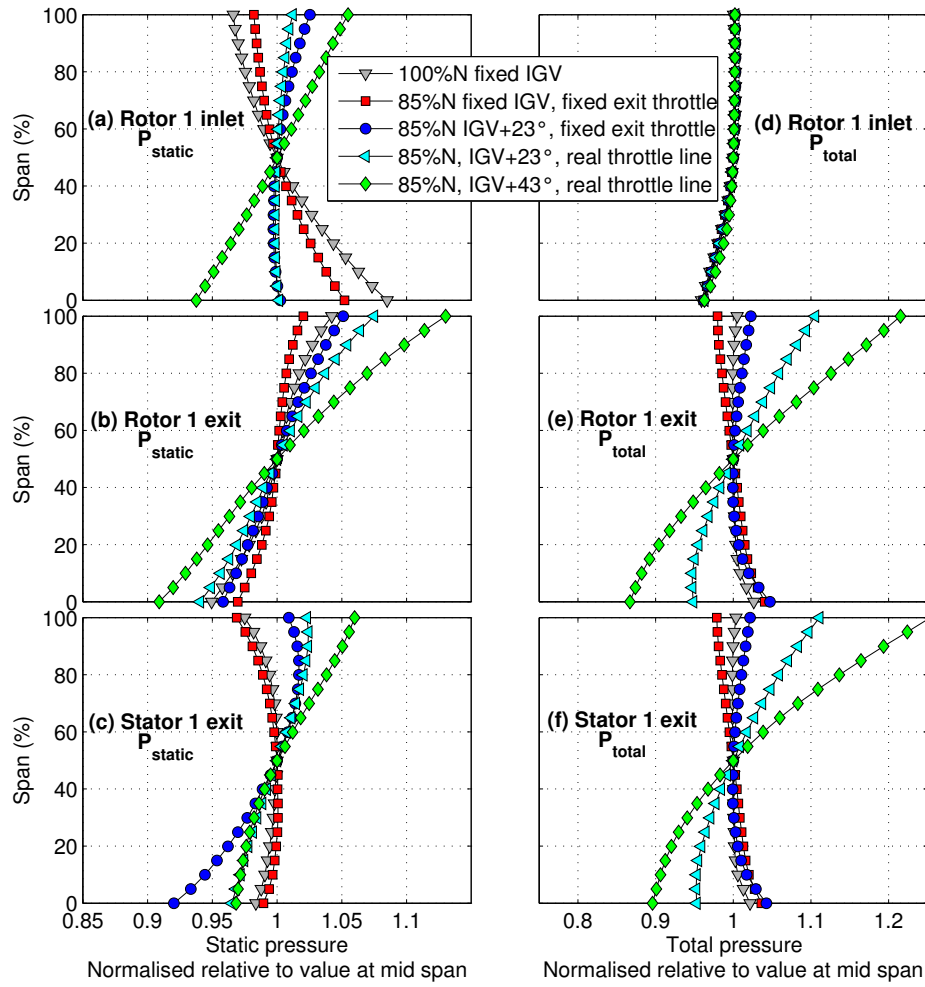


Figure 6.8: Throughflow model, radial profiles of static pressure and total pressure at R1 inlet, R1 exit and S1 exit as the IGV is closed down consistent with the points in Figure 6.6.

that a distorted hub-low total pressure profile also enters the downstream blade rows. This can be seen in Figure 6.8(d) to (f) which shows the corresponding radial profile of total pressure also at Rotor 1 inlet, Rotor 1 exit and Stator 1 exit. As the IGV is closed down, an increasingly distorted hub-low total pressure profile is seen at Rotor 1 exit, as the hub does very little pressure ratio with the IGV closed. This is also observed in the CFD simulations and experimental measurements shown earlier in Figure 6.3, which also both exhibited a hub-low total pressure profile at Stator 1 inlet, precisely due to this inviscid effect. This hub-low total pressure profile thus feeds into Stator 1 in addition to the already high hub diffusion levels discussed above. Consequently, the Stator 1 hub is overloaded as the IGV closes down, eventually becoming stalled.

In relating these inviscid mechanisms back to the observations from the RANS simulations, the propensity of the Stator 1 blades to stall will be driven by the levels of diffusion imposed on the viscous boundary layer flows on the blade and endwall surfaces. Figure 6.9 shows the Stator 1 diffusion factor as obtained from the throughflow code, diffusion factor being an appropriate parameter to quantify this effect. It is evident from Figure 6.9 that as the IGV is closed down *on the real throttle line*, the Stator 1 diffusion increases considerably at the hub. In reality, once the static pressure rise becomes sufficiently high, viscous effects will become dominant within the suction surface boundary layer. At this point the boundary layer will separate causing Stator 1 hub to stall. This also appears to be consistent with the RANS CFD simulations: at 85% speed, with the IGV closed down, the Stator 1 hub becomes separated, which is consistent with the measured radial profiles shown in Figure 6.3.

For completeness it should also be noted that the case with the IGV closed down at *fixed exit flow* results in a fall in Stator 1 diffusion factors below zero. This is simply because, as discussed earlier, Stator 1 is also closed down in this model by two-thirds of the IGV restagger, to represent a ganged VSV system. At a fixed exit flow function this means that the blade is doing progressively less turning and in this case reaches the point that it *accelerates* the flow, although this case is not of further significance to this thesis.

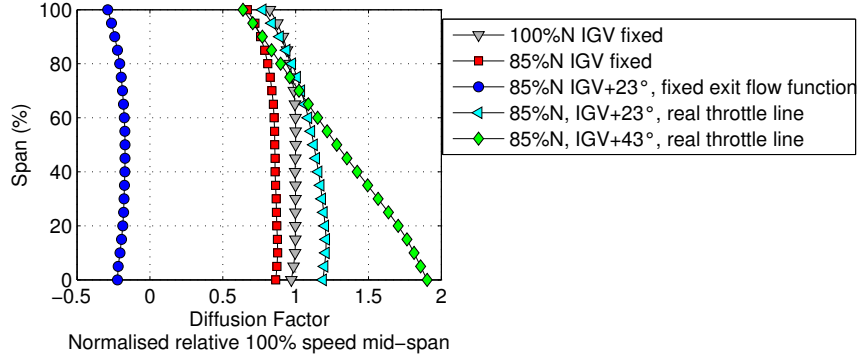


Figure 6.9: Throughflow model, radial profile of Stator 1 diffusion factor as the IGV is closed down consistent with the points in Figure 6.6.

Based on this simplistic inviscid reasoning and modelling approach, it is apparent therefore that closure of the IGV at part-speed has a powerful redistributive effect on the flow. Although the IGV closure protects Rotor 1 from a 1D point of view and hence improves the overall compressor stability, it preferentially protects the hub whilst the tip becomes stalled at a high incidence. Similarly, the distorted static pressure profile generated by the IGV closure is largely left for Stator 1 to eliminate, which has the effect of raising the amount of diffusion at the hub to levels which the blade and endwall boundary layers cannot tolerate. For this reason, it is quite possible for part-span stall to form within the variable stages to give a flow pattern similar to that in Figure 6.4(a). These inviscid effects are both common to variable geometry compressors particularly of low hub-to-tip ratio.

6.5 Concluding remarks

To summarise, therefore, this chapter first showed that the steady state simulations for this compressor appear credible in terms of stage characteristics and radial profiles. Further analysis of the solutions from this model shows the presence of two regions of separated flow at Rotor 1 tip and the Stator 1/Rotor 2 hub region. A link between these stalled regions and the observed “families” of unsteady activity observed in chapters 4 and 5 can immediately be drawn, which the next chapter seeks to build upon using unsteady simulation.

A final discussion point before completion of this chapter concerns the logical question of why the compressor does not surge in response to such severely separated flow in the front stages. Evidently in this case the compressor is heavily mismatched both in an axial sense (due to both the effect of speed and the deliberate closure of Stator 2) and in a radial sense (due to the radial equilibrium effects of the swirl imparted by IGV closure). The apparent stability of this condition can be best explained by recalling the work of Longley and Hynes [5], as discussed in chapter 2. During their experiment with a 3 stage compressor they showed how, by re-staggering closed the rear 2 stages of the compressor, the front could be forced to operate on a characteristic to the left of its peak pressure rise - i.e. in a stable but stalled condition. Testing this front stage in isolation showed that the same operating point could not be achieved. The conclusion of this experiment was that the rear 2 stages, operating on stable and strongly negatively sloped characteristics were able to support the front stage and allow it to operate in stall whilst the compressor as a whole remained stable.

Following similar logic, this 8 stage compressor behaves in a very similar manner. Whilst the front stages can become heavily stalled at part-speed local to the endwalls, the downstream stages together with the unstalled inner span of Rotor 1 remain on very stable (negatively sloped) characteristics and are able to maintain the overall stability of the compressor. For this reason, the mismatching effects discussed here force the stall to be of the localised part-span type, that persists at working line conditions. The following chapter explores this behaviour further using the unsteady model.

7 Unsteady CFD analysis

The previous chapter introduced the steady state CFD simulations for the test compressor, at several key conditions identified during the experimental study in chapters 4 and 5. The single passage models for the datum VSV configuration showed clearly the presence of a separated region close to the Rotor 1 tip together with a second region in the Stator 1/Rotor 2 hub, which may be linked with the two families of rotating stall observed at different frequencies in the experiment. Further analysis performed using the 2D streamline curvature model showed these separations to be driven by a combination of deliberate Stator 2 closure, which drives the upstream stages towards stall, together with the significant part-speed IGV closure, which imposes a strong spanwise static pressure gradient at Rotor 1 inlet. In this chapter, the steady models are expanded to full annulus sliding plane models, with the objective being to investigate how these regions of separation develop when an unsteady modelling approach is adopted.

7.1 Stator 1/Rotor 2 stall family (low cell count).

The earlier chapters showed how adjustment of the VSV settings relative to the datum configuration (specifically IGV closed, Stator 1 open and Stator 2 closed) provoked stall in the Stator 1/Rotor 2 region of the compressor. This gave rise to activity on the casing pressure transducers at a frequency below 5EO, which chapter 5 showed was due to rotating stall with 4 to 6 cells, with the cell count undergoing spontaneous transitions at certain speeds. The steady state analysis of the previous chapter suggests this is linked to stall originating in the hub region. In this section the effects of progressively pushing the Stator 1/Rotor 2 region into stall are investigated by performing unsteady simulations on Cases Z, A and D*.

An important consideration to be made here relates to the starting condition used for unsteady analysis. One logical approach would be to initially run steady single passage computations for each of the configurations of interest, which could each then be expanded to full annulus sliding plane computations and allowed to evolve in real time. This approach was rejected, as the steady starting solution for any stalled configuration would clearly be periodic around the circumference, with every blade/vane stalled in equal measure. The experimental results show that such a scenario is non-physical, as in reality rotating stall is non-periodic and has discrete sectors of the circumference with severely separated flow and others with attached flow. Any simulated transient stall inception process started from the stalled steady model would therefore have an unrealistic starting condition.

The numerical strategy adopted was therefore to initially run the full annulus unsteady model for 5 shaft revolutions with Stator 2 opened up by 5° (which is referred to as Case “Z”). This suppresses stall in the affected (front) blade rows as was shown in chapter 6 and allows a realistic unsteady solution to develop with minimal separation. This solution was then used as a starting point for analysis of the more stalled Case A, for which Stator 2 was closed back down. For brevity, the details of this unstalled starting condition are not presented here. With this starting solution obtained, Stator 2 was then adjusted back to its Case A setting and the calculation resumed. This instant is defined herein as revolution zero.

Figure 7.1 shows how the solution for Case A evolves transiently in terms of axial velocity at Stator 1 exit, over the subsequent 10 shaft revolutions from the initial closure of Stator 2 (Case Z to Case A). This starts from the unstalled initial condition. Initially, for the first half revolution, the flow is axisymmetric and the only circumferential disturbance is due to stator wakes. By 0.5 revolutions, blockage develops in the Stator 1 hub region, still in an axisymmetric manner. Subsequent to this, the blockage regions coalesce into multiple short lengthscale disturbances, rotating in the same sense as the shaft, with approximately 17 discrete disturbances detectable after 1.5 shaft revolutions. As the solution progresses these disturbances gradually merge together until the solution stabilizes naturally on a uniform pattern of 7 cells.

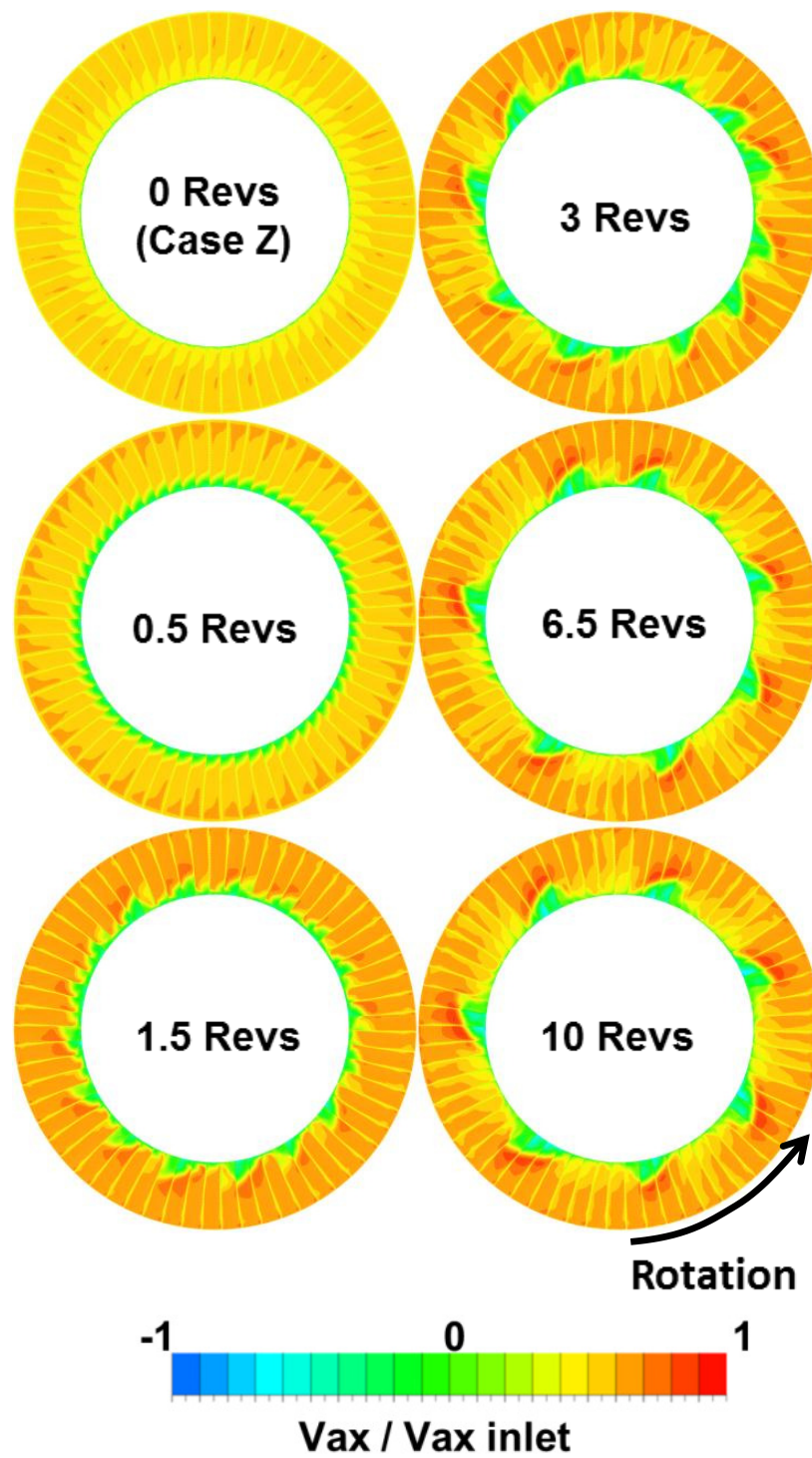


Figure 7.1: Stator 1 exit axial velocity contours computed during the transient from Case Z (unstalled) to Case A.

This transient process is also visualized in Figure 7.2 in an alternative manner. Figure 7.2 was created by extracting the axial velocity field from the solution on a circle at 10% span (around the whole circumference), also at Stator 1 trailing edge. By extracting this data at regular intervals as the solution evolved (approximately 30 times per shaft revolution), the time varying profiles of axial velocity against circumferential location (θ) can be arranged alongside each other and viewed in colour. This allows the whole process to be visualised, from the initial Stator 2 closure at revolution zero to the final stabilisation on 7 cells.

Initially, only horizontal (stationary) disturbances are present, i.e. the stator wakes. By 0.5 revolutions the vane passages fill with blockage until, by 1.5 revolutions, travelling disturbances emerge as diagonal lines. The gradual merging of these stall cells can then be observed over the next 10 revolutions until the calculation stabilizes on a 7 cell pattern, which propagates at 25% of the shaft rotational speed.

Once the Case A simulation was stable and periodic, at approximately 12.5 shaft revolutions, the variable vanes were moved again to represent Case D* (i.e. IGV closed 3.6° , Stator 1 opened 2.8° , Stator 2 closed 3.6°). Figures 7.3 and 7.4 show, in the same manner, how the 7 cell pattern reacts to the more stalled operating point. Almost instantaneously the stall cells grow in size and accelerate, as evident in the change of slope in Figure 7.3. Between revolution 14 and 15, merging of stall cells can be observed at two circumferential locations (approximately $+90^\circ$ and -90°). The calculation eventually stabilizes on a pattern of 5 cells, propagating more rapidly at 40% of shaft rotational speed. The relationship between cell size and propagation speed is left for the discussion where the results will be considered alongside experimental data.

More detailed attention is now given to the process of stall cell inception and formation in the first 2 shaft revolutions for Case A. Figures 7.1 to 7.4 clearly show rotating stall is present within the Stator 1/Rotor 2 hub region. The following section therefore seeks to understand the blade row in which these disturbances originate.

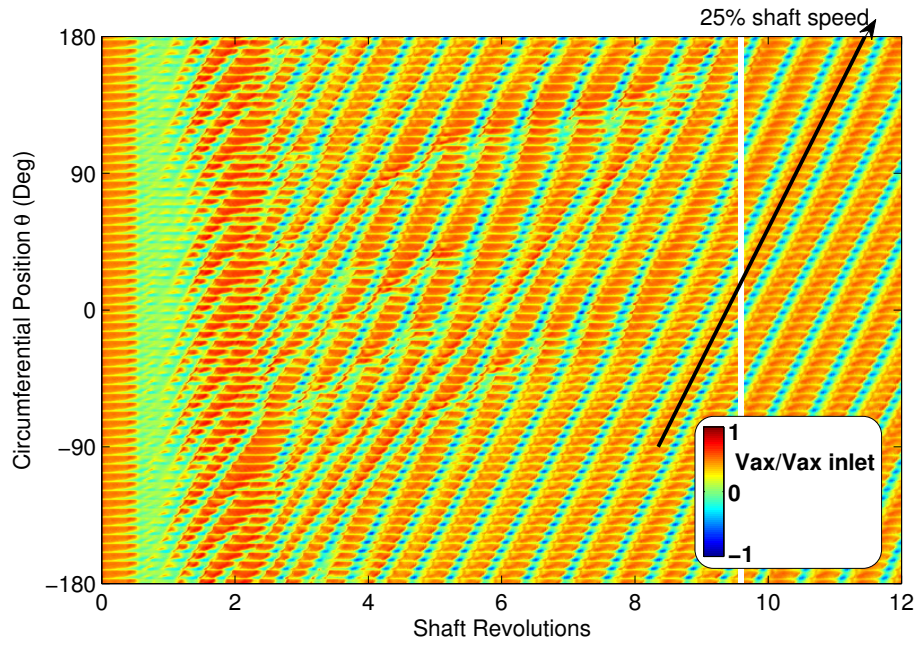


Figure 7.2: S1 exit axial velocity profile around circumference at 20% span, during calculated transient from Case Z (unstalled) to Case A.

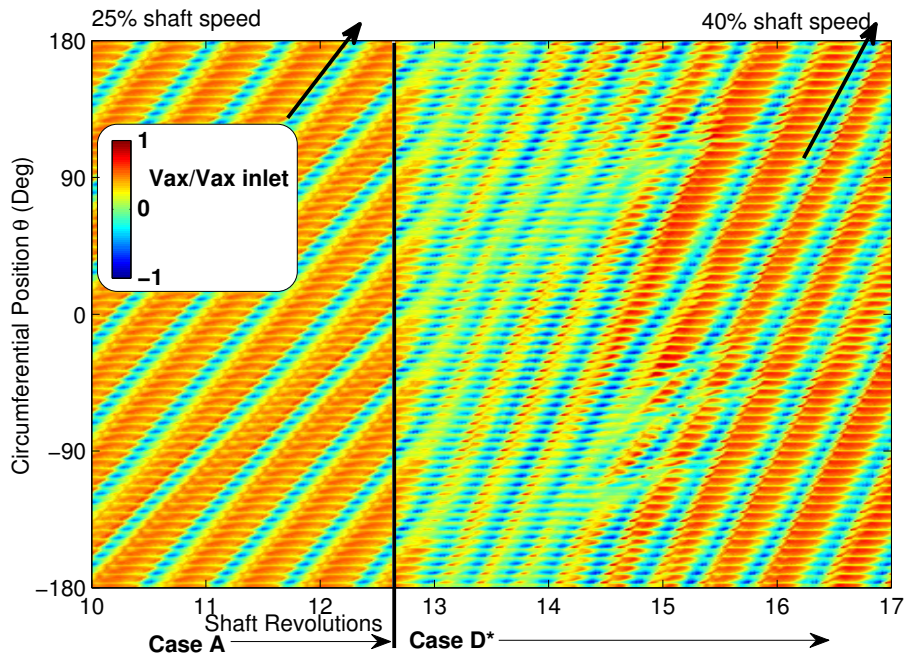


Figure 7.3: S1 exit axial velocity profile around circumference at 20% span, during calculated transient from Case A (stalled) to Case D* (severely stalled).

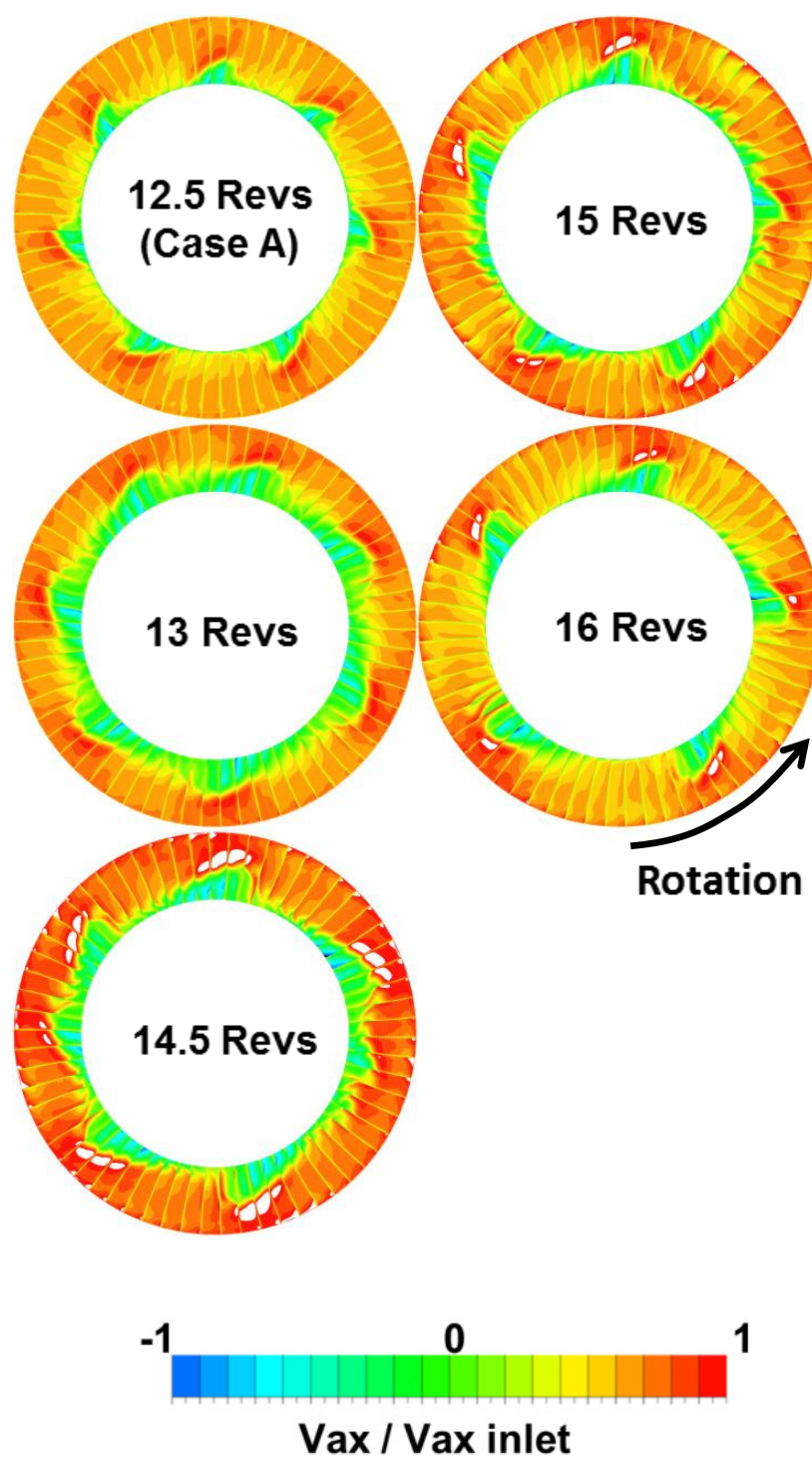


Figure 7.4: Stator 1 exit axial velocity contours computed during the transient from Case A (stalled) to Case D* (severely stalled).

