

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

**NUMERICAL AEROACOUSTICS INVESTIGATION OF THE EFFECT OF
AXIAL GAP LENGTH BETWEEN THE ROTOR AND STATOR OF A
TRANSONIC COMPRESSOR STAGE**



M.Sc. THESIS

Bora YAZGAN

Department of Mechanical Engineering

Heat and Fluid Programme

JUNE 2018

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**TRANSONİK KOMPRESÖRLERDEKİ ROTOR VE STATOR ARASI
EKSENEL BOŞLUK MESAFESİNİN AKUSTİK ETKİSİNİN İNCELEMESİ**

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To my family,



FOREWORD

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ABBREVIATIONS

B	: Number of Rotor Blades
BEM	: Boundary Element Method
BPF	: Blade Passing Frequency
CAA	: Computational Aeroacoustics
CFD	: Computational Fluid Dynamics
DES	: Detached Eddy Simulation
DNC	: Direct Noise Calculation
EPNL	: Effective Perceived Noise Level
FAA	: Federal Aviation Administration
FAR	: Federal Air Regulation
FFT	: Fast Fourier Transform
FW-H	: Ffowc Williams-Hawkings
HBR	: High By-Pass Ratio
ICAP	: International Civil Aviation Organization
LBR	: Low By-Pass Ratio
LES	: Large Eddy Simulation
MCA	: Multiple Circular Arc
OGV	: Outlet Guide Vane
PN	: Perceived Noiseness
RANS	: Reynolds Averaged Navier Stokes
RPM	: Revolution Per Minute
SPL	: Sound Pressure Level
SST	: Shear Stress Transport
STN	: Station
TVR	: Turbulence Viscosity Ratio
URANS	: Unsteady Reynolds Averaged Navier Stokes



SYMBOLS

c_0	: speed of sound
C_2	: Absolute Velocity of Bulk Flow
$C_{2,wake}$	: Absolute Velocity of Wake
C_p	: Pressure Coefficient
dB	: Decibel
k	: Turbulent Kinetic Energy
M	: Mach Number
n_i	: Surface Normal Vector
P	: Pressure
$p(t)_i$	: Instant Pressure
$\bar{p}$	: Time Averaged Mean Pressure
p'	: FW-H Calculated Pressure
$P_{0,ref}$	: Reference Pressure
P_{ij}	: Compressive Stress Tensor
p_L'	: Dipole Term
$\bar{p}_{normalized}$	: Averaged Unsteady Normalized Pressure
p_Q'	: Quadrupole Term
p_{ref}	: Reference Sound Pressure
p_{rms}	: Root Mean Squared Sound Pressure
p_T'	: Monopole Term
P_{total}	: Total Pressure
R	: Gas Constant
t	: Time
T	: Temperature
T_{ij}	: Lighthill's Stress Tensor
T_{ref}	: Reference Temperature
T_{total}	: Total Temperature
T_v	: Vane Passing Period
U	: Blade Speed
u_i	: Fluid Velocity Component in x_i Direction

$\mathbf{u}_j$	: Fluid Velocity Component in x_j Direction
$\mathbf{u}_n$	: Fluid Velocity Component Normal to the Surface
$\mathbf{v}_i$	: Surface Velocity Components in the x_i Direction
$\mathbf{V}_{\text{mean}}$	: Mean velocity of the flow
$\mathbf{v}_n$	: Surface Velocity Components Normal to the Surface
$\mathbf{V}_T$	: Tangential Blade Speed
$\mathbf{W}_2$	: Relative Velocity of Bulk Flow
$\mathbf{W}_{2,\text{wake}}$	: Relative Velocity of Wake
$\mathbf{x}'$	: FW-H Observer Location
$\mathbf{x}_i$	: x_i Direction
$\mathbf{x}_j$	: x_j Direction
y^+	: y plus
δ_{ij}	: Kronecker Delta
ρ	: Density
ρ_0	: Far Field Density
σ_{ij}	: Viscous Stress Tensor
ε	: Turbulent Dissipation Rate
η	: Polytropic Efficiency
∂	: Partial Derivative
Δ	: Increment Operator
∇	: Nabla Operator
λ	: Acoustic Wavelength
$\lambda_{\text{downstream}}$	: Dowsntream Acoustic Wavelength
$\lambda_{\text{upstream}}$	: Upstream Acoustic Wavelength
$\Delta \mathbf{x}$	: Element Size
Δt	: Time Step Size

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NUMERICAL AEROACOUSTICS INVESTIGATION OF THE EFFECT OF AXIAL GAP LENGTH BETWEEN THE ROTOR AND STATOR OF A TRANSONIC COMPRESSOR STAGE

SUMMARY

Modern civil aviation regulations stipulates the commercial airline companies not to exceed certain noise levels in the neighbourhood of airports which are generally located in the city. Strict noise regulations requires detecting the noise generations of the aircraft engine and applying an appropriate design or treatment of the engine component accordingly.

The main noise generation mechanisms of the aircraft engines are inlet turbofans, compressor stages, turbine stages, combustion and jet flow. Each component contributes their own effects on the noise spectrum of an aircraft; however, the primary noise sources in modern high by-pass pressure ratio aircraft engine is the turbofan among them. Turbofan noise, which constitutes the largest part in the engine noise spectrum, radiated in both upstream and downstream on the engine. However, the other sources, which are compressor stages, turbine stages, combustion and jet noise, radiate only in one direction.

The turbofan, compressor stages and turbine stages are axial turbomachinery applications. That is why, the largest component of the aircraft engine noise sources is turbomachinery originated. In order to keep the engine noise emission under the required level the flow physics of the turbomachinery should be profoundly understood and new designs should be formed accordingly.

Two main components of the turbomachinery noise are discrete (tonal) and broadband noise. The discrete noise appears at certain frequencies (e.g. blade passing frequency), while broadband noise covers the whole noise spectrum. The discrete noise is generally originated from the interactions of the turbomachinery components like rotor and stator. It stems from the stator potential field on the rotor blades and rotor wake interaction with the stator vanes. However, the broadband noise arises from the turbulent content in the flow domain.

In this study, discrete (tonal) noise generation of the transonic axial compressor turbomachinery application was studied. NASA Stage 37 test case was selected as the transonic axial compressor stage. The most significant tonal noise generation mechanism of the axial turbomachinery which is the rotor-stator interaction was investigated.

The magnitude of the axial gap between the compressor rotors blades and stator vanes are the primary factor of the rotor-stator interaction. The smaller the gap is, the intenser the rotor-stator interaction becomes. That is why, in order to observe the effect of the axial gap length on noise generation, the flow field of the compressor with three different axial gap length was investigated.

Computational fluid domain model of the transonic compressor stage with three different axial gap between the rotor and stator was formed. Unsteady Reynolds Averaged Navier Stokes Equations (URANS) analysis of three models were carried out. For the nearfield acoustic propagation unsteady pressure profiles were directly extracted from the URANS simulations. For the farfield acoustic propagation the unsteady rotor and stator blade pressure values were imported an acoustic solver.

The computational fluid dynamics results shows that the potential field between the rotor blades and stator vanes and interaction of the rotor wake and stator vanes strongly decayed as the axial gap between the rotor and stator blade incerases. This resulted in an observable reduction in the tonal noise generation at blade passing frequencies (BPF) and its higher harmonics.



TRANSONİK KOMPRESÖRLERDEKİ ROTOR VE STATOR ARASI EKSENEL BOŞLUK MESAFESİNİN AKUSTİK ETKİSİNİN İNCELEMESİ

ÖZET

Günümüz modern havacılık uygulamalarında hava yolu şirketlerinden sıkı bir şekilde gürültü regülasyonlarına uyulmaları istenmektedir. Zira şehir içlerinde bulunan havalimanlarında oluşan gürültü kirliliği insan sağlığını ciddi yönde etkilemektedir. Buna karşı önlem olarak havacılık regülasyonları uçak motoru kaynaklı gürültünün insan sağlığını etkilemeyecek düzeylere çekilmesini şart koşmaktadır. Bu sebeplerden dolayı modern uçak motoru üreticileri motor gürültüsünün kaynaklarını iyi bir şekilde belirleyip düzgün tasarımlarla aynı verimi sağlayabilecek fakat daha az gürültülü çalışabilecek motorlar geliştirmektedir.

Uçak motoru gürültüsünün istenilen seviyelere çekilebilmesi için öncelikle gürültüyü oluşturan motor komponentlerinin iyi bir şekilde analiz edilmesi gerekmektedir. Günümüz yüksek by-pass oranlı uçak motorlarında gürültüyü oluşturan ana etmenler arasında motor girişindeki turbofan, kompresör kademeleri, türbin kademeleri, yanma odası ve itki kuvvetini sağlayan jet akışı bulunmaktadır. Uçak gürültü diyagramına her bir komponent ayrı şekilde yansımaktadır. Bu komponentlerden turbofan kaynaklı gürültü uçuş esnasında hem motorun giriş yönünde hem de çıkış yönünde yayılmakta, kompresör kaynaklı gürültü motor giriş yönünde, türbin kaynaklı gürültü, yanma kaynaklı gürültü ve jet gürültüsü motor çıkış yönünde yayılmaktadır. Bu bağlamda bakıldığında modern yüksek by-pass oranlı uçak motorlarında gürültüyü oluşturan en kilit komponent giriş turbofanıdır.

Uçak motorlarındaki turbofan, kompresör kademeleri ve türbin kademeleri birer eksenel turbomakina uygulamalarıdır. Bu sebeple uçak motorlarında gürültüyü oluşturan en büyük etmen turbomakina kaynaklıdır. Uçak motorlarından yayılan gürültüyü istenilen gürültü seviyelerinin altına çekmek için turbomakinalardaki hava akışının fiziği çok iyi derecede çözülmeli ve buna bağlı olarak da yeni tasarımlar oluşturulmalıdır. Uçak motorlarında en önemli tasarım verim olduğu için tasarımcı sadece düşük gürültü üretecek tasarımla yetinmemelidir. Aynı zamanda yapacağı yeni tasarım uçak motorunun ağırlığını önemli derecede artırmamalı ve de yakıt verimini düşürmemelidir.

Turbomakinalarda hava kaynaklı gürültü iki çeşittir. Bunlar kanat geçiş frekansı gibi belirli frekanslarda görülen tonal gürültü ile gürültü spektrumundaki tüm frekanslarda görülen genişbantlı gürültüdür. Gürültü spektrumlarında tonal gürültü etkili olduğu frekanslarda pik yapmış bir şekilde gözükür. Turbomakina gürültü spektrumlarında tonal gürültünün decibel cinsinden büyüklüğü genişbantlı gürültüye göre genellikle yüksek olmaktadır. Özellikle kanat geçiş frekansındaki ve bunun harmonik katlarındaki tonal gürültü değerleri genişbantlı gürültüye oranla gözle görülebilir bir şekilde fazladır. Bu sebeple modern turbomakina tasarımlarında tonal gürültüyü daha düşük bir seviyeye çekebilecek ve turbomakinanın aerodinamik verimini azaltmayacak yeni tasarımlar oluşturulmaktadır.

Tonal gürültü genel olarak rotor ve stator gibi turbomakina bileşenlerinin motor shaft hızına ve kanat sayısına bağlı olarak belirli frekanslardaki etkileşimleri sonucunda oluşur. Bu etkileşim sırasında statorun rotor üzerinde bir akış potansiyeli oluşması sebebiyle stator geçiş frekanslarınca rotor üzerinde basınç dalgalanmaları oluşur. Ayrıca rotorun arkasında oluşan girdap bölgesinin stator üzerine rotor geçiş frekansınca çarpması sonucunda bu frekansta stator kanatları üzerinde basınç dalgaları oluşur. Bu basınç dalgalanmaları da tonal gürültünün ana etmenleridir. Öte yandan, genişbantlı gürültünün ana kaynağı akış bölgesindeki türbülans içeriğidir. Akış bölgesinde oluşan türbülanslı vorteksler geniş bir frekans ölçeğini kapsamaktadır, bu sebeple oluşan gürültü genişbantlı olarak isimlendirilmektedir.

Bu çalışmada aksel transonik kompresör kademesinde üretilen tonal gürültü incelenmiştir. Transonik kompresör kademesi olarak 36 adet rotor kanadı ve 46 adet stator kanadı olan NASA Kademe 37 kullanılmıştır. İncelemede turbomakinalardaki tonal gürültünün en önemli etkeni olan rotor ve stator kanatları arasındaki etkileşim ele alınmıştır.

Rotor ve stator kanatları arasındaki etkileşimi etkileyen en önemli etkenlerden birisi de rotor ve stator arasındaki aksel boşluktur. Bu boşluk ne kadar küçük olursa etkileşim de o kadar yoğun olmaktadır. Bu sebeple, boşluk boyutunun rotor-stator etkileşimindeki etkisini gözlemlemek için 3 farklı boşluk boyutu denenmiştir.

Hesaplamalı akışkanlar dinamiği analizlerinde ilk olarak NASA Kademe 37 içinde bulunan rotor kanadı önce tek başına sonra stator kanadı ile modellenmiştir. Zamandan bağımsız şekilde koşulan analizlerde elde edilen değerler NASA tarafından gerçekleştirilmiş olan test sonuçlarıyla kıyaslanarak oluşturulan hesaplamalı akışkanlar dinamiği modelinin doğruluğu ölçülmüştür.

Akustik analizler için bilgisayar ortamında NASA Kademe 37 geometrisi modifiye edilerek 3 farklı boşluk boyutu içeren 3 farklı hesaplamalı akışkanlar dinamiği modeli oluşturulmuştur. Çözüm kolaylığı sağlaması açısından stator kanadı sayısı 46'dan 36'ya indirilmiştir. Bu şekilde zamana bağlı analizlerde akış alanı daha küçük tutularak bilgisayar gücü ve hesaplama zamanından önemli derecede tasarruf edilmiştir. Zamana bağlı Reynolds ortalama Navier Stokes Denklemleri 3 farklı model için çözülmüştür. Etkileşime yakın bölgedeki akustik dalgaların çözümü hesaplamalı akışkanlar dinamiği programından basınç dalgalanmaları direk olarak çekilerek yapılmıştır. Uzak bölgedeki akustik dalgaların hesaplanması için ise hesaplamalı akışkanlar dinamiği çözümünden gelen rotor ve stator üzerinde basınç dalgalanmaları akustik denklem çözücüyeye transfer edilmiştir. Akustik denklem çözücü gürültü kaynağından yayılan akustik dalgaları uzak bölgede hesaplayarak bu bölgedeki gürültü seviyesini bulmuştur.

Hesaplamalı akışkanlar dinamiği analizleri sonucunda stator kanatlarının rotor kanatları üzerindeki potansiyel alanının aksel boşluk arttıkça hızla küçüldüğü gözlemlenmiştir. Özellikle rotor kanadı basınç tarafındaki basınç dalgalanmalarında önemli derecede küçülmeler görülmüştür. Basınç tarafında daha çok hissedilmesinin sebebi rotor kanatlarının diziliş yönü ve turbomakinanın dönme yönüyle ilgilidir. Dönüş sırasında stator kanatları ile ilk olarak rotor kanatlarının basınç tarafı etkileşim haline gelmektedir. Rotor kanatlarının emme tarafı ise stator kanatları ile daha az etkileşim halindedir. Bu sebeple de sadece kanat diplerindeki statora yakın bölgelerde iyileşmeler gözlemlenmiştir. Tüm bunlardan anlaşılabileceği üzere stator kanatlarının rotor kanatlarının performansı üzerinde önemli etkisi vardır.

Bir diğ er yandan, rotor kanatlarının arkasında oluřan d ř k hızlı girdap b lgesinin rotor kanadı ge iř frekansınca her bir stator kanadına  arpmasıyla oluřan etkileřimde de  nemli miktarda iyileřme g zlemlenmiřtir. Bunun i in stator kanadının emme ve basın  taraflarındaki y zeylerde zamana baėlı normalize edilmiř basın   alkantıları hesaplanmıř olup yapılan kıyaslamada en y ksek basın   alkantılarının en d ř k eksenel bořluklu modelde olduėu g zlemlenmiřtir. Buna sebep olarak da eksenel bořluk uzunluėunun artmasıyla birlikte rotor kanadı arkasında oluřan girdaplı b lgenin ana akıma daha  ok karıřması ve hızının artmasıyla stator kanadına daha d ř k bir h cum a ısıyla  arpmasıdır.

Stator kanadına  arpan akıřkanın h cum a ısı zamana baėlı olarak deėiřmektedir. Bunun sebebi roto arkasındaki d ř k hızlı girdap b lgesinin stator kanatlarının  zerinden kanat ge iř frekansında ge ip h cum a ısını artırmasıdır. Bu sebeple zamana baėlı h cum a ısı deėiřimi en  ok d ř k eksenel bořluklu durumda olmaktadır. H cum a ısının deėiřiminin  ok olması stator  zerindeki dinamik y k  artırmakta bu da daha  ok g r lt  oluřmasına sebep olmaktadır.

Stator kanatlarının  zerinde oluřan basın   alkantılarının genliėi rotor  zerindeki oluřan  alkantı geniliėine g re oranı daha b y kt r. Bundan dolayı stator kanatlarının g r lt  oluřumuna katkısının rotor kanatlarına g re daha fazla miktarda olduėu s ylenebilir.

 nceden de belirtildiėi gibi g r lt  azaltmak i in yapılan tasarımlarda turbomakinanın verimi de kesinlikle g z  n nde bulundurulmalıdır. G r lt y  iyileřtiren bir tasarım verimde ciddi bir kayıba sebep oluyorsa bu tasarım uygulanamaz.  nemli olan verimde bir k t leřmeye sebep olmadan g r lt y  azaltabilmektir. Analiz sonu larına g re eksenel aralıėın artması kompres r g r lt s n   nemli řekilde azaltsa da verimde de y zde 2 ye varan kayıplara sebep olmuřtur. Bu baėlamda eksenel sadece eksenel aralıėın artırılarak oluřturulacaėı bir tasarımın pratik olarak uygulanabilirliėi turbomakina uygulamasının ihtiya ına g re deėiřir. Eėer u ak motoru gibi verimin ve hafifliėin  ok  nemli olduėu bir alanda tasarım yapılıyorsa, eksenel aralıėın artırılması uygun deėildir. Fakat g r lt n n verimden daha  n plana  ıktıėı alanlarda bu uygulamaya gidilebilir.

Sonu  olarak rotor ve stator arasındaki etkileřimin eksenel aralıėın artmasıyla birlikte hafiflemesi sonucunda, rotor ge iř frekansındaki ve daha y ksek harmoniklerindeki tonal g r lt de  nemli azalmalar kaydedilmiřtir. Fakat aerodinamik verimde de kayıplar g zlemlenmiřtir.



1. INTRODUCTION

The invention of the jet engines caused a new noise problems and their utilization by commercial airline companies is an important inconvenience for the public living next to airports. As depicted in Figure 1.1 in the 1960s new jet engines are replaced with the conventional propeller thrust engines. That gave rise to six-fold increase in the aircraft engine noise emissions [1]. This dramatic noise increase stems from the energy conversion, within the jet engine, which requires massive amount of air to be sucked into engine to provide enough thrust. Vibrations of the air molecules forming sound waves draw some portion of this energy. These sound waves carry a small energy content, however their propagation cannot be restricted and they cannot be substantially damped.

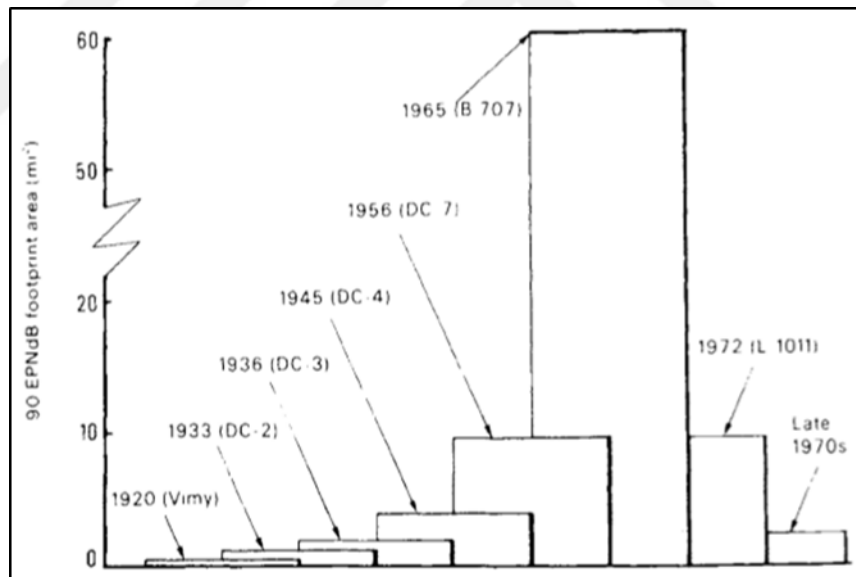


Figure 1.1: Noise types generation of different engine [1].

At first the dominant source of noise was generated by shear layer mixing of high speed jet. However, with the introduction of the low-bypass-ratio and high-bypass-ratio aircraft engines, this became the compressor and fan related noise respectively. The comparison of the dominant noise generation mechanisms between the 1960s turbojet (and low bypass turbofan) and modern high bypass ratio (HBR) turbofan design shown in Figure 1.2. Most obvious feature of the modern HBR design is the

substantial reduction of the jet noise. Although the HBR turbofan design purpose is to operate civil aircraft engine at higher efficiency, it provides important benefits for noise generation. It brings a large reduction of jet mixing noise by reducing the jet flow velocity. However, HBR turbofans introduces a considerable fan noise in the forward and rear [2].

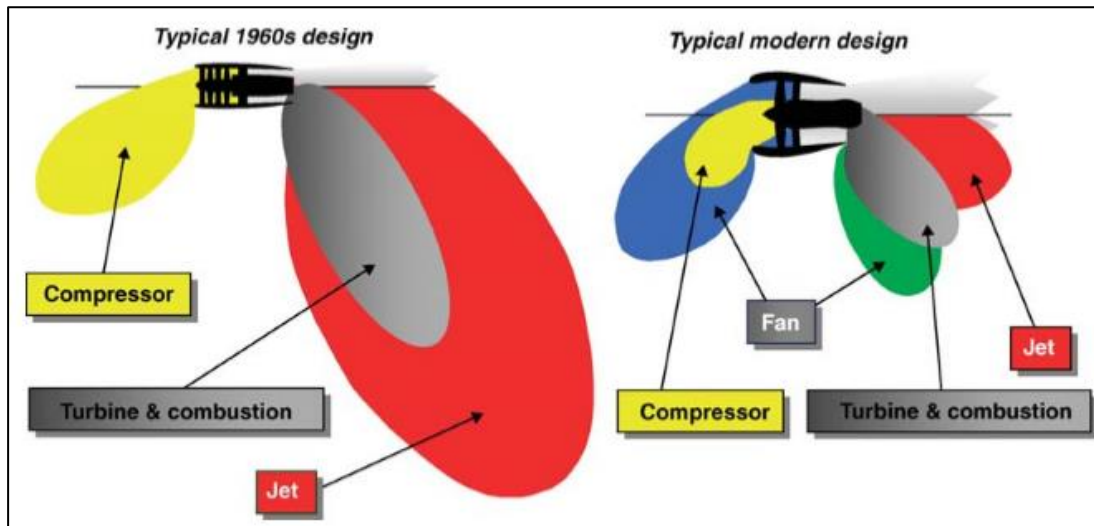


Figure 1.2: Dominant noise source of a jet engine [2].

In the 1960s as the utilization of turbojet and turbofan engines by civil airliners became widespread, noise regulations of aircraft engines became essential. A growing noise nuisance in the neighborhood of the airports took place. Firstly, in the late 1960s, US Federal Aviation Administration (FAA) enacted a certification of noise regulation. They designated the maximum permitted noise level for aircrafts with the Federal Air Regulation part 36, known as FAR 36. In the advancing years, all the regulations have evolved and the requirements have become more stringent and tended to a common specific standard. Recently designed airliner must obey the noise certification rules introduced in Annex 16 ICAP (International Civil Aviation Organization) and stage 4 of FAR 36 [2].

ISO (International Standardization Organization) recommended the unit of effective perceived noise level (EPNL) to determine the noise on the ground generated by the aircraft. Numerical values are given in decibels and written as EPNdB. In the noise standards, maximum permitted noise in terms of EPNL are measured at three reference locations: flyover (cutback), sideline and landing (approach) as seen in Figure 1.3. Upper limits for EPNd dB are assigned for each measurement points. Specifically,

modern commercial aircraft noise must not exceed 108 EPNdB at three reference measurement points.

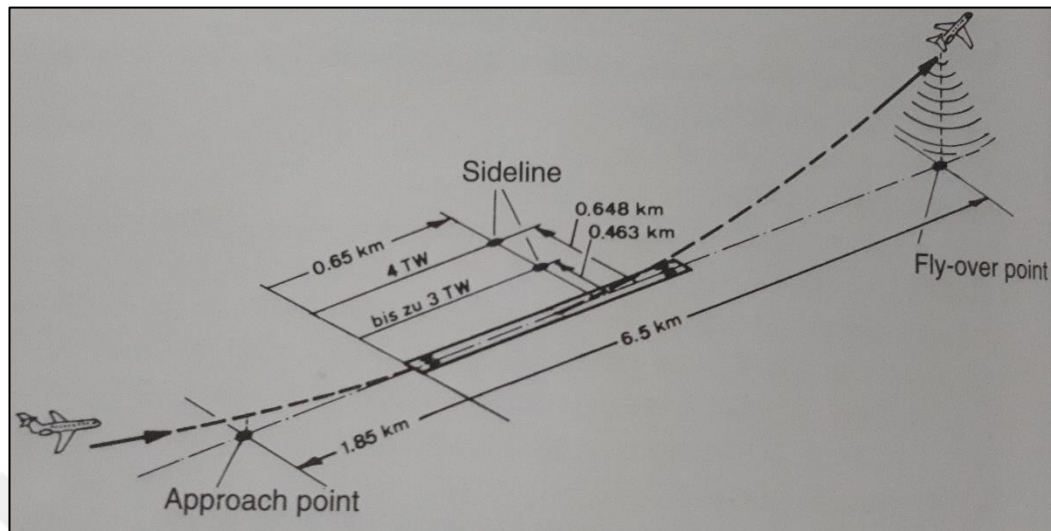


Figure 1.3: Noise measurement points of an aircraft [3].

1.1 Purpose of Thesis

The main objective of this study is to learn the aeroacoustics field of the turbomachinery applications. Obtaining a knowledge on the noise generation mechanisms in turbomachines and the noise propagation calculations are the fundamentals reasons for this research.

Additionally, learning how modern computational fluid dynamics codes are utilized for the turbomachinery aeroacoustics calculations and developing a methodology for further turbomachinery related designs and researches have a significant importance for this research.

1.2 Aircraft Engine Noise Sources

First turbojet engines had small flow area and relatively slower airflow inside the machine, yet the jet discharge velocities were higher than 600 m/s. Hence, the dominant noise source of first turbojet engines were mixing jet with the air.

In the sixties with the introduction of the low by-pass ratio turbofan engines, which were operated at 1.5 by-pass ratio, need to suck higher mass flow rate and release exhaust jet at lower speeds. This gave rise to generation of lower jet originated noise, whilst an increase of the turbomachinery noise.

Modern HBR turbofan engines use the most of the sucked air on the fan, while a small portion of the intake air flows through the core. Exhaust velocity of the fan flow is around 300 m/s and the core engine flow is around 400 m/s. At high exhaust jet velocities (about 600 m/s) propagated sound intensity increases as the eighth power of the jet stream velocity. However, lower exit velocities (about 200 to 300 m/s) in HBR reduces this effect down to second power of the exit velocity. This core engine flow exhaust velocity degrades the noise generation of jet compared to plain turbojet engines and makes the fan dominant noise source of the HBR turbofans. Figure 1.4 shows the different noise sources of a modern HBR turbofan engine.