7.1.1 Origin and location of stall cells

In order to gain further insight into the origin of this stall phenomenon, circumferential profiles of static pressure were extracted around the annulus from the solution at 10% span, immediately upstream of Rotor 1, Stator 1 and Rotor 2. This was repeated at several instantaneous points during the transient in Figure 7.2. These were then analyzed using a spatial Fourier decomposition to deduce the modal content within the flow. The resulting plots are shown in Figure 7.5.

The initial starting solution at revolution zero (Case Z, with Stator 2 opened 5°) has no stalled flow present, with the only visible modes due to the blades themselves and their harmonics, as labelled. By revolution 1, after the instantaneous closure of Stator 2 back to its Case A setting, activity appears upstream of Stator 1 with a mode order of approximately one third that of the stator itself, corresponding to the 17 discrete features in Figure 7.1 and 7.2. Importantly, this feature is not observed upstream of Rotor 1 or Rotor 2 it is only present upstream of Stator 1.

This suggests that circumferentially non-uniform flow structures (stall cells) are initially forming within Stator 1, with their blockage effect resulting in a circumferential variation in upstream static pressure. This is further supported by a sharp fall in the amplitude of the pressure field associated with Stator 1 itself, to half of its original (unstalled) level, implying a breakdown in the periodic flow from one blade passage to the next. By revolution 1.6 the stalling mode has grown to have the same amplitude as Stator 1 itself, remaining weak at the other locations. The solution at revolution 12 is also shown. By this time the stall signal has grown in wavelength both axially and circumferentially, with mode 7 becoming dominant at over twice the Stator 1 amplitude, clearly now also visible downstream at Rotor 2 leading edge. This implies that rotating stall initially forms locally in Stator 1, with shorter circumferential lengthscale of mode order 17. The stall cells then grow axially and circumferentially and merge to stabilize at mode order 7.

The equivalent Fourier decomposition of static pressure for Case D*, with 5 stall cells, is shown in Figure 7.6, also at Stator 1 leading edge, 10% span.

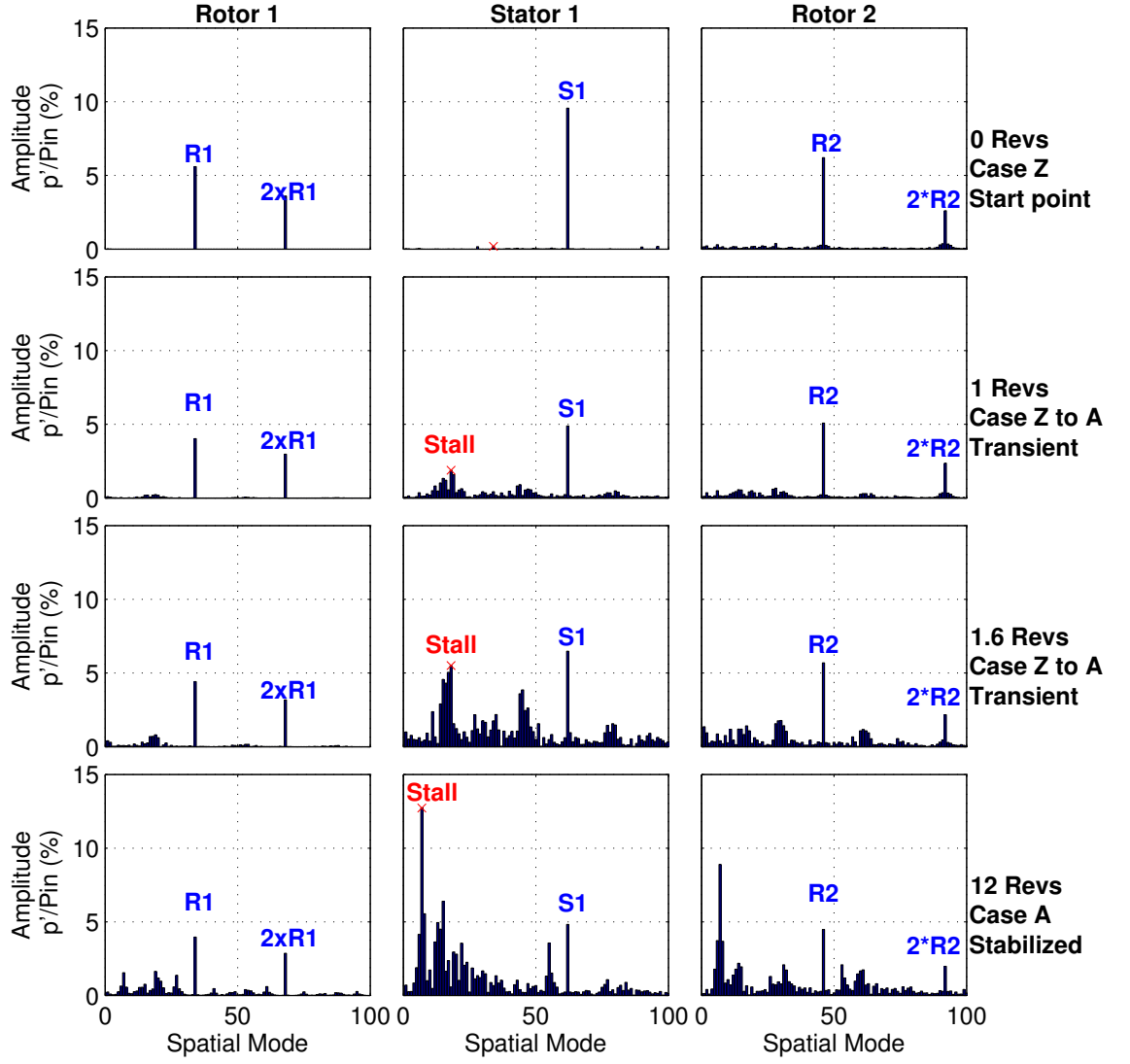


Figure 7.5: Spatial static pressure mode amplitudes at R1, S1 and R2 leading edge, 10%span, at several time points during computed transient from Case Z (unstalled) to Case A (stalled).

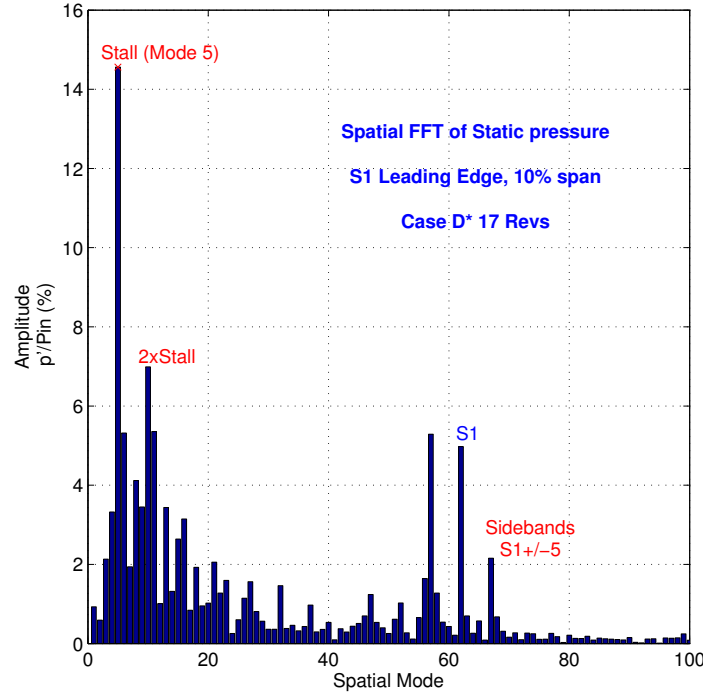


Figure 7.6: Case D* final stabilised solution, spatial static pressure mode amplitudes at Stator 1 leading edge, 10%span.

The dominant mode order is 5, consistent with 5 cells, with a first harmonic clearly visible. Modulation of the rotating stall with the Stator 1 signal is also evident in the form of sidebands.

The location of the stall cells is illustrated in the axial velocity contours in Figure 7.7(a), showing the final Case D* solution at 20% span, across Rotor 1, Stator 1 and Rotor 2. One of the five stall cells is clearly visible as a region of axially reversed flow. The cell is dominant in the Stator 1 region, with 4 passages completely filled with reverse flow. The cell appears to lose its structure across the adjacent rotors, implying that the fully developed stall cell phenomenon resides primarily within the stator.

The approximate location of the cell “trailing edge” boundary has also been added to Figure 7.7(a), noting that as the cell propagates in the circumferential direction (in the same sense as the shaft, as indicated), such that the cell trailing edge boundary is effectively a line of constant circumferential position (θ). At this location several comments can be made.

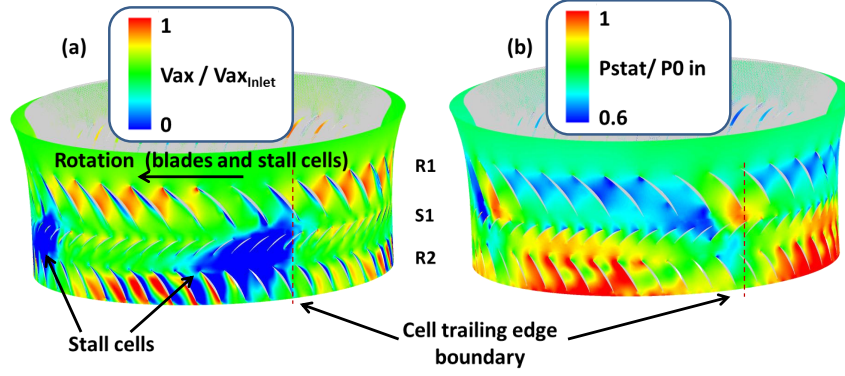


Figure 7.7: Case D*, slice through stall cells at 20% span showing R1(top), S1 and R2(bottom). (a)Axial velocity and (b)Static Pressure

First, the axial velocity shows a very abrupt transition at the trailing edge boundary between axially stationary/reversed flow (within the cell) and conventional forward flow. This transition occurs across a single blade passage. Similar behaviour can be seen in this solution when visualised as the Stator 1 exit axial velocity field in Figure 7.4, which also shows the cell has a clear asymmetric shape with an abrupt change in axial velocity across the trailing edge boundary. This is consistent with other reported observations for rotating stall cells[22][58].

Second, Figure 7.7(b) also shows the final Case D* solution at the same location in terms of static pressure, with the cell trailing edge boundary location also indicated. In this location steep static pressure gradients can be observed. Cumpsty and Greitzer[58] reported a similar static pressure discontinuity at the cell trailing edge boundary for a full-span stall case. By considering the behaviour in a frame of reference moving with the cell, such that the rotors and stators move in opposite directions across the cell boundary, the static pressure discontinuity was attributed to inertial effects resulting from abrupt changes in velocity in the blade passages as the flow immediately transitions between forward and zero flow. This observation was then used to provide an analytical for the rotating stall cell propagation speed[58].

Finally, the behaviour at the cell boundary which results in this static pressure discontinuity can be considered the main source of the periodic

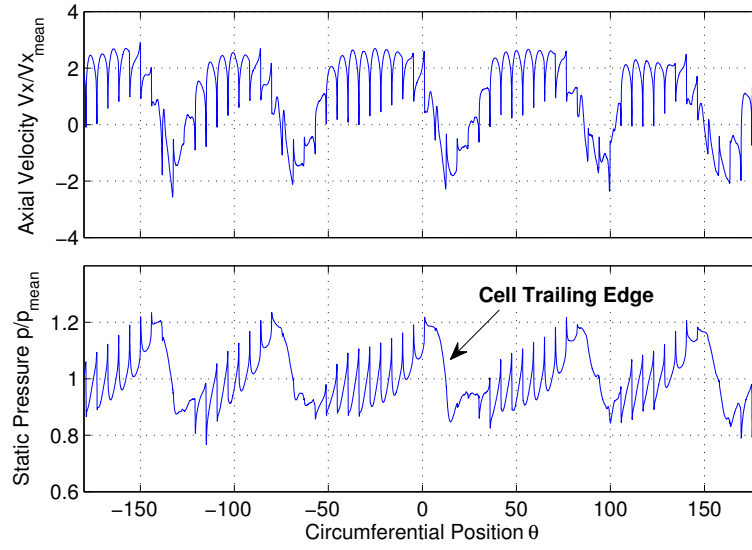


Figure 7.8: Case D* - 20% span circumferential profiles of (a) Axial velocity and (b) Static Pressure at Stator 1 leading edge.

static pressure variation seen in Figure 7.6. Stated another way, whilst the stall cell is considered a fluid dynamic disturbance with significant changes in velocity, it is the abrupt behaviour at the cell boundary gives rise to the largest static pressure perturbation.

Figure 7.8 shows the variation of axial velocity and static pressure around the circumference at Stator 1 leading edge at 20% span. In addition to the “short” wavelength features due to the blade disturbances, the 5 cells can very clearly be seen as regions of significant axial velocity deficit together with the pronounced static pressure change occurring only at the cell boundary, giving rise to a sawtooth-like waveform.

Using the spatial mode decomposition technique shown earlier for analysis of the stall inception phase, the axial and radial extent of the disturbance can be explored by determining the spatial modes at any chosen axial and radial location within the domain. It should be emphasized that a simplification is made here by considering only the static pressure perturbation induced by the stall cell, which ultimately will allow comparison with experiment. The more complex vortical and entropic modes are not considered here.

Figure 7.9 shows, for Case D* (with a 5 cell pattern in the simulation), how the amplitude of the 5th spatial Fourier mode of static pressure varies along the hub and casing at the leading edge of each row. The peak intensity clearly occurs close to the hub, upstream of Stator 1, consistent with Figures 7.7 and 7.8. The corresponding intensity at the casing is less than half the hub level. Upstream, the disturbance is attenuated rapidly across Rotor 1 hub, which, at this condition, operates locally on a negatively sloped (i.e. stable) characteristic. Recalling the 2D analysis presented chapter 6 it was shown that IGV closure significantly increases axial velocity onto the hub and drives it to negative incidence relative to its design point. Consequently, the rotor hub does not itself stall in response to the exit static pressure variation (as also seen in Figure 7.7), which prevents the stall cells from growing axially upstream into the Rotor 1 blade passages.

Downstream of this peak in the hub static pressure variation, shown in Figure 7.9, the static pressure variation is more gradually attenuated, with significant attenuation across the hub of Stator 1 itself, although the flow must still clearly have significant vortical content, as shown in the axial velocity contours in Figure 7.4 and Figure 7.7.

Further downstream the disturbance is artificially terminated in the calculation by the mixing plane at Rotor 3 exit. In order to confirm that this modeling assumption did not invalidate the findings already presented, the calculation was repeated with the mixing plane moved one stage downstream to Rotor 4 exit. The corresponding axial distribution is also shown in Figure 7.9. The solution also evolved into a pattern of 5 stall cells, implying the resulting behavior was not affected, with the resulting axial distribution showing only small differences¹.

It is also of interest to consider the static pressure variation at the casing, also shown in Figure 7.9. This arises from the redistribution of flow around the cells at the hub, locally raising the velocity towards the casing (as can be seen in Figure 7.4, where velocity is higher radially outboard of the cells) and lowering the static pressure. It appears, from Figure 7.9, that the casing peak occurs approximately one stage downstream of that at the hub.

¹A recent simulation for Case A with a full annulus domain throughout the *whole* compressor also shows no change to the final solution (which also resulted in 7 cells).

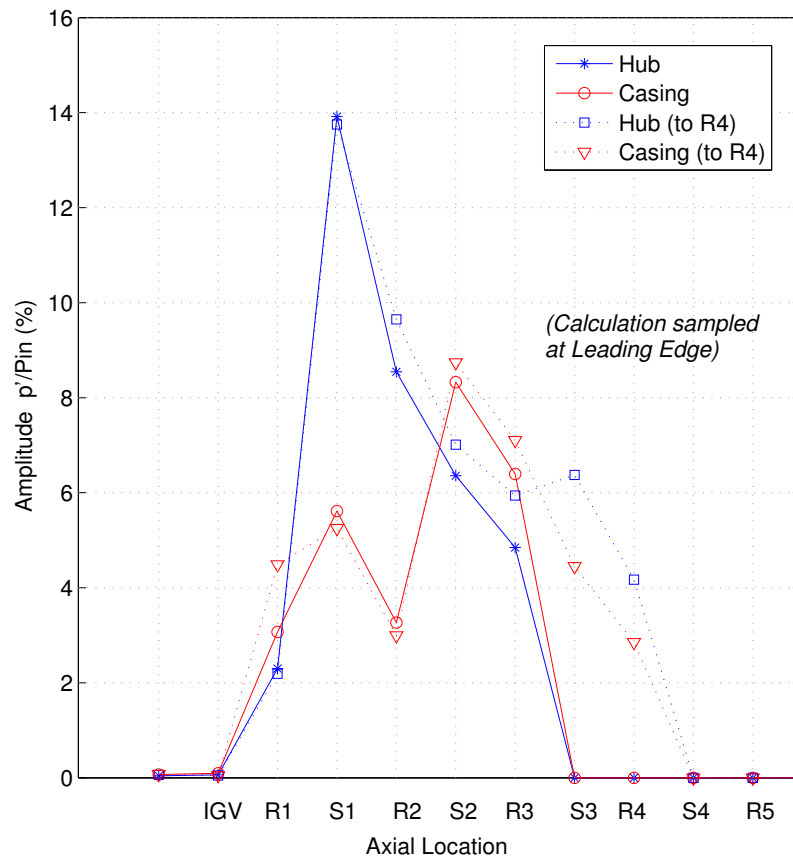


Figure 7.9: Case D* - axial distribution of mode 5 static pressure amplitude along hub and casing.

7.1.2 Comparison with experiment

It is now possible to relate this observed behavior back to the experimental data shown in chapter 5. The experimental results for Case D across the acceleration manoeuvre are repeated in Figure 7.10, in terms of the number of stall cells and propagation speed. Based on measurements taken on the casing transducers together with the rotating strain gauges it was found that at 85% speed, a 6 cell rotating stall pattern was clearly present, rotating at half of the shaft speed. At a slightly higher speed of 86%, this transitioned spontaneously into a 5 cell pattern rotating at 55% of the shaft speed. The corresponding stall cell count and propagation speed from the simulation for Case D* is also shown in Figure 7.10. Once stabilized, the CFD model clearly shows a 5 cell pattern rotating somewhat slower at 40% of shaft speed. This gives some confidence that unsteady RANS simulations are capable of predicting part-span rotating stall, although the propagation speeds are somewhat lower than measured.

The axial distribution of the disturbance obtained from the simulation is also compared with experiment in Figure 7.11, by performing modal analysis similar to Figure 7.9, now at the location of the casing pressure transducers. Experimental data is included for both 85% and 86% speeds (the former included due to being at the same speed for which the simulations were performed, the latter included as it enables a measured 5 cell disturbance to be compared directly with a simulated 5 cell disturbance albeit at slightly different shaft speeds). The axial distribution of the 5 cell disturbance obtained from the model agrees well with the experiment for the equivalent 5 cell stall measurement. Interestingly, the peak intensity clearly occurs in the Stator 2/Rotor 3 gap (at location P3), despite the stall cell itself being present at the hub one whole stage further upstream.

Returning therefore to Figure 7.10(c), the intensity of the stall phenomenon measured at the transducer locations P1-P4 (Figure 3.1) is shown and was discussed in chapter 5. At lower speed the maximum intensity of the Stator 1/Rotor 2 stall signal occurred in the “P2” location (at Stator 1 exit, shown in Figure 3.1), but moved downstream to the “P3” location (at Stator 2 exit) at higher speeds. As the Stator 2/Rotor 3 block operates well away from stall, this finding could not be explained in chapter 5 based on

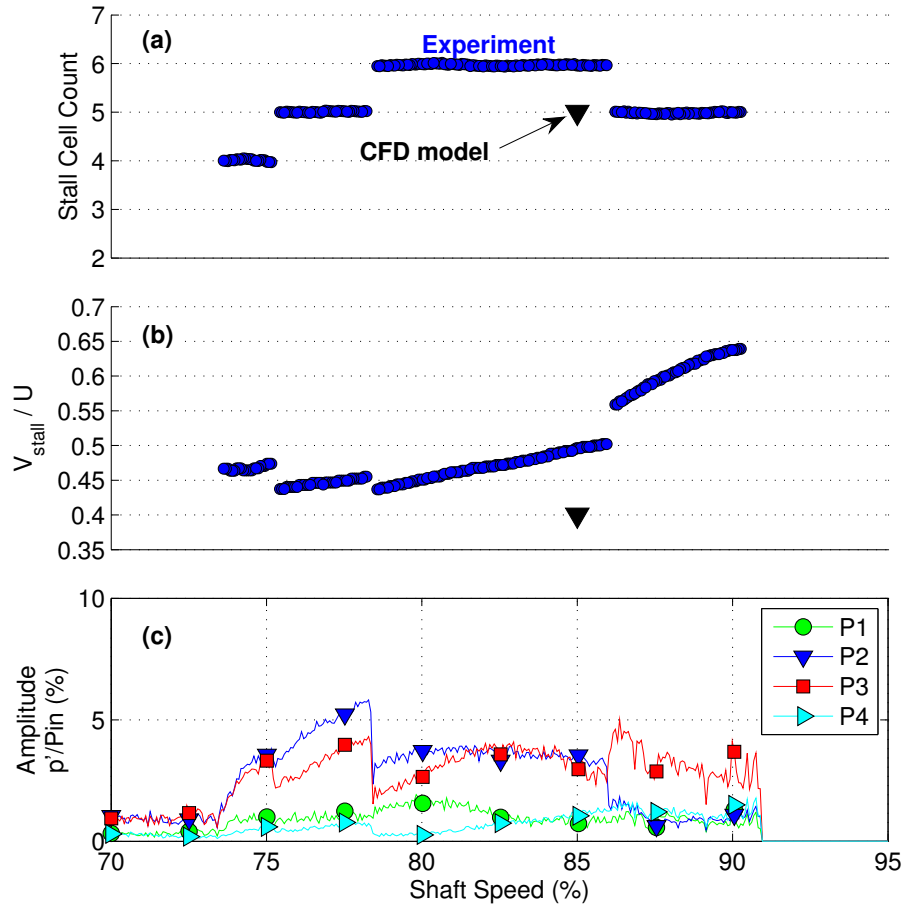


Figure 7.10: Repeat of Figure 5.2 with CFD computation for Case D* also shown: Case D - rotating stall observed in experiment through manoeuvre.

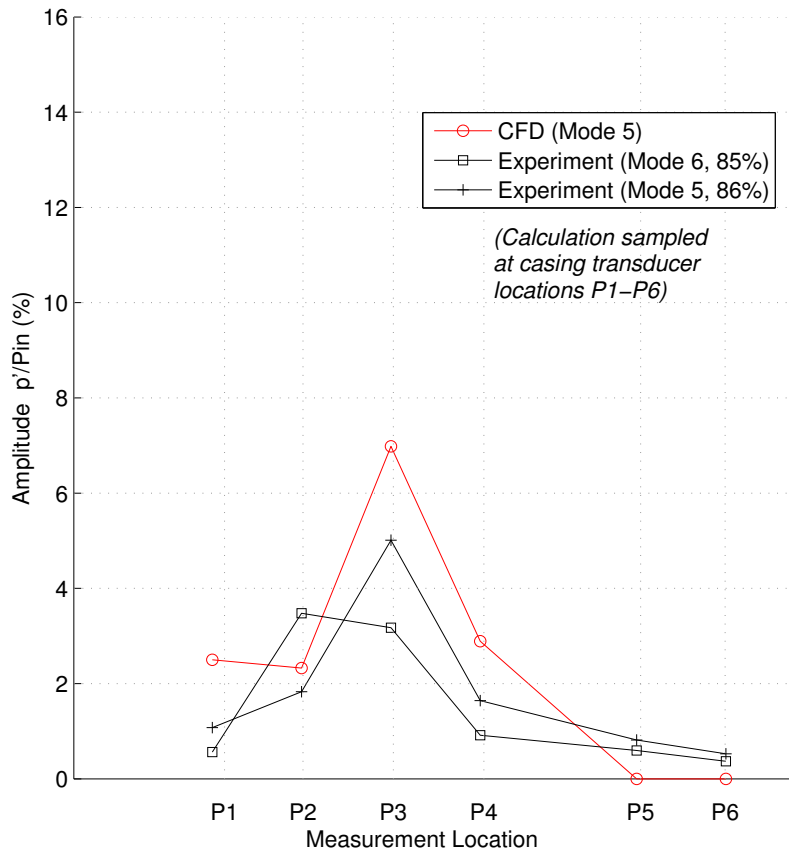


Figure 7.11: Case D* - axial distribution of static pressure mode 5 amplitude along the casing, compared with experimental data.

experimental evidence alone. Based upon the numerical simulations, this can now be understood from Figures 7.9 and 7.11. The stall cells reside in the hub of Stator 1/Rotor 2, such that the casing transducers do not measure the cell directly but only the effect of radial redistribution outboard around the cell. The increased velocity of the flow displaced outboard by the stall cells causes the static pressure variation to be measured. As Figures 7.9 and 7.11 show, this casing static pressure perturbation reaches its maximum amplitude further downstream than at the hub.

7.2 Rotor 1 stall family (high cell count)

Having explored the unsteady simulations of rotating stall associated with the Stator 1/Rotor 2 hub separation, attention now focuses on the numerical analysis Rotor 1 stall, which chapters 4 and 5 showed was associated with a higher number of cells and appeared on the pressure transducer at frequencies above 5 engine order. In this section the progression through Cases Z, A and G is studied pushing Rotor 1 into stall. The formation of rotating stall on Case A will be considered first and it should be noted that the solution presented for this condition here is *identical* to that presented in the previous section (i.e. stall is also forming at Stator 1/Rotor 2 hub simultaneously).

As stated in the previous section, the starting point for the unsteady solution was Case Z, for which Stator 2 was opened by 5° to ensure a stall free initial condition. Figure 7.12 shows, for Case A, how the axial velocity field develops around the circumference at 99% span, immediately upstream of Rotor 1 leading edge. This is shown in a frame of reference moving with the rotor but uses the same analysis methods presented in the previous section.

Initially, the horizontal lines seen represent the periodic velocity field induced by each blade. Over the first revolution, the blade passages fill with blockage in an axisymmetric manner until, at around 1.5 revolutions, travelling disturbances become evident, at a speed corresponding to 79% of shaft speed (absolute frame). These disturbances are of lengthscale comparable to that of a rotor pitch.

Beyond the 3 revolutions shown in Figure 7.12, some merging of the stall cells occurs, although they remain stable at very short lengthscale. In order to accurately determine the mode order of this disturbance once the calculation had stabilized at revolution 12, the circumferential profile of static pressure was extracted, also at 99% span, immediately upstream of Rotor 1. The spatial Fourier decomposition of this profile is shown in Figure 7.13, with a dominant mode order of 27. At this point the propagation speed had stabilized at 73% of shaft speed.

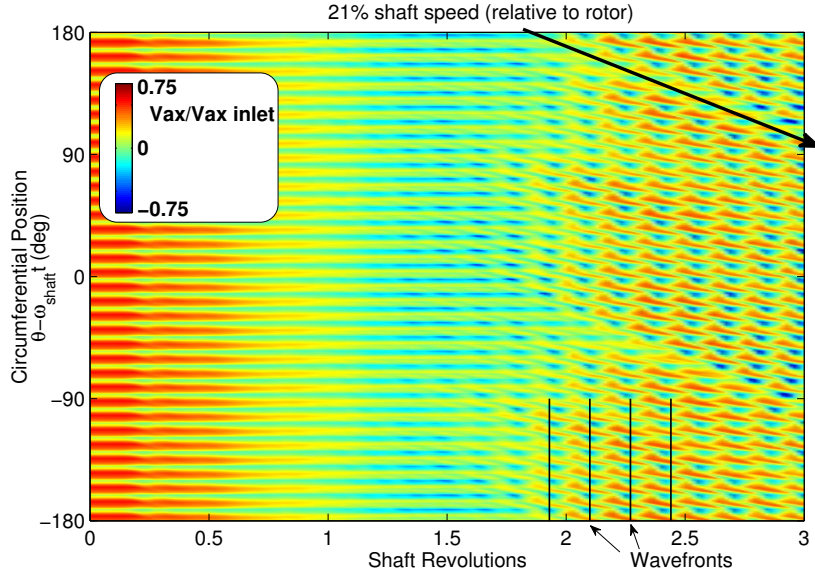


Figure 7.12: R1 inlet axial velocity profile around circumference at 99% span, during computed transient from Case Z to Case A.

In understanding the nature of these disturbances it is instructive to contrast these observations with other reported cases. Young et al.[16] showed experimentally the presence of similar stable disturbances, comparable to a blade pitch in scale. Unsteady RANS analysis of the stall inception process, performed by Pullan et al.[17], showed that these features could be explained by radial vortex like structures shedding from the blades as they stalled. These migrated across the passage, causing the adjacent blade to stall in the same manner, thus leading to a propagating disturbance. Following, therefore, the same visualization method as Pullan et al.[17], a snapshot of the final Case A solution is shown for Rotor 1 in Figure 7.14, displaying an iso-surface of the λ_2 vortex identification parameter[59]. Discrete vortex like structures can be observed, each with lengthscale comparable to a blade passage. The disturbances seen in Figure 7.12 are therefore believed to arise from these structures propagating relative to the rotors.

During the experimental analysis in chapter 5, it was shown that short lengthscale disturbances were also present, which interacted with the rotor to generate cut-on acoustic waves (Section 5.2.3). It is therefore of interest to investigate if such an acoustic source is present in this simulation.

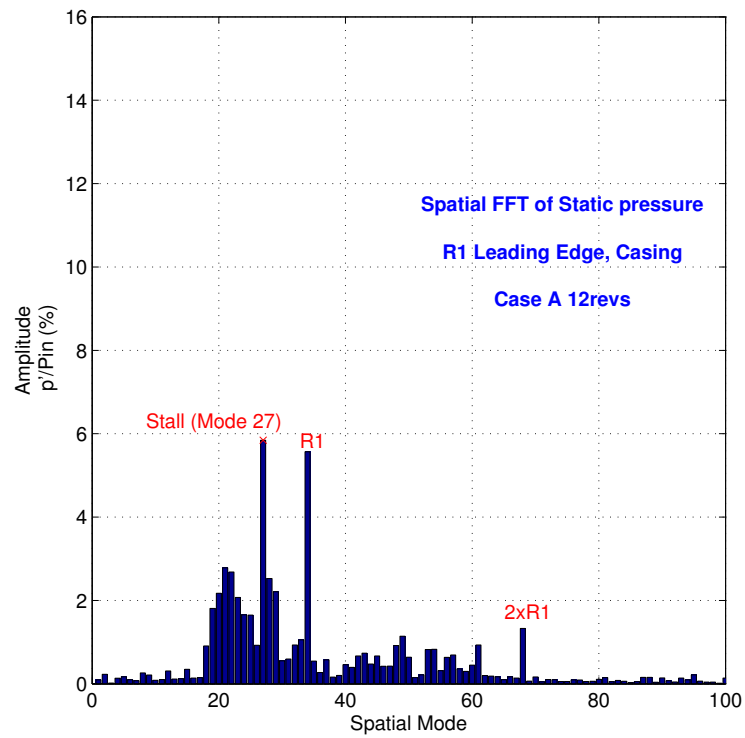


Figure 7.13: Case A final stabilised solution, spatial static pressure mode amplitudes at Rotor 1 leading edge, 99%span.

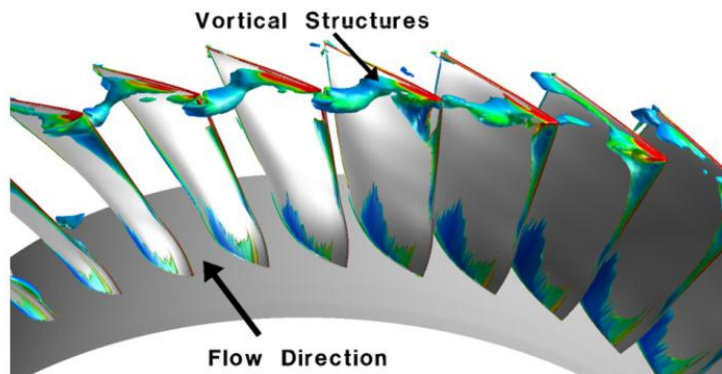


Figure 7.14: Case A - Isosurface of λ_2 parameter[59], viewing R1 from upstream.

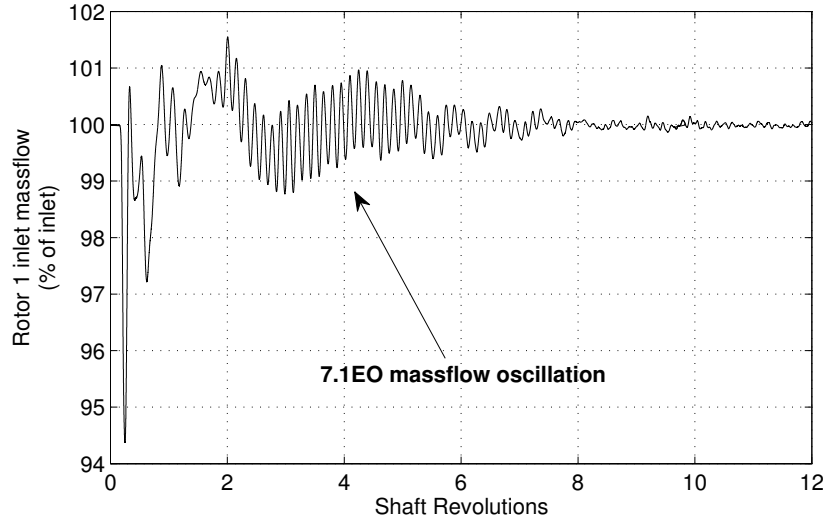


Figure 7.15: Rotor 1 inlet mass flow during computed transient from Case Z (unstalled) to Case A (stalled).

Initially it appears that, in Figure 7.12, one small disturbance is shed for each blade, propagating in the absolute frame of reference at approximately 79% of shaft speed. This implies two waveforms of identical wavelength are moving relative to each other (blade and stall cell). This will result in a difference tone with mode order zero i.e. purely axial waves (analogous to a siren). It is straightforward to show that, at this propagation speed, the two waveforms will “beat” at a frequency of $7.1EO^2$. This manifests itself in the calculation as a “piston like” oscillation in massflow at Rotor 1 inlet, as shown in Figure 7.15. The oscillation persists for several revolutions until the stall cells merge, diminishing the intensity of zeroth order wave. The generation of successive axial wavefronts is also highlighted in Figure 7.12 and can be seen most clearly each time a stall cell (diagonal running disturbance) intersects a disturbance associated with a rotor blade (horizontal line).

Attention now turns to the development of this phenomenon as the VSVs are adjusted to the more stalled Case G. Once the Case A solution was stable

²Relative to the rotor, a time period of 4.76 shaft revolutions is needed for a cell to travel the full circumference, which is equivalent to 0.14 shaft revolutions for a blade pitch (34 blades). Therefore the stall/rotor beat frequency will be $7.1EO$

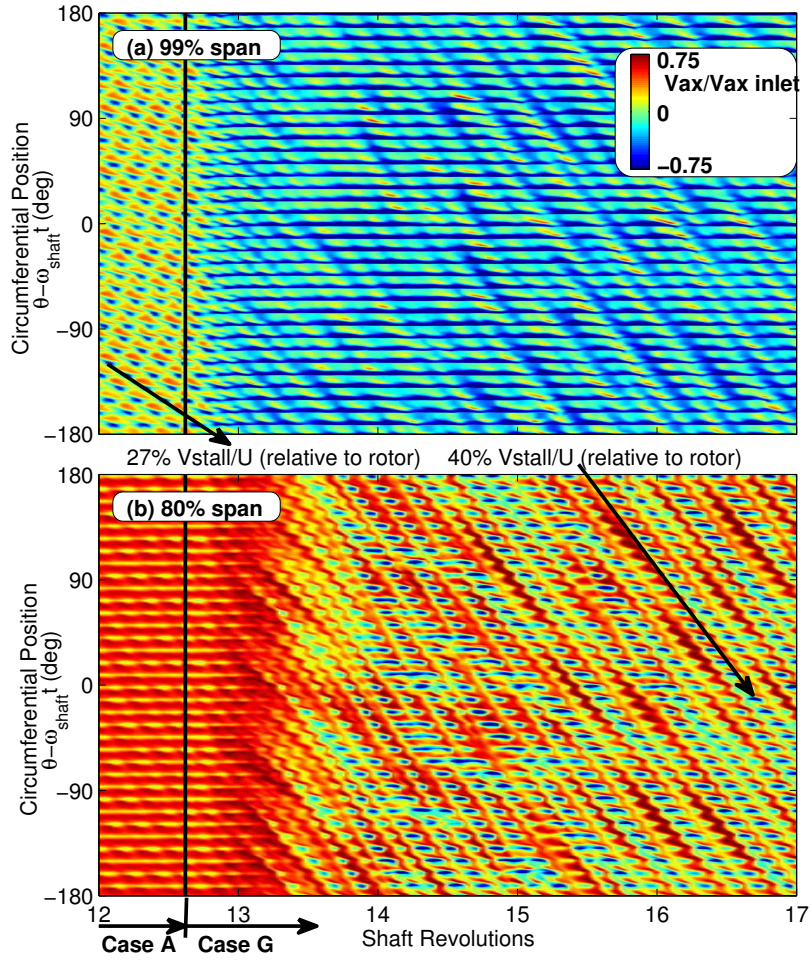


Figure 7.16: Transient solution at R1 leading edge, (a) 99% span and (b) 80% span, when vanes are moved from Case A to Case G.

at 12.5 revolutions (noting also that the hub stalling phenomena discussed in section 7.1 was also stable), the VSVs were moved to the Case G setting (i.e. IGV open 4.5° , S1 closed 3.5° , S2 closed 3.5°). Figure 7.16 shows how the flow in the outer span region develops in the rotating frame of reference.

At 99% span, Figure 7.16(a) shows the 27 short lengthscale disturbances are immediately engulfed by a rapid growth in blockage as the tip flow becomes axially reversed almost uniformly around the circumference. Shown beneath this in Figure 7.16(b) is the corresponding behavior at 80% span. Initially the flow is periodic, corresponding to the velocity field of the rotor

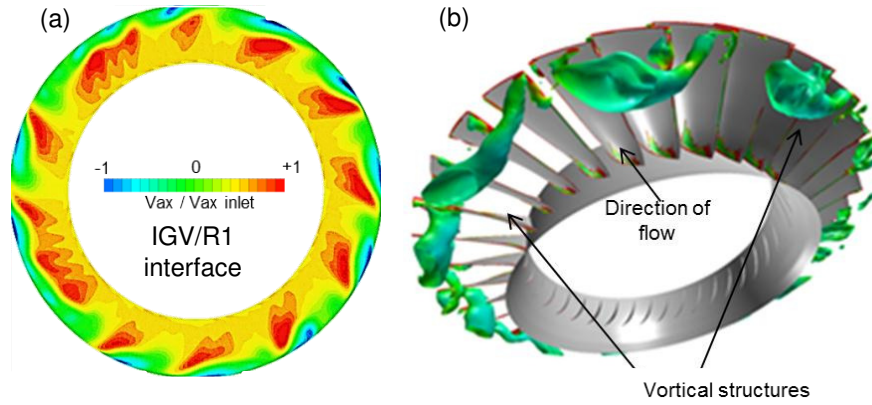


Figure 7.17: Case G solution (a) Axial velocity at IGV/R1 sliding plane, (b) Isosurface of λ_2 parameter[59].

blades alone, implying that the rotating stall at 99% span is not felt this far inboard. As the flow becomes axially reversed at 99% span, propagating disturbances appear at 80% span. This implies that rotating stall cells grown inboard from the tip. Furthermore, they have decelerated in the absolute frame to 60% of shaft speed.

The final Case G solution at revolution 22 is shown in Figure 7.17 in terms of the axial velocity upstream of Rotor 1 in Figure 7.17(a) and the λ_2 vortex parameter isosurface[59] in Figure 7.17(b). Clear vortex like structures, now of greater circumferential lengthscale, can be observed with approximately 12 features identifiable around the circumference.

The spatial Fourier transform of static pressure for this “final” fully developed solution is shown in Figure 7.18, at Rotor 1 leading edge, 80% span. The dominant circumferential mode implies 12 stall cells are present, consistent with the axial velocity contours in Figure 7.17(a). The associated static pressure variation is over twice the intensity of the Rotor 1 blade alone signal.

Performing the same modal analysis across chosen points in the computational domain allows the axial and radial distribution of the 12th spatial mode to be evaluated. Once again, this is limited to disturbances in static pressure. Figure 7.19 shows this distribution along the hub and casing. The

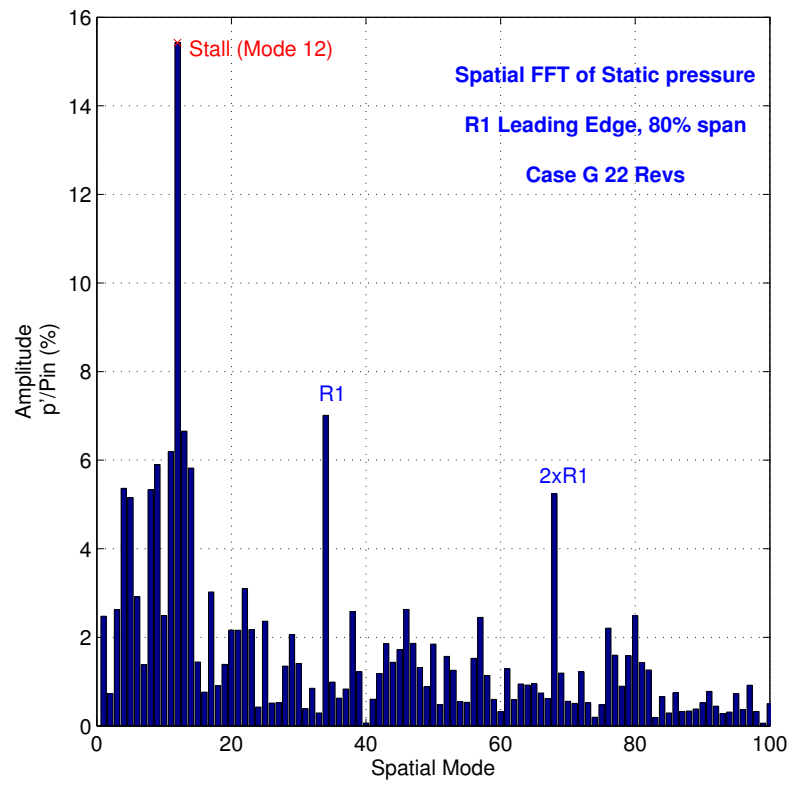


Figure 7.18: Case G final stabilised solution, spatial static pressure mode amplitudes at Rotor 1 leading edge, 80%span.

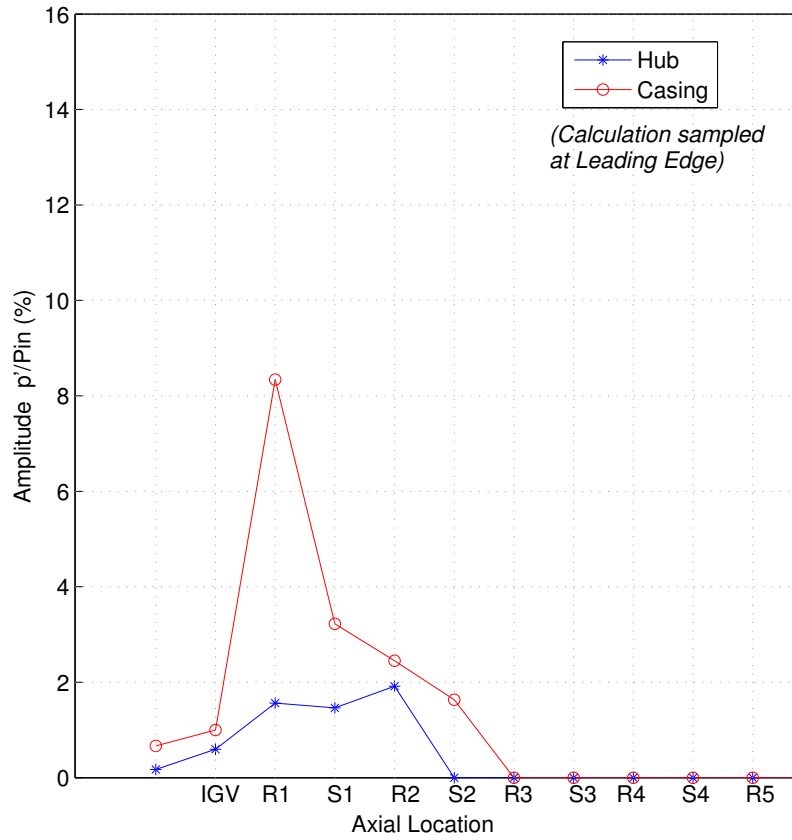


Figure 7.19: Case G - axial distribution of mode 12 static pressure amplitude along hub and casing.

amplitude is of highest intensity towards the casing in the IG/Rotor 1 gap. Upstream, the static pressure variation is attenuated almost completely by the IG. This can be understood by recalling that the IGs are accelerating (non-diffusing) vanes, which cannot stall in response to the variation in exit static pressure. For this reason, the stall cells are limited in terms of their axial growth as they cannot extend into the IG passages.

Finally, it is also seen in Figure 7.19 that, in a similar manner to that demonstrated for the hub stalling behaviour earlier in the chapter, the peak amplitude at the hub is downstream of that at the casing. This again is due to the the flow being diverted radially around the stall cells.

7.2.1 Comparison with experiment

It is now possible to relate the behaviour observed in the model back to the experimental observations presented in chapter 5.

For Case A, very short length scale rotating stall was observed during the experiment at 85% shaft speed, which could be seen in Figure 4.7, labelled “A1”. A detailed study of this phenomenon was provided for the more stalled Case F showing that the signal was associated with higher order modes, with 28-31 identified. This phenomenon was also linked to upstream noise propagation and Rotor 1 vibration in the second torsion mode. These disturbances propagated at approximately 65% of shaft speed in the absolute frame. A corresponding experimental study for Case A shows very similar behaviour in terms of mode order and propagation speed, however the overall intensity of this signal was below 1% of inlet total pressure, when measured using a transducer in the IGV-Rotor 1 gap.

The simulation for Case A shows good agreement in terms of the circumferential length scale of the rotating disturbances, with mode 27 apparent in the model. The stall propagation speed of 73% of shaft speed is also in good agreement with experiment. The amplitude of the resulting static pressure perturbation obtained from the model at Rotor 1 leading edge was 6% of inlet total pressure (see Figure 7.13), falling to 0.2% by IGV exit, consistent with exponential decay of a cut-off disturbance axially along the duct upstream of Rotor 1. This level is therefore in line with experiment, although this rapid decay in the axial direction implies that in reality, the measurement must be highly sensitive to the location of the transducer, thus making such comparisons prone to error.

For the more stalled Case G, spectral analysis of measurements presented in chapter 5 showed that, at 85% shaft speed, mode orders 13-16 were visible, propagating at 55% of the shaft speed in the absolute frame of reference, which is repeated in Figure 7.20 with the corresponding CFD simulation also added. The existence of multiple frequency bands in the measurement was also attributed to spatial and temporal non-uniformity in the pattern of stall cells. By comparison, the CFD model shows a good agreement, with a 12 cell pattern present rotating at 60% of shaft speed. Some degree of

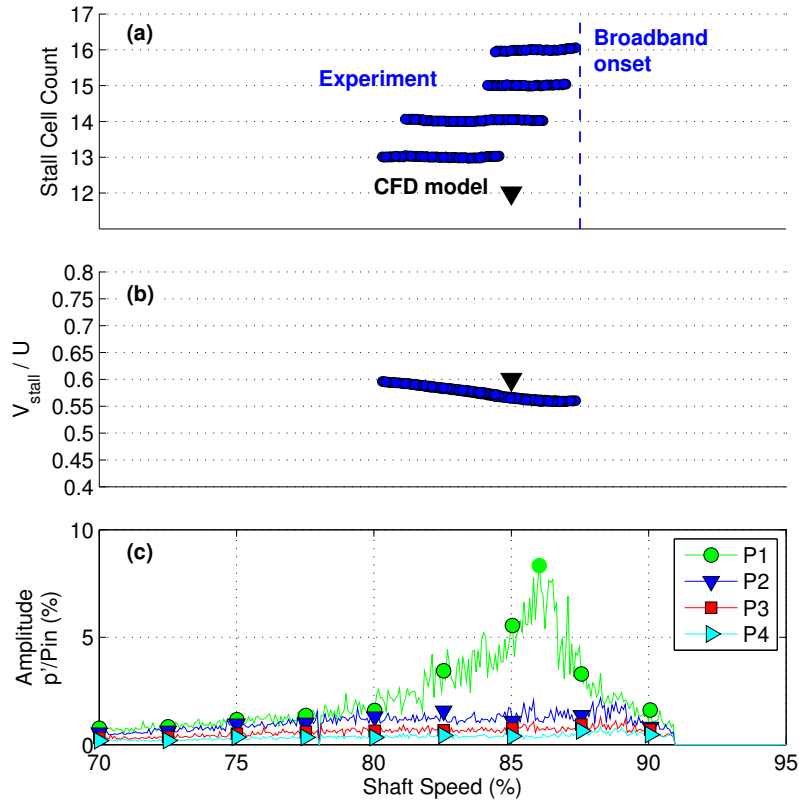


Figure 7.20: Repeat of Figure 5.6 with CFD computation for Case G also shown: Case G - rotating stall observed in experiment through manoeuvre.

spatial non-uniformity can be seen in Figure 7.17, in contrast with the highly uniform hub stalling behavior seen in the previous section (e.g. Figure 7.4). Scattering to adjacent spatial modes can also be observed in Figure 7.18, and is also indicative of non-uniformity.