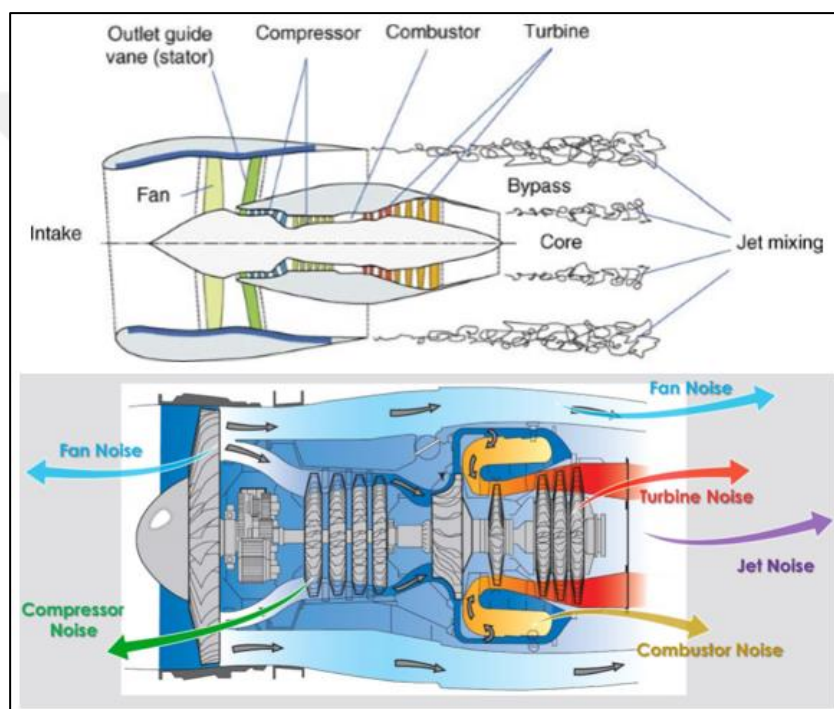


Figure 1.4: Noise sources of modern HBR turbofan.

Fan originated noise propagates through the intake and bypass ducts. Compressor noise radiates through the intake. Combustion and turbine generated noise propagates through the engine hot core as it is seen in Figure 1.2. By-pass jet and hot core jet mixes with ambient air and forms shear layers after leaving the engine.

Figure 1.5 illustrates the relative noise contribution of each engine components to the entire airplane EPNL at the three aforementioned reference measurement points. As seen, upstream and downstream fan noise propagation stands dominant for all cases. However, other noise sources can degrade and even vanishes at some measurements point. Additionally, the engine components airframe noise sources (fuselage, landing

gear, high lift device noise) reaches a significant noise level particularly for approach measurement.

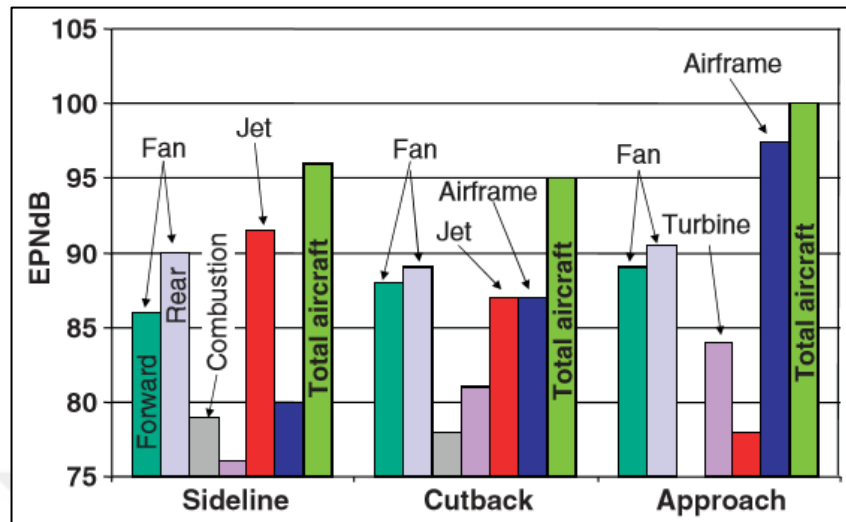


Figure 1.5: Noise contribution of the engine components [Astley, 2010].

As bypass-ratio of turbofan engines increases with the passing years, designers have faced raising level of fan noise, which is related to turbomachinery. Rotating fan blades shred eddies from the trailing edge in the flow causing broadband noise spectrum, this is called rotor self-noise. Moreover, impingement of the turbulent rotor wake on the stator blade, rotor tip-boundary layer interaction and ingested turbulent air on the rotor leading edge also causes broadband noise. The first three one outweigh than the last one in terms of broadband noise generation of the fan. In Figure 1.6, part (a) shows the fan broadband noise sources, part (b) decomposition of total broadband noise of fan [Astley, 2010].

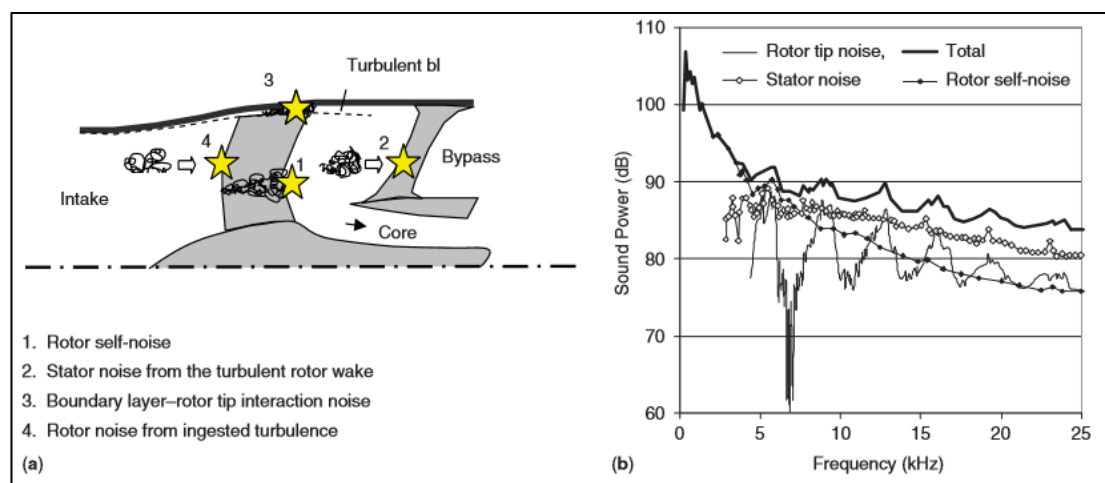


Figure 1.6: Fan broadband noise sources [Astley, 2010].

Additional to turbulence originated noise generation of the fan, due to the displacement of the rotor blade within the fluid, a discrete frequency noise (tonal noise), which is associated with the blade rotational speed, is generated. The discrete noise characteristics of the fan are generally associated with the axial gap between rotor and stator, rotor and stator blade number and tip speed of the rotor. Generated discrete noise of the fan can be reduced 6 to 8 PNdB by increasing the axial gap between the rotor and stator. Perceived noiseness (PN) is one of the measures of noise magnitude comprising subjective factors. PNdB stands for the unit of the perceived noiseness (PN) in dB. The main drawback of increasing axial gap is extending the length of the engine and as a result its weight. Moreover, larger axial spacing affects the aerodynamics in the engine and causes an efficiency loss of 1%. Number of rotor and stator blades affects the tonal noise characteristics of the fan. Fan efficiency requires relatively high number of blades due to the aerodynamic considerations. This is also beneficial for noise reduction, because each blade burden lower aerodynamic load, and noise increases with aerodynamic forces. In HBR turbofan engines, typically the ratio of stator blades to rotor blades is 2. Larger number of the stator blades can provide 5 PNdB noise reduction. Fan tip speed is another consideration which has effect on fan noise generation. Stage pressure ratio is related to tip speed of fan. Obtaining pressure ratio of 1.5 requires supersonic tip speed, however lowest noise generation is observed if the tip speed of fan is subsonic. In the commercial turbofan engines, keeping the tip speed of fan supersonic to achieve high-pressure ratio. That is why it is not a proper way to decrease the tip speed for noise regulations in modern turbofan engines.

Besides the fan which is the fundamental turbomachinery related noise of HBR turbofans, turbine and compressor stages also contributes to the turbomachinery noise. This noise source become significant especially in LBR turbofan engines which operates relatively smaller fan. Sound generation mechanism of compressor and turbine stages is same as fans, the main difference is having smaller axial gap compared to fan. As a result of this operating condition, generated noise by compressor and turbines is dominated by discrete interaction tones. As seen in Figure 1.5 the propagation direction of the compressor noise towards upstream, whilst turbine generated noise is transmitted through downstream. In compressor and turbine stages, the blade number of the rotors and vanes can be adjusted in order to emit cut-off modes which propagate weakly. Cut-off modes have subsonic phase speeds; in contrast cut-

on modes have supersonic phase speed and propagates strongly. In the first stages of turbine the temperature of air is high because of the incoming hot gases from the combustion chamber. Thus, the speed of sound is much greater than in the cold fluid stream. This enables the engineers to design rotor working with tip speed that provides cut-off modes. Aeroacoustic design is more crucial for the last few stages of the turbine which operate under relatively lower temperature than the first stage. A design is possible which includes a majority of the interaction-originated noise arising at high frequencies that are not harmful for human audience range.

Another noise source of turbofan engines is combustion event which is composed of direct and indirect noise. The innate noise of combustion process is called direct combustion noise. The noise, which stems from the non-uniform temperature distribution within the turbine and exhaust, is named as indirect noise. The turbulent flame frequency is the main factor affecting character of noise spectra which is inherently broadband and varies between 300 and 600 Hz. Figure 1.5 shows that combustion noise constitutes a small parts relatively turbomachinery and jet noise of engine.

1.3 Turbomachinery Noise Sources

Noise sources in turbomachines mathematically explained first by Ffowcs Williams & Hawkings (FW-H) equation, which is detailed explained in 4.1.2.3. The main property of the equation it takes into account the quadrupole sources from turbulence, dipole sources from pressure fluctuations on blades and monopole sources from the volume displacement of the fluid due to the moving blades.

1.3.1 Monopole noise sources

Sound wave of a monopole source radiates equally in all directions as seen in Figure 1.7. Combustion, jet streams, cavitations, piston movements are typical monopole noise sources of industrial fluid-working machines. In turbomachinery, blade thickness noise generated by volume displacement of fluid is a monopole source. A turbomachine blade has its own thickness and volume. As the rotor moves, the volume of the each rotor blade replace its position with the surrounding fluid volume and this periodic replacement fluctuates the pressure in the adjacent field, which causes noise generation [4]. For subsonic blade speeds, the propagated sound is relatively weak

because of local cancellation of out-of-phase regions near the source [5]. In case of low Mach number turbomachinery applications such as ventilation or industrial fans, radiated blade thickness pressure velocity is subsonic and the radiation efficiency is low. In contrast, blade thickness noise becomes significant under supersonic blade speed operations. This mechanism would become severe for helicopter rotors and propellers and it depends on the cross-sectional area of the blade. Another example of the monopole noise source of the supersonic blade tip speed is shock wave induced buzz-saw noise. This stems from the rotating shock waves and occurs at multiple of rotation speed. Physics of the buzz-saw noise will be explained in next chapter.

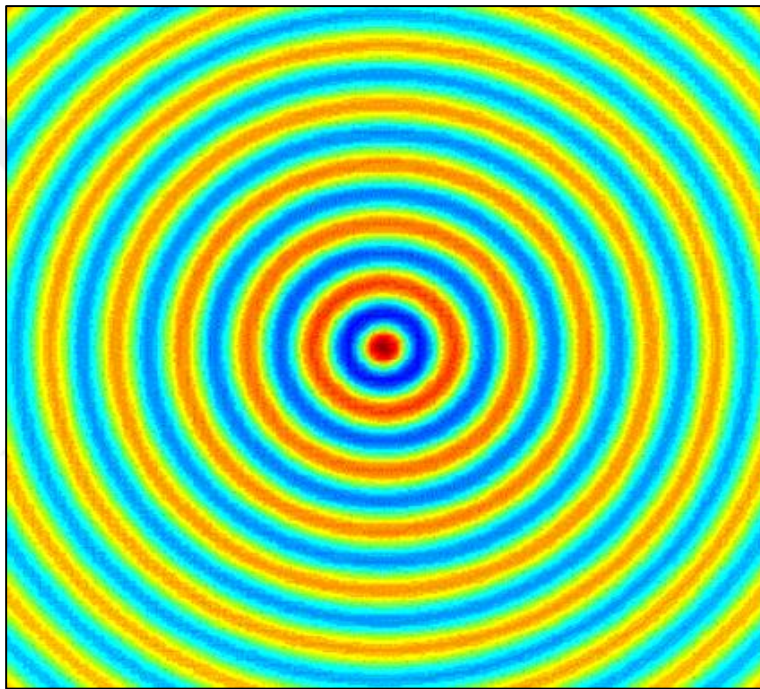


Figure 1.7: Sound propagation from a monopole noise source.

1.3.2 Dipole noise sources

Fluctuating aerodynamic forces on the solid boundaries, which interact with the fluid, generate dipole source. Unlike the monopole sources that have only one propagation direction, dipole sources have two directions including the radius from the source and one angle as seen in Figure 1.8. It consists of two contrary monopoles, which have equal strength, while one source expands the other one contracts. This yields in the fluid near the two sources shakes backward and forward to generate sound. A sphere, which oscillates back and forth, acts like a dipole source.

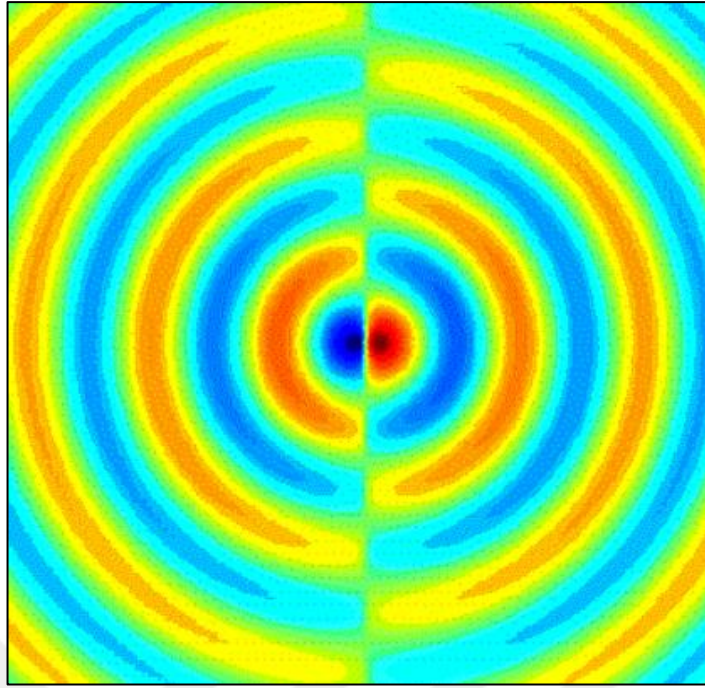


Figure 1.8: Sound propagation from a dipole noise source.

Dipole sources in turbomachines arise from rotation and non-uniform time and space inflow conditions. Tonal noise at blade passing frequency (BPF), blade tip clearance noise, noise from rotating stall, secondary flow (horseshoe vortices) effects, flow separation, stagnation points on the blade are typical dipole noise sources that appear in turbomachinery applications. Dipole noise source can generate both discrete and broadband noise according to their physical mechanism.

Uniform inlet flow and steady aerodynamic forces on the blade are very similar to monopole noise sources and cause a weak noise radiation. Non-uniform inlet flow forms non-uniform aerodynamic forces on rotating blades as their circumferential position changes. This generates aerodynamic noise at the BPF and its harmonics. Unsteady incoming flow causes random pressure distribution on the blades. Spatially non-uniform part of the incoming velocity generates discrete noise at BPF, while the velocity fluctuation generates broadband noise as seen in Figure 1.9. [6].

1.3.3 Quadrupole noise sources

Quadrupole noise sources are associated with the turbulent structure and the shear flow in the fluid domain. It consists of two dipole sources as seen in Figure 1.10 and can propagate in longitudinal, transverse and arbitrary directions.

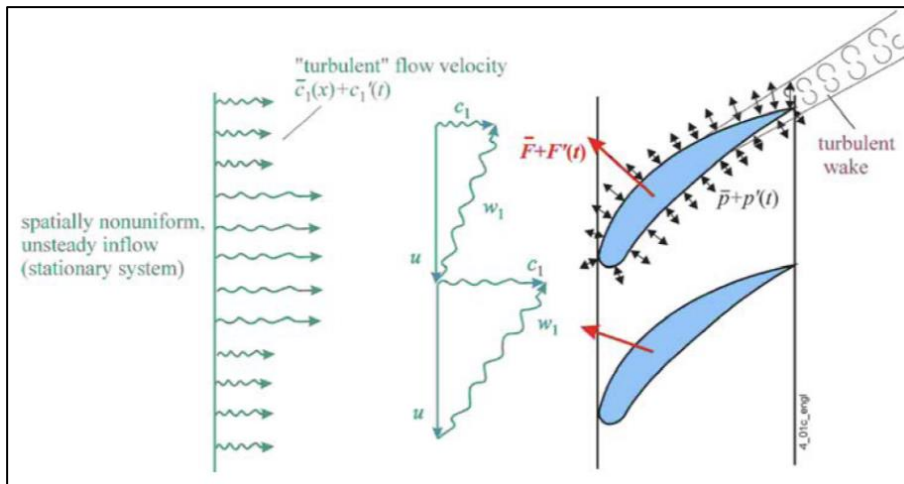


Figure 1.9: Noise generation on rotor blade [6].

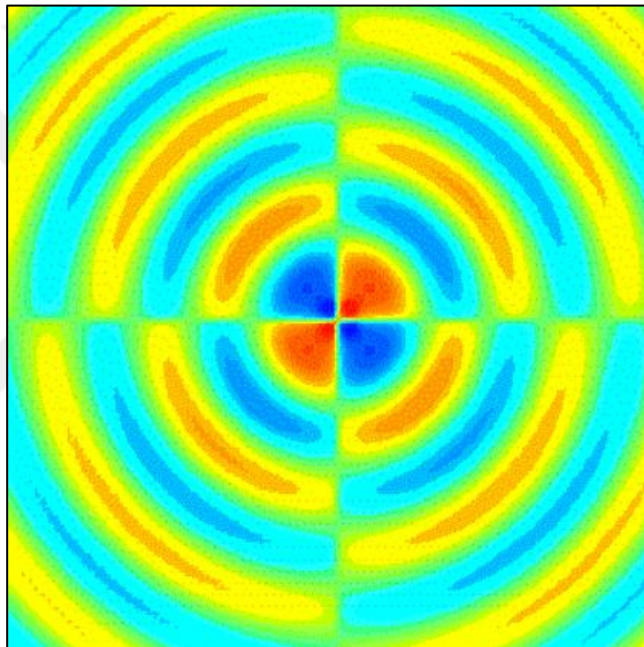


Figure 1.10: Sound propagation from a quadrupole noise source.

Random pressure fluctuations on the blade exists, even though the incoming flow be steady and uniform. This is caused by the turbulent boundary layer. These random force fluctuations generates some noise but it is not dominant.

Velocity fluctuations of the inlet flow due to the turbulent content generates random forces on blade and causes broadband noise as seen in Figure 1.9.

Vortex shedding from the trailing edge forms a wake in the downstream of the blade which contains high turbulence content and causes quadrupole noise source as in Figure 1.11.

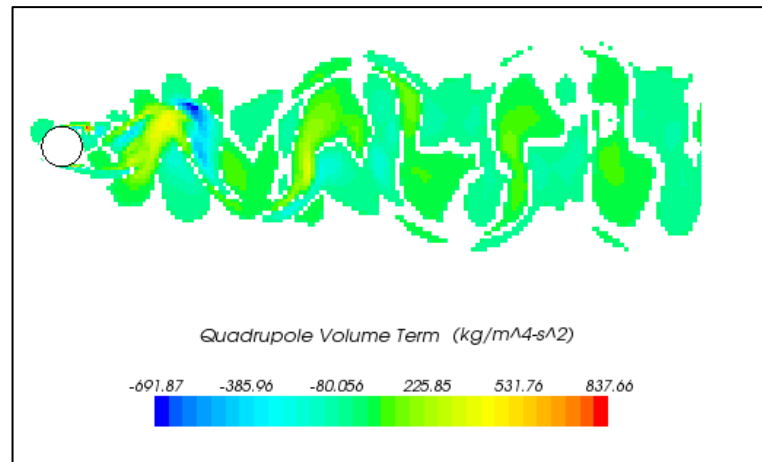


Figure 1.11: Quadrupole noise source extraction from a CFD simulation

1.4 Turbomachinery Noise Generation Mechanisms

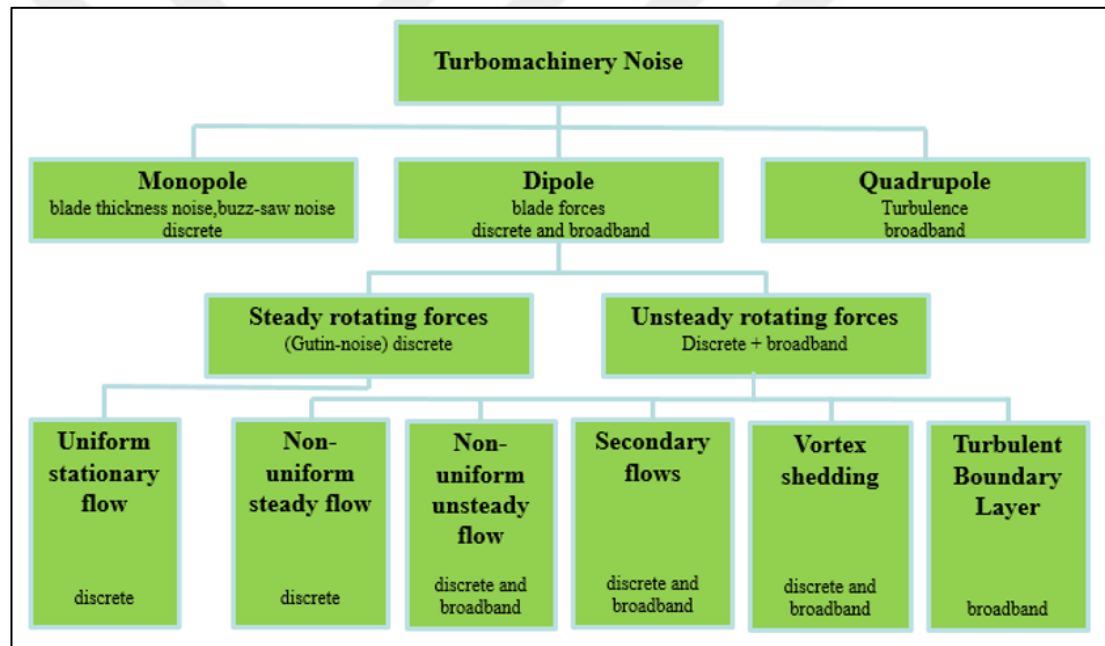


Figure 1.12: Turbomachinery noise generation mechanisms.

Figure 1.12 demonstrates the categorization of acoustic source mechanisms of turbomachinery noise. We can sort out the noise type of turbomachinery into two groups: broadband noise and tonal noise. Broadband noise has a continuous distribution on noise spectrum. The energy of the broadband noise is present over a wide range of frequencies. On the other hand, discrete noise only present at certain frequencies on the spectrum. Figure 1.13 is noise spectra of a subsonic compressor stage. As seen, discrete noise occurs at the BPF and its harmonics, whilst the broadband noise covers all the frequencies.

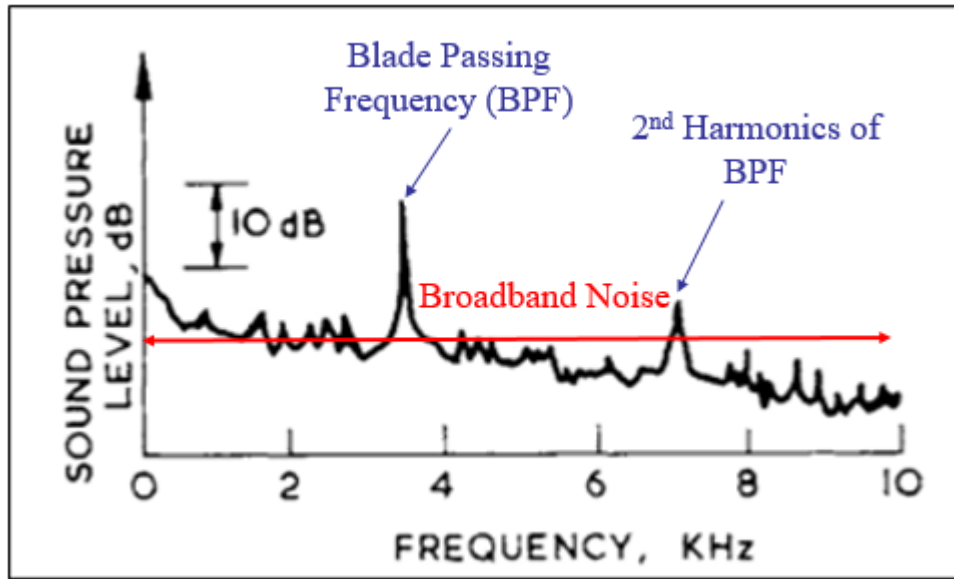


Figure 1.13: Noise spectrum a subsonic compressor stage (Cumpsty, 1985).

1.4.1 Turbomachinery broadband noise

Broadband noise is generated by random disturbances in the flow and characterized by a continuous noise spectrum. In Figure 1.14, generation mechanisms of turbomachinery broadband noise are illustrated on two-dimensional airfoil (Michel, 1994). Turbulent boundary layer, incident turbulence, vortex shedding, flow separation are typical broadband noise sources of turbomachinery applications.

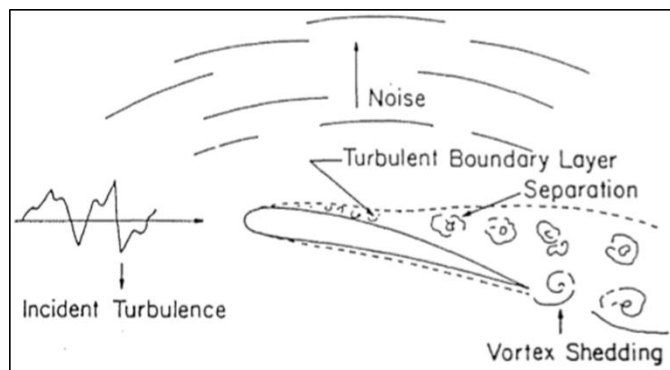


Figure 1.14: Broadband noise sources of an airfoil (Michel, 1994).

1.4.1.1 Turbulent boundary layer noise

Blades are exerted random pressure fluctuation when the turbulent boundary layer develops on blade surface. However, the propagation mechanism is not managed by the quadrupole sources in the turbulent boundary layer. The model which was proposed by Mugridge [7] resulted that the turbulent boundary layer noise is 10 dB lower than

the vortex shedding noise. That is why boundary layer noise can be considered as the lower limit of the broadband turbomachinery noise.

1.4.1.2 Incident turbulence noise

Velocity fluctuations due to the turbulent content of the incoming flow causes broadband noise spectrum when hit the blades. Sharland [8] conducted an experiment and placed the flat plate first exhaust of a jet flow and second further downstream of the jet. By this way, in the first case vortex shedding and second case incident turbulence dominated noise field.

1.4.1.3 Vortex shedding noise

Each vortex separation from a blade causes the flow to circulate around the blade in a different manner and hence pressure fluctuations on the blade arises. Two types of vortex shedding including laminar and turbulent boundary layer are significant for turbomachinery broadband noise.

The laminar vortex shedding noise stems from the instability waves in the laminar boundary layer, which travels along the suction side and causes a pressure disturbance while passing the trailing edge. This disturbance also propagates upstream again and establishes a loop as seen in Figure 1.15.