Taking the comparison further, the intensity of spatial mode 12 obtained from the simulation is compared with experimental data for mode 15 in Figure 7.21, at each measurement location. Experimental data is shown at both 85% and 86% speed, the latter speed included as the signal peaked sharply there. The agreement between experiment and model is good, in terms of both the absolute level and axial distribution.

In summary, this section shows that as the variable stator vanes are adjusted to push Rotor 1 towards stall (by opening the IGV and closing the

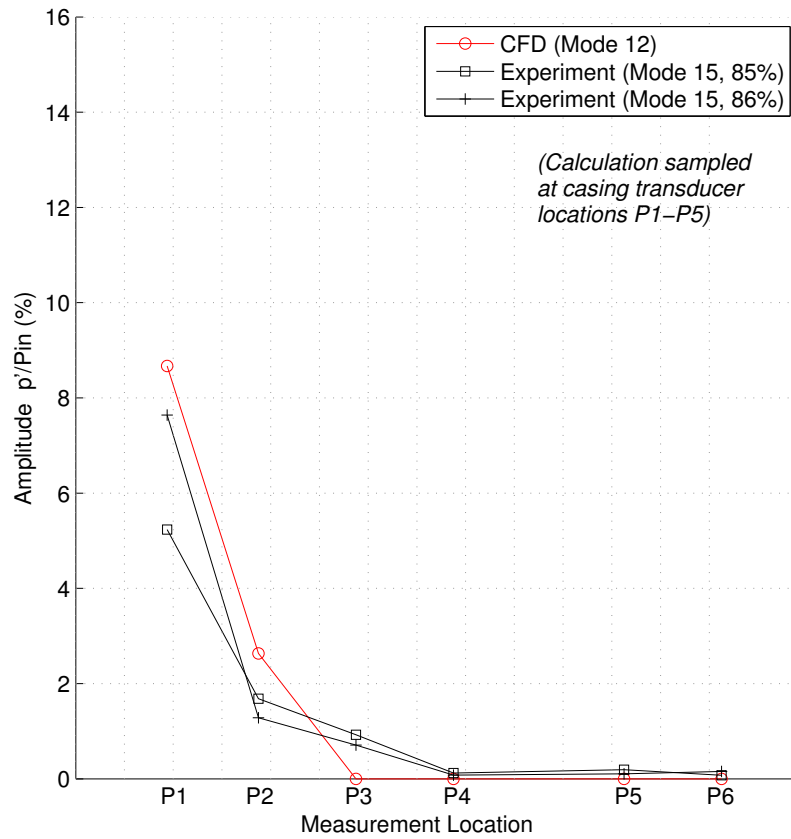


Figure 7.21: Case G - axial distribution of static pressure mode 12 amplitude along the casing, compared with experimental data.

downstream vanes), rotating stall cells form in the tip region. Initially these have very short lengthscale similar to that of a blade passage, which grow and merge as the rotor loading is increased further. These findings are consistent with the experimental observations.

7.3 Discussion on stall cell behaviour

Having demonstrated that the numerical model gives credible results, the discussion now focuses upon whether any more insight can be gained into the observations made in experimentally in chapter 5.

The experimental findings showed quite clearly that, for the family of

rotating stall linked to Stator 1/Rotor 2, the stall cell count appeared to spontaneously change several times during the acceleration manoeuvre (Figure 5.2). Of particular interest, the propagation speed always increased when the cell count reduced (the cells growing in lengthscale). This observation is supported by the numerical behaviour in Figure 7.3, which show that as 7 cells grow and merge into 5, they also accelerate from 25% to 40% of shaft speed (in the absolute frame).

This observation would at first sight appear to be contrary to past experience. During stall inception, for example, spike disturbances initially appear at high propagation speed and decelerate as they grow in size [12]. In the majority of these cases, however, it is the rotor tip which becomes unstable, causing spikes (or embryonic stall cells of very short lengthscale) to come into existence in the rotating frame of reference, accelerating relative to the rotor as they grow.

This numerical investigation demonstrates that, for the Stator 1/Rotor 2 family, the disturbance forms in the stator (which followed from Figure 7.5 and the accompanying discussion) and remains in this region as it grows to full size, with a much weaker presence in the neighbouring blade rows (as seen in Figure 7.7). Moreover, chapter 9 will show that similar behaviour is predicted when all bladerows downstream of Stator 1 are removed from the model. It is therefore appropriate in this case to define stall propagation speed relative to the stator.

Consequently, it follows that, for short lengthscale cells, the results obtained from the experiment and CFD simulations are entirely consistent with past experience when an appropriate “relative” propagation speed is used defined in a frame of reference appropriate to the blade row in question. The vast majority of reported cases involve spike formation in the rotor tip[12]. However, spikes forming near the leading edge of shrouded diffuser vanes have been reported in a centrifugal compressor [18], rotating at only 20% of shaft speed.

Turning to the Rotor 1 stall family. Figure 7.16 shows that, in the rotor frame of reference, the stall cells at Rotor 1 tip also accelerate as they grow in size. This demonstrates that both families of rotating stall behave in the same way.

The numerical results also show, in Figure 7.4, that rotating stall forms in the hub region, persisting there as a stable disturbance. Again, this is not commonly reported, with the majority of published cases of rotating stall occurring in the tip region. However, it was shown in chapter 6 (section 6.4.2) that radial equilibrium effects arising from the severe part-speed IGV closure, combined with the throttling effect of deliberately closing Stator 2, causes the hub of Stator 1 to stall whilst the outer span remains at quite a tolerable diffusion level, as shown in Figure 6.9. This allows rotating stall to form in the hub and also persist in this location, as the stable outer span prevents the stall cells from migrating radially outboard. This effect is likely to be unique to variable geometry compressors and highlights the fact that the majority of reported cases, which utilise fixed geometry, do not capture this effect.

7.4 Sensitivity to calculation time step size

For the unsteady calculations shown in this chapter so far, the calculation time step has been held constant at a value corresponding to 1200 iterations per shaft revolution. This was initially chosen based upon consideration of the frequency at which rotating stall was observed in the experimental study in chapter 5. In this section the effect of reducing the timestep is considered, by running the Case A solution (with stall cells already present) further with the timestep reduced to half of its original value to give 2400 timesteps per shaft revolution. The effect on the overall compressor performance, in terms of pressure ratio and inlet flow function is considered first, which is followed by a comparison at a detailed level on the effect of stall cell spatial/temporal lengthscales.

The overall performance map of the compressor at 85% speed was shown in the previous chapter in Figure 6.1. A close-up view near the working line is shown in Figure 7.22 which also now includes the *transient* operating point of the compressor during the unsteady simulation starting at Case Z and closing Stator 2 back to its Case A setting (i.e. the same transient shown in Figure 7.2). Figure 7.23 shows the corresponding transient in terms of inlet flow function variation against the iteration number.

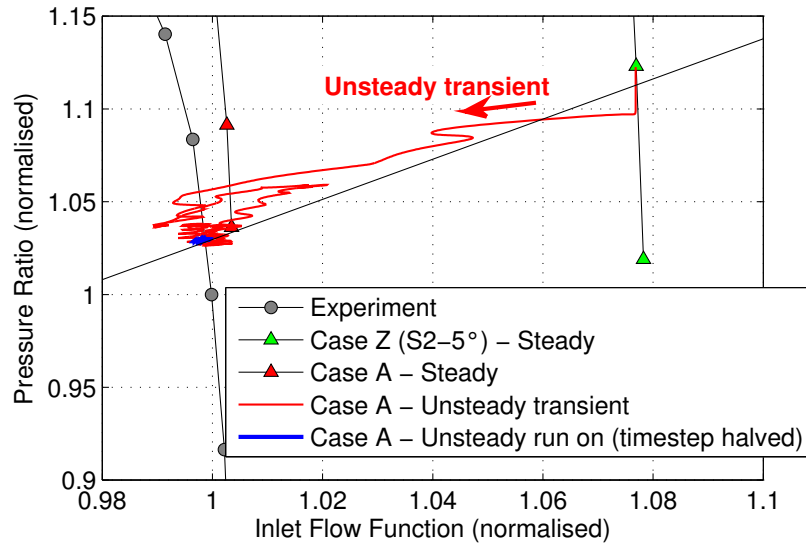


Figure 7.22: Overall characteristic showing computed transients for unsteady simulations. Effect of running Case A simulation further with a reduced timestep also shown.

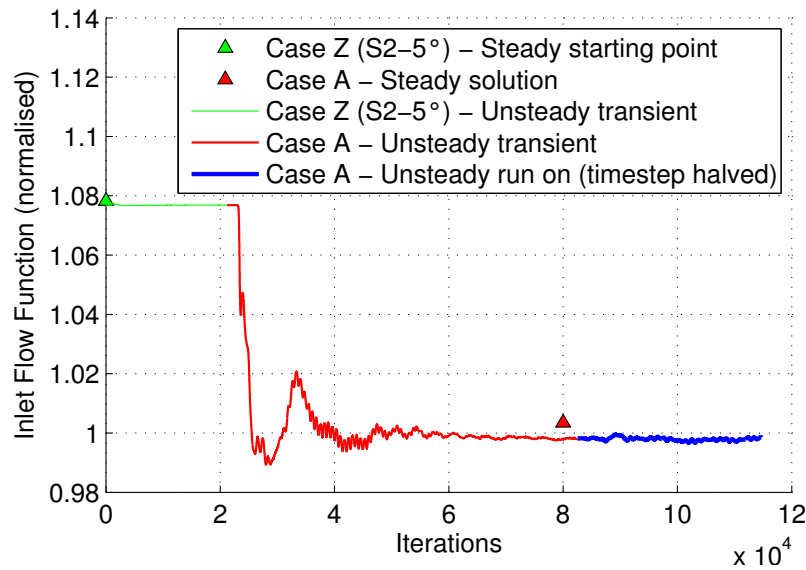


Figure 7.23: Transient variation in inlet massflow function during unsteady simulations. Effect of running Case A simulation further with a reduced timestep also shown.

Initially the pressure ratio drops at a near constant inlet flow from the initial Case Z starting point, followed by a fall in flow along a path which remains close to the working line. The inlet flow function then oscillates (corresponding to the zero nodal diameter oscillations in Figure 7.15), and eventually settles at a value slightly below the equivalent steady solution for Case A. This final inlet flow function for the unsteady simulation is also closer to the experiment. The calculation was then restarted with the timestep reduced to half of its original value, which from Figures 7.22 and 7.23 shows no further change to the operating point of the compressor.

At this condition (Case A), the previous sections of this chapter have already shown that 7 stall cells are simulated in the hub region of Stator 1/Rotor 2, together with a high number of cells in the Rotor 1 tip region (27 based on Figure 7.13). The effect of the reduced timestep size on these disturbances is shown in Figure 7.24 and 7.25 respectively, in the same format used throughout this chapter, with the point at which the calculation was restarted with the reduced timestep highlighted with a vertical line.

The 7 cell stall pattern associated with Stator 1/Rotor 2 hub, shown in Figure 7.24, remains unaffected by the reduced timestep and maintains a cell propagation speed equal to 25% of the shaft rotational speed, confirming that the simulation is not significantly affected by the smaller timestep.

At Rotor 1 tip, approximately 27 short lengthscale features (associated with radial vortices similar to that reported by Pullan[17]) were observed. Figure 7.25 shows these features persist with their lengthscale not significantly affected by the reduced timestep. The circumferential propagation speed does appear to increase slightly (by approximately $0.04U$ in the frame of reference moving with the rotor) with the timestep reduced, which does imply that a more accurate stall frequency could be obtained for this phenomenon by increasing the temporal resolution. It should also be expected that this high frequency disturbance will not be resolved as accurately in the simulation as the lower frequency stall pattern associated with Stator 1/Rotor 2. However, it can still be concluded that the fundamental behaviour is adequately captured by the simulation and remains robust to changes in the model setup.

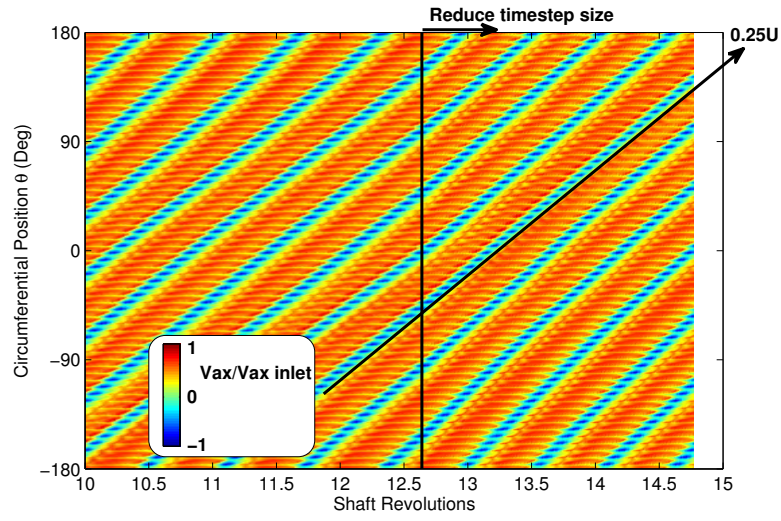


Figure 7.24: S1 exit axial velocity profile around circumference at 20% span showing effect of halving the calculation timestep.

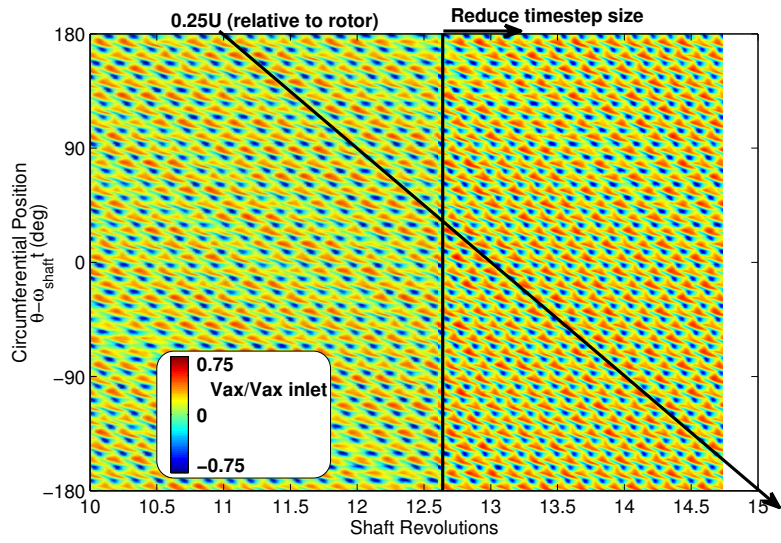


Figure 7.25: R1 leading edge axial velocity profile around circumference at 99% span showing effect of halving the calculation timestep.

7.5 Overall performance implications

Reliable information can be derived from the unsteady models for overall compressor performance providing the pitchwise averaging planes at inlet and exit are chosen to be far away from the regions of significant unsteady flow. No attempt has been made to derive the performance of individual stages where rotating stall is present, as an appropriate post-processing method (which averages the flowfield in a physically appropriate way) would first need to be developed which is beyond the scope of this thesis.

Figure 7.26 shows the overall performance of the compressor in terms of pressure ratio and efficiency vs inlet flow function. These points were extracted from the simulations for Cases A, Z, D* and G by pitchwise averaging at domain inlet and exit. Characteristics obtained from the steady single passage simulations for Cases A and Z are also shown together with the corresponding characteristic from the experiment for Case A. Steady state simulations performed for Cases D* and G are not included, as the severe separations lead to poor convergence in the steady model and potentially unreliable results.

The effect of VSV adjustment on flow and efficiency in Figure 7.26 is very powerful. Relative to Case A, for which both Rotor 1 tip and Stator 1/Rotor 2 hub stall are present, the effect of opening Stator 2 by 5° (to Case Z), which eliminates these stalled regions, results in a rise in efficiency in excess of 5%. Moreover, for the unstalled Case Z, the steady and unsteady simulations overlay exactly showing that in the absence of rotating stall the two methods give identical results. However for Case A, in which rotating stall is present in the simulation, the working line efficiency is approximately 0.7% lower in the unsteady model than in the steady model. This implies that the steady and unsteady models deviate once regions of separation exist which are sufficient to exhibit rotating stall.

It can be also seen in Figure 7.26 that adjustment of the VSVs to the more extreme Cases D* and G results in a significant fall in efficiency in excess of 10%. This shows how detrimental a badly matched compressor can be to performance, although the condition considered here is at part-speed, where the engine does not burn fuel for significant durations.

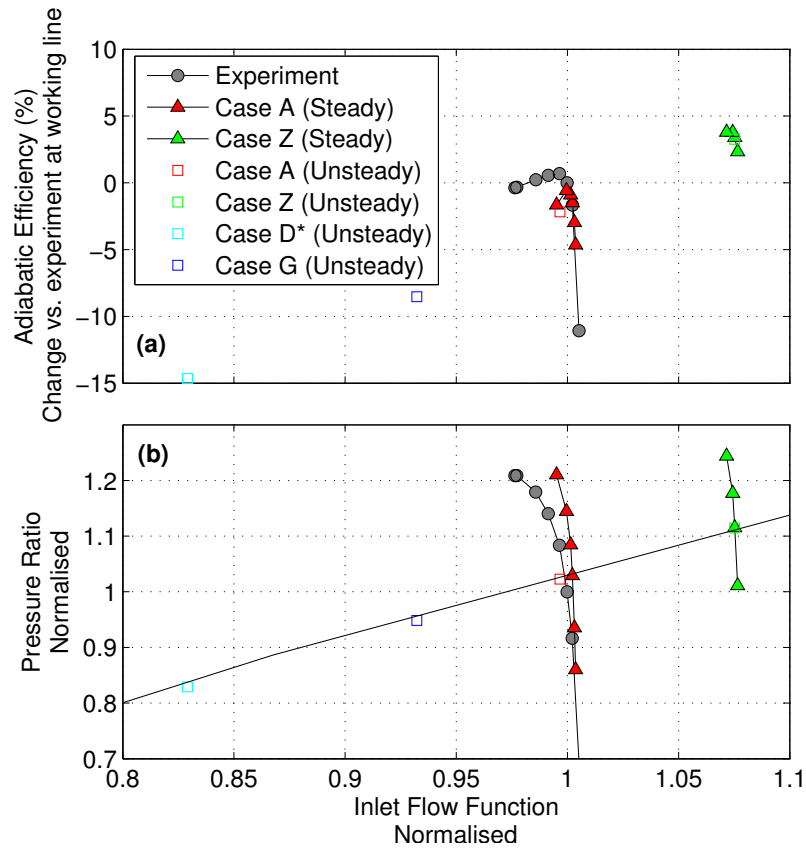


Figure 7.26: Overall performance parameters from the experiment, steady (Case A and Z) and unsteady (Cases A, Z, D* and G) models in terms of (a) Pressure Ratio and (b) Efficiency vs Inlet Flow Function

The effect of VSV changes on efficiency also strengthen the assumption that detailed geometric features such as shroud leakages, fillets and penny clearance gaps could be omitted from the model (as discussed in chapter 3). Wellborn[60] showed the effect of shroud leakage flow on efficiency follows a trend of approximately 1% efficiency loss for 1% increase in shroud clearance normalised by blade span. For a well built compressor with clearances of approximately 1% span the effect of shroud leakage, whilst very important at design point (where the majority of fuel is burned and changes of 0.1% efficiency become important), is of second order at part-speed in comparison to the mismatching effects introduced here with the VSVs.

7.6 Concluding remarks

In summary, the steady single passage solutions discussed in the previous chapter were expanded to full annulus unsteady cases and rotating stall allowed to evolved naturally from an unstalled starting condition.

The results show unequivocally that URANS CFD methods are able to capture part-span rotating stall in both the Rotor 1 tip and Stator 1/Rotor 2 hub region, with rotating stall cells forming quite naturally in the solution. Moreover, the alignment between these simulations and the experimental findings in chapter 5 confirm that this method gives credible results for both stall families in terms of the lengthscales, time scales and amplitudes of rotating stall. This also implies that the method can suitably be extended to investigate the noise and vibration aspects of the phenomenon, thus assisting the designer.

In the subsequent chapters, the experimental and numerical analyses are developed further to consider the behaviour at a more fundamental level. In particular, chapter 9 will use the unsteady modelling technique presented here to demonstrate the importance of flowfield coupling mechanisms (such as the mechanism reported by Longley and Hynes[5]) in determining the fully developed stall properties. This is shown to be because the “stable” parts of the compressor are able to suppress the growth of stall cells forming in the unstable parts of the compressor and set the final lengthscale of any rotating stall pattern. This behaviour is driven largely by simple inviscid

mechanisms, which follow from the discussions in the previous chapter concerning radial equilibrium effects and closure of the rearmost VSV (Stator 2). This offers some insight into the reasons why credible unsteady predictions of the fully developed stall phenomenon can be obtained from basic URANS modelling techniques.

8 The relative effects of changes in VSV setting and speed

It is recalled from the previous chapters that the test compressor operates with the variable stator vane settings altering across the speed range. For example, Figure 2.11 showed a typical variable stator vane (VSV) schedule, whereby the stagger angles of the IGV, Stator 1 and Stator 2 were increased progressively as the compressor speed was reduced from the design point. By utilising the flexibility in this test compressor rig, which has independently actuated VSVs, it was shown how stall could be provoked in the Rotor 1 tip and Stator 1/Rotor 2 hub region by further movement of the VSVs *relative to each other*, in addition to their normal scheduled movement across the speed range.

A common further constraint on a real engine application, however, requires that a single actuator be used to drive several stages together through a so-called “ganged” mechanism. This forces the vanes of each variable stage to always move in linear proportion to the other variable stages, and it is a key task to find the optimum ratio between stages to satisfy the different design attributes of the whole engine as the speed is varied. The optimum VSV schedule must achieve maximum efficiency, whilst still maintaining acceptable stability, vibration and noise levels. Further requirements such as adequate bleed offtake pressures may be an additional constraint. Commonly this optimisation requires a stand alone test rig with independent VSV actuation, such as the test case studied in this thesis.

In this chapter, the objective is to explore how rotating stall behaviour is affected by operating the compressor in a manner that simulates this ganged VSV mechanism. Rather than focusing on re-stagger of each row relative to the others, attention here focuses on how stall behaviour changes when

adjustments are made to the setting of the variable vane system as a whole *depending only on the operating speed*. For example, what is the effect of forcing the VSVs to “lag” relative to their nominal settings during an acceleration? In such a situation the vanes will be more closed than their nominal settings at any given speed but will still pass through the same physical settings at higher speeds. From an experimental point of view, this enables the effect of speed changes to be isolated from VSV setting changes as the following sections will show.

Firstly, an example is shown where the compressor is accelerated initially with the VSVs held fixed, followed by a further acceleration with the VSVs moving. This is shown to have a pronounced effect on the propagation of stall cells. Subsequent to this a more detailed study is presented by systematically offsetting the relationship between the vane angles and aerodynamic speed.

8.1 Effect of VSVs moving and fixed with speed

Recalling chapter 4, a group of 4 different VSV schedules was studied which push the Stator 1/Rotor 2 region into stall. These VSV schedules were referred to as Cases B, A, C and D, with the latter case (D) being the most extreme, as shown previously in Figure 4.5. More severe stall (relative to Case A, which is chosen as a datum schedule) was achieved by closing the IGV whilst opening Stator 1 and closing Stator 2. This gave rise to unsteadiness in the 0 to 5EO frequency range when observed on the casing mounted pressure transducers.

In this section attention focuses on the less severe Case C schedule, which still exhibited rotating stall of significant amplitude (casing pressure fluctuations of 4% of inlet total pressure were observed as shown in Figure 4.5).

With the VSVs set to Case C, the compressor was accelerated from 50% speed to 95% speed. During this experiment, the VSVs were deliberately held to fixed settings during the first part of the acceleration up to 71% speed, after which they were opened progressively.

Figure 8.1 shows the spectral analysis of the P2 (Stator 1 exit) pressure transducer and Rotor 2 strain gauge in a format that was used extensively in chapter 5 to study the other cases. Considering first the pressure transducer in Figure 8.1(a), a signal below 5EO can clearly be seen consistent with the Stator 1/Rotor 2 stall. The key point of interest here is the change in behaviour as the VSVs start to open at 71% speed. Below 71% speed, with the VSVs fixed, the signal appears at a frequency which remains almost perfectly proportional to the shaft speed at 2.18EO (as highlighted). However, as the VSVs open, the stall signal shows a pronounced “kink” as the frequency rises more steeply with speed. At 75% and 80% speed the stall is still present but at engine orders of 2.50 and 2.94 respectively.

Figure 8.1(b) shows the corresponding behaviour on the rotating strain gauge, where the same phenomenon is observed at different frequencies to Figure 8.1(a), due to the change in the frame of reference of the measurement as explained in Section 3.1.4. Below 71% speed with the VSVs held fixed, the stall signal has a constant engine order of 2.82. The slope of the signal then drops sharply as the VSVs begin to open and no longer follows an engine order, but falls to 2.5 and 2.06EO at 75% and 80% speed respectively.

Recalling the methodology presented in, Section 3.1.4 and used extensively in chapter 5 for analysis of rotating stall in 2 frames of reference, the results of tracking the signal from Figure 8.1 through the manoeuvre are plotted in Figure 8.2(a). It is clear that this phenomenon consists of 5 stall cells across the speed range tested. However, it is the stall propagation speed in Figure 8.2(b) that shows the most interesting behaviour. Whilst the VSVs are held fixed, V_{stall}/U remains at a near constant value of 0.43 to 0.45. However, as soon as the VSVs begin to open above 71% speed the stall propagation speed rises sharply, V_{stall}/U increasing to 0.65 by the time the stall signal decays away above 85% speed.

Physically this suggests that the effect of changing compressor speed alone has a relatively weak effect on the non-dimensional rotating stall behaviour, whilst the effect of allowing the VSVs to move is considerably more powerful. In the next section this observation is studied further by exploring the effect of adjusting the relationship between the vane angles and the speed.

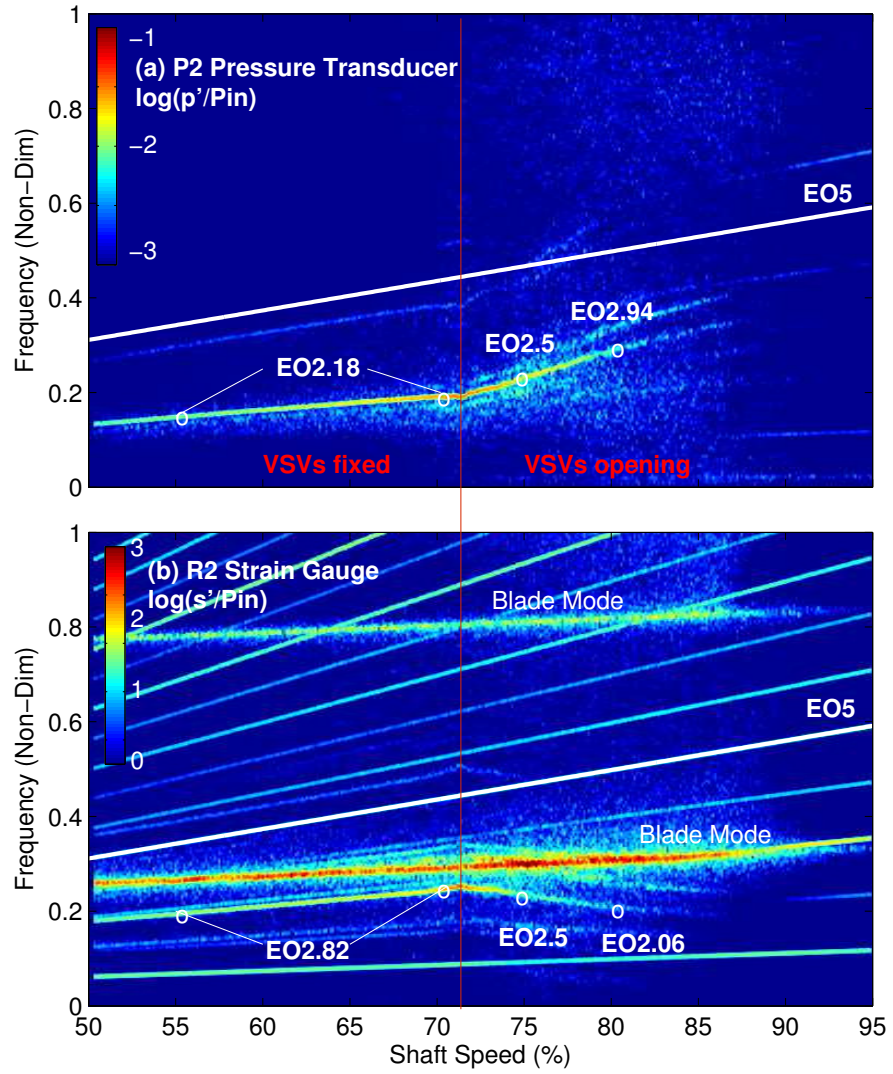


Figure 8.1: Case C - (a) P2 pressure transducer and (b) R2 strain gauge spectra during acceleration with VSVs fixed to 71% speed and progressively opening thereafter.

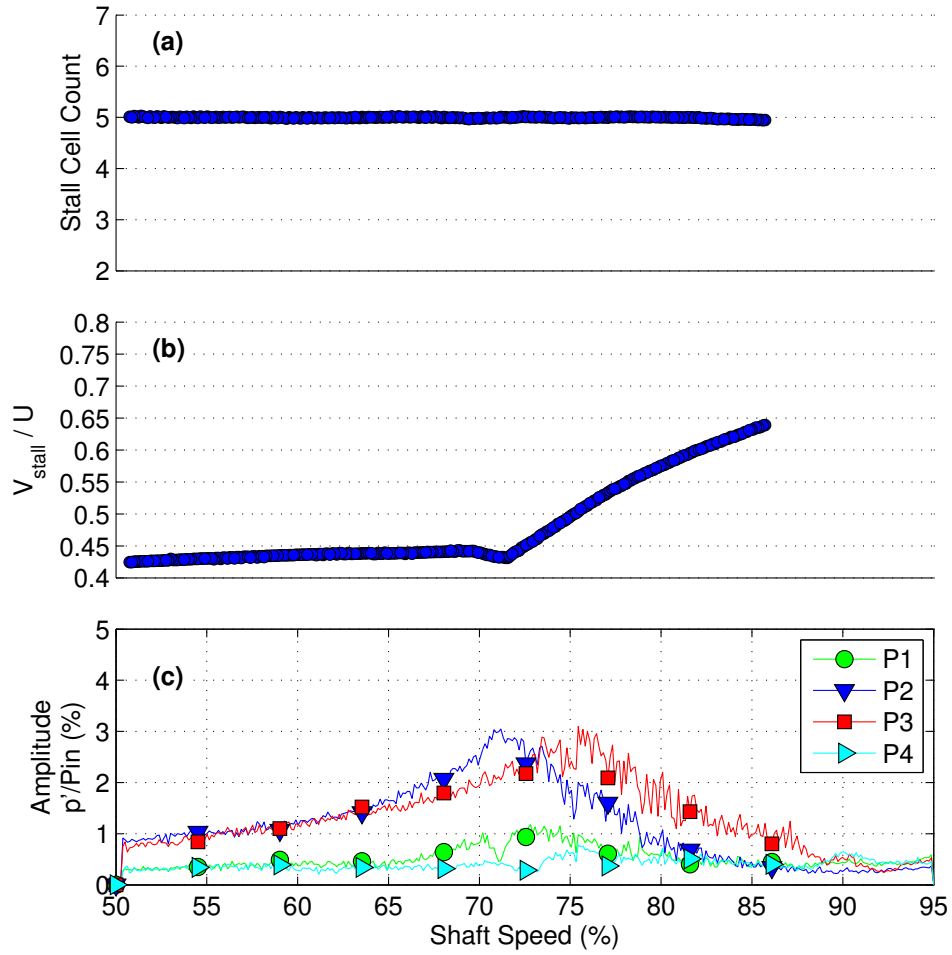


Figure 8.2: Case C acceleration showing (a) Derived stall cell count (b) Derived stall propagation speed and (c) Intensity across the first 4 stage pressure transducers.

8.2 Offsetting the VSV setting to speed relationship

The earlier chapters demonstrated how adjustments to the VSV schedule were able to trigger rotating stall in the compressor. The VSV schedule itself can be split into two separate aspects:

- How the VSVs are set to operate *relative to each other*, i.e. the proportionality in the setting/movement of the vanes for one stage relative to the others. In a real engine application, where the VSVs for several stages are often driven through a single actuator, the movement of each stage relative to the others is completely fixed by the mechanism. Only physical changes to the mechanism hardware can affect this proportionality. These are referred to as “unganged changes” to the schedule, for example the changes made in the earlier chapters which utilised the independent VSV actuation capability available on this experimental rig.
- How the VSVs are set to operate as a combined or “ganged” system *relative to the shaft speed*. Once the proportionality has been set between stages, the whole system can be adjusted to be ganged open or closed relative to a datum case. A ganged change to the schedule can therefore be made by a single actuator and in the context of a real engine could be implemented through the control system software. For example, forcing the vanes to “lag” behind their nominal settings during an acceleration/deceleration manoeuvre would cause the VSV system to be ganged closed (on the acceleration) and open (on the deceleration) respectively.

In this chapter the following terminology is used to quantify changes to the VSV schedule: An *unganged* IGV closure of 5° consists of 5° closure on the IGV only, with no adjustment to Stator 1 or Stator 2, such that proportionality between VSV stages is changed. A *ganged* closure of 5° would consist of the same 5° IGV closure, with additional closure of Stator 1 and 2 according to the schedule, with the proportionality maintained. This would result in stagger changes of less than 5° on Stators 1 and 2.

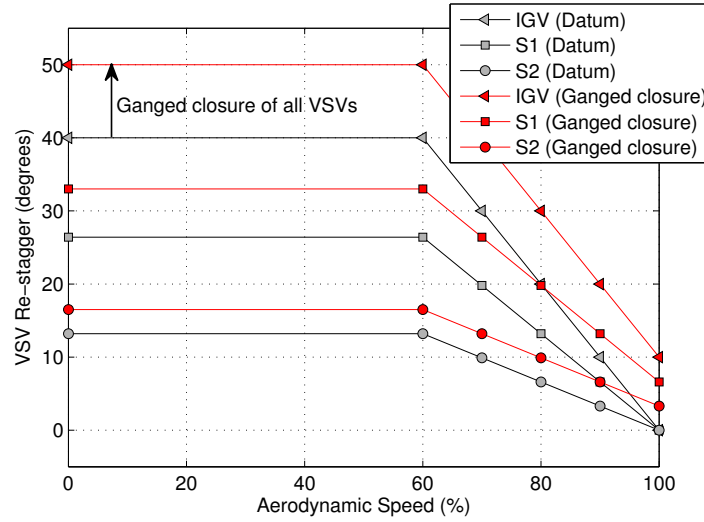


Figure 8.3: A typical VSV schedule showing an example of a *ganged* closed offset.

In the previous chapters the effects of unganged changes to the VSV schedule were studied, which were shown to have a significant aerodynamic mismatching effect on the compressor. In this section ganged changes are considered which conveniently allow the effect of aerodynamic speed/shaft speed to be isolated from the VSV settings. This is because ganged changes force the VSVs to move through the same physical settings but each time at different operating speeds. This can be seen in Figure 8.3 which shows a typical ganged closed offset. The settings at 70% speed on the datum schedule are identical (on all vanes) to the settings at 80% speed on the ganged closed VSV schedule.

For this part of the investigation, the Case A VSV schedule was used and accelerations were performed with the VSV system ganged closed by 10° and 5° and then ganged open 5° . The results in chapter 4 showed that for the Case A VSV schedule, *both* families of rotating stall were present. For example, Figure 4.5 shows the low frequency activity up to approximately 80% speed whilst Figure 4.7 shows the high frequency signal present at 80-85% speed (point ‘A1’). This means conveniently that the effect of ganged changes on both families may be investigated using the same acceleration manoeuvres.

8.2.1 Effect on Stator 1/Rotor 2 stall family

Accelerations were performed in the same manner as in section 8.1, initially with the VSVs held fixed up to a condition close to 70% speed, above which they were opened progressively in a procedure that was repeated with the different ganged offsets. The setting of the IGV through these manoeuvres is shown in Figure 8.4(a) against shaft speed. It is emphasized that Stators 1 and 2 are also moving in exactly the same manner but by a lesser amount to always maintain linear proportionality between stages. Figure 8.4(a) shows that all 4 manoeuvres pass through identical VSV settings but at different speeds. For example, the gang closed 10° case at 85% speed has identical VSV settings to the datum (Case A) at 76% speed and the gang open 5° case at 71% speed, as indicated in Figure 8.4(a).

Figure 8.4(b) and (c) show the derived rotating stall cell count and propagation speed (V_{stall}/U) for each of these cases. For brevity, the pressure transducer and strain gauge traces used to calculate these details are not presented here, but remain consistent with the findings of chapter 5 with the strongest pressure perturbations occurring at locations P2 (Stator 1 exit) and P3 (Stator 2 exit). The compressor consistently stalls in a pattern of 4 cells, which appears to be unaffected by ganged changes to the schedule or speed. The propagation speed (V_{stall}/U), by contrast, shows very large variation due to ganged adjustments. At 76% speed, the stall propagation speed varies from $0.45U$ (gang close 10°) to $0.62U$ (gang open 5°). The pronounced change in slope of Figure 8.4(c) as the VSVs begin to move from initially fixed settings is also seen, consistent with the first section.

This observation therefore implies a very significant link between the properties of rotating stall exhibited by this compressor and the setting of the VSV system as a whole at a given speed. It is quite logical to therefore question whether the effect may in fact be speed independent and driven entirely by the VSV settings. Figure 8.4(d) seeks to answer this by visualising the same derived stall propagation speed (V_{stall}/U) against the instantaneous IGV setting, rather than against shaft speed. When viewed in this way, it becomes clear that, for this set of conditions, V_{stall}/U is uniquely a function of the VSV settings and is indeed independent of speed. Stated another way, the variation in V_{stall}/U observed across the speed range in

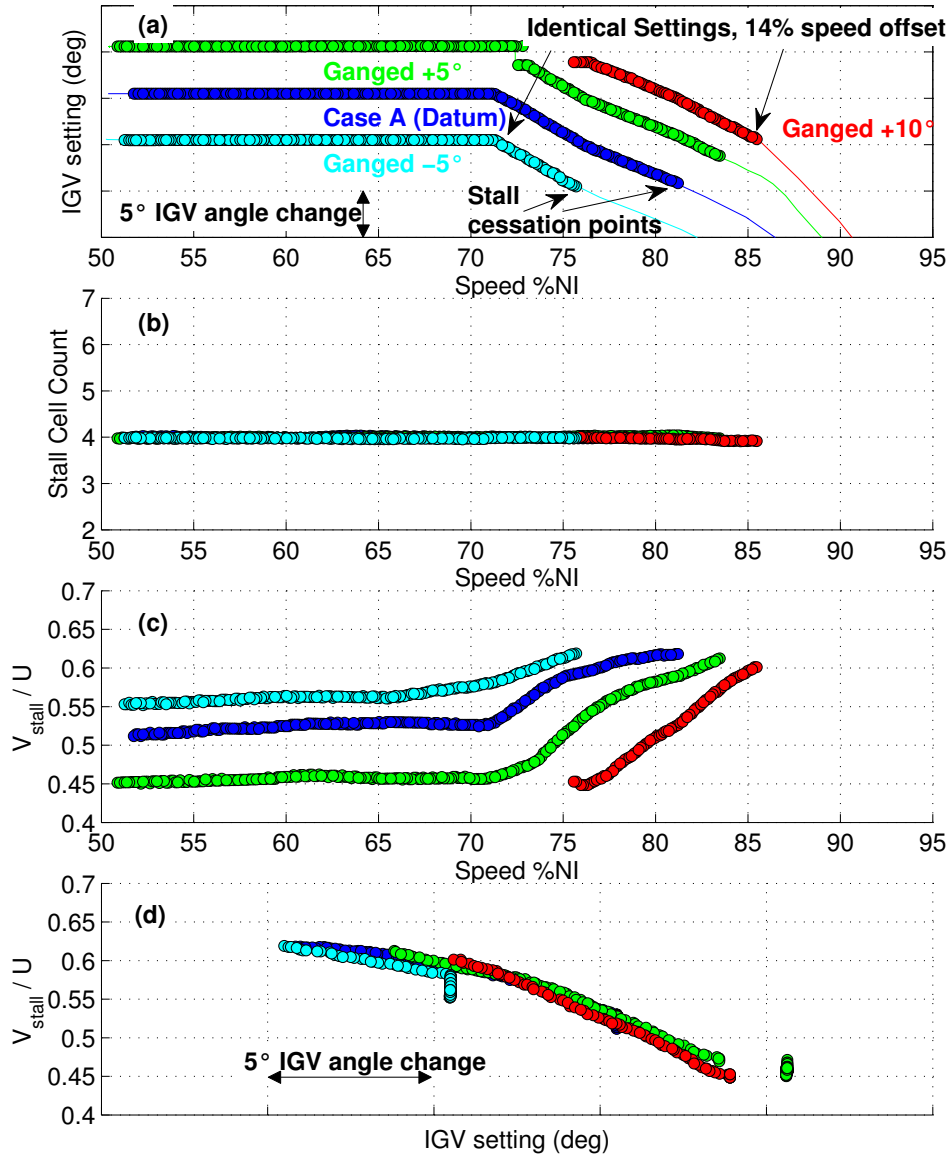


Figure 8.4: Stator 1/Rotor 2 stall, Case A acceleration with ganged offsets showing (a) IGv setting, (b) Derived stall cell count (c) Derived stall propagation speed and (d) Derived stall propagation speed plotted against IGv setting.

Figure 8.4(c) has little to do with speed itself, rather it is driven by the way the VSV settings are moved across the speed range according to a schedule. This implies that, as an approximation, similar non-dimensional behaviour to Figure 8.4(c) could in fact be obtained by holding the shaft speed fixed and simply sweeping the VSVs through the ranges shown in Figure 8.4(a).

It is also of interest to expand upon this observation by including all other observed cases of the Stator 1/Rotor 2 stall family. Of the 26 VSV schedules tested in this experiment and discussed in chapter 4, 11 showed very clear signals of sufficient clarity (i.e. signal to noise ratio) to allow stall cell counts and propagation speed to be derived. This group includes the Case C and D VSV schedules already presented (i.e. where unganged changes have also been made to the proportionality between VSV stages). The 11 clear cases are shown in Figure 8.5 in terms of (a) cell count against shaft speed, (b) V_{stall}/U against shaft speed and (c) V_{stall}/U against IGV setting, with the colours chosen to represent differing numbers of stall cells observed in each case. Figure 8.5(a) confirms that cell counts of between 3 and 6 are always observed from 50% to 90% speed and suggests that unganged changes are considerably more powerful than ganged changes in influencing the observed number of stall cells.

The variation in stall propagation speed (V_{stall}/U) in Figure 8.5(b) at a given shaft speed is considerable, with variations $0.4U$ to $0.6U$ quite typical at any given speed. When visualised in Figure 8.5(c) against the IGV setting, however, combined with also grouping the observed stall behaviour into 3,4,5 and 6 cell families, two trends emerge. Firstly, it appears that when there are fewer stall cells present, they travel faster even at a fixed VSV settings. Secondly, when the number of stall cells is fixed, it can be seen that the propagation speed is determined more by the VSV settings than by the shaft speed.

A key observation at this stage therefore is that as the VSVs are closed further, the stall propagation speed appears to also drop. Before presenting a more fundamental interpretation of this result, the next section explores whether the Rotor 1 family of rotating stall exhibits the same behaviour.

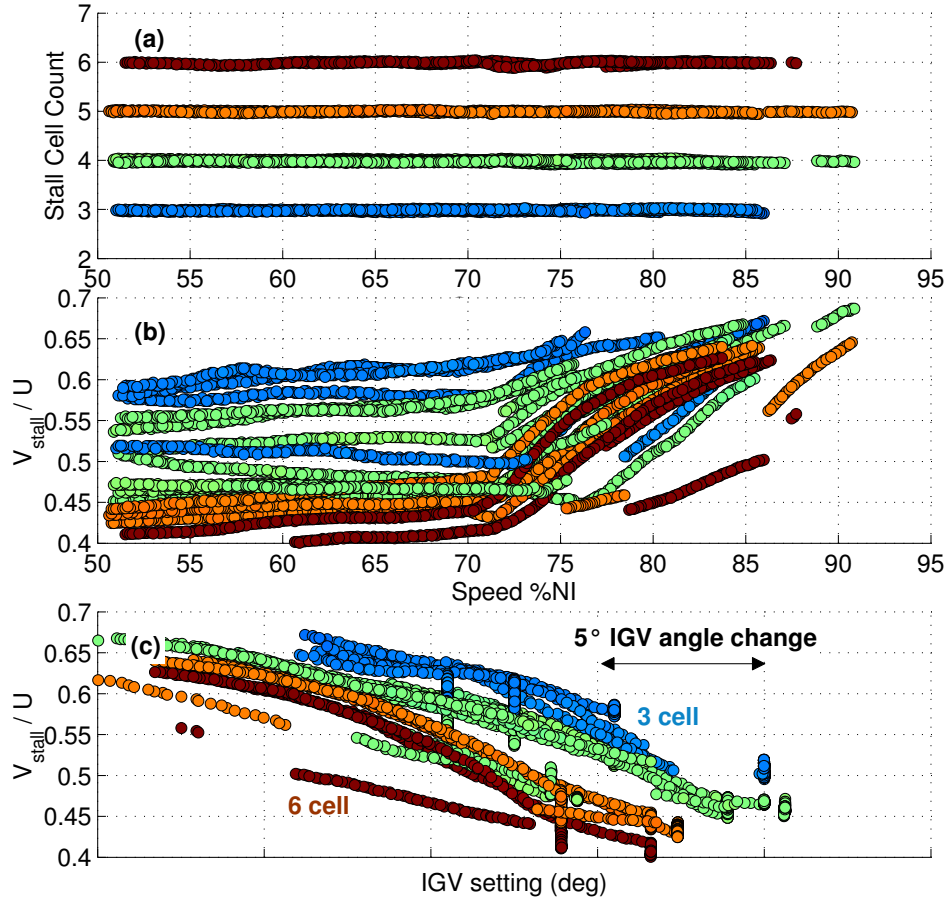


Figure 8.5: All observed cases of Stator 1/Rotor 2 stall phenomenon for acceleration manoeuvres with multiple VSV schedules showing (c) Derived stall cell count (b) Derived stall propagation speed and (d) Derived stall propagation speed plotted against IGV setting.

8.2.2 Effect on Rotor 1 stall family

The same experiment as that shown in section 8.2.1 is now studied at higher speed, where the higher frequency phenomenon associated with Rotor 1 stall (see chapter 5) was observed. For Case A and the ganged open/closed cases considered here, the physical stalling behaviour closely resembles that presented in Sections 5.2.3 and 5.2.4, which consisted of a very high number of cells, close to the Rotor 1 blade count and resulted in both noise propagation and vibration in the second torsion mode. For brevity, the pressure transducer and strain gauge data for each of these cases is not shown here and the investigation proceeds to the derived stall parameters in Figure 8.6.

Figure 8.6(a) shows the IGV setting against shaft speed for both the datum and ganged open/closed cases (recalling once again that the Stator 1 and 2 settings were also changing to maintain linear proportionality), for which it can be seen that the ganged closed 5° case at 90% speed has identical VSV settings to the datum case at 86% speed and the gang open 5° case at 81% speed. In Figure 8.6(a) solid points are shown where the stall activity could be observed (e.g. 80-88% speed for Case A), however the complete manoeuvres extended across a wider speed range, shown by dashed lines in the regions where Rotor 1 stall activity could not be observed.

The derived stall cell counts for these 3 cases are shown in Figure 8.6(b). Initially, for each case, a multiple mode response consisting of 25 to 30 lobes can be observed, still with remarkable clarity using the frequency shifting method presented in detail in the earlier chapters. The presence of multiple modes was discussed in Section 5.2.2 and is considered an indication of a spatially non-uniform and time varying stall cell pattern. Based on chapter 7, this disturbance appears to have a similar vortical structure to that described by Pullan[17]. For all 3 VSV settings shown, this stall activity abruptly changes above a certain speed, with the cell count jumping to a higher value equal to or even in excess of the blade count (34), which then falls again with further acceleration. In Section 5.2.4 this was tentatively linked to the stall phenomenon coupling with the second torsion mode and will be returned to later in this chapter. It is of interest at this stage to note that the 3 cases presented here with ganged open and closed offsets all show similar behaviour but at different speeds.