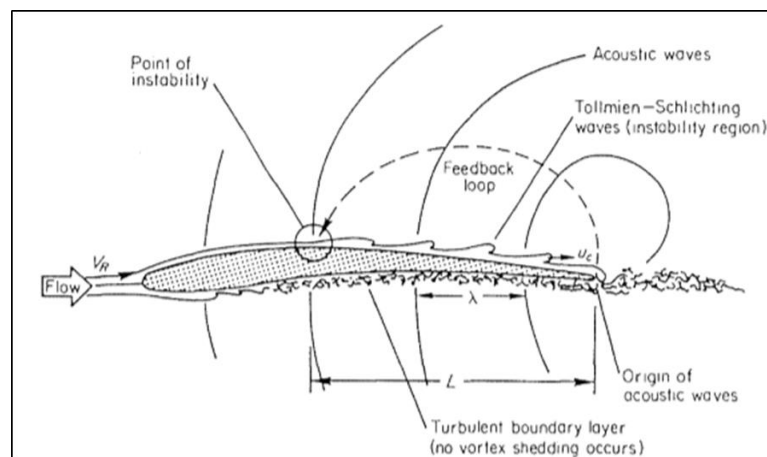


Figure 1.15: Laminar vortex shedding noise generation mechanism (Michel, 1994).

The turbulent vortex shedding shows its dominance at the high frequencies of the noise spectrum. Bluntness of the trailing edge is deterministic on the turbulent shedding noise.

1.4.1.4 Flow separation noise

Flow separation as seen in Figure 1.16 (a) causes random pressure fluctuations on the blades, which are the responsible for the broadband noise generation. Since the blade wall are in interaction with the random separations, the source type of the noise is dipole. The stalled rotor blade shows complete broadband noise characteristics on the spectrum. However, unstalled operation produces blade passing frequency and harmonics as seen in Figure 1.16 (b).

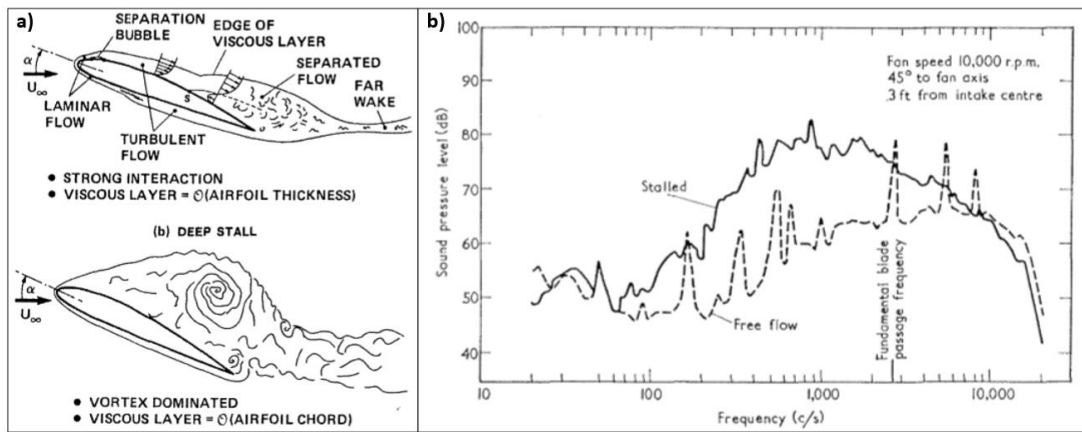


Figure 1.16: Airfoil unstalled & stalled flow (a), noise diagram of unstalled & stalled flow (b) (Michel, 1994).

1.4.1.5 Tip vortex noise

Pressure difference on the each side of the blade forces the fluid migrate from pressure side to suction side by passing the blade tip. A vortex on the suction side arises and causes unsteady blade forces. Tip clearance noise can be severe noise source for ducted axial machines depending on the height of tip clearance. As seen in Figure 1.17 secondary vortex pattern occurs in addition to the typical tip vortex and extends through the azimuthal direction.

1.4.1.6 Quadrupole noise

The pure quadrupole noise generation due to the turbulent velocity fluctuations can be generally neglected compared to the dipole radiated broadband noise. The reason for that, the fluctuation forces on the surfaces forms a sound intensity that changes as the sixth power of the characteristic velocity, whilst for quadrupole sound this relationship is eight power. In experiments, sixth power dependence was acquired as the broadband spectrum of the turbomachinery noise. Consequently, pure quadrupole noise

generation may be neglected compared to broadband noise of dipoles in turbomachinery applications.

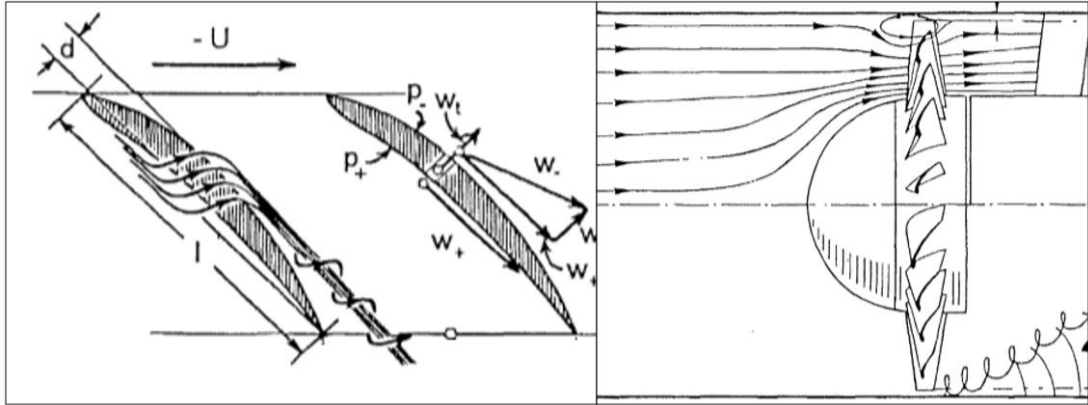


Figure 1.17: Tip vortex formation in axial turbomachinery applications (Michel, 1994).

1.4.2 Turbomachinery discrete (tonal) noise

Another turbomachinery noise component is discrete (tonal) noise. Tonal noise radiates at discrete frequencies and composed of three main sources. These are associated with the volume displacement, steady loading on rotating blades and dynamic loading of blades. The first two is can be calculated by steady state methods. However, the last one requires using transient methods to predict the interaction of the rotating and stationary blades [9]. Uniform steady flow on rotor, non-uniform steady flow on rotor, rotor-stator & rotor-casing interaction, impulsive noise, rotating stall and buzz-saw noise are the typical tonal noise generation mechanisms of rotating machines.

1.4.2.1 Noise generation due to uniform & steady rotor blade forces

Spatially uniform and steady flow field as seen in Figure 1.18 (a) forms steady forces on the blades. However, periodic pressure fluctuations at BPF and higher harmonics are observed in stationary reference frame. This noise generation mechanism is named after Gutin [10] who conducted a research on noise due to the steady forces exerted on a propeller. These forces were modeled as dipole sources and their radiation is weak in case if low Mach number operated turbomachinery applications. Wright [11] stated that radiation of this noise could be controlled by increasing the number of blades.

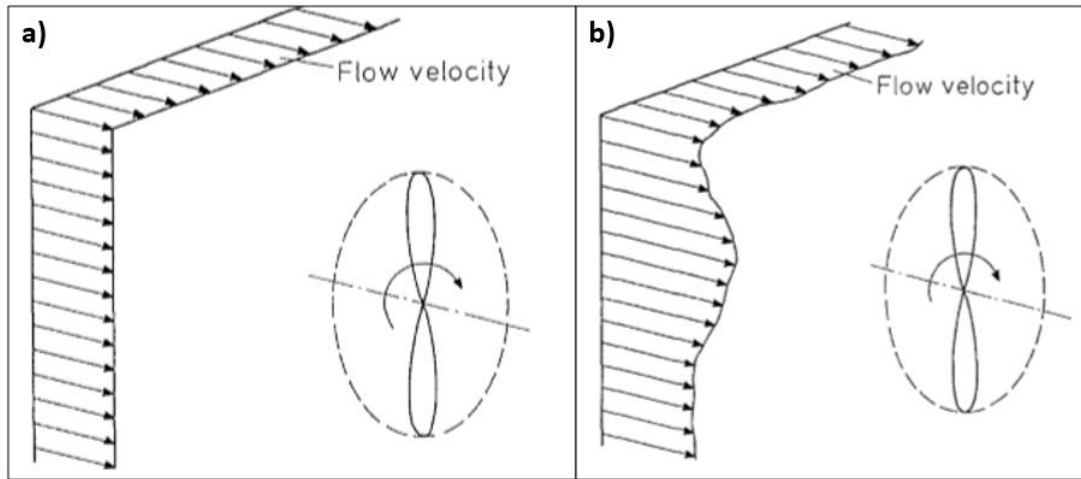


Figure 1.18: Spatially uniform (a) & spatially non-uniform (b) incoming flow (Michel, 1994).

1.4.2.2 Noise generation due to non-uniform & steady rotor blade forces

When a rotor undergoes a spatially non-uniform steady flow field, unsteady fluid forces occur on each blade, because the magnitude and the angle of attack of the incoming flow that each blade faces differ as the rotor changes its angular position as seen in Figure 1.18 (b). Noise generated by the interaction of the rotor and non-uniform steady flow field is tonal at the BPF and its harmonics. Because of high azimuthal direction velocity, acoustic propagation of unsteady blade forces is strong; hence, this constitutes an important turbomachinery noise.

1.4.2.3 Rotor-stator interactions

In multi-stage turbomachinery applications, an interaction takes place between the rotating blades (rotor) and stationary vanes (stator). This interaction may take place in different mechanisms including potential field of the blades, viscous wake, vortex shedding and shock wave interactions as seen in Figure 1.19.

Potential effect stems from the presence of adjacent blade rows. In axial compressors, the presence of the stator vanes in the downstream of the rotor blades forms fluctuations in the static pressure field in the neighborhood of the rotor and stator. These excitations are densely transient and depend on the blade passing frequency. The effects of the potential field on flow dynamics propagate with the speed of sound both upstream and downstream of the flow, unlike the vortical excitations. Potential effect can become severe when the axial gap between the rotor and stator is decreased. It is stated that 85% to 95% of the unsteady loading on the blades can be generated by the potential

field [12]. The potential field may interact with the wake of the blade and lead to non-linearities especially in transonic machines. Laborderie et al. [13] conducted numerical analysis of a subsonic rotor stage and found that stator potential field is the second major tonal noise source on the rotor blades. He further emphasized that right in front of the stator leading edge a deficit occurs in velocity field, yet the fluid accelerates through the passage between two stator blades. The field that this potential effect covers is important because it can maintain its effect towards rotor trailing edge and can be sensed between rotor blades. He observed that pressure fluctuations caused by potential effect of stator vane could reach up to 80% chord length of the rotor blade from the trailing edge of the rotor as seen in Figure 1.20.

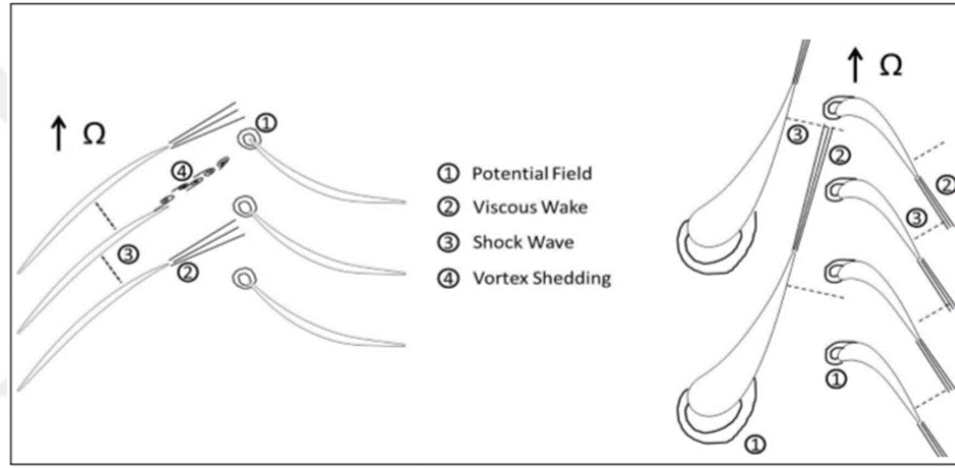


Figure 1.19: Rotor-stator interaction mechanisms in axial turbomachinery applications (Sadek, 2015).

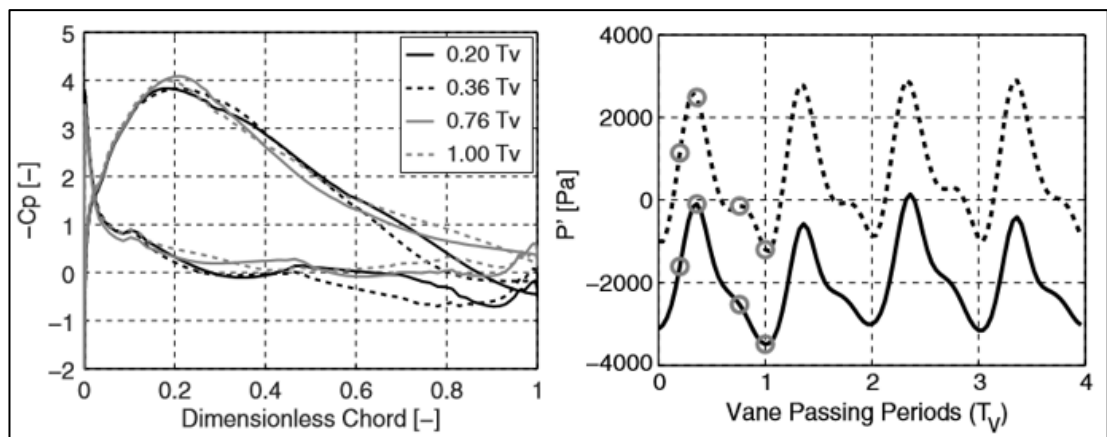


Figure 1.20: Stator potential effects on rotor blade for an axial transonic compressor [13].

Sanjose et al. [14] numerically investigated the potential effect of the outlet guide vanes (OGV) on rotor blades. He found that the presence of the potential effect has

effect on the rotor wake, consequently affects the wake excitations. Astley (2010) observed unsteady pressure components on the rotor due to the potential field of the downstream stator vanes.

Vortical effects are composed of vortex shedding, viscous wake, tip leakage and horseshoe vortices. They are defined as velocity deficit fields with total pressure loss and arise behind the obstacles in the flow. The forcing magnitude of the vortical effects depends on the strength and the width of the wake. The wake strength is associated with the engine operation order and the length from the source. As the viscous wake proceeds, it experiences a dissipation process, in which the kinetic energy in the wake is converted to internal energy irreversibly. Divergent fluid area between the compressor blades enables the wakes to stretch when the wakes proceed through compressor stages. Consequently, total pressure and efficiency can be recovered in compressors. This cannot appear in turbine stages due to converging area between blades. As seen in Figure 1.21 the incoming wake is compressed through the turbine blades rather stretching.

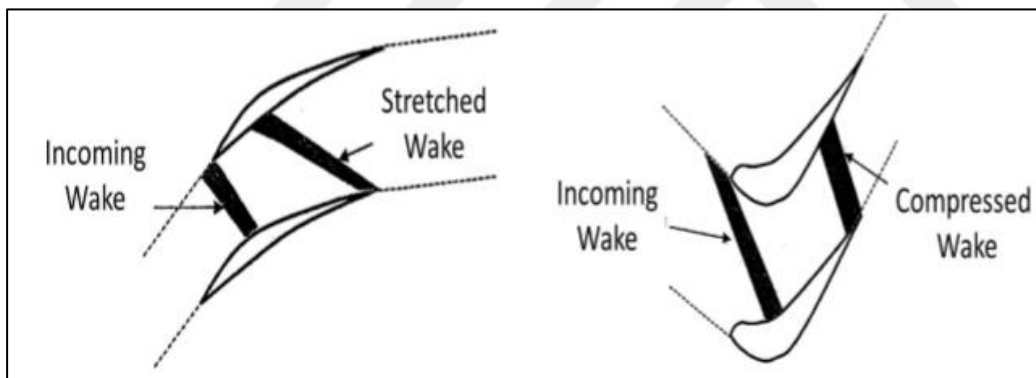


Figure 1.21: Wake stretching and compressing mechanisms (Sadek, 2015).

Laborderie et al. (2014) defined the rotor wake as deficit in axial velocity and increase in tangential velocity, which impinges the stator vane with a higher incident angle. This impingement takes place periodically at the frequency of blade passing (BPF) and results in pressure fluctuations having same frequency as BPF on the entire surface of the stators. These unsteady pressure fluctuations on the stator wall radiate as acoustic waves at BPF and its higher harmonics.

He further attributes the tonal noise generation mechanism of the wake to upwash velocity. Upwash velocity constitutes the normal velocity component to the stator vane. As seen on the velocity triangles in Figure 1.22 relative velocity of wake $W_{2, \text{wake}}$

is lower than the bulk flow relative velocity W_2 . That is why the absolute velocity of the wake $C_{2, \text{wake}}$ impinges the stator blade with a higher incidence than the absolute velocity of the bulk flow C_2 does. Thus, a velocity component that is normal to the stator vanes occurs as called upwash velocity $C_{2, \text{upwash}}$. This unsteady velocity component of $C_{2, \text{upwash}}$ impinges stator leading edge at the BPF and generates pressure fluctuations at the stator surface by changing angle of attack.

Gallus et al. [15] stated that the rotor wake-stator interaction rather than the stator potential field dominantly generate unsteady forces on the blades. Kohlhaas et al. [16] observed blade wake produced upwash velocity had significant effect on second harmonic on BPF, yet fundamental BPF is strongly determined by potential flow field of downstream stator, hub and shroud vortices. He concluded that controlling the rotor wake might only result in eliminating second harmonic of BPF tones, but not the BPF itself. Walker [17] claimed that viscous wake interactions were the predominant tonal noise source for the blade rows used in fans and compressors. He experimentally studied the interaction of blade wakes of two adjacent axial compressor stages and found that the blade wake interactions in multistage axial turbomachinery applications reduce the velocity deficit in the downstream wakes and consequently generates less tonal noise at BPF.

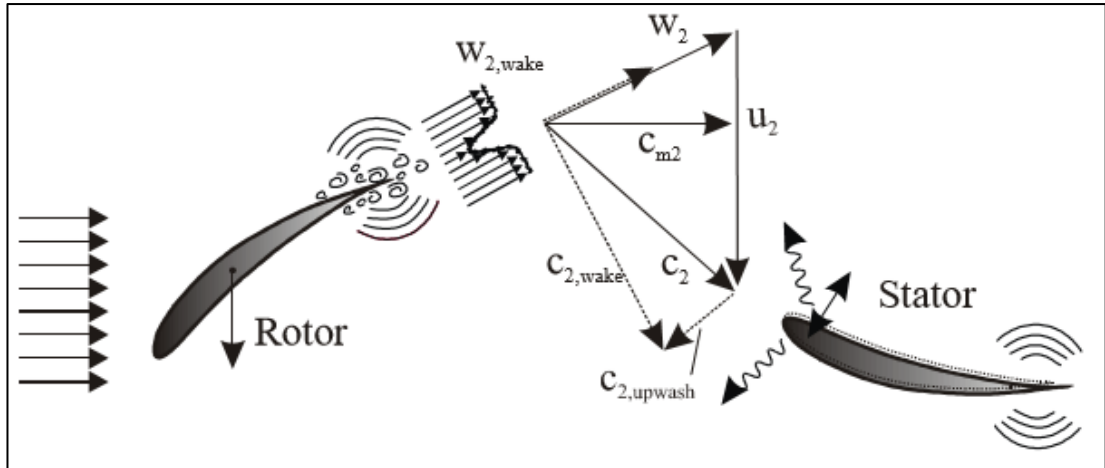


Figure 1.22: Upwash velocity in axial compressor (Kohlhaas et al, 2013).

In centrifugal fans, the unsteady pressure fluctuations take place at the cutoff (tongue) part of the casing due to the periodically leaving flow from the rotating impellers unless outlet guide vanes are present. This mechanism constitutes the main source of the tonal noise in centrifugal compressor without outlet guide vanes. Cai et al. [18]

numerically studied the cut-off impeller interaction tonal noise. He observed that inclined cut off shape with larger cut-off impeller clearance resulted in lower BPF pressure fluctuations. Darvish et al. [19] numerically investigated the effect of volute tongue shape on the tonal noise generated by centrifugal fan. He conducted three dimensional CFD analysis of centrifugal fans different stepped tongue configurations (see Figure 1.23) and achieved to reduce the tonal noise around 8 dB at BPF.

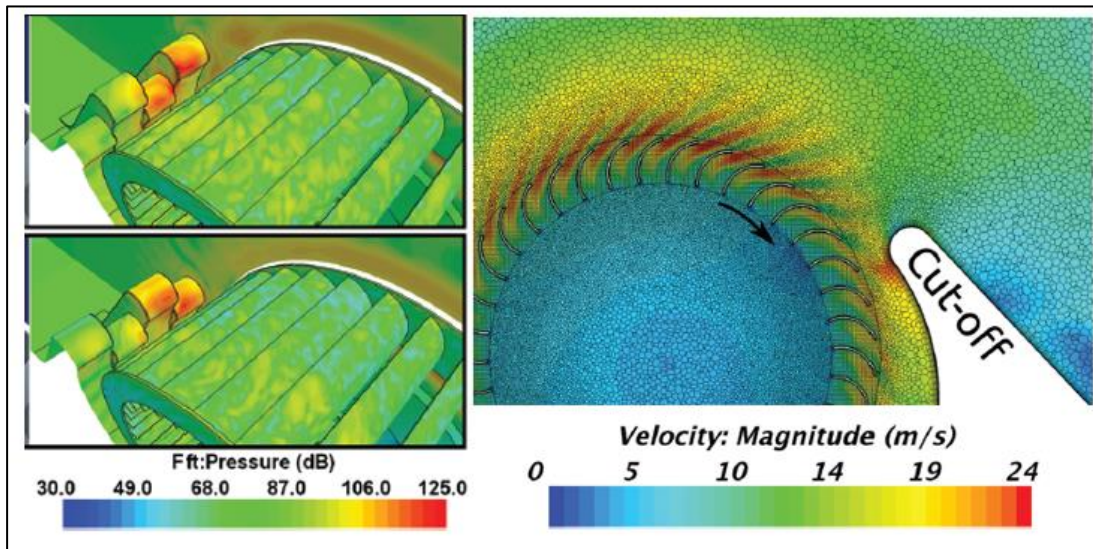


Figure 1.23: CFD investigation of effect of cut-off design on centrifugal fan noise (Cai et al, 2010).

In transonic and supersonic operating conditions, shock waves occur on rotating and stationary blades. Shock waves generally appear in three forms including; bow, oblique and normal shocks. Besides their propagation in the downstream direction, the adjacent blades also reflect shock waves. When the shock wave interacts with the adjacent blade's boundary layer shock boundary layer interaction phenomena takes place. This concludes with boundary layer separation and entropy generation on blade surface as seen in Figure 1.24.

1.4.3 Rotating stall noise

Rotating stall arises in a low flow rate operation of a fan or compressors with local flow disturbances that trigger stalls on one of the blades. In Figure 1.25, the suction side of the blade in the middle undergoes a stall event. This forces the mean flow deviate from its original route and it is diverted away in the opposite direction of the rotation. This results in an increase of the angle of attack on the first adjacent blade on the suction side of the originally stalled blade. Because of increase in angle of attack,

this blade undergoes stall. On the other hand, the angle of attack decreases for the first blade near to the pressure side of the originally stalled blade, hence the flow is attached on this blade. In this manner, the stalled region migrates around the blades in the opposite direction of rotation at a frequency of 0.3 to 0.5 of the shaft. Consequently, unsteady pressure fluctuations on the blades arise and noise is generated.

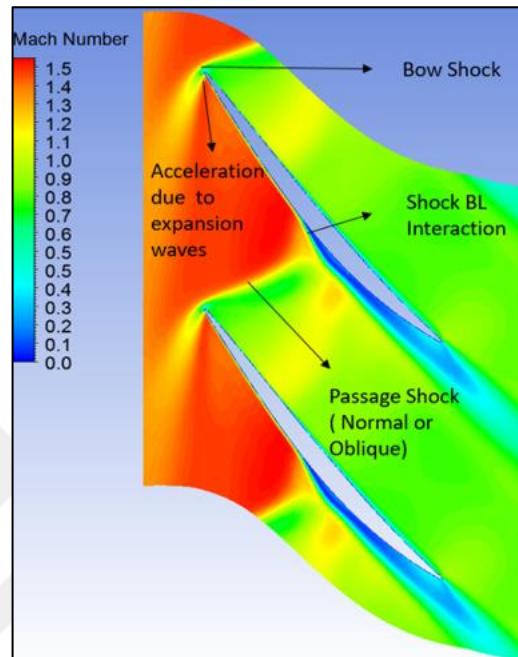


Figure 1.24: Shock mechanism in on a transonic compressor rotor.

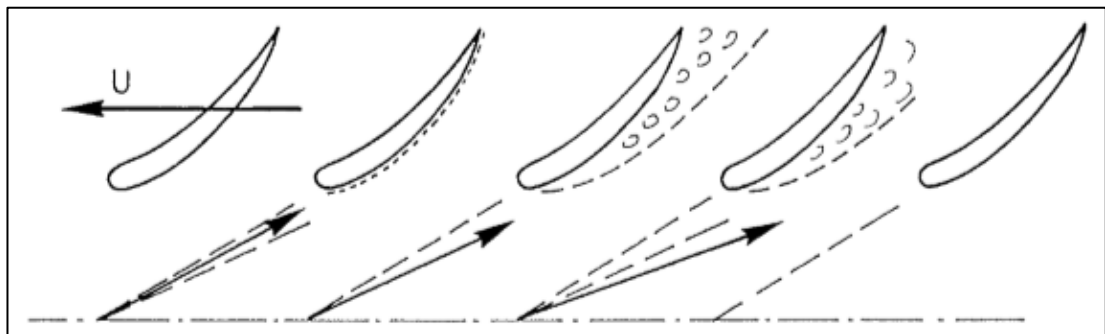


Figure 1.25: Schematic representation of rotating stall in an axial compressor (Michel, 1994).

1.4.4 Buzz-saw (multiple pure tone) noise

In supersonic operating conditions shock waves occurs in the upstream region of the rotor blades. These shock waves are identical in case of perfectly periodic alignment of the rotor blades. In Figure 1.26, a probe in the upstream of the rotor blades measures exactly periodic pressure time history leading to sound generation at the BPF and its harmonics. However, in practice, rotor blades are not periodically aligned as designed

due to the small manufacturing imperfections. Each blade might have different geometry, spacing, stagger angles which perturbs the regular shock patterns. Shock waves with higher amplitudes radiates faster than the ones with lower amplitudes because of the nonlinear effects. Irregularities observed in the shock waves increases as the waves propagate away from the rotor blades. Consequently, an uneven combination of the shock waves arises and multiple tones are radiated at the rotor shaft rotating frequency. This noise type is called as ‘buzz-saw’ noise and observed only in the upstream side of the rotor since the flow in the downstream side of the rotor becomes subsonic.

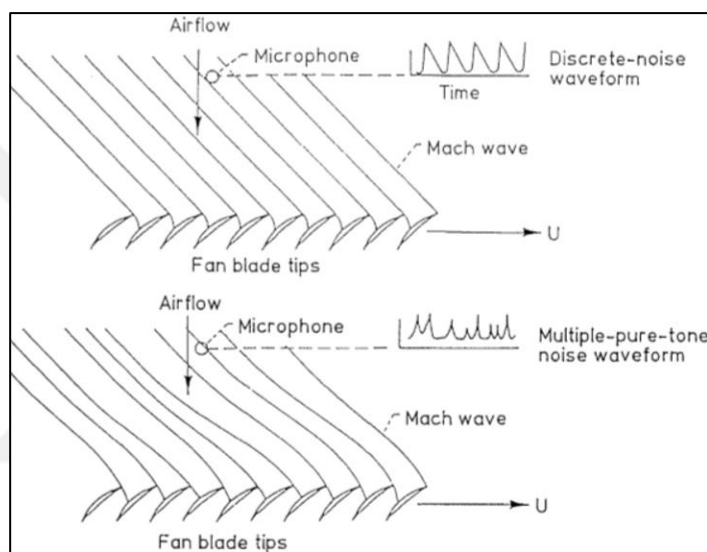


Figure 1.26: Regular discrete noise (above) and buzz-saw noise (below) (Michel, 1994.)

2. ROTOR-STATOR INTERACTION NOISE REDUCTION METHODS

In axial turbomachinery applications the interaction between the successive rotor and stator rows are the fundamental source of the tonal noise. The potential effect of the presence of the downstream blade rows and the wakes shedding from the rotating or stationary blades interacts with the adjacent blade rows and causes strong unsteady pressure fluctuations on blade surfaces. As the noise regulations become stricter, designers develop aerodynamics methodologies to reduce the noise generated by rotor-stator interaction without sacrificing the efficiency of the machine. Increasing axial gap between rotor and stator, optimizing rotor and stator blade geometry, wake filling technology, porous stator surface skin are several applied methodologies for reducing tonal noise due to rotor-stator interaction.

2.1 Increasing Rotor-Stator Axial Gap

In case of multi-stage turbomachines, successive rotor and stator stages interact each other and generate non-uniform bulk flow conditions. The presences of the upstream and downstream blade rows yields in unsteady interaction forces, even if the fluid domain is considered as inviscid. However, the effect of the potential field decreases strongly as the axial gap between the rotor and stator row increases, and hence the interaction of the potential field is significant when the length of the axial gap becomes short. On the contrary, wakes from upstream rotor or stator blades do not decay as rapidly as the potential effect, and thus unsteady pressure fluctuations on the blades which generates tonal noise are dominantly build up by the wake interactions. Gallus et al. (1997) studied the axial gap distance effect on the potential field and wake interactions. He stated that amplitude of the potential effect-generated unsteady pressure fluctuation vanishes quickly with the increasing axial gap. In contrast, wake interactions shows a slightly decreasing trend as shown in Figure 2.1. Moreover, he experimentally observed that the stimuli forces which represents the dynamic blade forces decreases with the increasing axial gap.

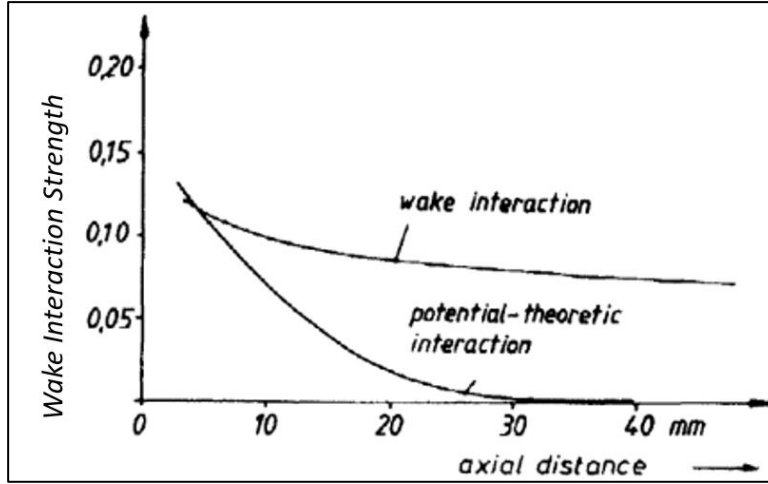


Figure 2.1: Wake interaction strength versus axial distance between rotor and stator in an axial compressor stage (Gallus et al, 1997).

Saren et al. [20] observed that potential effect disturbances can reach up to one pitch upstream from the potential field generating blade row. According to their research level of the unsteady disturbance is severely related to axial gap between the rotor and stator blades. Sanjose et al. (2015) numerically studied different rotor stator axial gap configurations and observed potential effect of the stator on rotor blades reduces as the axial gap increases as seen in Figure 2.2.

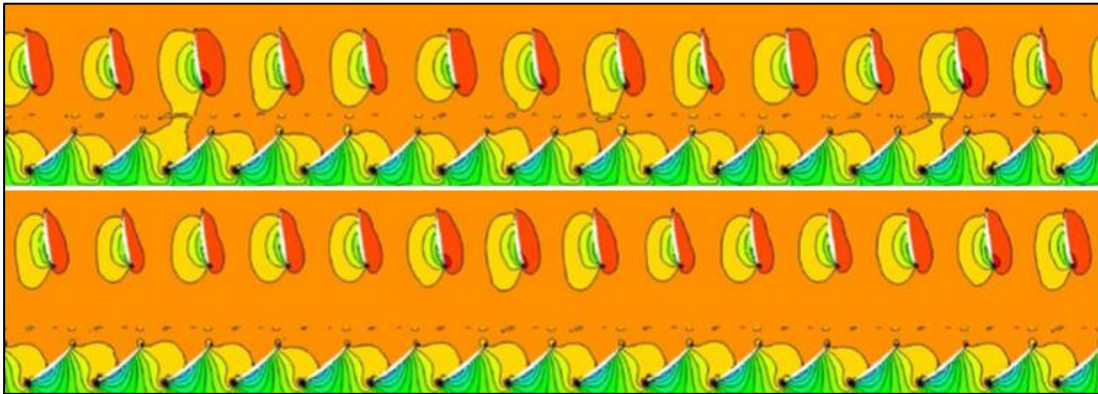


Figure 2.2: Pressure contours for different axial gap configurations (Sanjose et al, 2015).

Balombin et al. [21] experimentally studied the effects of rotor-stator axial spacing on the noise generation on a 1.83-meter-diameter turbofan without inlet guide vanes. The axial gap between rotor and stator was changed from one to two-rotor chord length. They observed no reduction in the fundamental tone since it was designed to be cut-off (rotor-stator interaction would not effect this tone). However, remaining higher harmonics of the BPF decreased approximately at a rate of 6 dB per doubling of axial

gap. Benzakein [22] stated doubling axial gap between the rotor and stator resulted in approximately 4 dB noise reduction in case of the axial gap length is shorter than a blade chord, whilst the noise reduction level is around 2 dB by each axial gap doubling when the axial gap is larger than a blade chord. Hünecke (1997) stated in his book that, noise reduction of 6 to 8 dB is possible by increasing the axial gap. However, increase in axial gap may result in a loss of fan efficiency of 1%. Sousa [23] conducted flat plate experiments to study the effect of axial gap distance on the tonal noise. He investigated three different axial gap configurations including one, two and three chord length. He observed that BPF tone decreased approximately 6 dB from one chord length case relative to the two and three chord length. While two chord length and three chord length cases showed same value at BPF, since the wake decayed already. For the higher harmonics the reduction in the noise level stayed in the uncertainty level expect for 7th harmonic as seen in Figure 2.3.

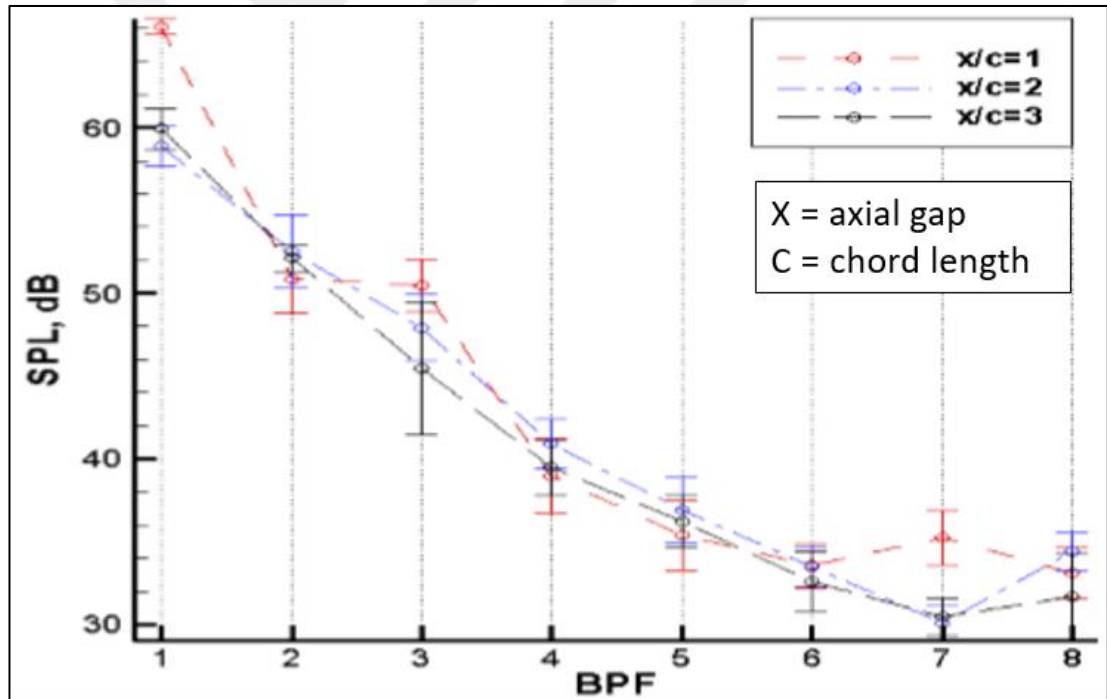


Figure 2.3: Tonal noise measurements at BPF for three different axial gap configurations (Sousa, 2005).

2.2 Leaned & Swept Stator and Rotor Blades

One of the design concept to reduce the noise generated by the rotor wake-stator interaction is using leaned and swept stator geometry. Stator sweep is the axial displacement of the stator in radial direction such that the tip of the stator blade is

located forward position downstream relative to hub. Lean is the azimuthal displacement of the stator blade alignment line relative to the radial coordinate as seen in Figure 2.4. These two stator configurations have been suggested to reduce to unsteady pressure fluctuations on the stator with its response to the rotor downwash. Noise reduction mechanism is the local cancellation of the unsteady pressure sources on the stator blade. Benzakein (1972) studied the effect of leaned stator on noise generation. With a lean angle of 30° BPF tone noise and broadband noise decreased 12 dB and 8 dB respectively even for a aircraft engine fan stage with a quite large axial gap which damps the potential effects of the stator. Kazin [24] reported 1.5 to 3.5 dB noise reduction at the second harmonics tone with a 30° leaned stator blade in the rotating direction. Woodward et al. [25] tested a high bypass ratio turbofan and demonstrated integration of the stator sweep and lean combinations may efficiently reduce both tonal and broadband noise. They compared the swept & leaned stator to the conventional radial stator row and reported a significant noise reduction, yet the aerodynamic losses oppositely somewhat increased.

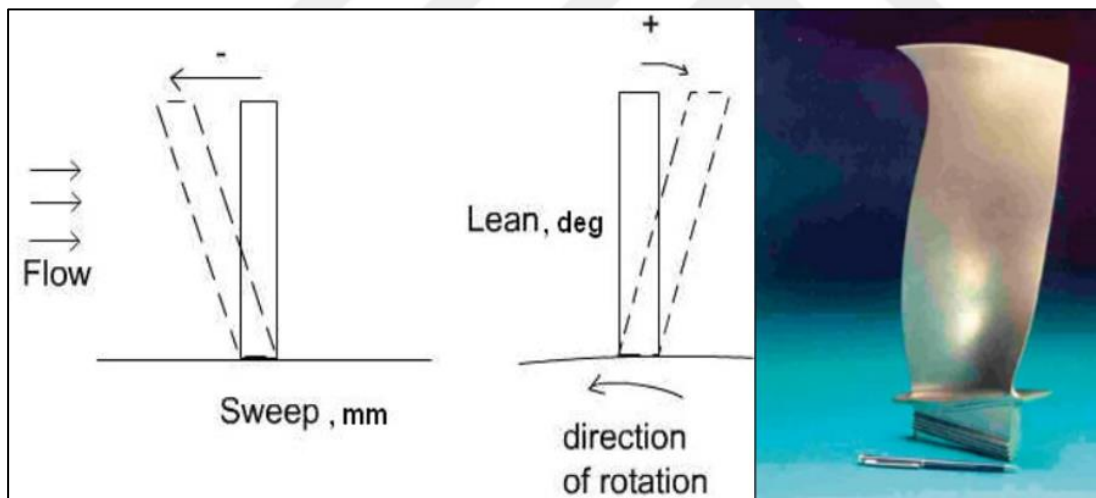


Figure 2.4: Swept and leaned stator and rotor modifications on axial turbomachinery applications.

Contemporary computation aided engineering design tools and manufacturing methods enable engineer to design and optimize new rotor blades in order to reduce tonal noise without sacrificing aerodynamic performance. For instance, rotor sweep is a way to reduce the shock strength on the fan rotors. A modern fan blade design, which was developed by computational fluid dynamics and computational aeroacoustics methods for lower tonal noise generation is illustrated in Figure 2.4.

2.3 Wake Filling Method

As explained in the previous chapters, upwash velocity due to the momentum deficit in the rotor wakes from the rotor blades is responsible for the rotor-stator interaction noise. Disturbance in the rotor wake periodically hits the stator surface giving rise to unsteady pressure fluctuations on the stator. The deficits in the rotor wake yield in different absolute velocities component in the wake relative to main flow. The flow field having different velocities hits the stator leading edge at blade passing frequency. Studies in the literature proposed that wake filling might bring about significant reduction of tonal noise reduction generated by rotor-stator interaction. Kohlhaas et al. (2013) tested a wake filling method called ‘trailing edge blowing’ on an axial compressor. This technique can be defined as controlling flow on the blade trailing edge to re-energize the momentum deficit in the wake. Controlling fluid is fed from the hub of the blade and flows through the internal passage in the rotor blade until it is sprayed from the rotor trailing edge as seen in Figure 2.5. Using this technique they observed that the second harmonics of the BPF was almost eliminated, however tone at BPF did not change. Brookfield et al. [26] achieved to reduce the 85% of the first three blade passing frequency harmonics tone amplitude compared to original rotor configuration case using the trailing edge blowing method.

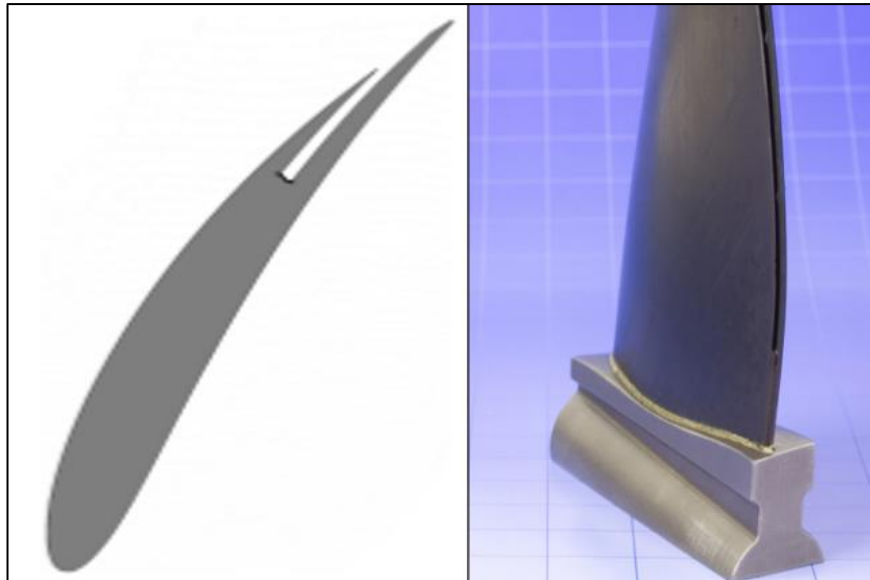


Figure 2.5: Wake filling technology applications on rotor blade [26].

2.4 Arranging Appropriate Number of Rotor and Stator Blades

Choosing an appropriate blade/vane ratio is another method to obtain the cut-off phenomena for propagation of blade passing frequency tone. Cut-off can be defined as the degradation of radiation of the fundamental tone by controlling the rotor stator interaction with appropriate number of blade and vanes. Although the fundamental tone is theoretically cut-off, it propagates in the far field, yet with weakened amplitude. Balombin et al. (1974) investigated the QF-5 fan stage with the appropriate ratio of blade/vane providing cut-off at the blade passing frequency tone. He measured the acoustic spectrum of the fan and observed that the amplitude of the second harmonic was larger than the BPF tone as shown in Figure 2.6.

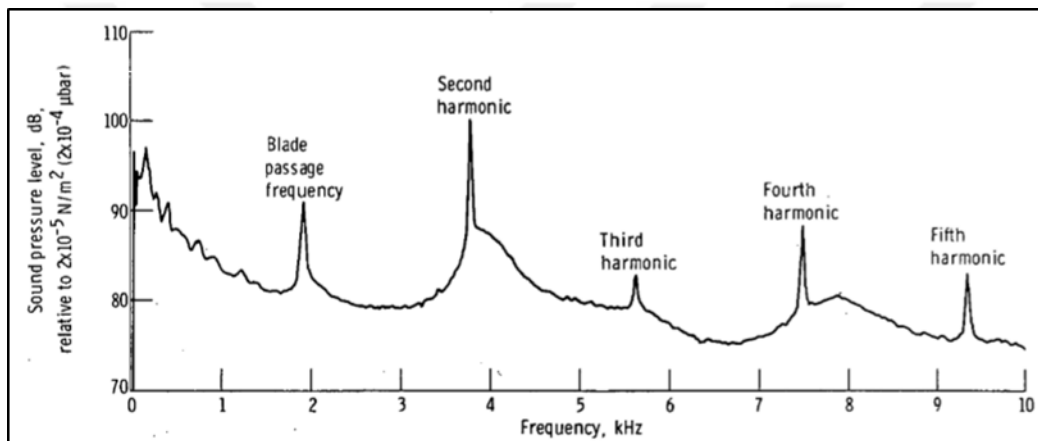


Figure 2.6: Noise spectrum of QF-5 turbofan (Balombin et al, 1974).

Sousa (2005) studied the number of the rotor blades on the effect of tonal noise generations. He tested the rotor configurations consisting of 3, 4 and 9 blades respectively. It was reported that 3-bladed result showed higher tonal noise amplitudes at BPF and some higher harmonics. It can be stemmed from the smaller amplitude of the stator response to the BPF with increased number of rotor blades. However, broadband noise increased beyond a certain frequency for 9-bladed rotor case since the overall turbulence content in the rotor downstream augmented.

2.5 Porous Skin on Stator Surface

Porous skin coating on the stator surface, as seen in Figure 2.7, permits a mass flux between high and low pressure regions, thus the pressure distribution on the surface is modified and the unsteady interaction effects on the stator can be attenuated. Reducing the unsteady forces acting on the stator results in alleviating the radiated rotor-stator

interaction tonal noise. Raymond et al. [27] reported that the porous coating allows formation of a smoother pressure distribution. Unsteady lift force was reduced 18% consequently 2 dB tonal noise reduction was observed. Tinetti [28] optimized the porous coat structure on the stator surface, which provides approximately 1 dB reduction of the tonal noise without severely changing lift and drag characteristics.

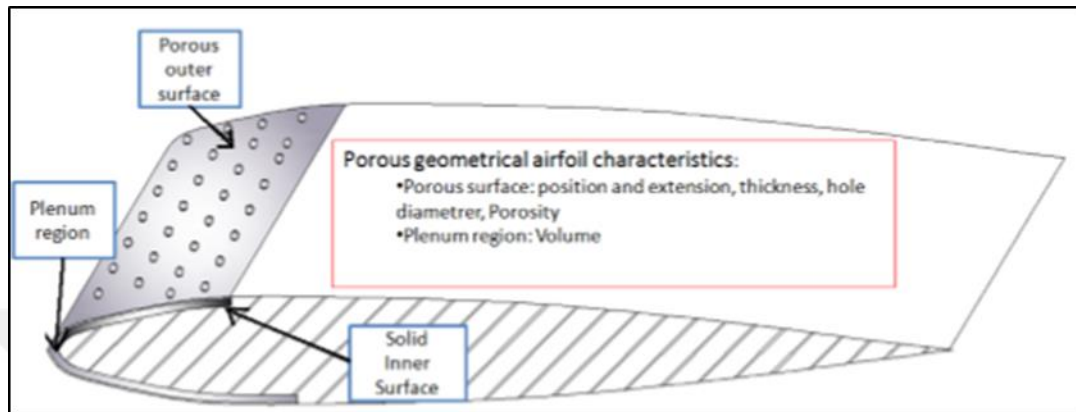


Figure 2.7: Porous skin on stator surface (Tinetti, 2001).



3. STEADY STATE CFD ANALYSES OF NASA ROTOR 37 & STAGE 37

In the present study aerodynamic flow analysis of a transonic compressor stage was performed. The main investigation point was the tonal noise generation due to the rotor-stator interaction under different rotor stator axial spacing configurations. As a transonic axial compressor, NASA Stage 37 was selected for this research. Before conducting transient aeroacoustics analysis, steady state aerodynamic performance analysis of rotor alone and stage configurations were completed for the validation of the numerical method and understanding the characteristics of the flow field.

3.1 NASA Rotor 37 & Stage 37 Design

Rotor 37 is a worldwide known axial transonic compressor rotor test case designed by NASA Lewis Research Center. It was designed to investigate the 3D viscous flow pattern in transonic axial compressors. The meridional view of the Rotor 37 design is demonstrated in Figure 3.1.

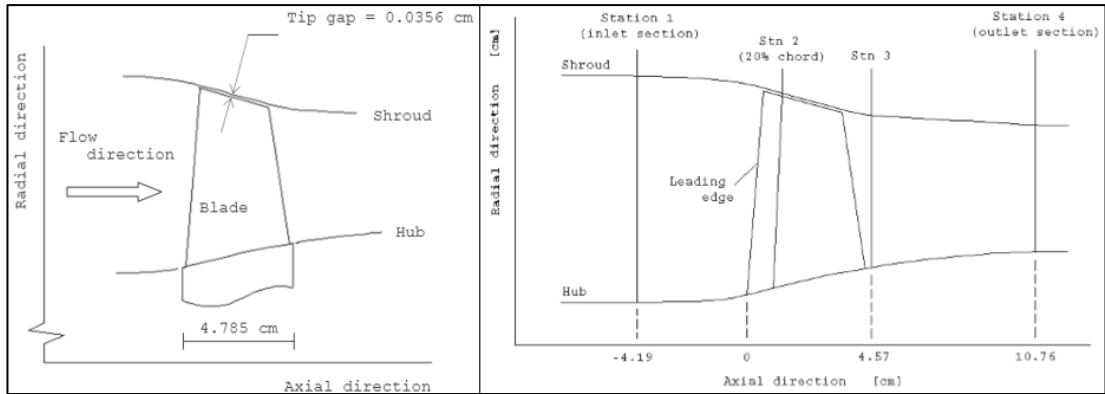


Figure 3.1: NASA rotor 37 meridional position.

The Rotor 37 consists of 36 Multiple-Circular-Arc (MCA) blades with hub-tip diameter ratio of 0.7 at the inlet. The aspect ratio of the blade and the tip solidity is 1.19 and 1.29 respectively. Operating rotor tip clearance is 0.356 mm corresponding to 0.5% of inlet and 0.7% of outlet span. The hub relative Mach number at the inlet is 1.13 and the tip relative Mach number at the inlet is 1.48 corresponding to 454 m/s at

the design speed of 17188 rpm. The design total pressure ratio and the isentropic efficiency are 2.106 and 0.877 respectively at a mass flow rate of 20.19 kg/s. Overall aerodynamic design parameters of Rotor 37 is illustrated in Table 3.1.

Table 3.1: Rotor 37 aerodynamic design parameters.

Rotor 37 Design Specifications	
Number of Blades	36
Leading Edge Tip Diameter	0.5074 m
Leading Edge Hub Diameter	0.3576 m
Rotational Speed (corrected)	17188.7 rpm
Tip Solidity	1.288
Tip Clearance	0.356 mm
Tip Speed	454.14 m/s
Total Pressure Ratio	2.106
Mass Flow Rate (corrected)	20.19 kg/s
Blading	Multiple Circular Arcs

The test case of the Rotor 37 was run at different rotational speeds and mass flow rates from surge to choking point. Aerodynamic measurement stations of the test is shown in Figure 3.1. Total pressure ratio, total temperature ratio and the isentropic efficiency of the rotor 37 is measured at station 4. Measurements were taken at 100%, 90% and 70% of the design speed which were 1800 rad/s, 1620 rad/s and 1260 rad/s respectively. Total pressure ratio, total temperature ratio and isentropic efficiency results are reported in Figure 3.2. The choking point and the surge margin of the Rotor 37 test case for three different rotational speed are distinct as seen. Total pressure ratio is calculated by dividing the average total pressure measured at station 4 (P_4) to reference pressure which is sea-level atmosphere value of $P_{ref} = 101325$ Pa. In the same manner, total temperature ratio is found by dividing the average total temperature measured at station 4 (T_4) to reference temperature of $T_{ref} = 288.15$ K. Spanwise variation of mass-flow averaged flow features at measurement station 4 for 98% of choking mass flow rate is illustrated in Figure 3.3. Reduction in the adiabatic efficiency over 80%span is remarkable since irreversibilities such as strong shock wave interactions with the casing boundary layer and tip leakage flow take place near the hub casing. This resulted in a temperature increase in this region as expected.

The Stator 37 has also Multiple Circular Arc blade shape. The Stator 37 row consists of 46 vanes with a tip solidity of 1.3 and aspect ratio of 1.26. Stage 37 which includes 36 Rotor 37 blades in the first row and 46 Stator 37 vanes in the following row as outlet guide vane. The design total pressure ratio of the Stage 37 is 2.05 at an airflow

rate of 20.2 kg/s. Relative Mach number at the rotor inlet varies from 1.25 at the hub to 1.493 at the top corresponding speed of 454 m/s. On the stator side, it varies from 0.736 to 0.810. Figure 3.4 shows the meridional flow path.

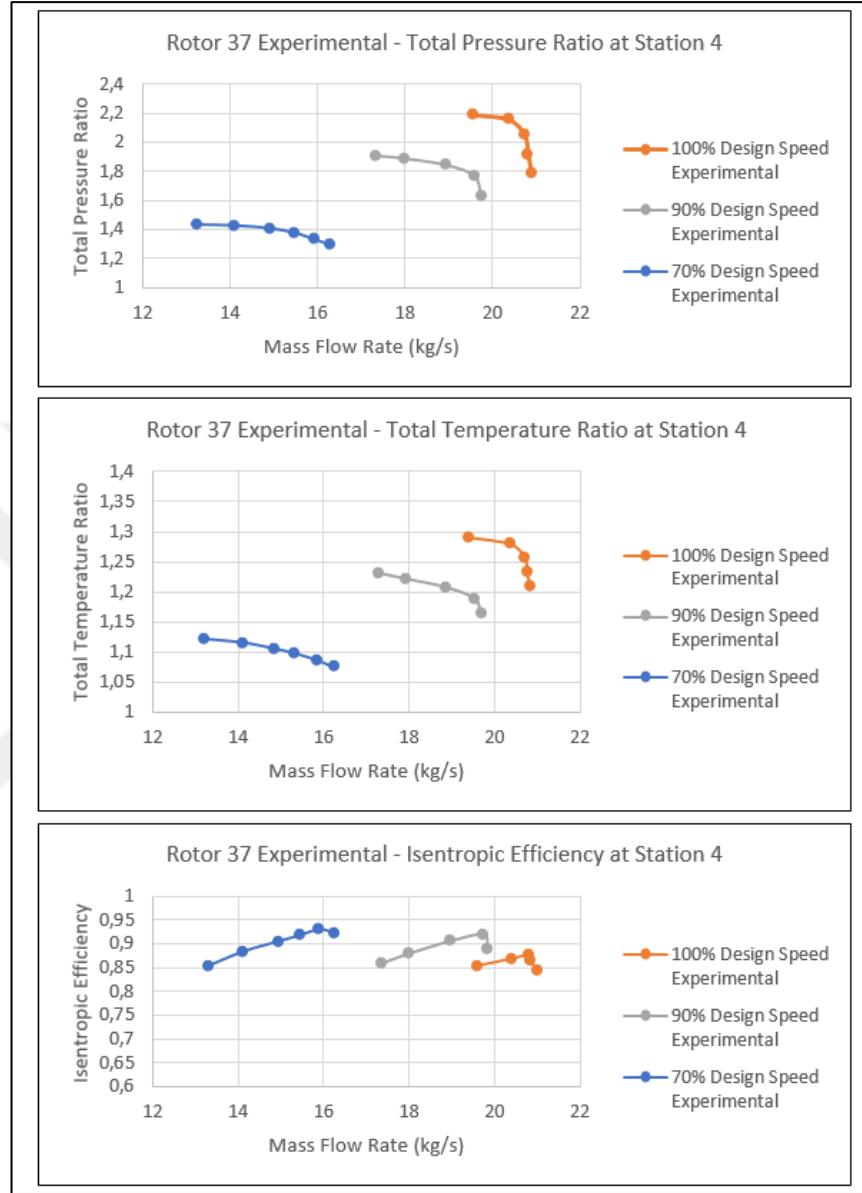


Figure 3.2: Rotor 37 experimental measurement results at station 4.