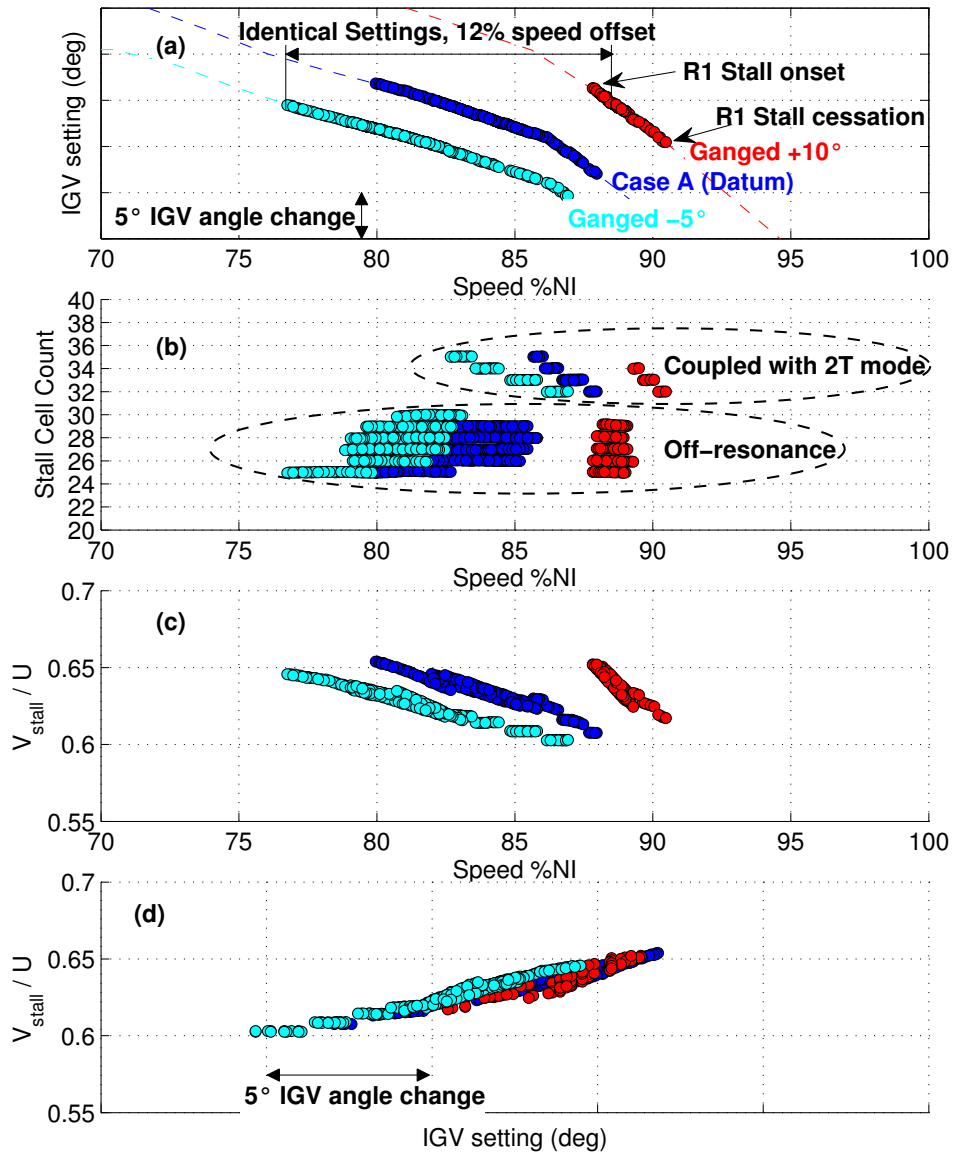


Figure 8.6: Rotor 1 stall, Case A acceleration with ganged offsets showing (a) IGv setting (dashed lines indicate complete manoeuvre), (b) Derived stall cell count (c) Derived stall propagation speed and (d) Derived stall propagation speed plotted against IGv setting.

Figure 8.6 also shows the derived stall propagation speed V_{stall}/U against (c) shaft speed and (d) IGV setting. The behaviour again is evidently similar to that in the previous section - the stall propagation speed is almost uniquely set by the VSV settings and is independent of the speed of the compressor. This confirms that both observed families of rotating stall show the same behaviour, though importantly Rotor 1 shows an *increase* in V_{stall}/U as the VSVs are closed, in contrast to the Stator 1/Rotor 2 stall family in Figure 8.4(d), for which V_{stall}/U *falls*. The next section considers this dependence on the VSV settings at a more fundamental level.

8.3 Physical interpretation

As was discussed in chapter 2, the “idealised” characteristic for a compressor stage can be derived from velocity triangles to give Equation 8.1.

$$\frac{\Delta H}{U^2} = 1 - \frac{V_x}{U}(\tan\alpha_0 + \tan\alpha_2) \quad (8.1)$$

This relates the work coefficient $\frac{\Delta H}{U^2}$ to the flow coefficient $\frac{V_x}{U}$ along a characteristic, where the flow angles α_0 and α_2 are the absolute rotor inlet and relative rotor exit values respectively (see Figure 2.1). For simplicity this is considered here only at mid-span.

Where the stagger angles are fixed, it can be assumed that the stator exit angle (α_0) and rotor exit relative swirl angle (α_2) remain constant (i.e. fixed deviation). It can be seen from Equation 8.1 that this case results in an ideal characteristic consisting of a straight line passing through a work coefficient of 1 at zero flow. For a variable stagger stator, α_0 will be set purely by the vane setting angle, again assuming the vane deviation is constant, a value which *is* varied during this experiment (the setting angle changes according to the VSV scheduled against speed and ganged offsets were also applied to this schedule). Closure of the variable stator will therefore cause α_0 to rise, moving the operating point moving onto a characteristic at lower flow coefficient.

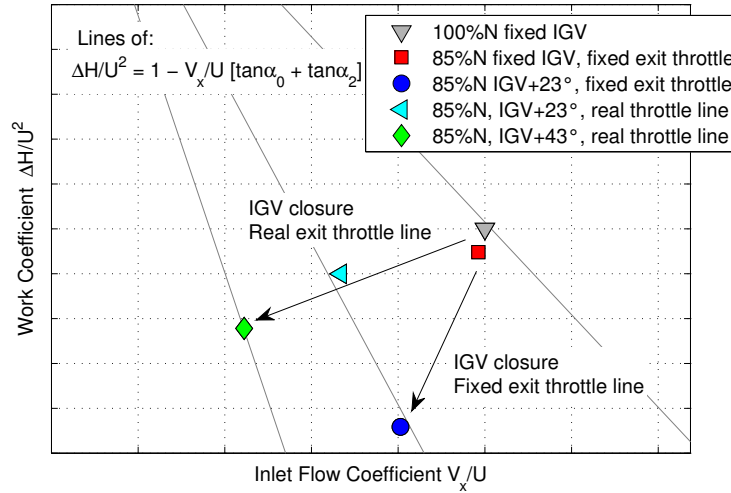


Figure 8.7: Throughflow model of front stage only, with operating points at 100% speed and 85% . IGV settings from 0-43° modelled.

A key benefit of considering the operation of compressors in this more fundamental manner is that the characteristic upon which the stage operates is also *independent of speed*. This fundamental behaviour has already been presented in chapter 6, using the 2D throughflow model, for which the non-dimensional map of stage 1 is repeated in Figure 8.7. With the IGV held fixed, the points for both 100% and 85% speed (with the stage 1 exit flow function held fixed) lie on the same characteristic and the change in flow coefficient is very small. However, closure of the IGV at a fixed speed causes a pronounced shift to a lower flow coefficient. When viewed this way the characteristics become speed independent but are heavily influenced by the VSV setting (though the VSV setting is normally varied with speed).

Consider, therefore, how the operation of the front stages changes through the manoeuvres presented earlier in the chapter. Using transient measurements of inlet massflow recorded during each of the accelerations, the approximate flow coefficient can be derived. Figure 8.8(a) shows this against speed for the open and closed cases, in addition to the Case A schedule, where the VSVs are moved together (ganged). At any given speed a difference in flow coefficient exists between the different ganged VSV offset cases and this is seen across the speed range covering both stall families. The approximate boundary is shown at which the low frequency Stator 1/Rotor

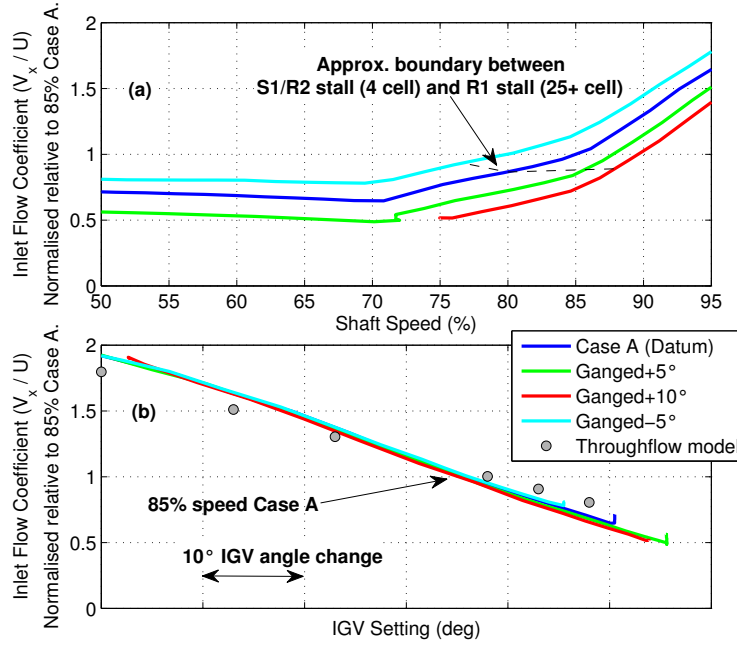


Figure 8.8: Instantaneous inlet flow coefficient derived from transient mass-flow measurements plotted against (a) Shaft Speed and (b) VSV setting. Flow coefficient is normalised relative to 85% speed.

2 stall signal clears and the higher frequency Rotor 1 stall signal appears (shown by the broken black line). Interestingly this boundary is closer to a line of constant flow coefficient than to a line of constant speed, which again supports the idea that the behaviour is largely independent of speed.

Figure 8.8(b) shows the same measured flow coefficients plotted against the IGV setting (with Stators 1 and 2 also moving according to the ganged system proportionality). The ganged changes all collapse onto a single line and the flow coefficient is almost uniquely determined by the VSV setting angles. Also shown in Figure 8.8(b) are several 2D throughflow simulations performed across a range of VSV settings, all at 85% speed, covering most of the range of VSV settings tested in the experiment. This demonstrates that the throughflow model (in which the viscous effects of as loss, deviation and blockage are held constant at each point) is accurately matching the measured rate of reduction in meanline inlet flow coefficient with VSV closure. These 2D simulations will be used later in this section to consider the effects local to Rotor 1 tip and Stator 1/Rotor 2 hub.

Figure 8.8(b) therefore shows that, with the VSVs held fixed in a particular setting, the compressor front stages will be forced to operate on a fixed characteristic at near constant flow coefficient V_x/U over a range of speeds. This in turn also determines the non-dimensional velocity triangles throughout the front stages and fixes the swirl velocity V_θ/U . The observations in the earlier sections of this chapter likewise show that the stall velocity V_{stall}/U is also nearly constant in this situation for cases where the number of stall cells remains fixed. As the VSVs move, these non-dimensional parameters change and each assume a different unique value. This suggests that a fundamental link exists between the stall velocity and some characteristic flow velocity. Historically V_{stall}/U has been widely presented as the non-dimensional form for stall propagation speed. The remainder of this section addresses the question of which flow velocity V_{stall} scales with, such that an alternative non-dimensional form may be obtained for stall propagation speed, which becomes invariant to changes in the VSV settings.

Using the throughflow calculations for the 6 different VSV settings as were shown alongside measurements in Figure 8.8(b), the *local* flow velocity components (V_θ and V_x) at inlet to Rotor 1 tip and Stator 1 hub can be obtained. This results in the relationship shown in Figure 8.9 between the *local* V_x and V_θ components against IGV setting angle. Note that these results are consistent with the radial profiles at Rotor 1 inlet shown previously in Figures 6.7(a) and (c), as can be seen by comparing values at the tip.

At both Rotor 1 tip and Stator 1 hub, Figure 8.9 shows that the local flow coefficient (V_x/U) is falling as the VSVs are closed down, which is consistent with the behaviour shown in a one-dimensional sense in Figure 8.8(b). The correspondingly swirl velocity V_θ/U always increases at both locations as the VSVs are closed, with the value also rising across the rotor as expected to give a higher swirl at Stator 1 inlet. Note that the axial and swirl velocity components are linked to each other by the velocity triangles and a simple hand calculation confirms the relationship seen in Figure 8.9 at both locations.

Returning to the stall behaviour, it is recalled that the stall cells of both families propagate circumferentially in the direction of shaft rotation, which is also the direction of absolute swirl at all locations. It is therefore logical to

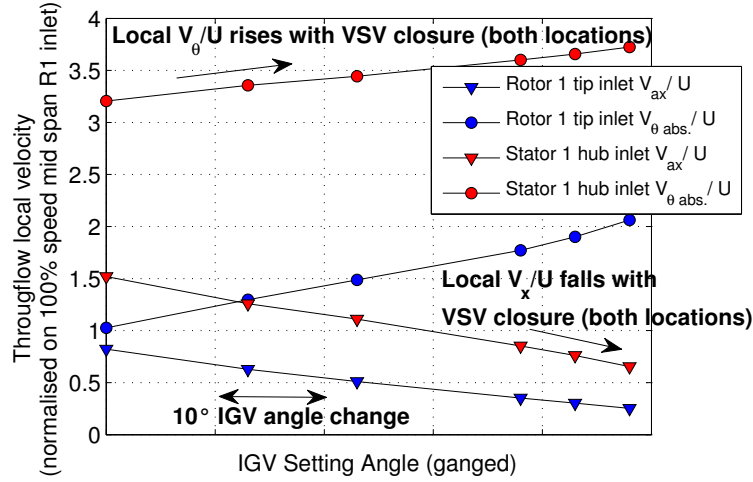


Figure 8.9: Throughflow models at 6 different IGV positions showing the change in V_{θ}/U and V_x/U at inlet to Rotor 1 tip and Stator 1 hub.

“test” whether the changes in stall cell propagation speed presented could be linked to changes in swirl velocity occurring as the VSV settings are adjusted. This can now be investigated using Figure 8.9 which shows that IGV closure results in an increase in absolute V_{θ}/U at both locations.

For the Rotor 1 tip stall, Figure 8.6(d) showed that the stall propagation speed V_{stall}/U rises as the VSVs are closed (independently of speed), which aligns with an increase in local V_{θ}/U in Figure 8.9. This alone would tend to support the view the changes in stall propagation speed are linked to changes in swirl. However, this does not remain true when analysing the lower frequency stall behaviour seen at Stator 1/Rotor 2 hub. Figure 8.4(d) shows that for this family, closure of the VSVs results in a *fall* in V_{stall}/U , whilst Figure 8.9 shows the corresponding value of V_{θ}/U at Stator 1 inlet is *increasing*. This implies that the changes in stall propagation speed at different VSV settings cannot be attributed to changes in swirl velocity.

However, in this case, a potential link between stall propagation speed and local *axial* velocity may still be established. At Stator 1 inlet, the falling value of V_x/U (particularly at the hub) as the VSVs are closed down is accompanied by an observed fall in the stall propagation speed in Figure 8.4(d). As presented in the unsteady CFD analysis in chapter 7, this

family of rotating stall was shown to form in the Stator, with the dominant velocity and pressure gradients generated by the stall cells occurring at Stator 1 leading edge. In chapter 9 it will also be shown that similar results can be obtained by removing Rotor 2 from the model, also suggesting again that the behaviour observed is driven by the stator. For this disturbance it is therefore appropriate to define the stall propagation speed in the absolute frame of reference. Moving to the Rotor 1 family, the stall cells are of very short lengthscale restricted to the tip region. For this disturbance it is appropriate to define stall propagation speed relative to the rotor. By reconsidering the Rotor 1 behaviour in this way the trend of stall speed against V_x/U becomes consistent for both stall families. Clearly, the axial velocity is independent of the frame of reference chosen, which shows that VSV closure results in a fall in V_x/U and a fall in V'_{stall}/U for *both* stall families, where V'_{stall} is defined in the appropriate frame of reference.

Using the throughflow model results in Figure 8.9, the local axial velocity V_x/U at both locations is compared with the measurements of V'_{stall}/U for both stall families in Figure 8.10, on the same non-dimensional scale, with the Rotor 1 values now defined in the rotating frame of reference. For each stall family, all of the ganged offset cases presented in the previous sections are grouped and for ease of viewing here are given the same color. At both Rotor 1 tip and Stator 1 hub, a correlation can now be observed between the *local* V_x/U obtained from the throughflow, and the corresponding V'_{stall}/U observed in the experiment.

The correlation of stall cell speed to the local V_x/U from throughflow model supports the view that swirl velocity is not significant and implies that V'_{stall}/V_x may be a more fundamental form for stall propagation speed. Stated in this form, the normalised propagation speed remains closer to a universal value *even when the VSVs are changed by a ganged offset*. It is conceded that a further complexity not addressed here is the effect of changes in the number of stall cells, such as that shown in Figure 8.5(d), which clearly has a strong influence on V_{stall}/U . It is possible that further analysis using the CFD methods may provide additional insight into this at a fluid dynamic level.

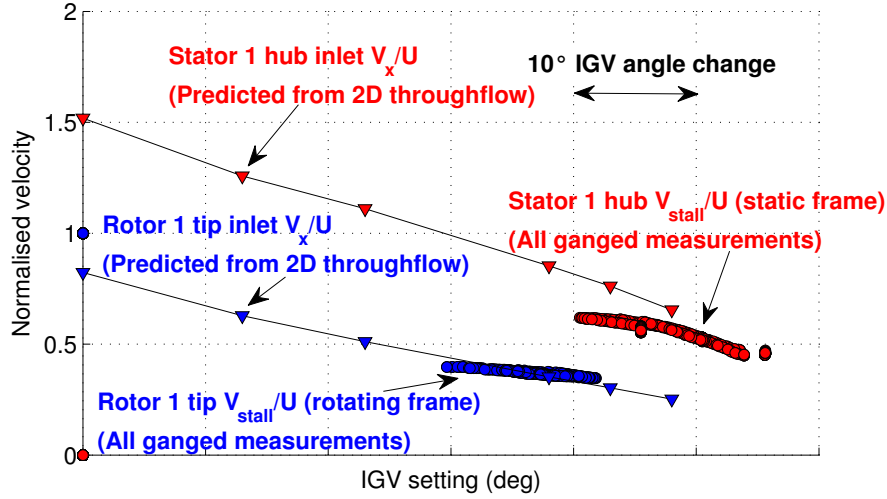


Figure 8.10: Comparison at Rotor 1 tip and Stator 1 hub of V_{stall}/U (measured) and V_x/U derived from throughflow model (normalised relative to design speed value at R1 inlet).

With the VSVs held fixed the inlet Mach number will still rise considerably across the speed range studied here which, as shown, has a weak effect on stall behaviour. This insensitivity to aerodynamic speed suggests that the behaviour is largely independent of Mach number and therefore not strongly linked to compressibility effects in the front stages.

It is also worthwhile to contrast this with other reported studies into the propagation speed of stall cells. The majority of reported experimental studies have considered propagation of *full-span* stall, consisting usually of a single cell. This prevents a direct comparison to the test case presented here which exhibits *part-span* stall, however, some useful insight into the behaviour here can still be offered. Cumpsty and Greitzer[58] proposed a model for the propagation of full-span cells, highlighting the importance of inertial effects arising from the rapid changes in momentum as the blade passages move across the boundary of the stall cell. This was shown to result in a local change in static pressure difference across the blade passages which in turn depends upon the velocity at which the cell propagates. The requirement for the local pressure changes to still sum to the overall pressure rise across the compressor was then used to derive an expression for the speed at which the cell must rotate for these effects to remain in balance.

Recalling the simulations in chapter 7, Figures 7.7 and 7.8 showed the most abrupt changes in axial velocity and static pressure also occur at the cell boundary, consistent with Cumpsty and Greitzer[58]. It is therefore suspected that a similar mechanism controls the propagation speed of the *part-span* cells in this test case. Specifically, the stall propagation speed V_{stall}/U must always be such that static pressure change induced at the cell boundary remains consistent with the overall pressure rise. As this static pressure change is driven by the difference in axial velocity (V_x/U) between stationary/reverse flow within the cell and conventional forward flow around the cell, it supports the existence of a physical link between the stall propagation speed and the local axial velocity. This offers a potential reason for the behaviour presented in this chapter and summarised in Figure 8.10.

8.4 Stall induced vibration on Rotor 1

Experimentation with ganged VSV offsets also allows some further insight to be gained into the stall induced vibration seen on Rotor 1. In chapter 5, a condition was identified whereby the Rotor 1 tip stall exhibited a large number of cells, close to the number of blades. At this condition it was also discussed in section 5.2.3 how modulation between the stall cells and the rotor resulted in a cut-on acoustic interaction which propagated upstream along the duct as a noise source. Also, section 5.2.4 discussed how the interaction of the stall cells with the second torsion (2T) mode resulted in a considerable rise in vibratory stress on Rotor 1. In this case, it was also observed how the stall pattern abruptly changed its cell count once mechanical resonance occurred. This leads to the hypothesis that a “lock-on” mechanism may be occurring in which the aerodynamics (i.e. stall cell properties) naturally adjusts to remain coupled with the mechanical vibration. Such behaviour is often reported for the problem of vortex shedding from bluff bodies and is a common cause of high cycle fatigue.

In this chapter, Figure 8.6(b) showed similar behaviour occurs on both the datum VSV schedule and the cases whereby the VSVs were operated ganged open and closed. An abrupt change in stall count was observed as the compressor was accelerated beyond a certain point.

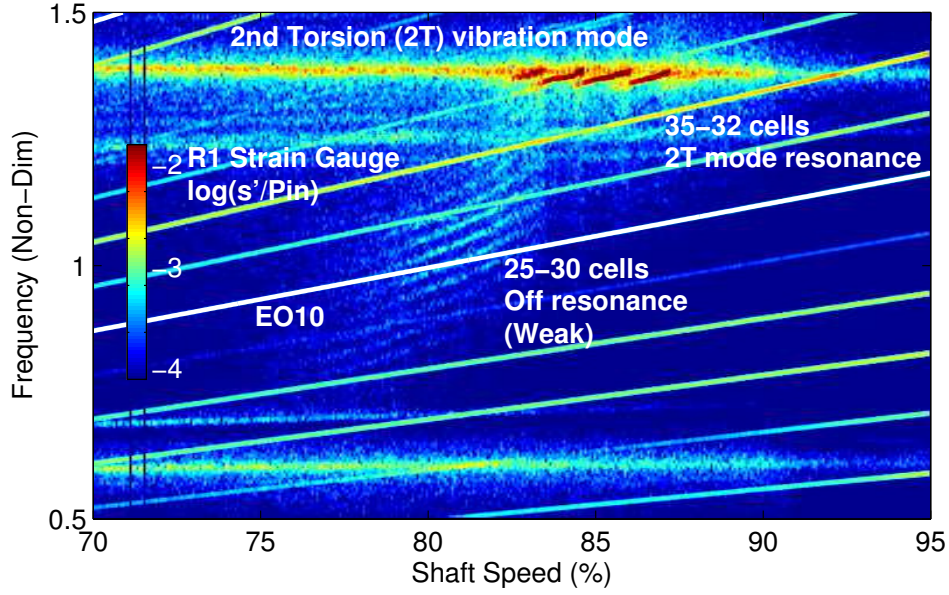


Figure 8.11: Case A ganged open 5° acceleration shown on Rotor 1 strain gauge showing evidence of rotating stall coupling with the second torsion mode.

An example of the rotor strain gauge spectrum for the ganged open 5° VSV schedule is shown in Figure 8.11, for which the interaction with the torsional mode can be seen most clearly. Between 77-83% speed a multiple mode pattern of 25-30 lobes can be identified (with the cell count confirmed by alignment to corresponding features observed on the stationary pressure transducer). As discussed in chapter 5, the presence of multiple modes is attributed to spatial and temporal non-uniformity in the stall cell pattern. Whilst this non-uniform pattern persists until 83% speed the torsional mode activity remains at low amplitude, as the stall pattern occurs with a frequency sufficiently far from the vibration mode.

However, above 83% speed, the stall activity “jumps” to become dominant at a higher frequency, now interacting with the second torsion vibration mode (which appears as a near horizontal line in Figure 8.11). In total 4 “streaks” can be observed to cross this mode, corresponding to a 35 cell pattern followed by 34, 33 and 32 cells (again readily confirmed by the frequency shifting method used throughout this thesis).

A key point here is that below 83% speed the stall phenomenon occupies a wide frequency range and is incoherent in the sense that multiple modes appear to coexist. At higher speeds, the stall phenomenon appears to become constrained to a fixed frequency in the rotating frame aligned exactly with the torsional mode. Figure 8.6(b) shows that exactly the same behaviour occurs on the ganged open and closed offset manoeuvres, but shifted to lower and higher speeds respectively.

Figure 8.12(a) shows these same measurements in terms of Engine Order in the rotating frame (i.e. frequency observed on the strain gauge normalised by shaft speed). For both the datum and ganged open/closed cases, there is initially a collection of points corresponding to the off-resonant phase with modes 25-30 present. However, at higher speeds, all 3 configurations show an abrupt jump where the cell count increases and the second torsion vibration appears. In all 3 cases the “on-resonant” behaviour always falls within a narrow band, as highlighted in Figure 8.12(a), upon which lines of constant frequency in the rotating frame have also been added. These lines correspond to the second torsion mode frequency $\pm 3\%$. Once resonance occurs, the observed stall frequency always moves into this band, with the stall phenomenon adjusting its cell count as necessary to maintain its frequency close to this constant value (of the vibration mode). This feedback from the mechanical vibration to the surrounding aerodynamics supports the hypothesis that this is a “lock-on” mechanism.

Recalling chapter 4, 26 different VSV schedules were initially tested, with unganged changed made in addition to the ganged changed shown in Figure 8.12(a). Of these cases, 13 showed this high frequency activity and torsional vibration with sufficient clarity to allow the stall cell count and propagation speed to be derived. The corresponding data is presented in Figure 8.12(b) in exactly the same format as Figure 8.12(a) in addition to the ganged offsets. It is again evident from Figure 8.12(b) that there exists an off-resonant phase during which stall is observed across a range of frequencies/mode orders, which then abruptly locks onto to the torsional mode at constant frequency within the narrow band indicated. This further confirms the presence of a lock-on mechanism and shows that the behaviour generalises to multiple VSV schedules.

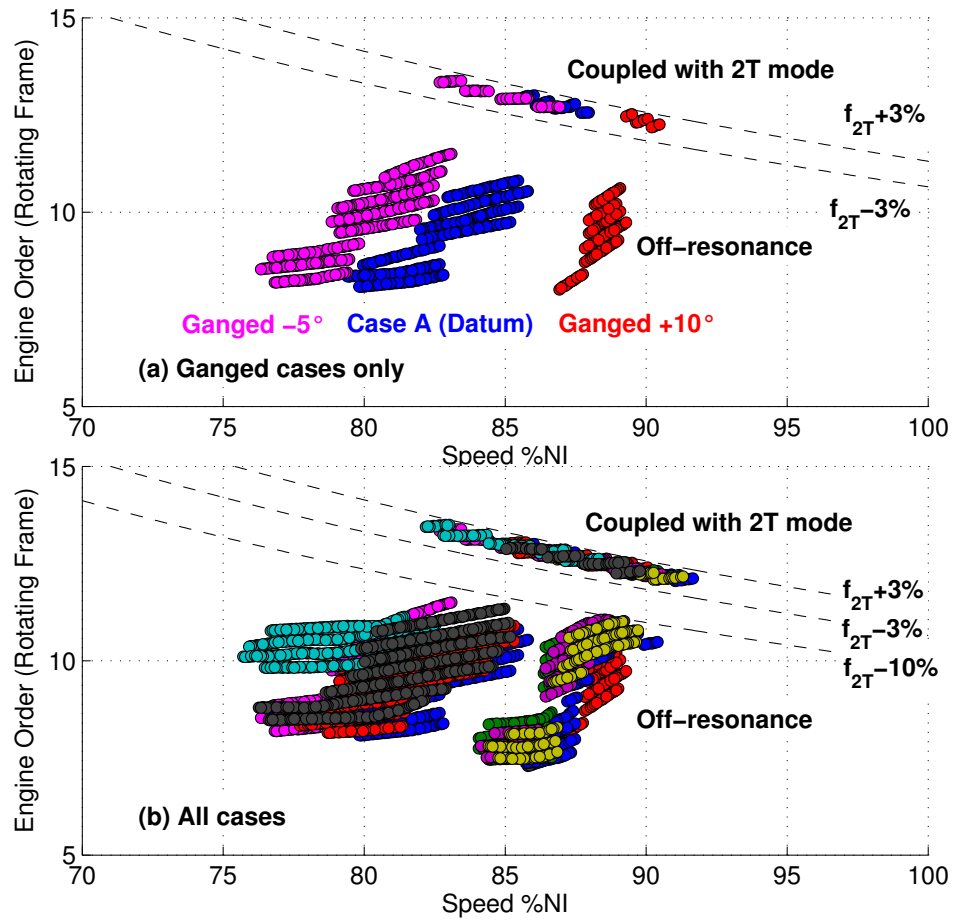


Figure 8.12: Rotor 1 high frequency stall phenomenon - observed engine order vs shaft speed for (a) Case A and ganged offsets and (b) All observed cases. Lines of constant frequency (in rotating frame are also added).

This therefore explains the reason why the cell count initially rises to 35 (see Figure 8.6(b)) and then drops again with further acceleration of the shaft - once the stall pattern has become coupled with the vibration mode the increase in shaft speed is offset by a stepwise fall in the number of cells to maintain a near constant frequency. The stall pattern decouples from the vibration mode at higher speed when the aerodynamic matching of the compressor has improved, approaching its design condition, at which point the stall cells vanish and the rotor vibration disappears.

It is also of interest to quantify the frequency separation required between the stall phenomenon and the vibration mode for “lock-on” to occur. Figure 8.12(b) shows also a line corresponding to a constant frequency 10% lower than the second torsion mode. It can be seen that off-resonant activity may occur close to this frequency, however there appears to be no stable stall activity within 10% frequency of the vibration mode which does not also become resonant. This quantifies the approximate level of frequency margin that must be achieved between the stall and vibration mode to ensure non-integral vibration does not occur. It is important to stress that whilst this frequency separation appears to be common to different VSV schedules, the 10% value is likely to be specific to this test case and also potentially even quite specific to this vibration mode.

8.5 Concluding remarks

In this chapter further experimental results have been presented and discussed concerning both the Stator 1/Rotor 2 (low frequency) and Rotor 1 (high frequency) stall families. The experiment consisted of performing acceleration manoeuvres with the VSVs of all stages “ganged” open and closed relative to the Case A schedule, allowing the effect of VSV setting to be isolated from the effect of speed when studying the resulting stall behaviour.

Evidence was presented for both families of rotating stall showing that the stall propagation speed V_{stall}/U has an almost unique value for a given VSV setting, which importantly was found to be independent of the aerodynamic speed/shaft speed of the compressor. Moreover, this also implies insensitivity to the Mach number which suggests the physics governing the propagation of stall cells can be considered to be largely independent of compressibility. The unique relationship between V_{stall}/U and the VSV settings was also linked to the operation on the work coefficient/flow coefficient map, which suggests that a more fundamental non-dimensional relationship may exist between the observed V_{stall}/U and the instantaneous V_x/U local to the cell.

Finally, the coupling between the high frequency Rotor 1 stall phenomenon and the second torsion mode was studied by considering the effect of ganged VSV offsets. This showed that the higher frequency stall activity associated with Rotor 1 initially appears across a wide frequency range, but spontaneously appears to “lock-on” to the second torsion mode. Under this coupled condition the stall pattern appears to naturally adjust its cell count spontaneously, such that the frequency in the rotating frame remains constant. A basic frequency separation criterion of 10% between the stall phenomenon and the vibration mode can also be estimated from the measurements and forcing within this range does not appear to be stable without coupling with the mode.

9 Numerical study into flowfield coupling mechanisms.

This chapter uses the validated unsteady model presented in chapter 6, to seek a deeper insight into the nature of rotating stall and the physical mechanisms which govern the dynamics of stall cells. Particular consideration is given to understanding the physics that dictates the cell count and length-scale upon which the rotating stall pattern stabilises, which the previous chapters have shown may occur across a wide range depending on the VSV settings.

9.1 Physical mechanisms

It is useful to discuss two related concepts reported in the open literature that are explored further in this chapter.

Firstly, compressors operating on negatively sloped characteristics are able to attenuate circumferential distortion. For example, Plourde and Stenning[61] showed how a compressor subject to circumferential total pressure distortion at inlet, which delivers to uniform static pressure at exit, must produce a static pressure perturbation at the compressor inlet face. A reduction in static pressure in the low total pressure sector allows the velocity fluctuations associated with distortion pattern to be suppressed. The effect naturally must decay progressively upstream where static pressure must become uniform. This provides a useful insight into how stages operating on a very stable characteristic may have a natural calming effect on circumferential distortion patterns.

Secondly, flowfield coupling exists between components of a compression system and between stages within a compressor. For example, Longley and Hynes [5] showed experimentally how the front stage of a compressor could be operated well beyond the stall point encountered when tested either as an isolated stage or within a well matched compressor. This was achieved by deliberately mismatching the compressor through re-stagger (closure) of the blades in the downstream stages to operate at a lower flow coefficient and thus on strongly negatively sloped characteristics. Their interpretation was that, for the mismatched build, stall cells forming in the front stage were prevented from growing beyond short lengthscale by the attenuating effect of the stable downstream stages.

These cases offer useful physical insight into how stalled stages of a compressor may behave, in terms of the lengthscales of rotating stall, when paired with stable stages which operate on a negative characteristic slope.

Such a situation arises in the experiment presented in this thesis. For example, chapter 4 showed how deliberate closure of Stator 2 causes a reduction in the work done by Rotor 3, with a subsequent fall in flow at inlet to the Stator 2/Rotor 3 block which moves the upstream stages along their characteristics towards stall. Block characteristics obtained from the 1D meanline model showing this effect were presented in chapter 4 and are repeated in Figure 9.1. It can be seen that the deliberate closure of Stator 2 also moves Stator 2/Rotor 3 onto a steeper, more stable characteristic at part-speed whilst simultaneously moving the upstream blade rows into stall.

This behaviour is analogous to that presented by Longley and Hynes[5] as the stalled front stages are paired with a very stable downstream stage which operates on a negatively sloped characteristic. It is therefore of interest to question whether some form of dynamic coupling exists between the destabilising effect of stall cells which form and grow in the front two blocks, and the stabilising effect of the strong Stator 2/Rotor 3 characteristic.

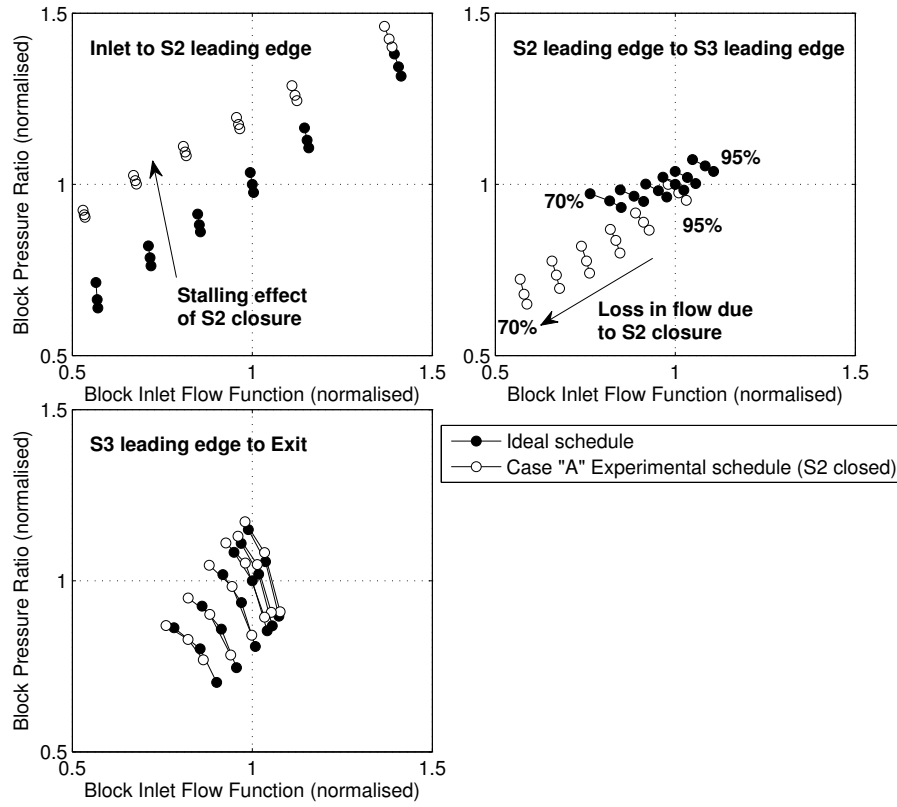


Figure 9.1: Repeat of Figure 4.2: 1D meanline model, operating maps for front stages at six speeds, Stator 2/Rotor 3 and rear stages. Shown on idealised VSV schedule and experimental schedule with S2 overclosure.

9.2 Axial coupling with downstream stage

In order to study the possibility of flowfield coupling mechanisms in this test case, a steady state numerical strategy was developed in which the Stator 3 vane was treated as an additional VSV and re-staggered within the model. The previous chapters showed how deliberate closure of Stator 2 can be used to trigger stall in the upstream stages.. However, a similar axial matching effect can also be achieved by instead closing Stator 3, which also raises the operating point of Stator 2/Rotor 3 up its characteristic. By careful iterative choice of the Stator 2 and 3 vane angles, the slope of the Stator 2/Rotor 3 characteristic can be manipulated whilst the upstream blocks (where stall originates) are always held fixed at the Case A operating point. This allows the coupling mechanisms between stages to be studied.

Figure 9.2 shows the procedure adopted, which was performed using the single passage steady state CFD model. Starting from Case A, Stator 2 was opened by 5° (to Case Z), for which the Stator 2/Rotor 3 inlet flow coefficient increases. This also has the effect of pulling the upstream blade rows down their characteristics and suppressing stall. Following this change, the Stator 3 vane was then closed in steps until the flow coefficient at inlet to the front block was reduced back to its Case A level. In this situation the separated regions of interest, at Rotor 1 tip and Stator 1/Rotor 2 hub, are both operating at aerodynamic conditions identical to Case A (i.e. identical velocity triangles). The downstream Stator 2/Rotor 3 block, however, is now operating on a weaker (less negatively sloped) characteristic. This process was performed twice to give three different settings for comparison: Case A (Stator 2/Rotor 3 most stable), Case A2 (Stator 2/Rotor 3 less stable) and Case A3 (Stator 2/Rotor 3 least stable). Naturally, this process simultaneously results in the opposite trend on Stator 3/Rotor 4, with the characteristic for this block becoming progressively more stable.

Having established these three VSV configurations, unsteady models were generated for each according to the methodology presented in chapter 7. It should be noted that for this study the interface in the calculation domain from full annulus unsteady to single passage steady was also moved one stage further downstream to Rotor 4 exit.

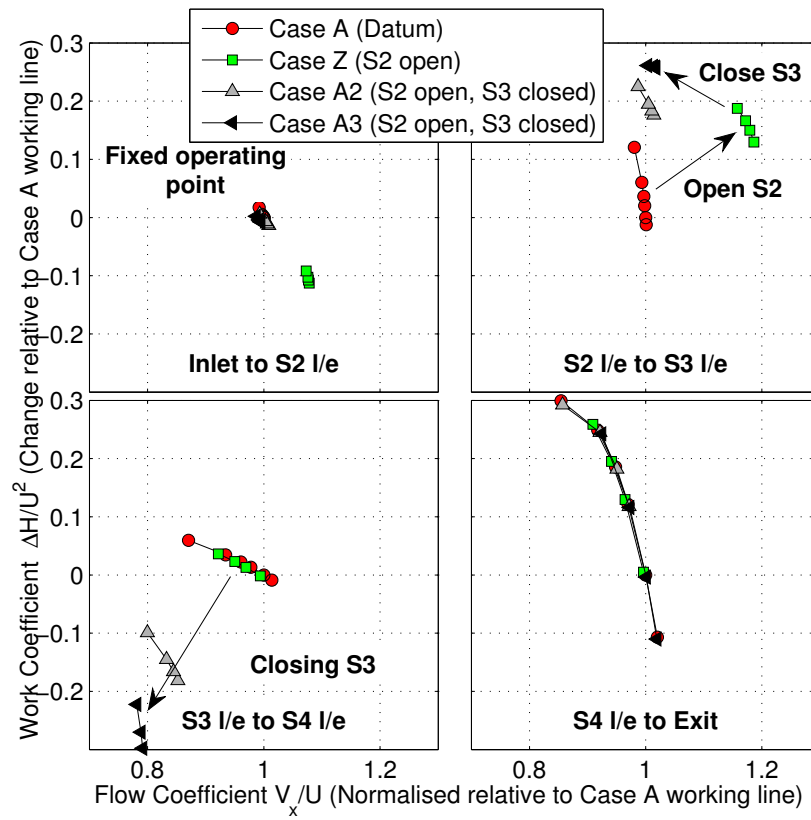


Figure 9.2: Steady state model 85% speed, block characteristics showing the effect of closing S3 and opening S2 relative to Case A.

Figure 9.3 shows the resulting flowfield obtained from these three calculations at Stator 1 and 2 trailing edge in terms of axial velocity. For Case A the simulation results in a 7 cell pattern, exactly consistent with the findings of chapter 7 and simply confirms that the extension of the unsteady part of the domain by one stage has no impact on the resulting stall cell behaviour. This stall pattern is no longer visible in the corresponding flowfield at Stator 2 exit confirming that the very stable Stator 2/Rotor 3 block is able to suppress the growth of stall cells downstream.

The more important finding for this study is the trend from Case A to A2 to A3, for which the Stator 2/Rotor 3 characteristic becomes progressively less stable. From Figure 9.3 it can very clearly be seen that this effect results in fewer cells forming, from 7 to 5 to 3. Noting that the Figure 9.2 showed that the operating point of Stator 1/Rotor 2 was held constant through this sequence, the behaviour predicted here suggests that some form of axial coupling between stages does indeed exist and can strongly influence the aerodynamics of the flow once stalled. The corresponding flowfield at Stator 2 exit for Case A3, for which Stator 2/Rotor 3 has been destabilised considerably, shows that the 3 stall cell pattern extends further axially rearwards. Very faint evidence of the 5 cell pattern can also be seen at Stator 2 exit for Case A2.

The following interpretation is therefore made of this predicted behaviour. As the Stator 1 hub region moves towards stall due to the throttling effect of downstream vane closure (either Stator 2 or Stator 3), stall cells initially form once the static pressure rise demanded of the vane hub becomes excessive. This initially occurs with cells at very short lengthscale (i.e. flow breakdown in the individual blade passages) which grow both axially, radially and circumferentially. This growth continues as an unstable transient process until a stabilising effect, in this case the downstream stage, resists further growth and imposes a lengthscale on the forming stall cells. This interaction persists until the combined stalled and unstalled stage reach an equilibrium, consisting of a “most stable” stall cell arrangement.

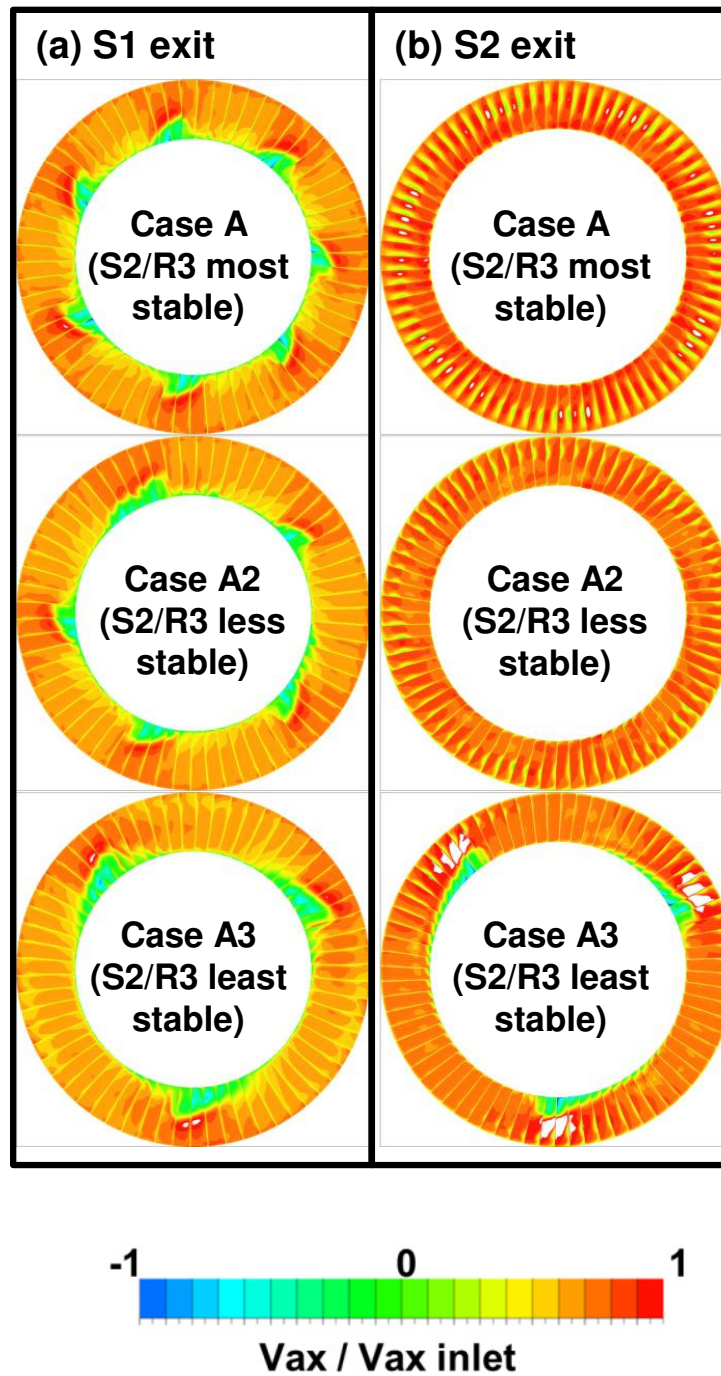


Figure 9.3: Unsteady model at 85% speed, Stator 1 and 2 exit axial velocity field, showing the effect of closing S3 and opening S2 relative to Case A. Both Stage 1 and 2 are operating at nominally identical conditions.

9.3 Front stage only model

The previous section provides evidence that the stalled regions of the compressor are stabilised by the neighbouring unstalled regions. Moreover, the predicted behaviour suggests that the fundamental properties of rotating stall, such as the number of cells and their spatial extent, are influenced by this coupling mechanism as the stable regions prevent the growth of cells beyond a certain size. Whilst the effect of Stator 2/Rotor 3 is shown to be powerful, it is quite feasible that the stall cell lengthscale may be influenced by other regions of the compressor.

Recalling chapter 6, the radial equilibrium effects arising from VSV closure at part-speed were studied. The swirl imparted by the IGV results in a hub-low static pressure gradient, which reduces the axial velocity at the outer span causing Rotor 1 tip to stall. Furthermore, the static pressure gradient is not attenuated by Rotor 1, across which the swirl rises further, but instead is left largely for Stator 1 to eliminate. This process was shown to cause simultaneous stall of the Rotor 1 tip together with the Stator 1 hub. However, the same effect also results in Rotor 1 hub moving to very negative incidence, together with the outer span of Stator 1 remaining at quite tolerable diffusion levels, as shown in Figures 6.7(d) and 6.9 respectively. Both of these regions will also have a stabilising effect on the formation of stall cells. For example, stall forming in Stator 1 hub will be prevented from growing axially upstream into the stable Rotor 1 hub passages. Similarly any radial growth of the cells will be limited once sufficiently far outboard that the local Stator 1 aerodynamics becomes favourable.

The potential for flowfield coupling of this type can be most easily explored by removing all blade rows downstream of Stator 1 from the model such that the Stator 2/Rotor 3 effect discussed in the previous section is no longer possible. Slight adjustment of the Stator 1 domain was made to blend the hub and casing lines into a parallel annulus duct which is extended downstream of the domain as shown in Figure 9.4. A gridded nozzle placed at the exit of this domain was then used to position this front stage only model at various points on its characteristic. The subsequent sections present the steady and unsteady predictions obtained from this model.



Figure 9.4: Single passage domain for simplified model of front stage only.

9.3.1 Comparison - single stage to 8 stage steady simulations.

The steady state predictions obtained with the IGV and Stator 1 at the Case A settings, using this simplified model of the compressor front stage, are shown in Figure 9.5 with the throttle closed in steps to produce a characteristic. A subset of points along this characteristic are labelled 1 to 6 for discussion purposes and the corresponding data from the 8 stage steady computations for Cases A and Z also shown. In order to differentiate between stall of the rotor and stator, two characteristics are shown, with Figure 9.5(a) presenting the performance of the front *stage* in terms of total pressure rise coefficient ($\Delta P_0/\rho U^2$) against stage inlet flow coefficient (V_x/U) and Figure 9.5(b) presenting the performance of the *Stator 1 alone* in terms of static pressure rise coefficient ($\Delta P_s/D$) against Stator 1 inlet flow coefficient (V_x/U).

Considering the stage performance first, Figure 9.5(a) shows how closure of the gridded exit nozzle behind this front stage model moves the operating point along a characteristic passing exactly through the 8 stage CFD simulations for Cases Z and A. These 8 stage model characteristics were generated by adjustment of Stator 2 which resulted in a drop in the flow swallowed by Rotor 3, thus reducing the flow function at exit to the front stage in a similar manner.

Figure 9.5(b) shows how the Stator 1 stalls abruptly once throttled beyond point 2 and drops to a considerably lower static pressure rise. This does not appear to significantly impair the ability of the rotor to deliver work as the stage as a whole remains on an unstalled characteristic. Stated another way, the loss in stator pressure rise between points 2 and 3 only has a relatively small throttling effect on the operating point of Rotor 1.