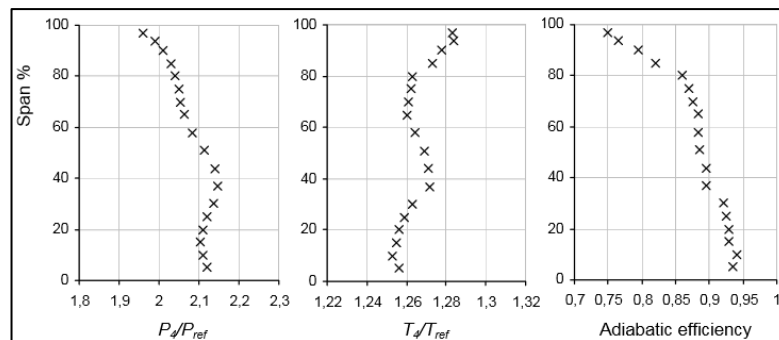


Figure 3.3: Spanwise aerodynamic measurement of rotor 37 test case [29].

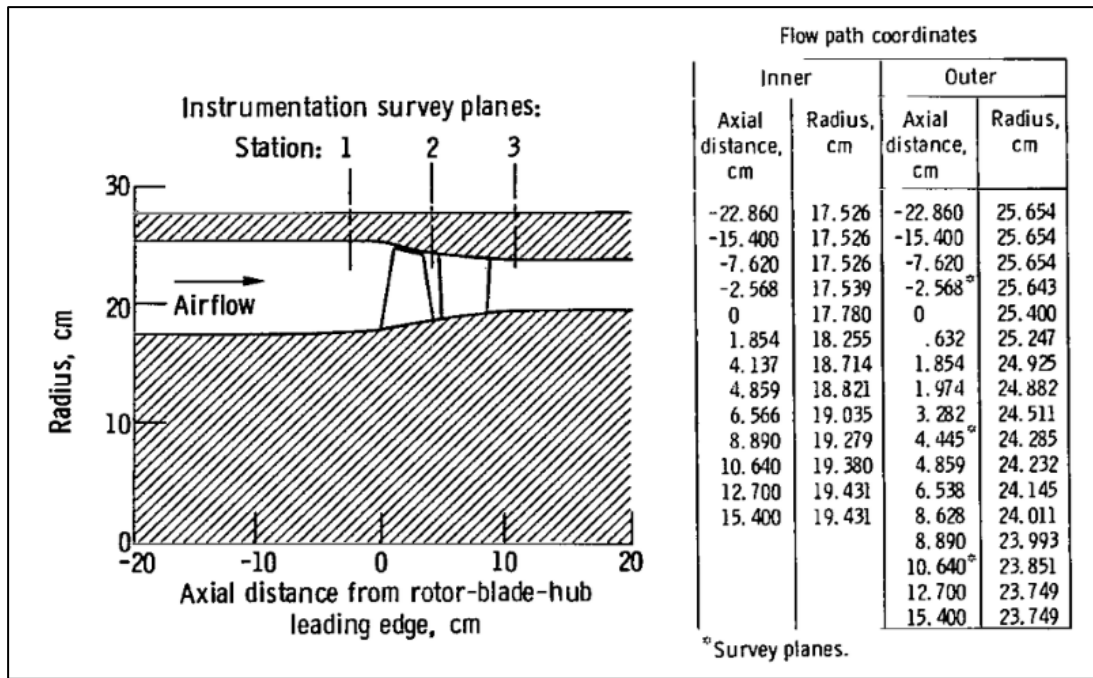


Figure 3.4: Stage 37 meridional flow path geometry [30].

Aerodynamic investigation tests were carried out at three different speed lines as done in Rotor 37 test case. Total pressure ratio, total temperature ratio and adiabatic efficiency of the Stage 37 was calculated as done for the Rotor 37 case. Average total pressure and temperature values were recorded at station 3. Overall aerodynamic performance results of the tests are tabulated in Figure 3.5.

3.2 Flow Structure Across Transonic Axial Compressor Rotors

Transonic axial compressor rotors are widespread utilized within the single stage fans of high bypass ratio jet engines, multistage fans within low bypass ratio engines and in front stages of multistage compressors to provide maximum pressure ratios per stage while keeping the efficiency at the same levels. It is possible to reduce the engine weight and size by designing axial compressor stages which gives high pressure.

Performance of the conventional subsonic compressors begins to deteriorate when the relative Mach number at the inlet exceeds 0.7, because through the blade passage relative Mach number exceeds one and flow become supersonic. This causes extra losses due to the shock waves and thickened boundary layer. Since the supersonic flows are more sensitive to any upstream changes the operation range of the compressor decreases. However, high Mach number compressor stages brings two main advantages. First, high relative Mach number permits high mass flow rate, thus

enables engineers to design more compact machine. Secondly, high relative Mach number contributes greater work input to flow and consequently provides higher pressure ratios. Equation 3.1 shows that high stage pressure ratios can be reached by increasing the speed of the blade. In modern transonic compressors, relative Mach number can reach up to 1.7 and the single stage pressure ratios of 2.0 can be achieved.

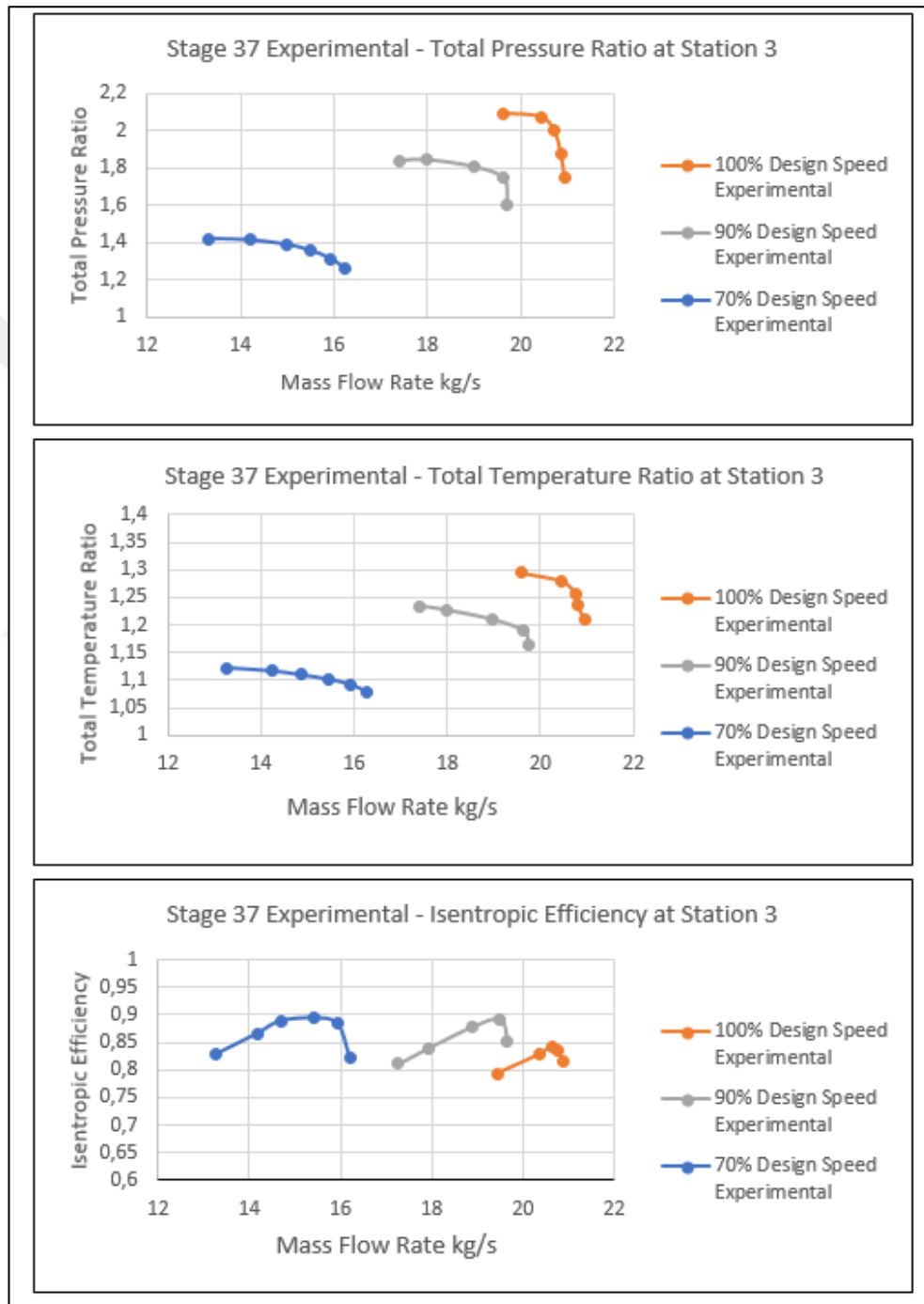


Figure 3.5: Stage 37 experimental measurement results at station 4.

$$Pressure\ Ratio = \left(1 + \eta \frac{U \Delta V_T}{c_p T_{ref}}\right)^{c_p/R} \quad (3.1)$$

To handle the complex flow introduced by high Mach number different blade profiles must be selected instead of the subsonic profile blades. Very thin profile blades that have very low percent thickness-to-chord ratio are used in order to reduce blockage effect. Moreover, the blades have very low camber with a slight turning of a few degrees, so that the Mach number does not exceed a certain value. The tip region of the blade look sharp, thin and almost like a flat plate. Figure 3.6 shows the typical design of transonic rotor blades.

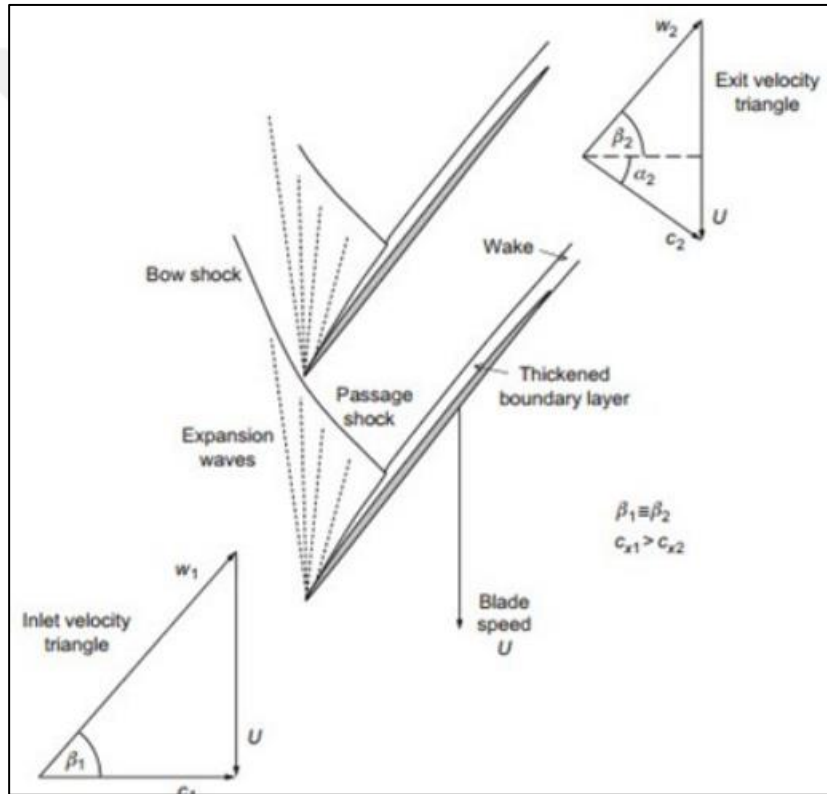


Figure 3.6: Typical operation principal of a transonic compressor rotor [31].

Although transonic rotor blades introduces many advantages for multistage compressors, their efficiency and operating range is slightly lower than the conventional subsonic compressor blades. In order to understand this disadvantage the flow field within the transonic compressor rotor should be carefully investigated. The most distinctive flow phenomena in the transonic blade operations is the presence of the shock waves. As seen in Figure 3.6 shock wave occurs in the upstream of the blade. The position of the shock wave depends on the compressor operation point. When the

compressor operates near the choking point the shock wave moves towards the blade passage, whilst near the surge the shock wave appears in the further upstream of the blades. The shock wave is close the blade leading edge for the case of best efficiency point. All these shock wave patterns are represented in Figure 3.7. The shock wave is an irreversible process, which give rise to an entropy increase in the flow field that consequently reduces the overall efficiency of the stage. Moreover, the shock wave interacts with the boundary layer, tip leakage flows, and effect the flow field negatively by disturbing the main flow. For instance, shock wave-tip clearance flow interaction might trigger stall process and shock wave-boundary layer interaction causes boundary layer separation and thickening of the boundary layer that disrupts the flow and generates entropy.

Figure 3.6 illustrates how the shock wave pattern introduces a very high work input to the flow. Across the passage shock wave, the direction of the flow does not change substantially, but the density rises drastically. Inherently the relative flow velocity downstream of the shock is much lower than upstream. Assuming the inlet and outlet blade angles are same, the turning of the flow in the absolute frame is only due to the slowing down the velocity in the relative frame by the shock wave. On the contrary, subsonic compressors turns the flow both in absolute and relative frames in order to give a work input within flow.

If the risks of separations on boundary layer could be avoided, utilizing shock wave for pressure rise in compressors is a highly efficient way.

3.3 Numerical Modeling of Rotor 37 & Stator 37 Simulations

Numerical models of the Rotor 37 and Stage 37 transonic compressor rotor and stage were constructed in ANSYS Workbench environment. In order to predict the aerodynamic flow field of the transonic compressor commercial CFD code CFX was used. Three-dimensional Reynolds-Averaged-Navier-Stokes equations with the finite volume discretization technique were solved. Algebraic Multigrid Method which is implemented in CFX was selected to accelerate the convergence. CFX uses coupled implicit steady state solver with false transient time marching approach. Time scale behaves like timestep as in unsteady solver and carries the numerical solution to the final step. Auto time scale in CFX was selected for the convergence since it is the recommended time scale for turbomachinery simulations.

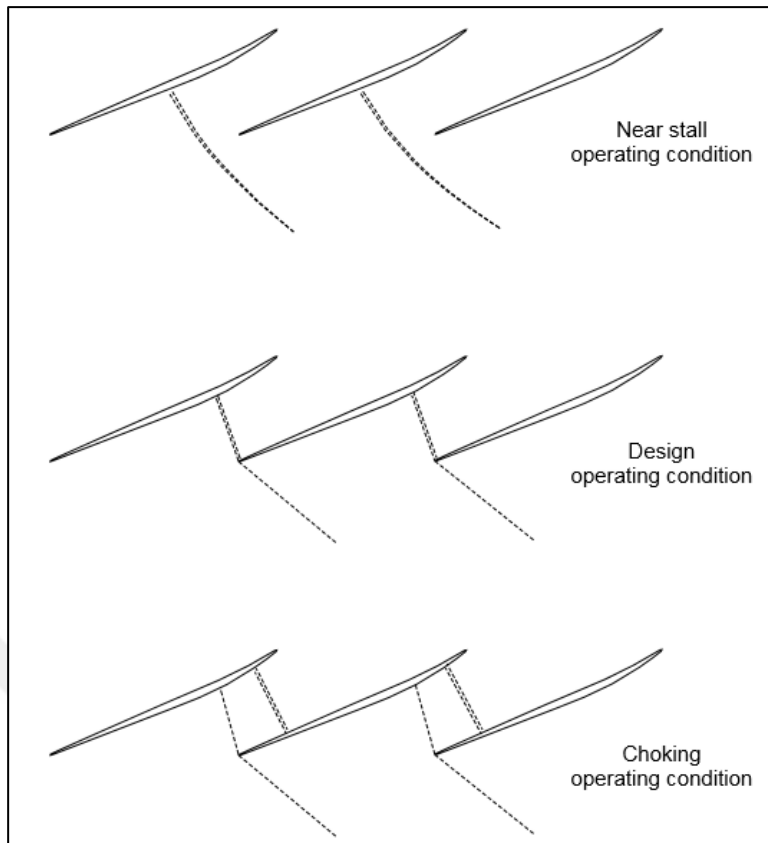


Figure 3.7: Shock waves structure for different operation conditions (Biollo, 2008).

High-resolution option is used for the spatial discretization which yields in more accurate results than the first order techniques. Coupled solvers are capable handling the high Mach number flows since they solve the continuity, momentum and energy equations at the same time. That is why selecting coupled solver would be the appropriate for solving high Mach number transonic compressor flow. Moreover, CFX solver is equipped with advanced high-speed solver setting which applies numerical treatment in shock regions of the flow field. The fluid is selected as air ideal gas in order to take into account the compressibility effects on the flow regions which contain Mach number higher than 0.3. Convergence criteria for the residuals was set as reaching below 10^{-6} . Moreover, small residual values do not satisfy the convergence of the solution alone, thus the convergence of the engineering values such as total pressure ratio, total temperature ratio, isentropic efficiency, maximum Mach number in the flow field were also monitored to assure the convergence of the solution. Stagnation inlet and mass flow outlet boundary conditions were assigned in the simulation. Absolute total pressure of 101325 Pa and total temperature of 288.15 K in absolute reference frame were set at the inlet boundary condition. The mass flow rate

at the outlet boundary was changed from surge to choking point with small increments. In this manner, the CFD solution of the compressor map can be obtained. The CFD solution was run for three different speed lines; including 17180, 15462 and 12031 rpm as in the experimental work. Moving reference frame technique was used for the rotating parts in the simulations. This technique allows steady state modeling of rotating parts by assigning a rotating frame in the selected domain. Mixing plane interface (Stage in CFX) was used for connecting rotating and stationary regions. In order to test the precision of the model the effects of the mesh density and the different turbulence models on the flow field were investigated.

3.4 Flow Domain & Meshing

Flow domain of Rotor 37 simulation consists of three regions, which are inlet domain in stationary frame, rotor domain in rotating frame and outlet domain in stationary frame. Full wheel of Rotor 37 is composed of 36 rotor blades. For the simplicity and reducing the simulation time only one blade is modeled which covers 10° in the circumferential direction. The flow domain of Stage 37 consists of three regions including: inlet domain in stationary frame, rotor domain in rotating frame and stator domain in stationary frame. Full wheel of Stator 37 consists of 46 stator vanes, but the Stage 37 model has only one stator vane covering 7.82° in azimuthal direction by means of periodic boundary conditions. The flow domain used in both analysis is pictured in Figure 3.8.

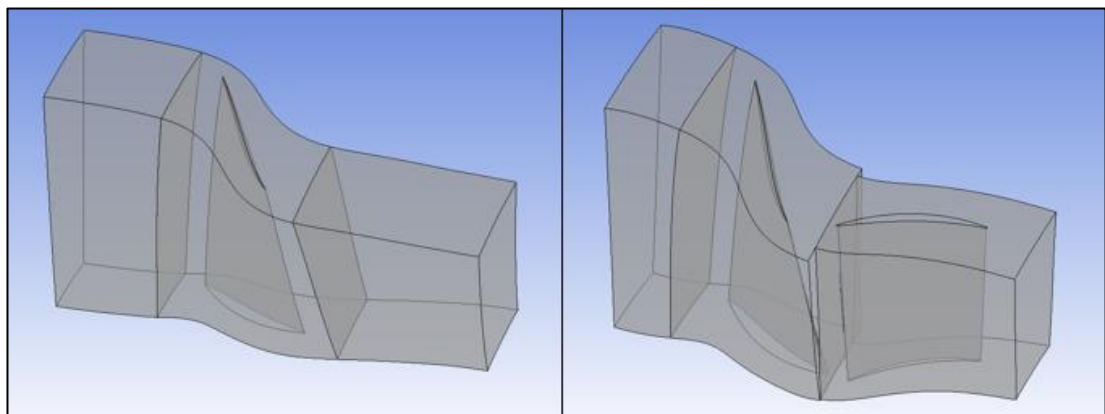


Figure 3.8: CFD model flow domain of rotor 37 (left) & stage 37 (right).

The flow domain is constrained with the boundary conditions. The side boundaries are set as periodic boundary conditions around the rotation axis. The rotor blade, stator blade, hub and shroud are assigned as wall boundary conditions. However, counter

rotating wall option is enabled for the shroud of the rotating region to satisfy the stationary state in absolute frame. Stagnation inlet and mass flow outlet boundary conditions are applied for the inlet and outlet of the model. The boundaries between the rotating and stationary domains are set as mixing plane interface as said before. Mixing plane interface (Stage in CFX) performs a circumferential averaging of fluxes across the interface surfaces when modeling change of frame or pitch. In the rotor alone case there is no pitch change, however, in the stage case the pitch ratio is 1.27 since the full rotor wheel consists of 36 blades and the stator 46 blades. The yellow boundaries represent the stage interface components between the rotating and stationary domain in the Figure 3.9.

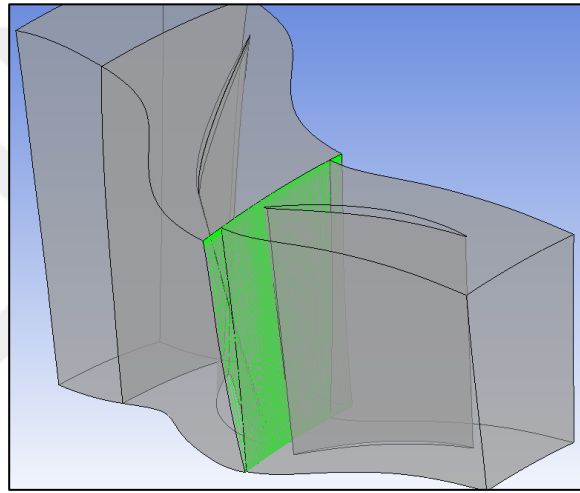


Figure 3.9: Mixing plane interface between rotating and stationary domains

Meshing process was done in ANSYS TurboGrid environment. Multi-block structured mesh with only hexahedral elements was used for each case. In order to resolve the complex phenomena such as shock waves, shock-boundary layer interactions, boundary layer separations, expansion waves the mesh structure was carefully constructed. On the rotor blade, rotor tip, hub and shroud the mesh structure was intentionally refined as seen in Figure 3.10 in order to keep the y^+ value on the boundary below one. This provides a good resolution of the gradients that take place on these boundaries. Moreover, three different grid structures were constructed for mesh dependency analysis on Rotor 37 simulation. The mesh structure from the coarsest to the finest grid consists of 450,000, 750,000 and 1,100,000 elements respectively. Experimental results of spanwise total pressure and total temperature profiles measured at station 4 were taken in consideration to evaluate the sufficiency of the mesh refinement. K- ω SST Turbulence model was selected for mesh

dependency analysis. CFD results and experimental results obtained at 17188 rpm and 98% of choking mass flow rate corresponding to 20.51 kg/s were compared to determine the optimum mesh structure. Comparison data is tabulated Figure 3.11 that shows the superimposed non-dimensional total pressure and total temperature profiles. Non-dimensionalized profiles were obtained by dividing the calculated and measured total pressure and total temperature by reference pressure and temperature respectively. Yellow, blue and red lines represent the CFD results of grid 1, grid 2 and grid 3 respectively, while the black line is experimental measurement. As seen all three CFD results almost captured the experimental data. However, slight discrepancy can be observed over the region of 70% span for both pressure and temperature profiles. This might be stemmed from the complex flow structure adjacent region to the rotor tip. Du et al. [32] numerically investigated the tip region flow of the transonic compressor and observed complicated flow structure, which includes tip clearance vortex, the shock and the interaction between them. Overall, each of the three different grid yielded in almost same result, even though the second and third one were slightly closer to the experimental measurement. That is why grid 2 is selected and the CFD compressor map analysis of Rotor 37 was conducted with grid 2. In addition, for the CFD analysis of Stage 37, stator 37 geometry was meshed with approximately 500,000 elements and combined with the grid 2 of Rotor 37. Consequently steady state Stage 37 analysis was conducted with approximately 1,400,000 elements.

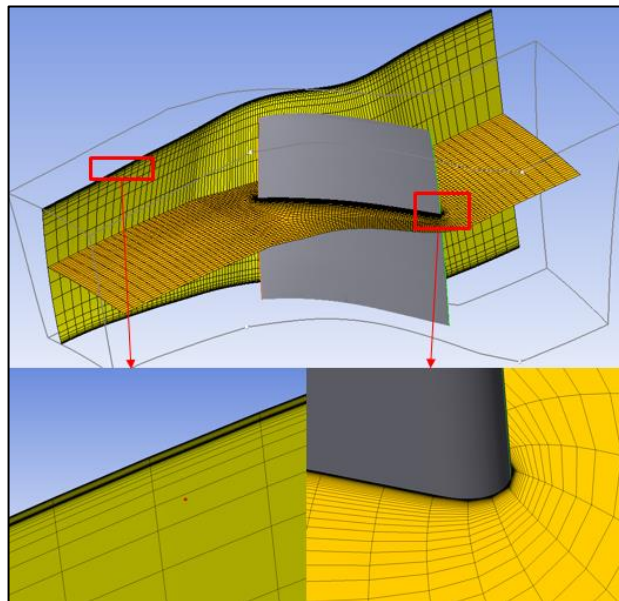


Figure 3.10: Rotor 37 CFD simulation mesh structure.