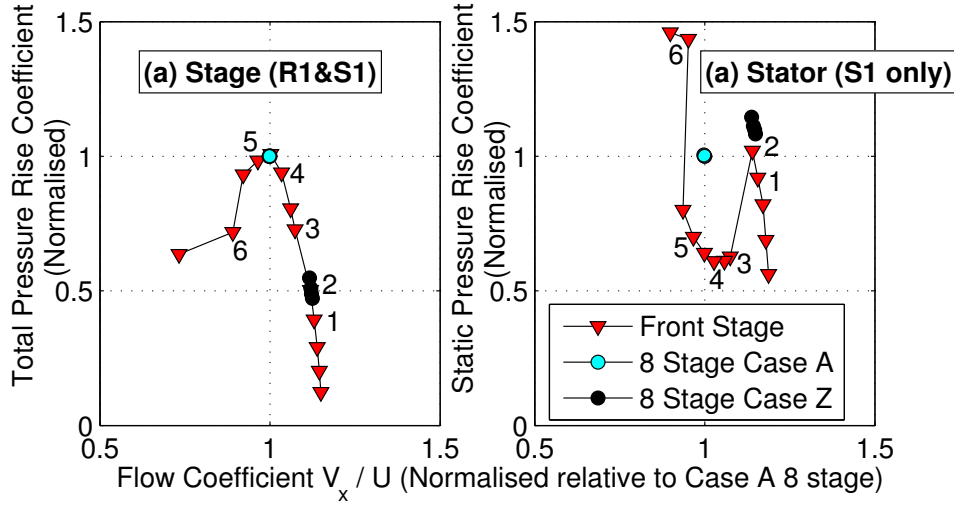


Figure 9.5: Steady state predictions at 85% speed for front stage only model compared to 8 stage computations of Case A and Z (a) Stage total pressure rise, (b) Stator 1 static pressure rise.

However, in the full compressor case where Rotor 2 is present downstream of Stator 1, the effect of the stator stalling will become considerably more powerful on Rotor 1, as the pressure ratio of Rotor 2 would also be reduced, throttling the front stage considerably more.

Further closure of the throttle moves the stage further up its characteristic whilst the Stator 1 static pressure rise remains at a near constant low value. Between points 4, 5 and 6 the stage characteristic flattens and rolls over onto a positive slope, consistent with the Rotor 1 tip becoming stalled. Interestingly, once this occurs it appears that Stator 1 is able to recover and deliver a considerably higher pressure rise. Physically this is because the blockage effect of Rotor 1 tip stalling causes flow to be diverted radially inboard, which eliminates the Stator 1 hub stall.

Comparison of the single stage model to the 8 stage models shows that Case Z operates at flow coefficient above the Rotor 1 and Stator 1 stalling point and thus is free of stall, whilst Case A clearly operates with Stator 1 stalled (though the drop in static pressure rise is less pronounced on the 8 stage model), with Rotor 1 operating near the top of its characteristic.

9.3.2 Comparison - single stage steady to unsteady model.

It is of interest to expand this simplified model into a full annulus domain and explore the behaviour in an unsteady sense. Figure 9.6 shows the characteristics (with the same views as shown in Figure 9.5) comparing the steady and unsteady models. The full annulus cases were initialised using the lowest point on the characteristic which was free of stall and the exit throttle closed to allow stalled flow to evolve in an unsteady manner. For discussion purposes the unsteady solutions are labelled ‘ 1_u - 8_u ’. The simulations at each point were left to evolve naturally and were stopped approximately 3 revolutions after rotating stall was observed to stabilise. For the most extreme points (7_u and 8_u), no stable solution could be obtained in a practical timescale.

The stage characteristic predicted by the unsteady model overlays the steady model almost exactly from points 1_u to 6_u , until Rotor 1 becomes stalled. Beyond the stall point it appears that the steady model drops abruptly to a lower pressure rise, a drop which the unsteady model does not appear to show. Instead, the unsteady model rolls over in a much more gradual sense as Rotor 1 stalls. Considering the Stator 1 behaviour, both models stall abruptly beyond point 2_u , with the steady model falling to a lower static pressure rise. Although, the stator-only characteristics for the unsteady model must be treated with caution as they are determined from a pitch-wise average of the Stator 1 inlet flowfield in which stall cells are present and the simple pitchwise averaging process may not adequately capture the correct mechanisms. Due to the more gradual stall of Rotor 1 tip, the sudden recovery of Stator 1 seen in the steady model (due to blockage effects as discussed earlier) does not appear in the unsteady case.

The differences seen between the two methods imply that the Rotor 1 tip stalling behaviour predicted by the steady model is likely to be pessimistic. Such behaviour is not surprising as the flow naturally stalls in an asymmetric manner, whilst the steady single passage model is forced to stall at a certain lengthscale (i.e. in an axisymmetric manner). Consequently, this does not allow for any redistribution of flow in the circumferential direction as would occur during rotating stall.

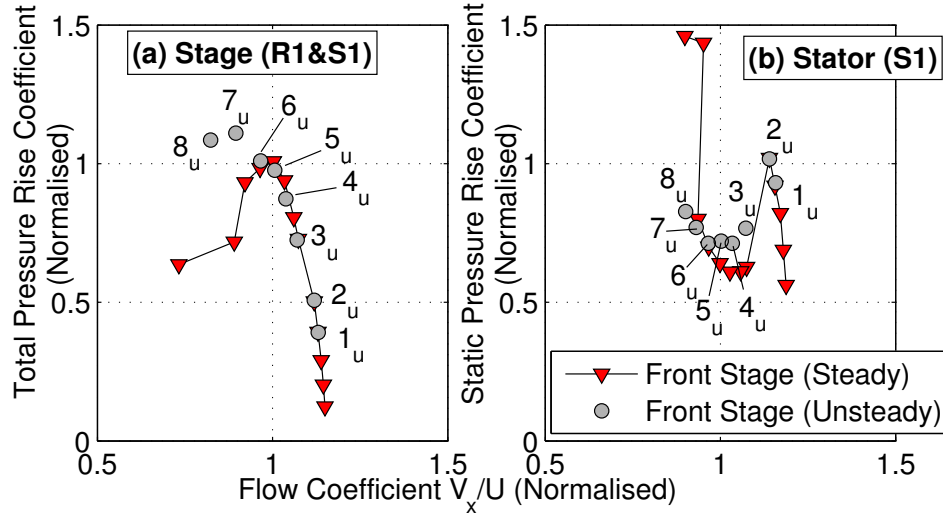


Figure 9.6: Steady vs Unsteady predictions at 85% speed for front stage only model (a) Stage total pressure rise, (b) Stator 1 static pressure rise.

9.3.3 Rotating stall in the single stage model.

Considering now the detailed fluid behavior associated with both Rotor 1 and Stator 1 stalling, Figures 9.7 and 9.8 show the behaviour at Rotor 1 leading edge and Stator 1 trailing edge moving up the unsteady characteristic.

Initially for points 1_u and 2_u the flow is axisymmetric and the only disturbances present consist of the blades themselves. By point 3_u, the characteristics presented in Figure 9.6, and the Stator 1 exit flowfield in Figure 9.8 show that Stator 1 has become stalled with 6 cells forming. This immediately implies that rotating stall may form in the stator even in the absence of downstream blade rows. Moreover, as the stall pattern stabilises on a uniform pattern of cells, it is also implied that some other stabilising mechanism must indeed be present. This must arise either due to the stable Rotor 1 hub or the outer span of Stator 1 suppressing further growth of the cells. It could justifiably be argued that some form of dynamic coupling may exist with the downstream nozzle, although this can be ruled out by repeating this calculation with an extended duct between the stator and the nozzle, for which the results are identical.

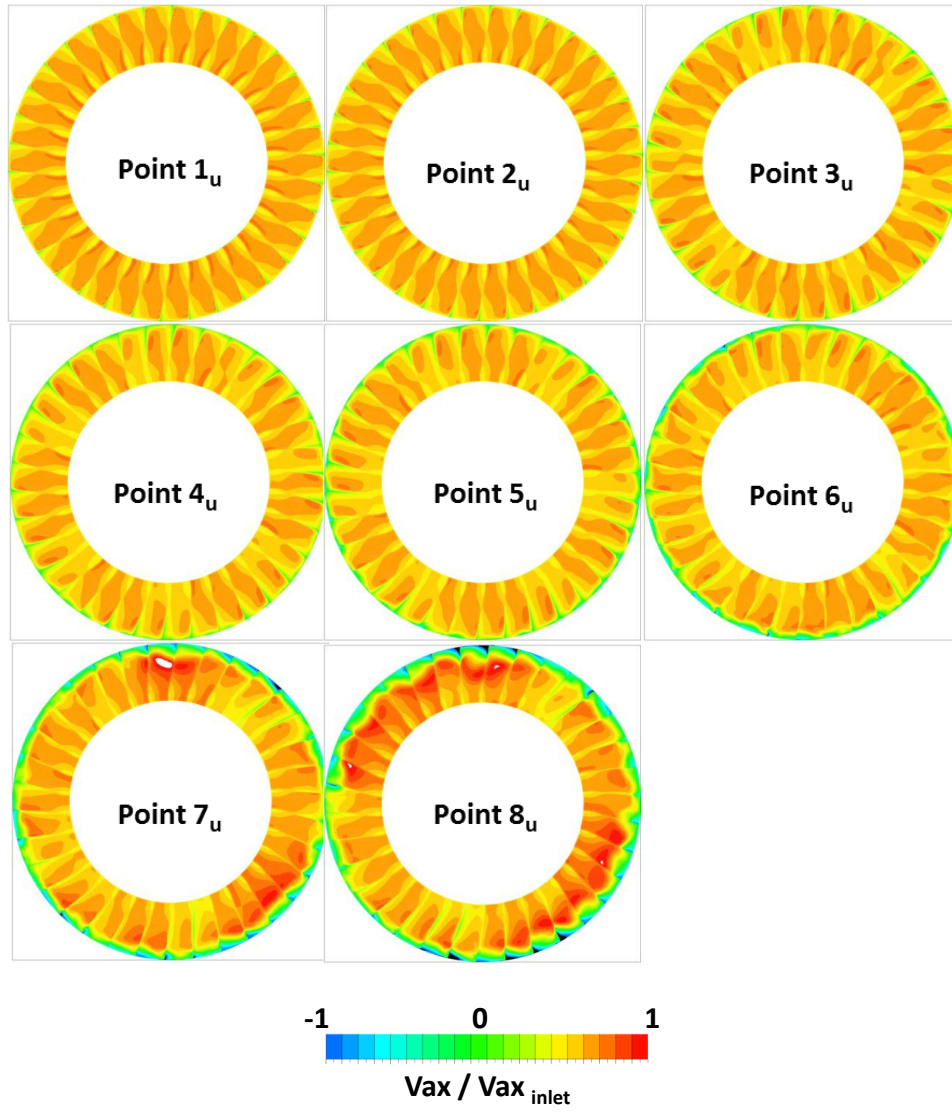


Figure 9.7: Unsteady model of front stage only, Rotor 1 leading edge axial velocity flowfield progressing up characteristic.

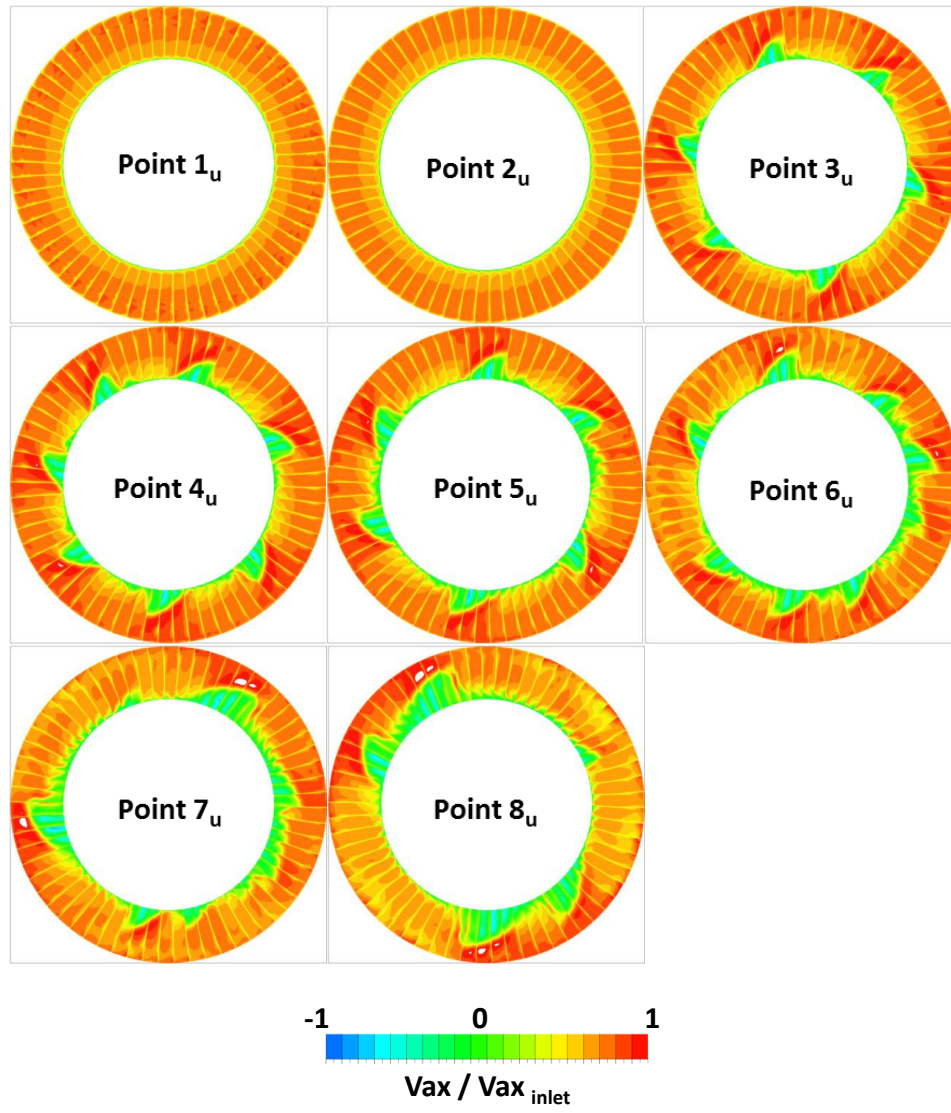


Figure 9.8: Unsteady model of front stage only, Stator 1 trailing edge axial velocity flowfield progressing up characteristic.

For points 4, 5, and 6 the stall cell pattern at Stator 1 hub persists with either 6 or 7 cells present, whilst there is evidence of a gradually deteriorating velocity profile at Rotor 1 tip with no abrupt stall or reverse flow. This is consistent with the stage characteristic in Figure 9.6 starting to flatten. By point 7, however, regions of axially reversed flow become more pronounced at Rotor 1 tip, which grow progressively inboard with further throttling to point 8. At this point the stage characteristic is moving over its peak as the rotor is losing its ability to deliver useful work and pressure rise. Interestingly, as the Rotor 1 tip starts to stall, the Stator 1 hub stall loses its coherent structure. The calculations for point 7 and 8 were continued for over 20 revolutions without stabilising upon a uniform stall pattern in either the rotor or the stator. Whilst it is quite feasible that 20 shaft revolutions is not a sufficient duration for stabilisation to occur, it is also possible that a stable solution simply does not exist for these boundary conditions. For example, the experimental results presented in chapter 4 showed how at some conditions (e.g. Case G in Figure 4.7) the Rotor 1 stall family had a “broadband like” response, for which significant fluctuations were still measured on the pressure transducer, but with no coherent structure or signal. This implies intermittent flow structures exist which break up and reform in a random manner.

9.4 Concluding remarks

This chapter builds upon the unsteady CFD simulations presented in chapter 7 to explore some more fundamental aspects of rotating stall. In particular, the effect of flowfield coupling within the compressor was considered and proposed as a potential mechanism for determining the lengthscale and cell count that occurs when the front stages exhibit rotating stall.

This effect has been demonstrated in two ways. Firstly, by simultaneously opening Stator 2 and closing Stator 3 (which is treated as an additional variable stator for this investigation), the operating point of the stalled front stages can be held constant whilst the downstream Stator 2/Rotor 3 block is moved progressively from a very stable characteristic (negative slope) onto a less stable characteristic. This results in the front stages stalling

in a pattern with fewer cells as the downstream stage becomes less stable. It is inferred from this result that the stabilising effect of the downstream stage is able to prevent the growth of stall cells beyond a certain lengthscale and force rotating stall to occur with a particular number of cells. This is consistent with the experimental research of Longley and Hynes[5].

Second, the stages from Rotor 2 rearwards were removed from the model and replaced with a choked nozzle to give a simplified model of the front stage alone. It was demonstrated how this model also exhibits rotating stall in the stator (even in the absence of the downstream rotor). This result implies some form of flowfield coupling also occurs within the front stage, which can be attributed to the very stable Rotor 1 hub and the unstalled outer span of Stator 1, which also prevent the growth of stall cells beyond a certain size.

Whilst the separation of a boundary layer (i.e. during stall inception at very short lengthscale) is unlikely to be predicted accurately with URANS based methods, it is of importance to recognise that this coupling mechanism applies to the fully developed stall phenomenon and does *not* involve complex viscous effects (e.g. related to mixing and turbulence). As shown in the earlier chapters, the aerodynamics of the VSV stages are strongly influenced by inviscid reasoning, such as the radial-equilibrium effect of the IGV closure, and the effect of Stator 2 closure. Both of these combine stalled regions with very stable regions which interact to determine the most stable stall pattern, which may change as the VSVs are adjusted (for example as seen in the spontaneous changes in Stator 1/Rotor 2 cell count in Figure 5.1). When viewed this way some insight can be offered into why credible predictions of fully developed rotating stall can be obtained using relatively crude URANS turbulence models.

10 Conclusions

The problem of compressor stall and surge is a significant field of research and understanding has been greatly enhanced by developments over the last 75 years[1]. However, research on the specific subject of variable stator vanes (VSVs) is limited, despite their widespread use. This thesis presents a detailed investigation into the effect VSV settings have on part-span rotating stall, in the front stages of an aeroengine style compressor. Having presented and discussed these experimental and numerical investigations in detail, this chapter draws together the main conclusions. Finally, this chapter concludes with a series of recommendation for future work.

10.1 Summary of findings

This thesis considered the behaviour of stalled flow in the front stages of a high-speed compressor, which is typical of that used in an aeroengine application. An experiment was presented whereby the compressor was accelerated slowly with multiple combinations of VSV settings, which redistribute aerodynamic loading across the front two stages. Chapter 4 revealed that two “families” of rotating stall were observed in the compressor, depending upon the VSV settings chosen, with each family occurring at a characteristic frequency when observed on the unsteady instrumentation. A one-dimensional meanline model was used to show how these stall families could be linked to the first and second stage of the compressor, with the stall pressure amplitude showing a strong correlation to well established loading parameters obtained from the model.

Further analysis of the experimental data was then presented in chapter 5, where behaviour observed with the most extreme VSV settings for each stall family was considered. This showed how the first stage stall at high frequency was due to a large number of cells of very short lengthscale (comparable to at most a few blade passages), which were both spatially non-uniform and time varying. The second stage stall at lower frequency consisted of fewer cells, with three to six cells present depending upon the VSV settings chosen. Interestingly, the most extreme example of this case showed spontaneous changes in cell count at certain speeds.

The front stage stall phenomenon was also shown to result in noise propagation, when the mode order of the stall cell pattern became close to that of the blade. This results in a cut-on interaction tone seen on pressure transducers far upstream of the compressor. Chapter 8 also showed how the aeroelastic interaction of this stall phenomenon with the front stage rotor results in a “lock-on” mechanism, in which the stall pattern naturally adjusts its cell count to remain coupled with the second torsion mode. Based on the measurements, a minimum frequency margin between the stall phenomenon and the vibration mode of 10% was estimated, below which the coupling was unavoidable.

The steady state RANS CFD results at part-speed in chapter 6 revealed the presence of stalled regions in both the first and second stages, at the tip and hub respectively. It was also demonstrated that this flowfield can be attributed to the largely inviscid effect of the IGV closing down at part-speed, which raises the swirl at rotor inlet and introduces a strong spanwise static pressure gradient. This results in stall of the front stage rotor tip and the downstream stator hub and is an effect that may be considered applicable to all variable geometry compressors.

In chapter 7 the full annulus unsteady (URANS) simulations confirmed that these stalled regions do indeed exhibit rotating stall and give a credible match to the experiment in terms of stall cell count and propagation speed. This confirms that for this category of problem (fully developed part-span rotating stall in the variable geometry stages), URANS techniques are quite capable of capturing the physical behaviour and provides a detailed description of how such modelling may be performed.

The effect of “ganged” changes to the VSV schedule were also considered in chapter 8, which allow speed changes to be isolated from changes to the VSV settings when studying stall behaviour. This showed clearly for both families that the cell propagation speed, stated in its common non-dimensional form (V_{stall}/U), is largely unique to the instantaneous VSV settings and is independent of the blade speed. Further analysis suggested that changes in stall propagation speed scale approximately with changes in the local axial velocity, which implies that stall propagation speed can be stated in a more universally constant non-dimensional form.

Finally, chapter 9 used the unsteady numerical model to demonstrate the effects of flowfield coupling on rotating stall behaviour. This builds upon the research of Longley and Hynes[5], who showed how the stable stages of a compressor can impose a lengthscale on stall cells forming in the unstable stages, thus limiting their growth beyond a certain size. In chapter 9, this was demonstrated by holding the front two stages at a constant operating point whilst the third stage was moved onto a progressively less stable characteristic, changing the nature of coupling between the stalled front two stages and the unstalled third stage. This resulted in clear changes in the number of stall cells forming. Importantly, this coupling effect is driven largely by changes to the VSV settings, resulting in inviscid static pressure changes (radially and axially) and is not due to more complex viscous effects. This gives some insight into why URANS based models are sufficient to capture fully developed rotating stall in this test case.

10.2 Potential exploitation

One key benefit of this research concerns the interpretation of unsteady signals from pressure transducers and strain gauges, as presented in chapters 4 and 5. This thesis presents documented examples of how rotating stall appears on such instrumentation and also what the potential risks are (for example noise and vibration). During testing a range of other phenomena may occur, particularly when the compressor becomes mismatched, such as flutter or acoustic resonance. The test case presented here thus provides useful guidance in accurately diagnosing such issues.

The use of one-dimensional loading parameters in chapter 4 proved successful in assessing the effects of different VSV settings on the resulting stall behaviour. This was evident in the strong correlations between diffusion factor[8] and the resulting unsteady pressures for both stall families. These correlations also extended to the noise and vibration aspects of the phenomenon. Whilst this method does not account for any spanwise flow variation, it is important to recognise that an assessment of this type may still be of significant benefit during the preliminary design phase when higher fidelity methods are neither available or practical. For example, the designer could make use of the noise and vibration correlation in Figure 4.11 to compare different preliminary design concepts. Furthermore, the process of VSV optimisation is very time consuming and requires costly experimental testing. The meanline model has a further potential use here in terms of finding the optimal VSV schedule with minimum vibration and noise.

The unsteady CFD analysis in chapter 7 confirms that URANS modelling is quite capable of providing credible simulations of rotating stall. This gives confidence that such a method may also be used during the design iterations on detailed blade geometry. It is logical to expect that this will lead to designs which are both aerodynamically and aeromechanically more robust and may allow conservatism to be removed from the design. For example, pessimistic stress assumptions which determine the blade thickness could be relaxed, with a potential improvement to design-speed efficiency.

The trends observed with ganged VSV changes in chapter 8 offer some basic rules concerning the effects of adjustments to the shaft speed and VSV settings on the resulting stall propagation speed and hence the potential excitation frequency. Whilst in some cases elimination of rotating stall may not be possible, these rules may allow changes to the VSVs to be made in order to shift the stall phenomenon to a speed and frequency where it has a smaller impact. For example, the stall frequency may be shifted away from any vibration modes, or to a speed where any noise is more tolerable in the flight cycle. The observation in chapter 8 concerning the stall/vibration “lock-on” phenomenon also provides a basic criteria for the designer when considering the blade Campbell diagrams, specifically that a 10% frequency margin between stall and blade natural frequency is desirable to avoid such coupling.

10.3 Recommendations for future research

This thesis clearly demonstrates the importance of VSV settings on the behaviour of rotating stall in this test case, and the behaviour presented is likely to be applicable to a wide range of other compressors due to the wide spread use of variable stator vanes. The study of further test cases would be beneficial, particularly in an experimental environment with greater flexibility in terms of accessibility and instrumentation. For example, the ability to perform unsteady measurements of these stall phenomena at multiple radial heights would offer significant further insight into the structure of part-span stall cells.

The unsteady CFD simulations show a credible agreement with the experimental data for the purely aerodynamic aspects of rotating stall. This provides confidence that this modelling technique can be extended further to investigate other aspects of this problem. For example, it would be desirable to investigate the aeroelastic behaviour where rotating stall forces the blade vibration modes. This test case offers an ideal opportunity for further study, where the “lock-on” mechanism has been demonstrated. The same modelling techniques may also be extended to capture the more complex unstable rotating stall process occurring prior to surge, which is of interest from both an aerodynamic and aeroelastic point of view, by performing the full annulus simulations further up the characteristic until the flow breaks down in an asymmetric manner. Finally, it would be of significant benefit to explore more systematically the differences between unsteady and steady simulations. For example, could the steady computations be corrected or improved to reflect the unsteady behaviour presented here?

The more fundamental aspects of rotating stall discussed in chapters 8 and 9 would benefit from investigation in more detail using the unsteady CFD model. For example, the stall propagation speed was found to scale well with local axial velocity based on experimental observations made as the VSVs were adjusted. However, this behaviour has not been explained from a fundamental point of view based on the fluid dynamics of the stall cells. Further interrogation of the unsteady model provides an ideal means by which to develop this understanding.

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