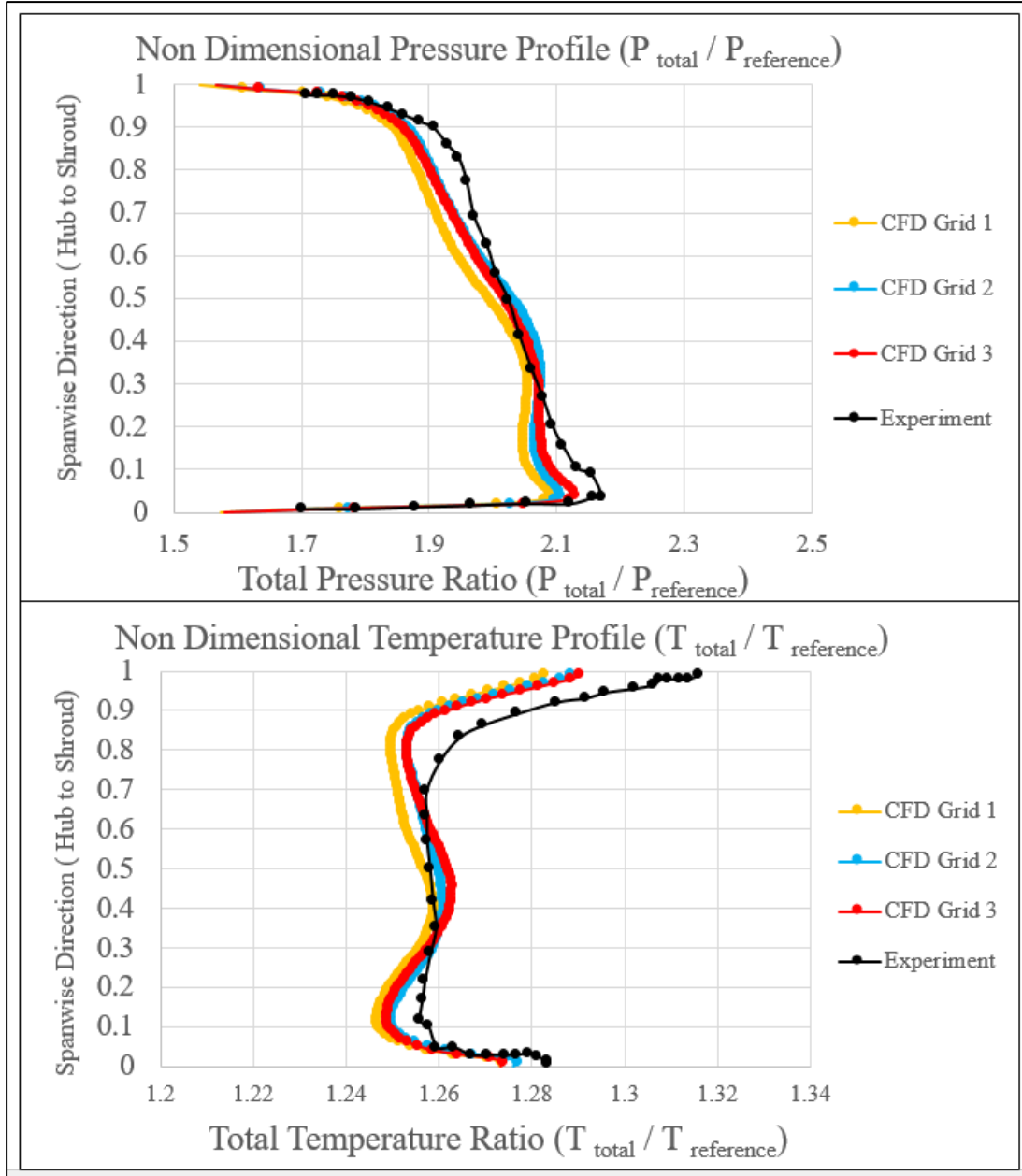


Figure 3.11: Non-dimensionalized total pressure (above) & total temperature CFD results for three different grid structure.

3.5 Turbulence Model

Turbulence closure is achieved by selecting a turbulence model for solving RANS equations. In CFD simulations $k-\epsilon$ and Shear Stress Transport (SST) $k-\omega$ model are widely used turbulence models. Standard $k-\epsilon$ model is the robust industry standard model and insensitive to inflow conditions, it may fail for the cases including separations and swirls and it requires near wall treatment. SST $k-\omega$ model is formulated by blending standard $k-\omega$ near the wall with $k-\epsilon$ in bulk flow. It is widely used in aerospace, Formula 1 and turbomachinery simulations. It performs well for separated,

wall bounded flows and adverse pressure gradients. It does not require wall treatment, the grid should be carefully constructed near the wall since the solution is sensitive to mesh refinement [33].

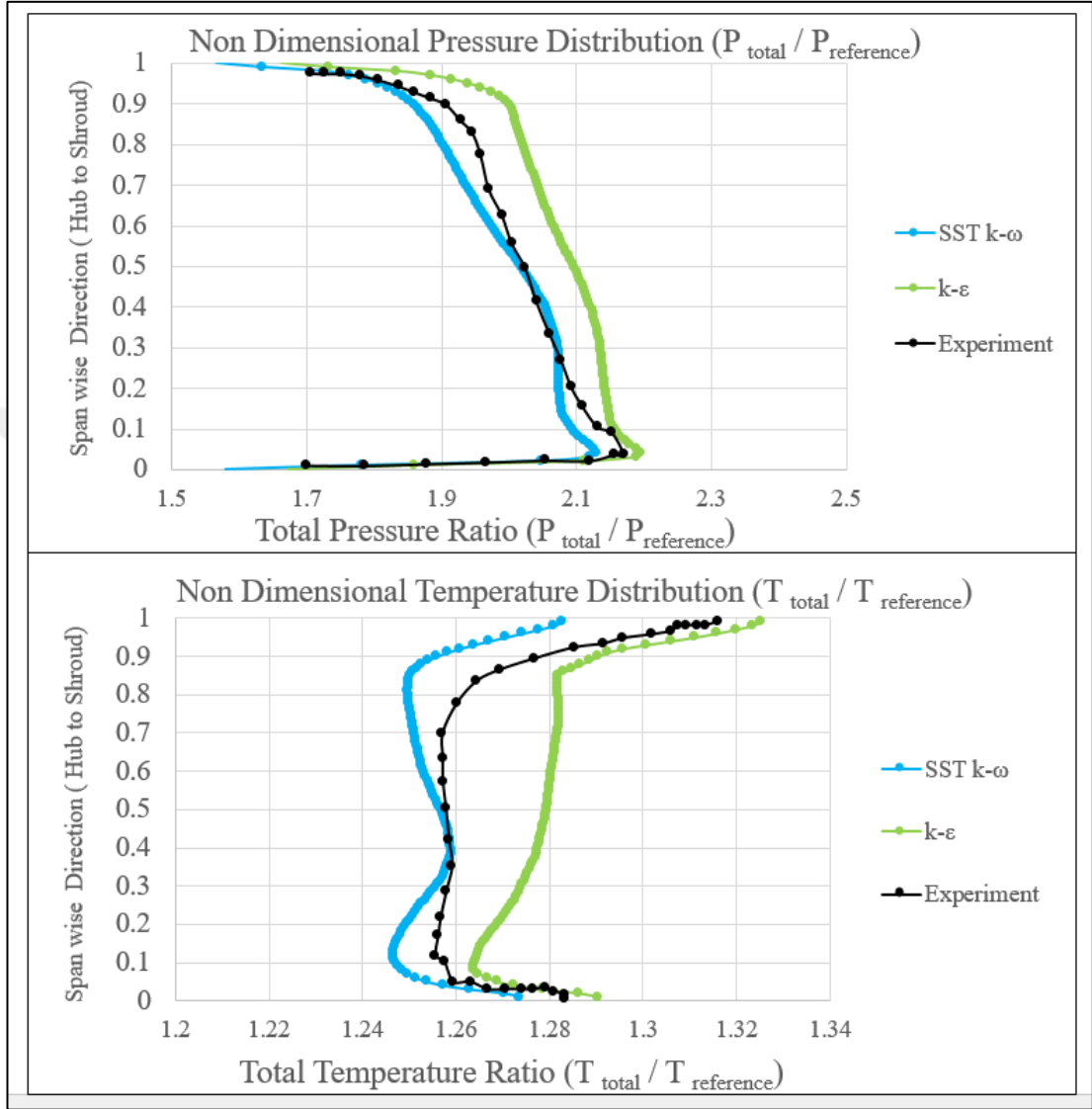


Figure 3.12: Non-dimensionalized total pressure (above) & total temperature CFD results for two different turbulence models.

Turbulence model comparison was carried out by applying same procedure as in mesh dependency analysis. Outlet non-dimensional total pressure and total temperature profiles obtained at 17188 rpm for 98% percent of choked mass flow with k- ϵ and SST k- ω were compared with the experimental results as seen in Figure 3.12. K- ϵ turbulence model over predicted the total pressure and total temperature profiles from 20%span to 80% span, whilst SST k- ω shows a better trend than k- ϵ in this region both for pressure and temperature profiles. However as seen in temperature plot,

temperature increase in the rotor tip region due to high entropy generation was better captured by k- ϵ . Overall, SST k- ω was chosen for the CFD compressor map of Rotor 37 and Stage 37 analysis.

3.6 CFD Calculated Compressor Map of Rotor 37 & Stage 37

After optimizing the mesh grid and selecting appropriate turbulence model, numerical model validation was done comparing calculated and experimental compressor map of Rotor 37 & Stage 37. Experimental compressor maps of Rotor 37 and Stage 37 were already displayed in the previous chapters. CFD compressor maps were obtained in ANYSY CFX in terms of total pressure ratio, total temperature ratio and isentropic efficiency. Comparison of the experimental results and CFD results of Rotor 37 case is shown in Figure 3.13, Figure 3.14 and Figure 3.15. As seen, the operating range becomes narrower as the rotating speed increases. CFD results and the experimental measurements perfectly match for the case of 70% of design speed. However, as the rotating speed increases, predicting the surge margin of the compressor map becomes difficult. First point of the experimental measurement at 90% of design speed could not be obtained by the CFD because of the numerical instabilities arising in simulation. Moreover, the largest discrepancy in terms of total pressure between the experimental data and the CFD simulation was seen in the surge margin of the 100% design speed. Measured total pressure ratio and CFD calculated total pressure ratio are 2.19 and 2.02 respectively that yields in a 7.7% discrepancy between the results. This might be stemmed from the complex flow field which becomes more difficult to be captured by CFD simulation as the tip speed increases. Moreover, surge margin of the compressor shows highly transient flow structures. Steady state simulations are not capable of capturing this transient structures and this may result in underestimation or overestimation of the results. Additionally, in the surge margin the simulation is sensitive to the incoming flow properties. Average total pressure boundary condition applied on inlet section might have not accurately represent the total pressure profile in the experiments. Nevertheless, 7.7% discrepancy is in acceptable range for a simulation result. In contrast to surge margin, CFD could capture the choke region perfectly. Choked region simulation convergence trend was easier compared to surge simulations, since the choked flow shows steady flow patterns. Khalil et al. [34] also

calculated lower total pressure ratio with the CFD solution in surge margin, however the calculation matches with the experimental data in choked region.

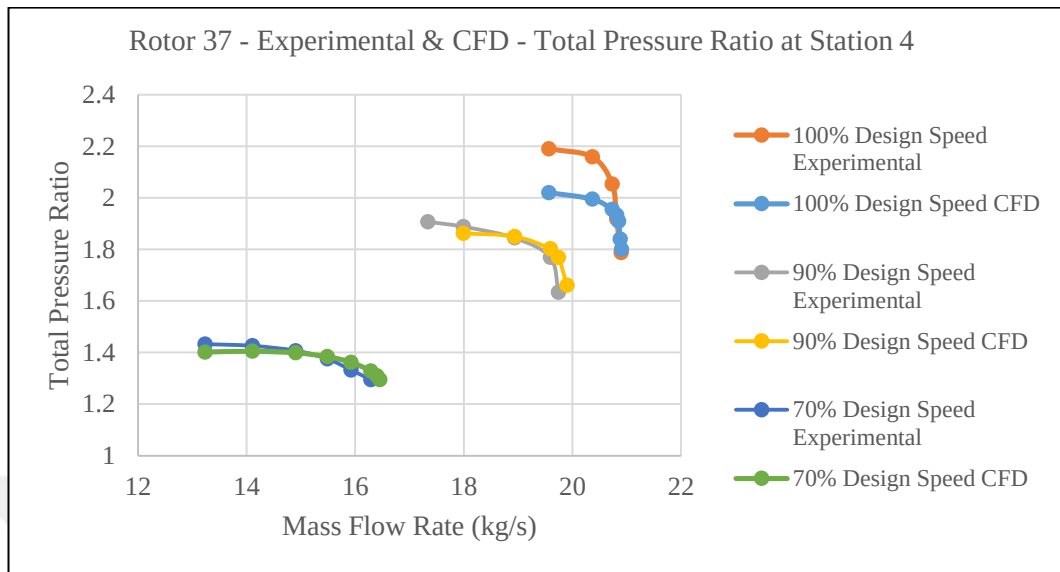


Figure 3.13: Rotor 37 experimental & CFD total pressure ratio comparison.

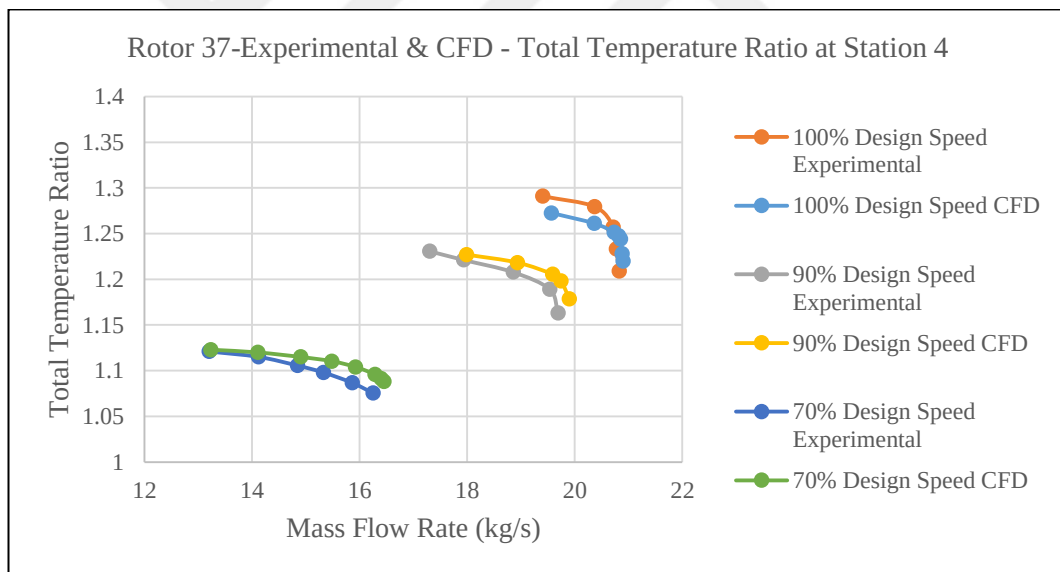


Figure 3.14: Rotor 37 experimental & CFD total temperature ratio comparison.

CFD calculation of total temperature ratio for three different speed line matches with the measurements better than the total pressure ratio results. A CFD overestimation of the 70% and 90% design speed total temperature results is observable. However, CFD underestimated the total temperature in the surge region of the 100% design speed line. In terms of isentropic efficiency lines, largest discrepancy is observed as 3% for 100% design speed. This stems from the CFD underestimation of the total pressure and total

temperature ratios. Only the 70% of design speed efficiency line is over the measured line because calculated total pressure and total temperature ratio were larger than the measurements.

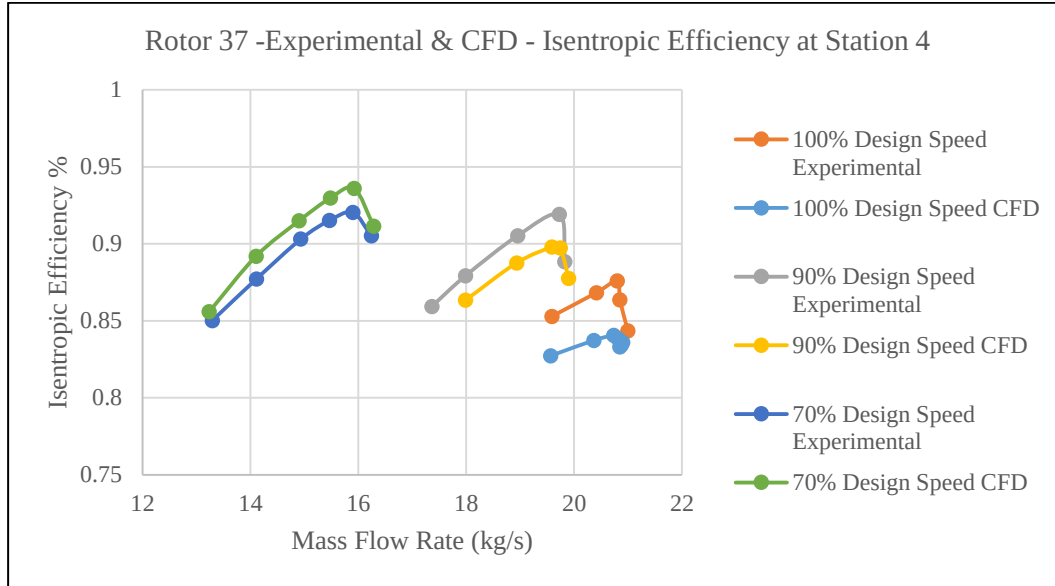


Figure 3.15: Rotor 37 experimental & CFD isentropic efficiency comparison.

CFD and experimental comparison of the total pressure ratio, total temperature ratio and the isentropic efficiency lines of Stage 37 analysis is shown in Figure 3.16, Figure 3.17 and Figure 3.18. It is observable that the total pressure ratio performance decreased with the addition of the downstream stator vane. However, the total temperature ratio lines stayed at the same level. Numerical results show the same trend as obtained in the Rotor 37 simulations. The surge line of the 100% design speed was underestimated by CFD for this case too. Numerical results of total pressure ratio and total temperature ratio at 90% speed line case are slightly higher than the measurements.

Overall, steady state CFD results shows a good agreement with the experimental results. This validates the basic steady state model of Rotor 37 and Stage 37 for the further more complicated transient analyses.

3.7 Flow Structure Investigation of Rotor 37 & Stage 37 Simulations

The most distinct flow feature in transonic compressors is shock waves arising in the upstream of the rotor blade. Main purpose of the simulation to capture this shock wave, because its interaction with the other flow features constitutes the main flow structures

in the machine. Apart from the shock waves, developing boundary layer on rotor and stator, flow separations, shock-boundary layer interactions, entropy generations, expansion waves, turbulent flow, pressure and temperature distributions were investigated. CFD analysis results of the Rotor 37 and Stage 37 simulations at design mass flow rate operation are demonstrated in next pages.

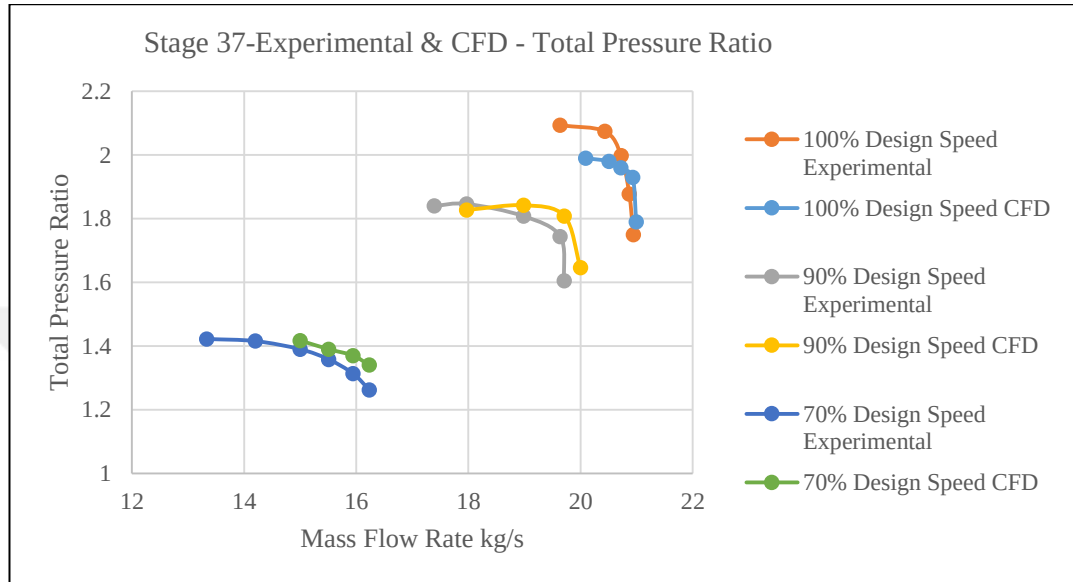


Figure 3.16: Stage 37 experimental & CFD total pressure ratio comparison

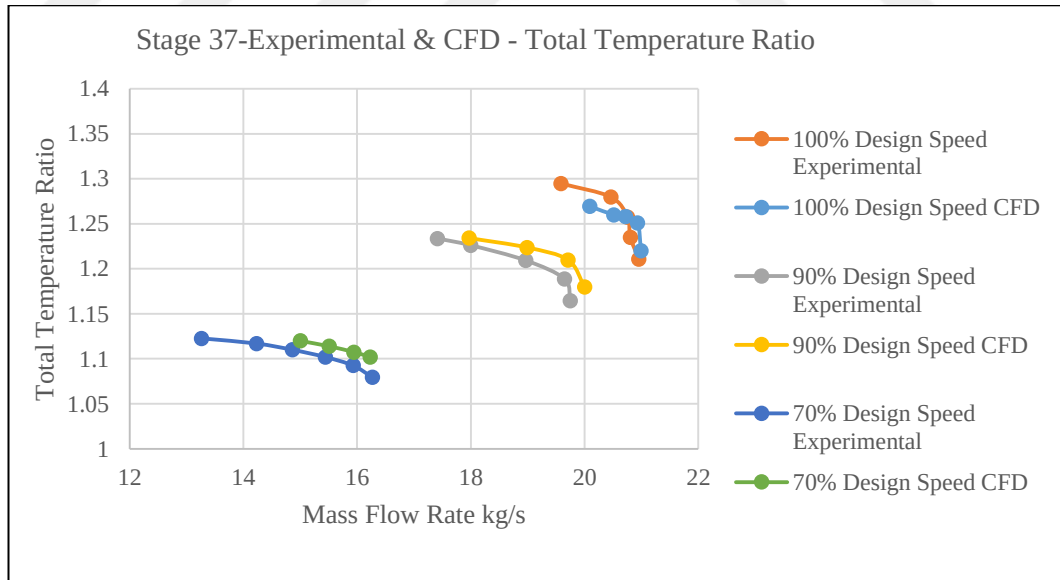


Figure 3.17: Stage 37 experimental & CFD total temperature ratio comparison

Local Mach number distribution at different spans of the rotor blade is shown in Figure 3.19. As it seen, the relative Mach number increases with the increasing span. For each case, bow shock appears in the upstream of the blade and it becomes stronger in spanwise direction.

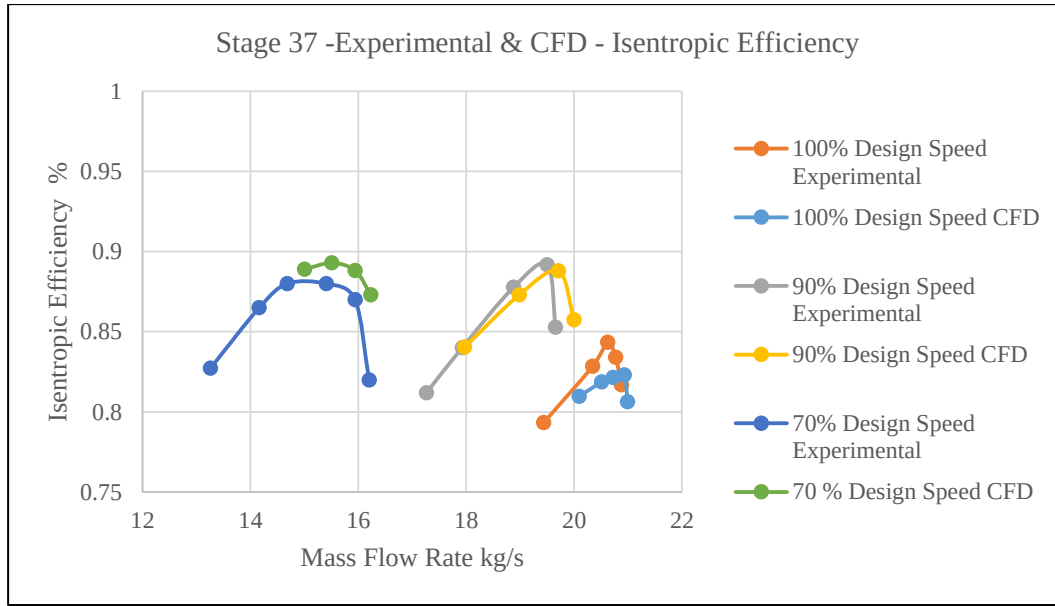


Figure 3.18: Stage 37 experimental & CFD isentropic efficiency comparison.

As a bow shock characteristics, the bow shock comes closer to the rotor blade as the tip Mach number of the blade increases. As seen, the bow shock interacts with the boundary layer of the adjacent rotor blade. The weakest interaction is at the 10% span, but for the higher spans, the interaction becomes stronger, thickens the boundary layer and eventually triggers boundary layer separation. The shock wave reflects from the interaction point and increases the pressure on the pressure side of the adjacent rotor blade. These are shown by the yellow contours on the pressure side of the blade, which has an upstream bow shock. The flow accelerates by the expansion waves on the suction side of the leading edge and reaches the maximum Mach number on the blade before the passage shock.

Figure 3.20 theoretically explains the shock-boundary layer interaction mechanism and the obtained CFD results at the mid-span of the blade. According to theory, incident shock wave disrupts laminar boundary layer and a separation bubble arises at the interaction point. Afterwards, as regards to the strength of the interaction the fluid reattaches on the wall and thickens the boundary layer or stays separated. As seen in the Figure 3.20, on the suction side of the compressor rotor blade shock-boundary layer interaction, which is visualized with velocity vectors and streamlines, takes place. The shock wave is seen as sudden reduction in the velocity in free stream and the interaction is seen as the vortices and bubble near the blade. Consequently, CFD predicted the interaction very well and captured the separation bubble.

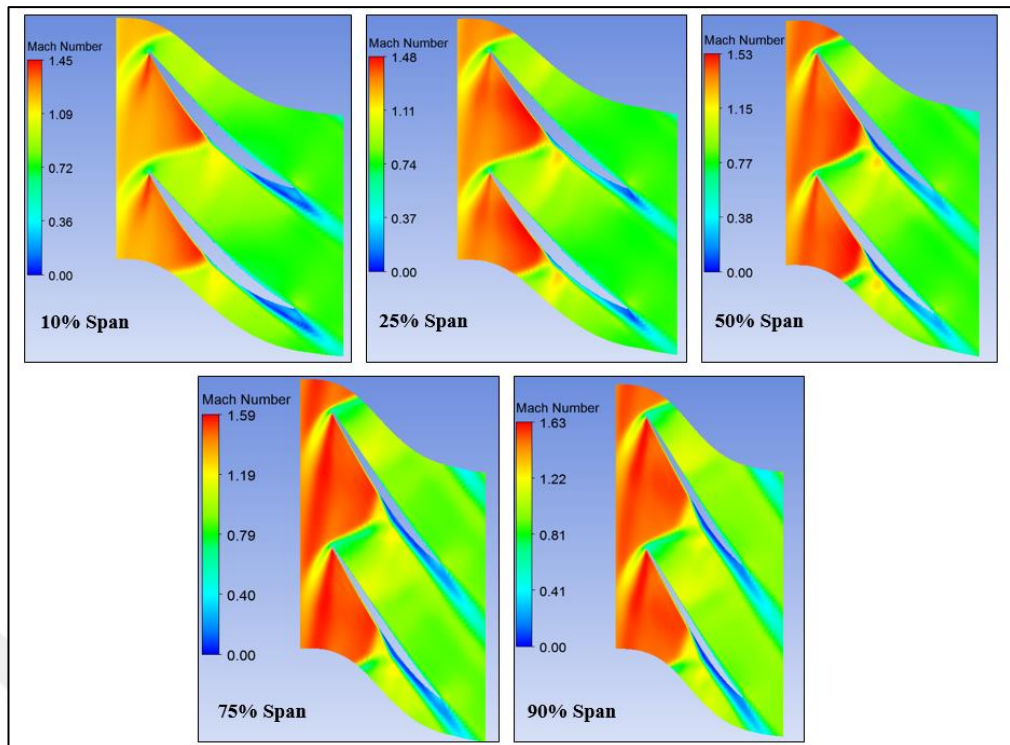


Figure 3.19: Local mach number of distribution at different span

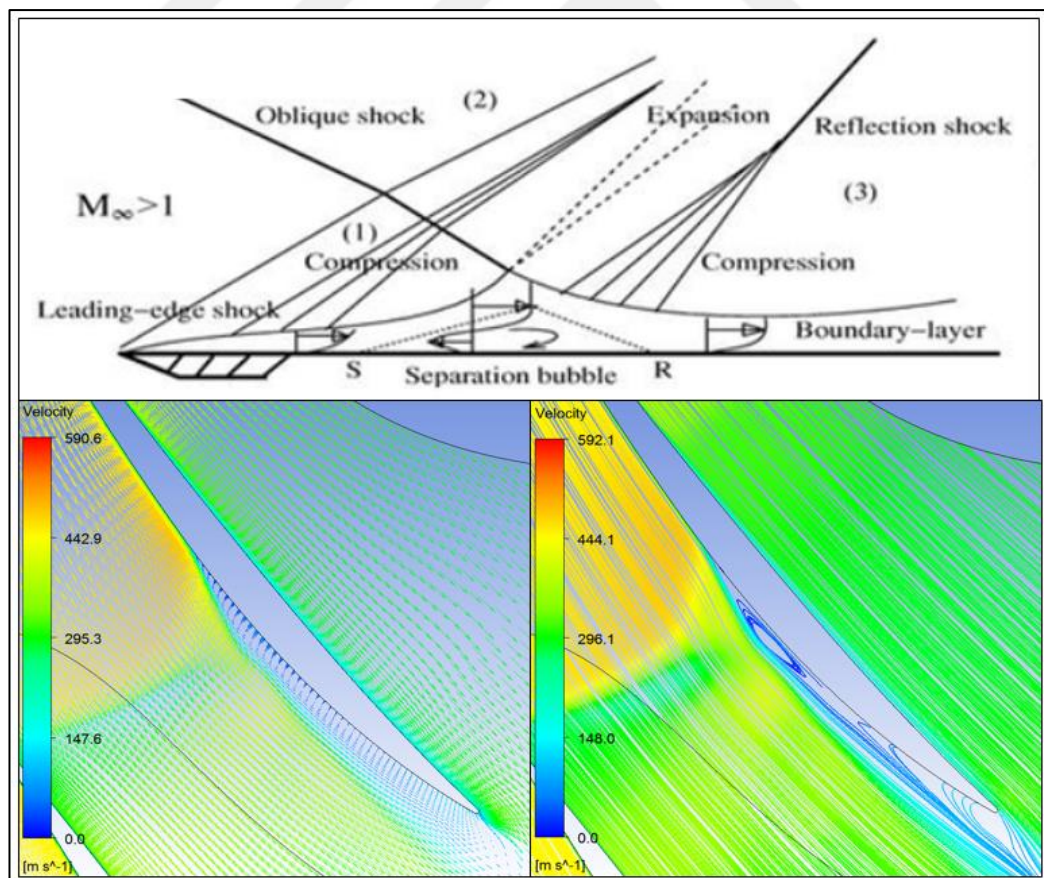


Figure 3.20: Shock-boundary layer interaction.

The shock wave boundary layer interaction is a significant factor for entropy generation in the rotor wake. Thickened turbulent boundary layer increases the dissipation on the boundary layer and the temperature reaches its maximum value in the flow. It also disrupts the smoothness of the streamlines near to blade and increases the drag. As seen in Figure 3.21, incoming streamlines before the shock wave-boundary layer interaction are smooth, whilst with the interaction the streamlines becomes irregular due to the separation. Figure 3.22 shows the static temperature and eddy viscosity distribution within the flow domain. As explained, the maximum temperature is observed in the stagnation point on the leading edge, boundary layer and rotor wake after the shock wave-boundary layer interaction. In the boundary layer region and rotor wake, turbulent eddies introduces dissipation in the flow, produces entropy and consequently the temperature rises.

Figure 3.23 (a) shows the static pressure distribution at the mid-span. Before the passage shock wave between the two rotor blades, the pressure decreases due to the expansion waves, which take place on the suction side of the rotor blade. In this region, the velocity reaches its maximum due to the expansion until it encounters the shock wave. As seen, across the shock wave static pressure increases sharply. After the shock wave, the pressure keeps increasing through the blade row due to the diffuser shaped cross section between the blades. Pressure increase via a shock wave and the divergent cross sectional area between the rotor blades are the main mechanisms to increase the pressure of the fluid in transonic axial compressors. The maximum pressure is at the stagnation point.

The picture on the right top in Figure 3.23 shows the pressure increase on the rotor blade where the shock wave interacts with the blade. As seen, pressure increase is shown as a spanwise line on rotor blade. Lastly, the plot in Figure 3.24 shows the pressure distribution at 50% span of the blade. Sudden pressure increase on both the suction side and the pressure side of the blade is observable. The one on the suction side is a bit sharper and caused by the interacting shock wave. The one on the pressure side is guessed to be formed by the reflection of the shock wave from the adjacent blade onto the pressure side of the mentioned blade. Apart from these, the pressure increases on the suction side and the pressure side of the compressor rotor in the streamwise direction via the mentioned mechanisms.

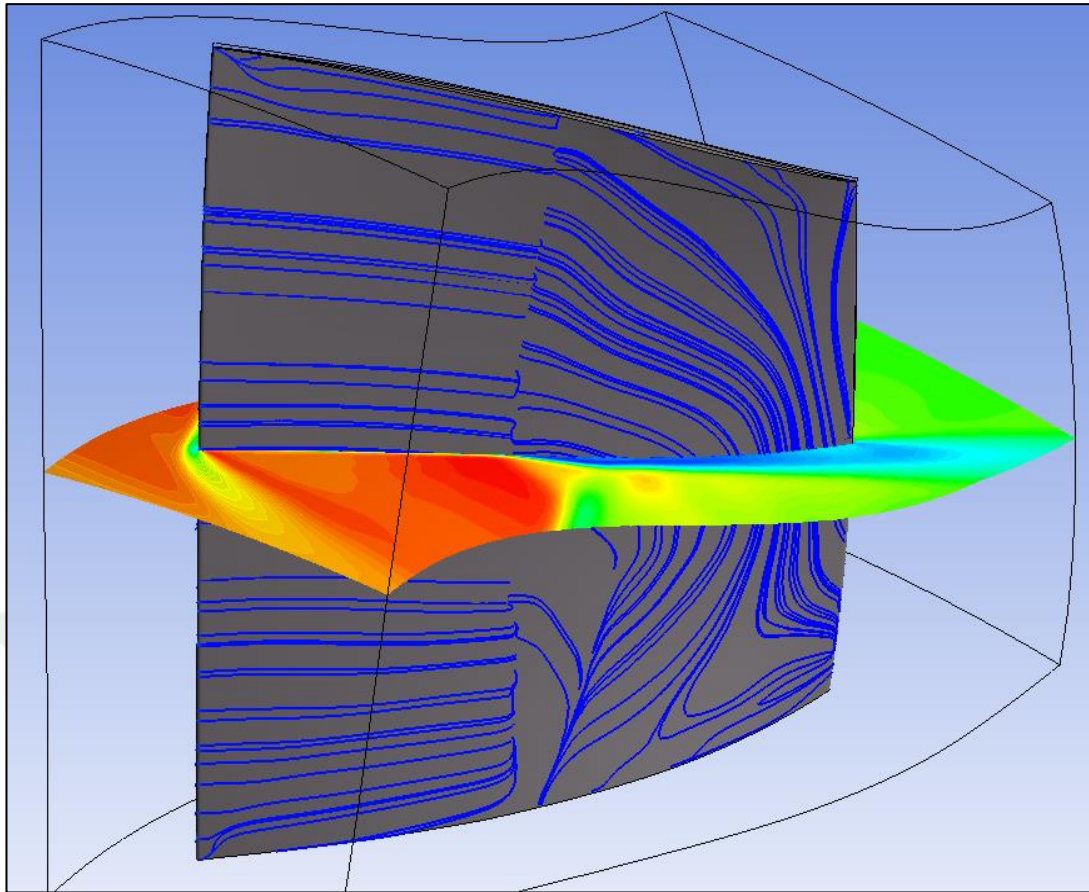


Figure 3.21: Distruption of the streamlines on the rotor surface due to shock-boundary layer interaction

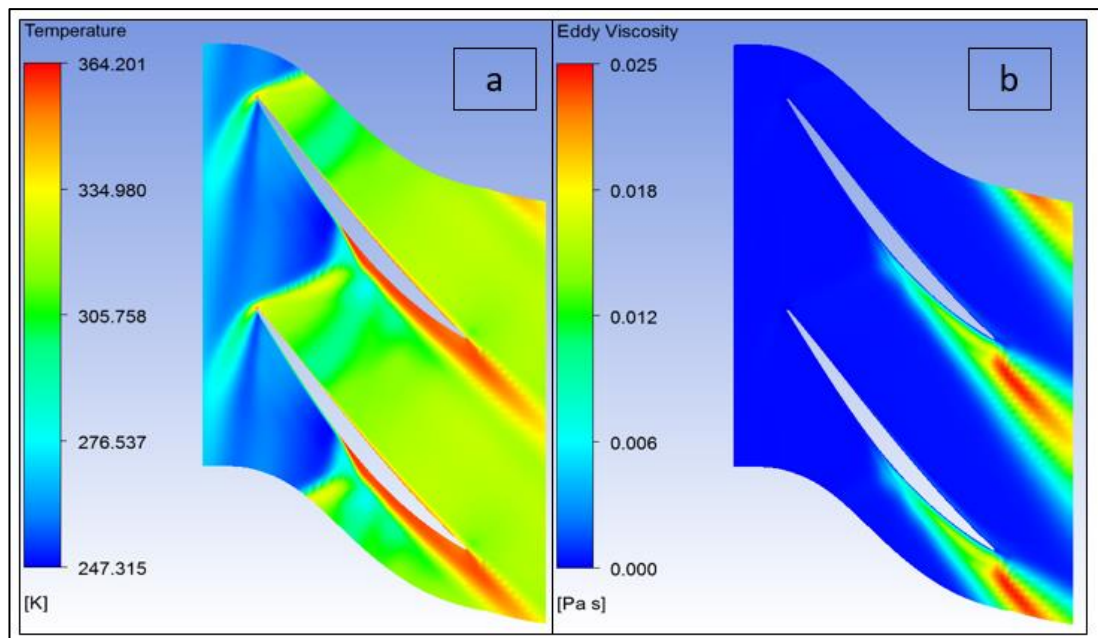


Figure 3.22: Static temperature (a) & eddy viscosity (b) distribution at rotor midspan.

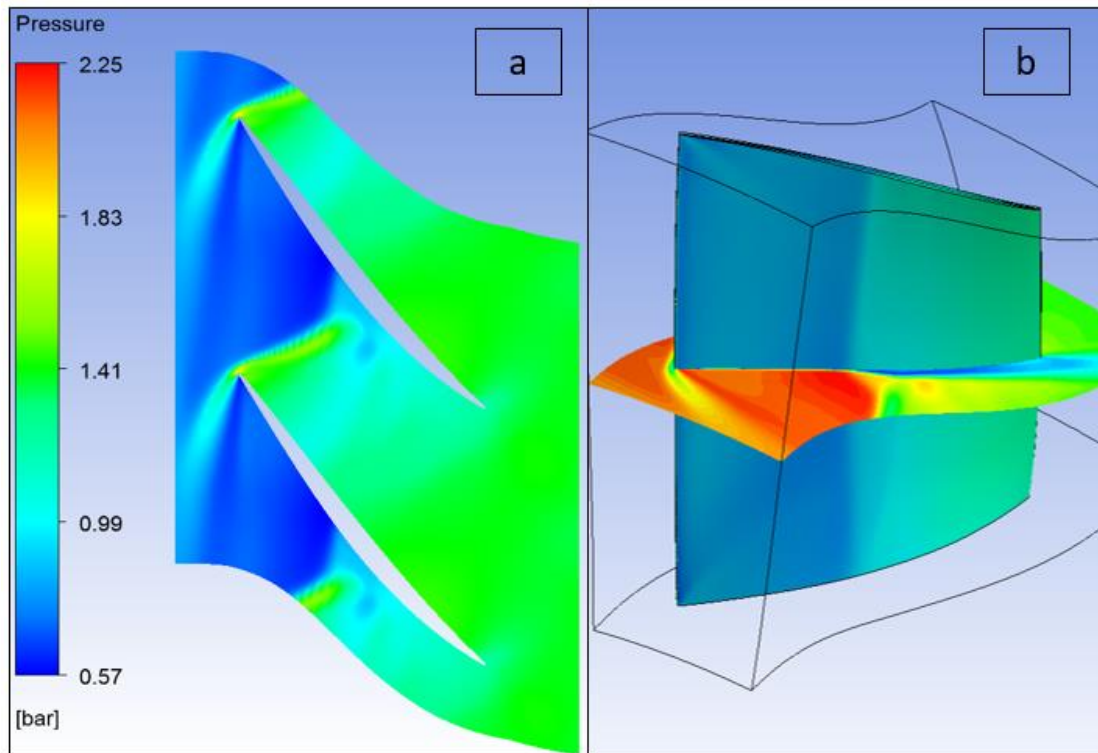


Figure 3.23: Midspan pressure distribution of rotor (a) & rotor surface pressure distribution with shock wave-boundary layer interaction (b).

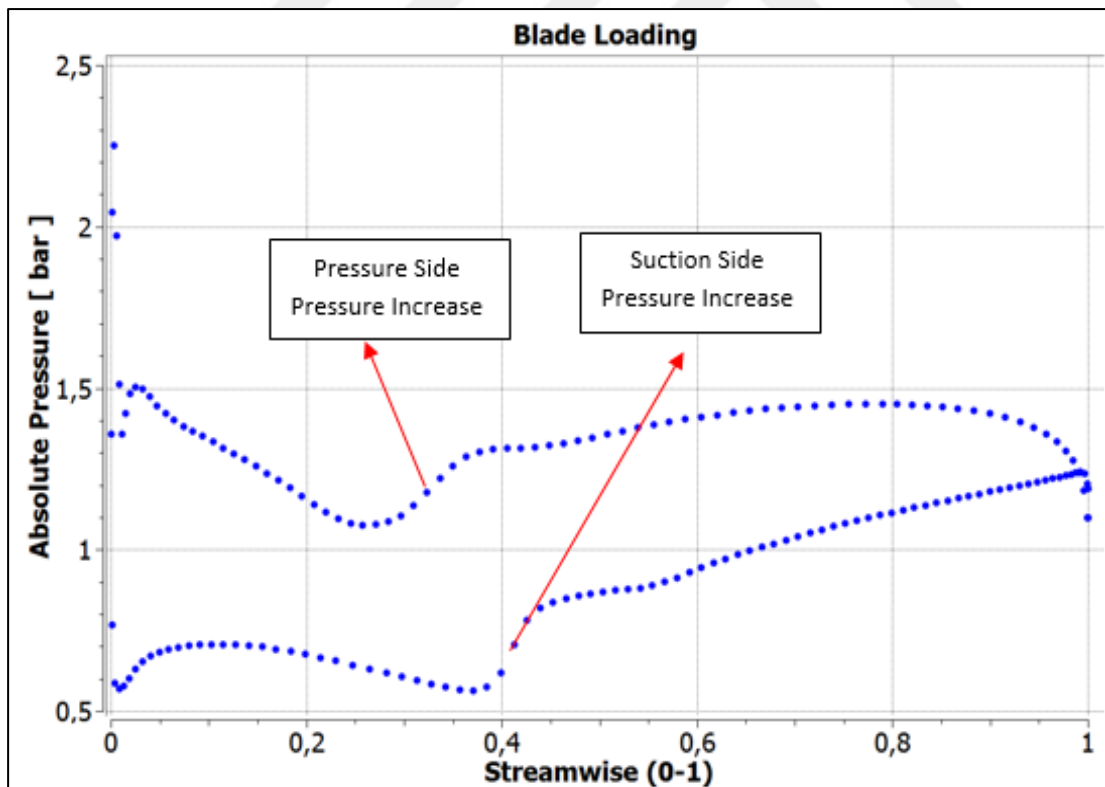


Figure 3.24: Rotor midspan pressure distribution.

3.8 Stage 37 CFD Simulation CFX & Star CCM+ Comparison

Numerical simulation of the Stage 37 was conducted both in ANSYS CFX and Star CCM+. In this manner, simulation tools were compared each other for the reliability of the numerical results. Moreover, the aeroacoustics transient analysis of Stage 37, as will be presented in next chapters, were carried out in Star CCM+ due to its sophisticated aeroacoustics tools. The numerical approach applied in Star CCM+ is same as done in CFX. Coupled steady state solver with same boundary conditions, material, turbulence model, interface model and mesh were used in Star CCM+. Mach number, pressure, temperature, density, entropy distributions within the fluid domain for both cases were compared as illustrated below. CFX solution stands on the left, while the Star CCM+ solution stands on the right. As seen in the comparison cases, numerical solutions of the CFX and Star CCM+ yield in almost identical results. Consequently, the validation study of the numerical codes of solving the aerodynamics of the Rotor 37 and Stage 37 were done by comparing the numerical solutions each other and with the experimental measurements. The results showed that, the numerical model of transonic compressor was successful for representing the aerodynamic features, and can be used for the more complicated transient aeroacoustics CFD analysis.

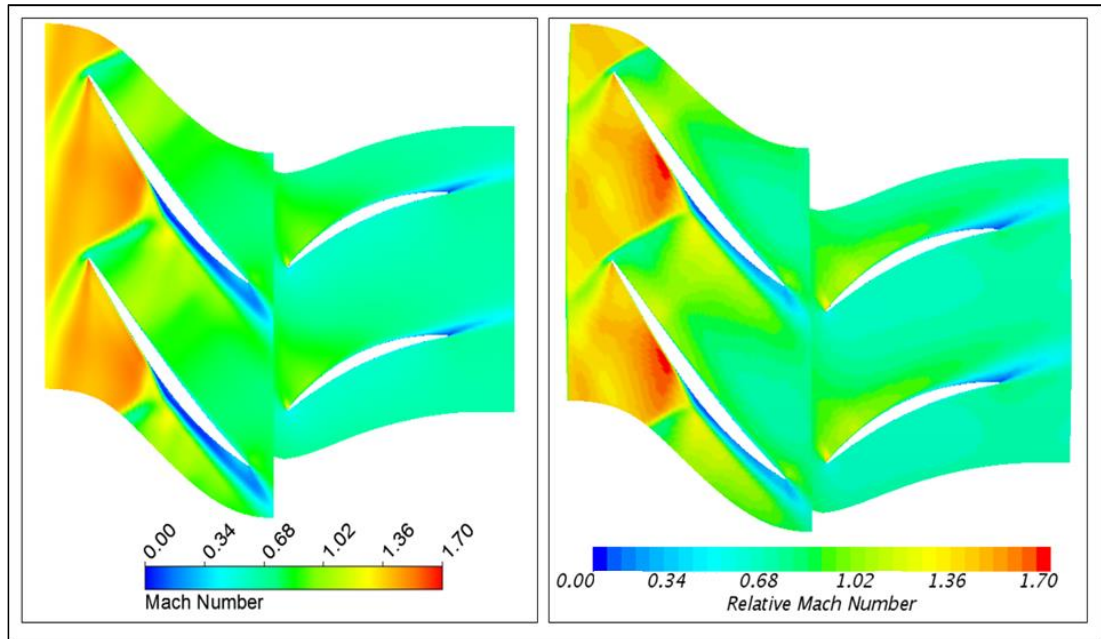


Figure 3.25 Mach number distribution at midspan (CFX & Star CCM+).

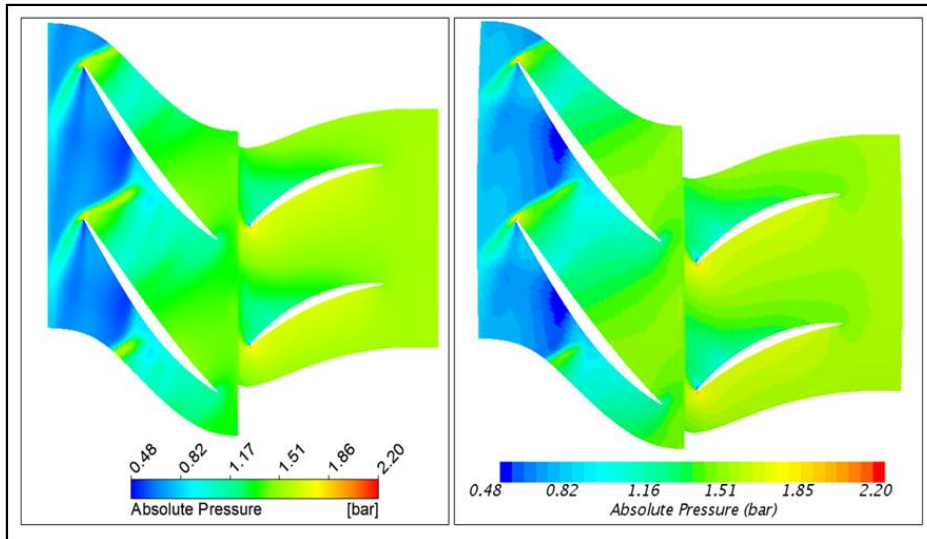


Figure 3.26: Absolute pressure distribution at midspan (CFX & Star CCM+).

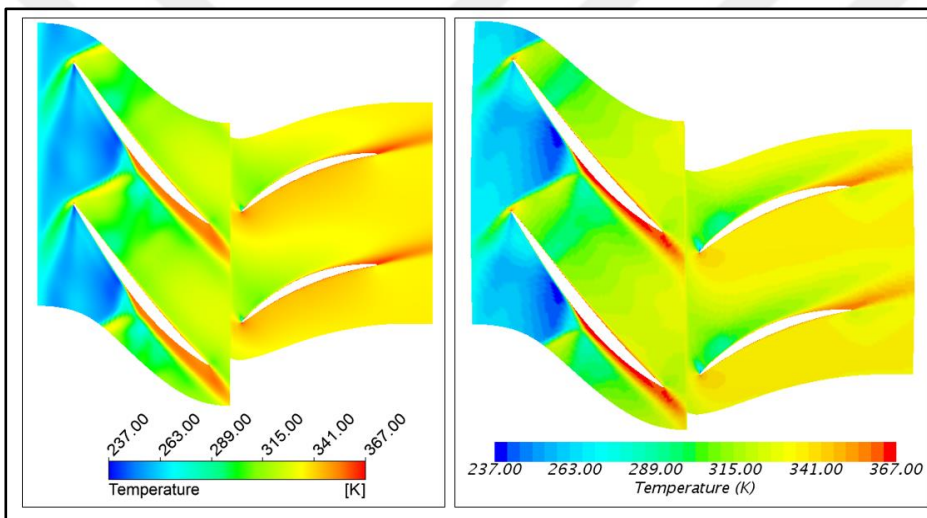


Figure 3.27: Temperature distribution at midspan (CFX & Star CCM+).

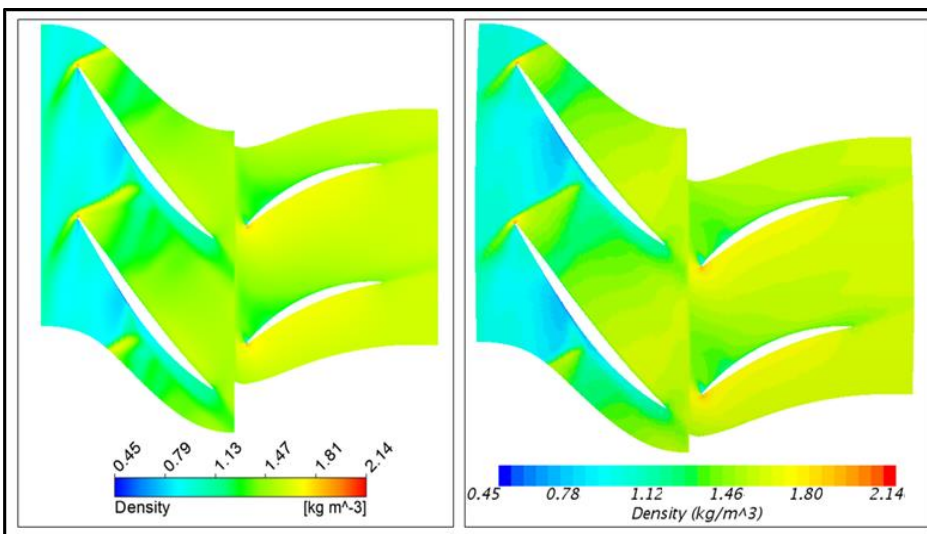


Figure 3.28: Density distribution at midspan (CFX & Star CCM+).

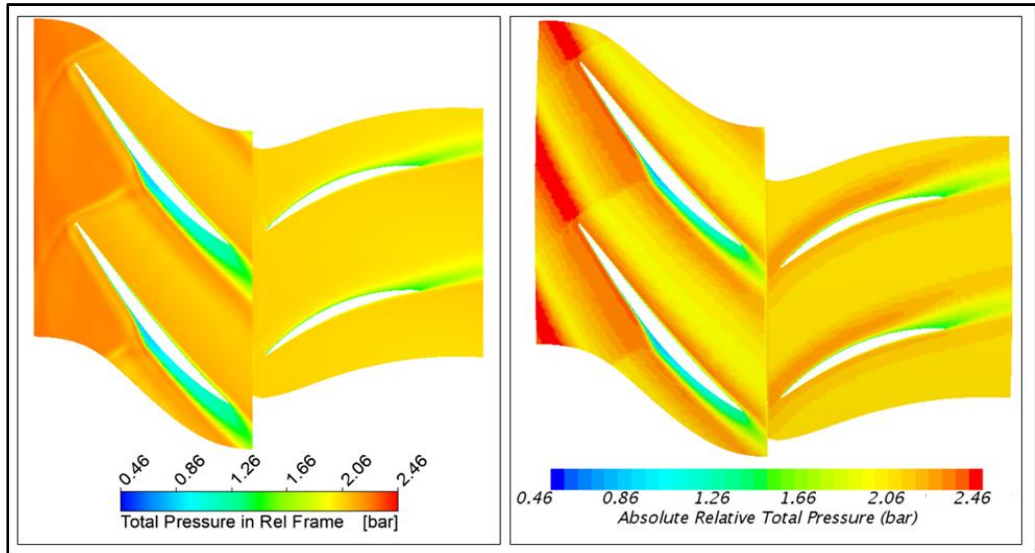


Figure 3.29: Total pressure distribution at midspan (CFX & Star CCM+).

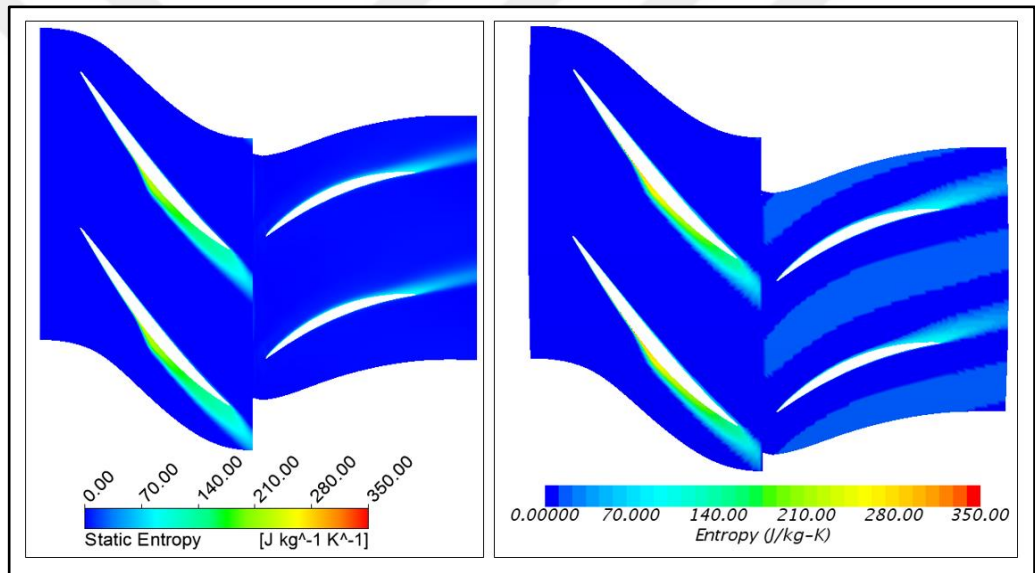


Figure 3.30: Entropy distribution at midspan (CFX & Star CCM+).

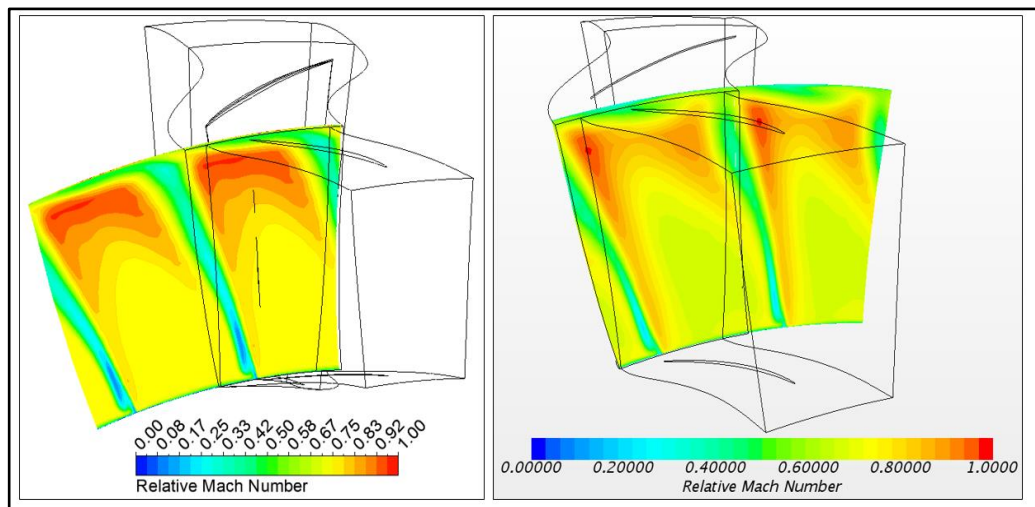


Figure 3.31: Relative mach number distribution (CFX & Star CCM+).



4. AEROACOUSTICS

Aeroacoustics is a research field of acoustics that investigates the aerodynamic generation of sound associated with either aerodynamic forces interacting with solid bodies within flow or turbulent fluid motion. In aeroacoustics, the magnitude of the relevant quantities, such as, acoustic pressure, velocity, and so on can be many orders of magnitude smaller than the respective aerodynamic flow variables. For example, for transient aerodynamics of predicting the main drag or lift coefficient of a wing the relevant pressure fluctuations are generally of an order of tens or hundreds of Pascals, whilst in aeroacoustics this can become 0.01 Pascals for the acoustic pressure fluctuations. This discipline was established by Lighthill in the earlyies of 1950 when noise generation of jet engines became a significant research topic. Lighthill rearranged the Navier-Stokes equations, which represents the flow of a compressible viscous fluid, into an inhomogeneous wave equation in order to connect the fluid mechanics to acoustics. This is called Lighthill's analogy replacing the noise generating flow field with an equivalent system of noise sources. Then using standard acoustic propagation equations far field noise is calculated on a stagnant fluid. In 1955, Curle improved the Lighthill's analogy to contain the sound generation from flow-body interaction. In 1969, Ffowcs Williams and Hawkings extended the Lighthill's analogy for the surface motion. This formulation can be used for noise prediction in rotor blades aerodynamics, such as turbomachinery applications. The main mechanisms of sound generation were introduced in previous chapters.

Inherently, there are four difficult aspects in directly solving acoustic fields using Navier-Stokes equations including:

- The field of acoustic waves is larger than the flow-field.
- Energy content of the acoustic-field can be relatively much smaller than the flow field.
- Numerical discretization becomes much sensitive.
- Placing the inlet and outlet boundaries in remote upstream and downstream locations from the noise source in order to prevent the acoustic reflections.

4.1 Computational Aeroacoustics

Any computational method related the computation of sound generation associated with the fluid flow can be defined as a method of computational aeroacoustics (CAA). These methods arise with the development of the new CFD methods and they are strictly linked to CFD. Each CAA method uses its specified technique to resolve the sources of the sound generations and/or the propagating sound waves. Three main techniques can be used to solve aeroacoustics including direct noise calculations, hybrid approaches and steady state methods. A direct noise calculation (DNC) is a simulation comprising all aeroacoustics variables are calculated within one single analysis. Hybrid approaches can be utilized to reduce computational effort by using acoustic analogy methods for calculating far field noise. Steady state methods can give only preliminary design ideas and determines the mesh refinement regions for transient analysis by processing RANS simulation. In this study, all these methods were used to investigate the aeroacoustics flow field.

4.1.1 Direct noise calculations

Unsteady flow responsible for noise generation and the propagation of the sound waves within the related entire domain are simulated using the compressible Navier-Stokes equations. Unsteady compressible solvers using LES & DES turbulence models (URANS only for tonal noise) simulate the turbulent broadband and tonal noise sources. It requires very accurate numerical discretization scheme to both capture the small turbulent eddies and prevent the damping of the high frequency-low wavelength acoustic waves. Non-reflecting boundary treatments are needed in order to prevent the artificial reflections of the acoustic waves on inlet and outlet boundaries. Since it is very computationally demanding, it can be only used for short sound propagation distances and strong fluid-acoustic two-way couplings. It is not reasonable to use direct noise calculations for far-field noise calculations since the computational work would be tremendous. Typical applications are flow-duct whistling, buffeting, trailing-edge-self-sustained oscillations.

Overall, few modeling assumptions and less work to set up are the benefits, on the other hand long computational times, high requirements on numerical accuracy, large mesh sizes, high requirement on cell quality, non-reflecting boundary treatments and post processing for acoustic analysis are the drawbacks of this technique.

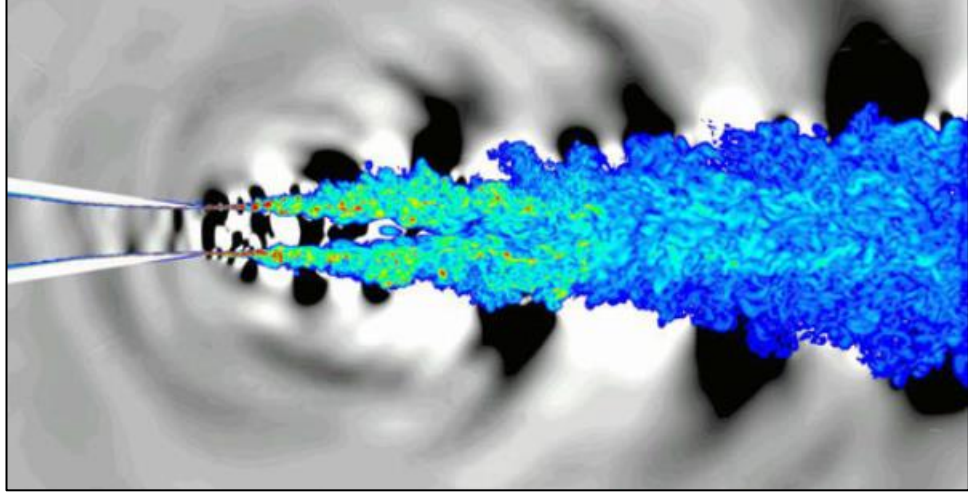


Figure 4.1: Direct noise calculation of a jet flow

4.1.2 Hybrid methods

This approach divides the simulation into several steps. In the first step, either steady state or transient CFD is conducted to capture the noise sources in the flow field. In the second step, an acoustic solver calculates the noise propagation to an observer location which is away from the source. Acoustic solvers which are based on acoustic analogy that assumes the aeroacoustic solution can be decoupled from flow field. As explained, acoustic analogy were derived by rearranging Navier-Stokes equations into acoustic wave equation. After obtaining the steady or transient flow field by CFD this analogy is imposed to propagate the noise from sources to the receiver location by solving wave equation. Since the general solution of wave propagation in free space represents the integral over all sources, these methods are called as integral methods. Lighthill's analogy, Curle's analogy, FW-H analogy, Kirchhoff integral and Boundary Element Method (BEM) are some of the integral methods.

4.1.2.1 Lighthill's acoustic analogy

As mentioned before, first acoustic analogy study was established by Lighthill (1952). He rewrote the Navier-Stokes equations, in form of wave operator representing acoustic propagation on left hand side and quadrupole source term on the right hand side, resulting in inhomogeneous wave equation as below.

$$\frac{\partial^2 \rho}{\partial t^2} - c_0^2 \nabla^2 \rho = \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j} \quad (4.1)$$

$$T_{ij} = \rho u_i u_j + \delta_{ij} [(p - p_0) - c_0^2 (\rho - \rho_0)] - \sigma_{ij} \quad (4.2)$$

The term T_{ij} is the Lighthill's stress tensor and represents the quadrupole sources in the flow. Since the analogy directly derived from the Navier-Stokes equations without any simplification, source term can represent any quadrupole type sources. The sources can be classified as turbulent or laminar noise. The left hand side which is the wave operator can be applied to either density or pressure perturbation. The sound pressure in far-field region is then calculated as the volume integral of the source domain.

4.1.2.2 Curle's acoustic analogy

Extended version of the Lighthill's analogy was proposed by Curle (1955). Addition to the fluid based quadrupole term; influence of the presence of the stationary solid boundaries is included. Sound field is composed of two sources including quadrupole sources adjacent to the solid boundaries and dipole sources generated by pressure fluctuations on solid boundaries. For the low Mach number flows, the dipole sources outweigh the quadrupole sources. Moreover, the surfaces within the fluid alters the characteristics of the sound waves by diffracting and reflecting them. As a result, for the low Mach number flows the quadrupole term can be neglected and only dipole source is taken into account to calculate the acoustic field.

4.1.2.3 Ffowcs williams-hawkings acoustic analogy (FW-H)

Ffowcs Williams and Hawkings (1969) derived a more sophisticated acoustic analogy than the Lighthill and Curle. In addition to the Curle's analogy, where only stationary solid boundaries only accounted for; FW-H analogy includes the effect of the motion of the solid boundaries in the flow. FW-H calculation represents the quadrupole sources with volume integral, dipole and monopole sources with surface integral. It is a preferred way to predict the far-field noise. FW-H analogy processes the near-field data from CFD and calculates the noise at the far-field location of the receiver. Noise prediction of turbomachinery applications such as fans, helicopter blades, compressors, turbines are the typical field of FW-H analogy. In this thesis, this analogy was used for the far-field noise calculations radiated from rotor and stator blades.

FW-H Acoustic model uses impermeable formula for surface terms, which are the monopole and dipole sources on the boundaries. Surface integral is solved on boundary of the object and the far-field noise is calculated. Fan, compressor, propeller noise are typical applications.

Volumetric quadrupole sources are evaluated with permeable formula. In this method all volumetric sources are surrounded within a control surface. Resulting far-field noise due to the volumetric term is then calculated. Calculating the far-field noise of a jet noise with high turbulent shear layer is a typical example. The FW-H equation for pressure that is propagated into medium is as shown below. The first two terms are surface integrals which are monopole and dipole sources respectively. The last term which involves the volumetric integral is the quadrupole term related to turbulence noise.

$$p'(x',t) = p'_T(x',t) + p'_L(x',t) + p'_Q(x',t) \quad (4.3)$$

$$p'_T(x',t) = \frac{1}{4\pi} \left[\left(-\frac{\partial}{\partial t} \right) \int \left(\frac{Q}{r(1-M_r)} \right) dS \right] \quad (4.4)$$

$$p'_L(x',t) = \frac{1}{4\pi} \left[\left(-\frac{\partial}{\partial x_i} \right) \int \left(\frac{L_i}{r(1-M_r)} \right) dS \right] \quad (4.5)$$

$$p'_T(x',t) = \frac{1}{4\pi} \left[\left(\frac{\partial^2}{(\partial x_i)(\partial x_j)} \right) \int \left(\frac{T_{ij}}{r(1-M_r)} \right) dV \right] \quad (4.6)$$

$$Q = \rho_0 U_i n_i \quad (4.7)$$

$$U_i = \left(1 - \frac{\rho}{\rho_0} \right) v_i + \frac{\rho u_i}{\rho_0} \quad (4.8)$$

$$L_i = P_{ij} n_j + \rho u_i (u_n - v_n) \quad (4.9)$$

$$P_{ij} = (p - p_0) \delta_{ij} - \sigma_{ij} \quad (4.10)$$

4.2 Aeroacoustics Simulation Workflow

As explained above, modeling aeracoustics requires choosing the appropriate computational aeroacoustics methods, since each method has its own calculation technique. Aeroacoustics simulations involve the near-field and far-field investigations of the sound. Direct noise calculations are preferred for near-field acoustics. LES or DES based turbulence models that requires high numerical resolution are used to calculate acoustic pressures in the flow domain. High mesh resolution and sufficiently small time step are needed to prevent the damping of the investigated acoustic waves in the flow medium. URANS models are only preferred when the flow features are certainly tonal such as blade passing frequency or coherent shedding. In this study, for the near-field noise calculation, direct noise calculation.

Since this thesis is related to tonal noise investigation, URANS simulation was conducted to capture the tonal noise sources and their propagation in the near field. Since, in practice, resolving all fluid features from source to the far-field requires too much computational effort, hybrid methods are resorted. In this study, FW-H acoustic analogy was utilized to calculate the far-field noise. First, URANS simulation had been carried out to collect the transient data, afterwards FW-H analogy processed transient pressure data and calculated the sound level at the far-field observer locations. In the far-field noise calculation the quadrupole term which is related to turbulent broadband noise was neglected, since URANS simulation is not capable of resolving turbulent eddies. Only the monopole term of blade thickness noise and the dipole term of the pressure fluctuations on the rotor and stator blades were taken into account. The transient pressure data on rotor and stator blades were processed for the far-field noise calculations.

Aeroacoustics calculation strategy used in this study is shown as below.

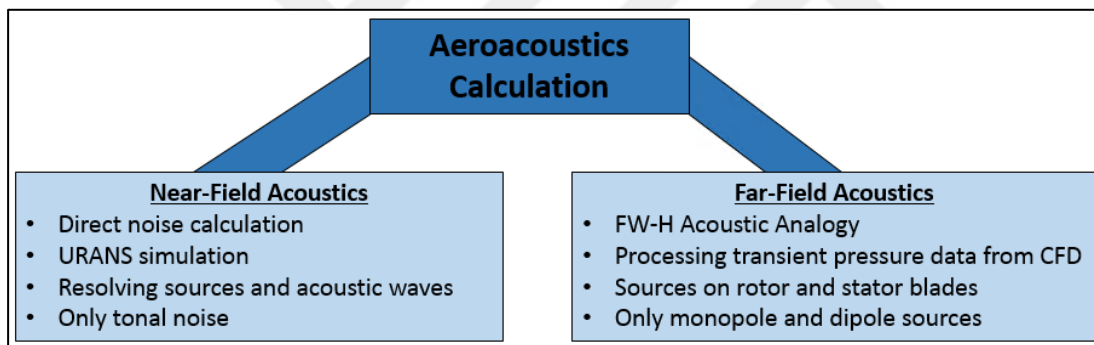


Figure 4.2: Aeroacoustic calculation strategy.

5. AEROACOUSTICS SIMULATIONS

In this study, the purpose of the aeroacoustics simulations to resolve the near-field and far-field acoustic field properties generated by the rotor-stator interaction. Owing to its wide range aeracoustic model selections, Star CCM+ was utilized for the unsteady aeroacoustic simulations. As explained in the previous chapter, near-field acoustic field was captured by the direct noise calculation which is the general URANS simulation and the far-field acoustic field was modeled with the FW-H acoustic analogy. Unsteady pressure on rotor blade and stator blade were collected during two entire wheel revolutions. These unsteady pressures were then imported to FW-H acoustic solver as an input for the FW-H equations. This input was processed to predict the far-field noise which the direct calculation simulation is not capable of solving.

The effect of axial gap between the rotor-stator on the noise generation due to the rotor-stator interaction was investigated with three different transonic axial compressor stage, each having different axial gap between rotor and stator. Flow field aerodynamics of each case, especially the stator potential field on rotor and the rotor wake-stator interaction, was investigated in order to understand how the noise sources and noise propagation are affected by the flow.

Case 1 that is the original geometry of the NASA Stage 37 has the axial gap 8 mm between rotor blades and stator vanes at 50% span. Case 2 and case 3 have the axial gap of 16 mm and 32 mm respectively as seen in Table 5.1.

Table 5.1: Rotor-stator axial gap length at 50%span.

Case Number	Axial Gap
Case 1	8 mm
Case 2	16 mm
Case 3	32 mm

Axial gap between the rotor and stator of the each case is represented in Figure 5.1. As seen in the figures, rotor domain stay as same, however, the upstream part of the stator domain is extended in order to increase the axial gap between the rotor and stator.

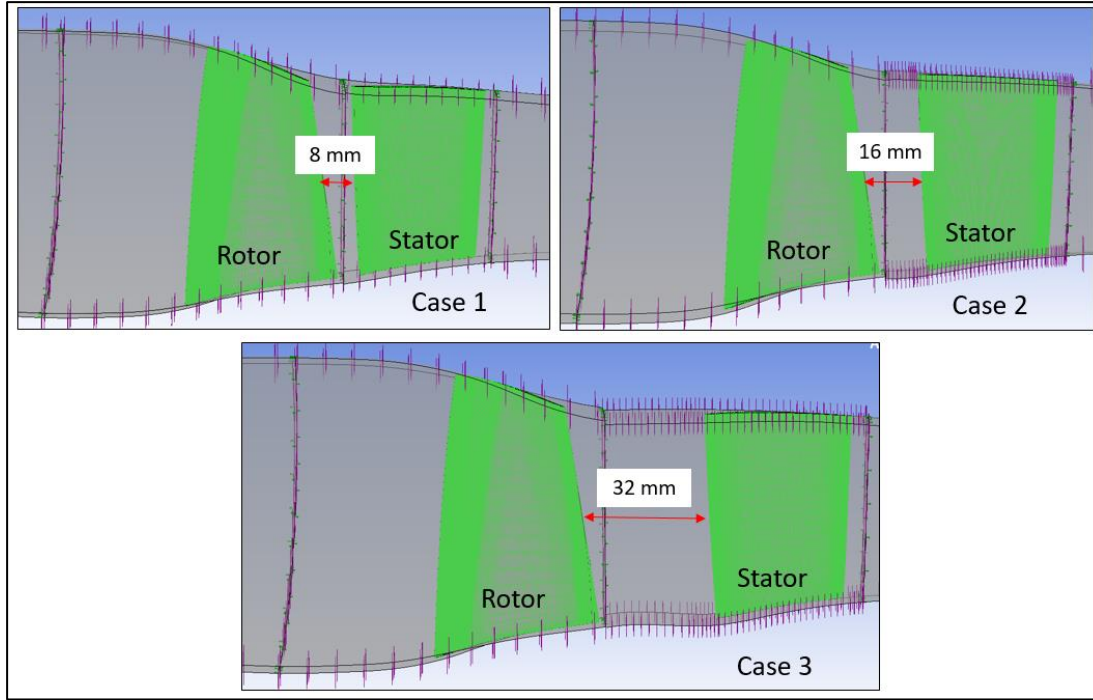


Figure 5.1: Rotor-stator axial gap at 50 %span.

5.1 Numerical Modeling

Inherent unsteady flow characteristics of the aeroacoustic problems requires to model the physics with unsteady CFD simulation. Numerical models of the unsteady direct noise calculation simulations and the FW-H acoustic analogy simulations were constructed in Star CCM+ environment. Unsteady Reynolds Averaged Navier Stokes Equation solver of the Star CCM+ was utilized for the direct noise calculation simulations. As explained in section 4, since the only the tonal noise generated by blade passing frequency was investigated, URANS solver was used instead of LES and DES. By this way, computational effort was significantly reduced. In Star CCM+ coupled implicit unsteady solver were utilized for the direct noise calculations simulations. As stated before, coupled solvers are capable of capturing high Mach number flows. Additionally, implicit solvers are better to use for stability of the solution. In order to capture the unsteady flow features and high frequency acoustic waves the time step size was carefully determined. The time step size of unsteady implicit solver corresponds to one percent of the rotor blade passing duration. That means each rotor blade passing was modeled with 100 time steps. Gourdain et al. [35] stated that 51 time step per rotor blade passing was sufficient for capturing transient

flow within an axial flow compressor. Time step calculation method is as shown below:

$$BPF = \frac{RPM \times B}{60} [s^{-1}] \quad (5.1)$$

$$Time\ Step\ Size = \frac{BPF^{-1}}{100} [s] \quad (5.2)$$

Air ideal gas was selected for material as done in the steady state simulations. Ideal gas equation of state enables the solver to include the compressibility effect, thus the acoustic waves propagated in the air can be properly modeled. Aeroacoustics simulation requires special boundary treatments on inlet and outlet boundary conditions to prevent the reflections of the acoustic waves. Non-reflective boundary conditions are needed in order to eliminate these spurious acoustic waves emanating from the inlet and outlet boundaries. However, Star CCM+ does not include these boundary treatment for URANS simulation. In order to eliminate these acoustic waves the upstream and downstream domains were extended and the element size towards the inlet and outlet boundaries were coarsened. Stagnation inlet and static pressure outlet boundary conditions were defined on inlet and outlet boundaries respectively. As in the steady state analysis, total pressure was set as 101325 Pa and total temperature was set as 288.15 K at the inlet boundary condition. Outlet static pressure was set to 145000 Pa, so that the calculated mass flow rate lies between surge margin and choked margin of the compressor map and provides a stable operation. Rotating speed was assigned as 17188 rpm as this compressor's original operation speed. Rigid body motion selection was selected for the time dependent movement of the rotor domain. The rotor domain rotates 0.1° with the calculated time step. SST k- ω turbulence model was utilized to resolve the developing gradients, separations, shock-boundary layer interactions and expansion waves on blade surfaces. Numerical modeling specifications are tabulated as seen in Table 5.2.

FW-H acoustic analogy modeling set up is relatively simpler than the DNC simulations. Since the acoustic analogy solver collects the unsteady pressure data from the direct calculation simulation, it is only activated after the DNC simulation. In this study, after the DNC simulation had been run for collecting sufficient amount of pressure data sample on rotor and stator blades, the FW-H acoustic solver was activated. Rotor blade and stator vane surfaces were set as the noise sources. Finally,

in the far-field region at the observer location, noise probes were assigned, so that the FW-H solver calculates the noise level at these probes.

Table 5.2: Numerical model details.

URANS Simulation Numerical Model Details	
Solver	Unsteady Implicit Coupled Solver
Rotation Modeling	Rigid Body Motion
Material	Air Ideal Gas
Time Step Size	9.696×10^{-7} s
Rotational Speed	17188.7 rpm
Inlet Absolute Total Pressure	101325 Pa
Inlet Total Temperature	288.15 K
Outlet Absolute Static Pressure	145000 Pa
Turbulence Model	SST k- ω

5.2 Flow Domain and Meshing

The unsteady direct noise simulation of transonic compressor stage requires resolving unsteady properties of rotor-stator interaction. In order to capture these unsteady properties, the pitch ratio of the rotor domain to the stator domain should be equal to one. This can be provided by modeling full rotor wheel and stator vanes. However, its computational effort would become tremendous. Instead of resorting this method, if the number of rotor blades and stator vanes have a common divisor, the number of the rotor blades and stator vanes are divided by the common divisor and the model consists of the calculated number of the rotor blades and stator vanes. For example, if the rotor wheel and stator consists of 36 blades and 48 vanes, their number would be divided into 12 and the resulting number of 3 rotor blades and 4 stator vanes would form a model with a pitch ratio of 1, since, in tangential direction, 3 rotor blades and 4 stator vanes sweep 30° . The compressor stage investigated in this study comprises 36 rotor blades and 46 stator vanes. This means that, in order to obtain the pitch ratio of one, 18 rotor blades and 13 stator vanes should be modeled. This modeling, which contains the half of the rotor wheel and the stator vanes, would result in a tremendous computational effort for simulation since the aeroacoustics modeling requires large number of element on mesh grid and long transient simulation times for sample data collection. In order to eliminate this computational effort and form a model which is reasonable for modifying and simulating, in this study, the number of stator vanes were decreased from 46 to 36. In this way, the rotor blades number and stator vanes number

were equalized and the transient compressor stage model was composed of only one rotor blade and one stator vane. Using 36 vaned stator row instead of original 46 vaned configuration resulted in some differences in aspect of aerodynamic field properties such as, total pressure ratio, total temperature ratio and isentropic efficiency of the compressor stage. Steady state CFD simulations of the case 1, case 2 and case 3 with 36 vaned and 46 vaned stator row were run to observe this difference in aerodynamic performance. The results are tabulated as shown below. According to these results, there is no drastic change in the performance of the compressor stage.

Table 5.3: Effect of stator vane number on flow properties for case 1.

Case 1	Total Pressure Ratio	Total Temperature Ratio	Rotor Blade Torque [N*m]	Mass Flow Rate [kg/s]	Isentropic Efficiency [%]
36 Vaned Stator	1.956	1.259	24.29	20.75	82.04
46 Vaned Stator	1.941	1.251	23.46	20.73	83.16

Table 5.4: Effect of stator vane number on flow properties for case 2.

Case 2	Total Pressure Ratio	Total Temperature Ratio	Rotor Blade Torque [N*m]	Mass Flow Rate [kg/s]	Isentropic Efficiency [%]
36 Vaned Stator	1.945	1.254	23.83	20.70	81.17
46 Vaned Stator	1.932	1.249	23.31	20.72	82.35

Table 5.5: Effect of stator vane number on flow properties for case 3.

Case 3	Total Pressure Ratio	Total Temperature Ratio	Rotor Blade Torque [N*m]	Mass Flow Rate [kg/s]	Isentropic Efficiency [%]
36 Vaned Stator	1.936	1.250	23.62	20.71	80.19
46 Vaned Stator	1.925	1.245	23.22	20.73	81.24

As shown in Figure 5.2 and Figure 5.3 which represents the mach number distribution in the rotor and stator domain respectively. The results shows that the flow fields are almost identical.

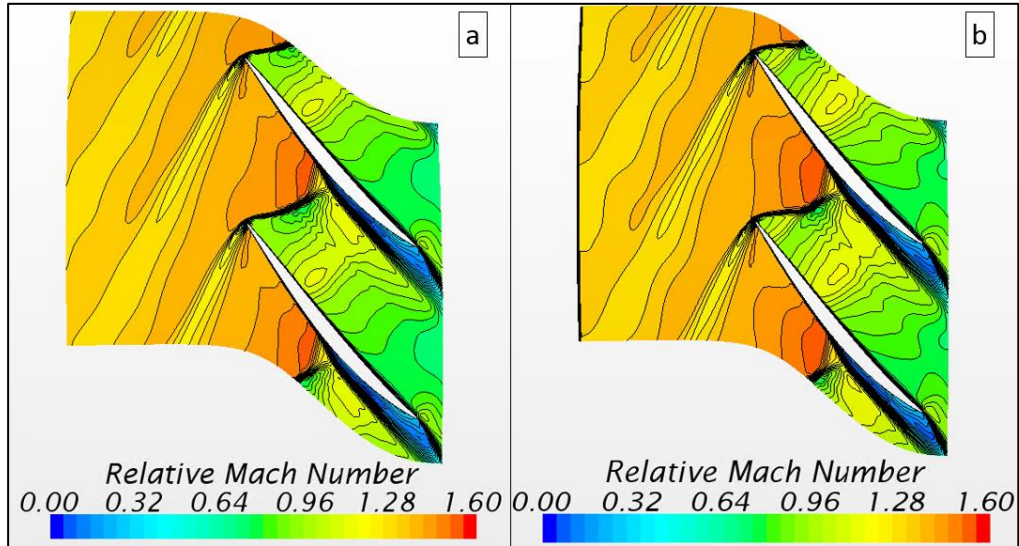


Figure 5.2: Rotor domain relative mach number, a) 36 vaned stator row, b) 46 vaned stator row.

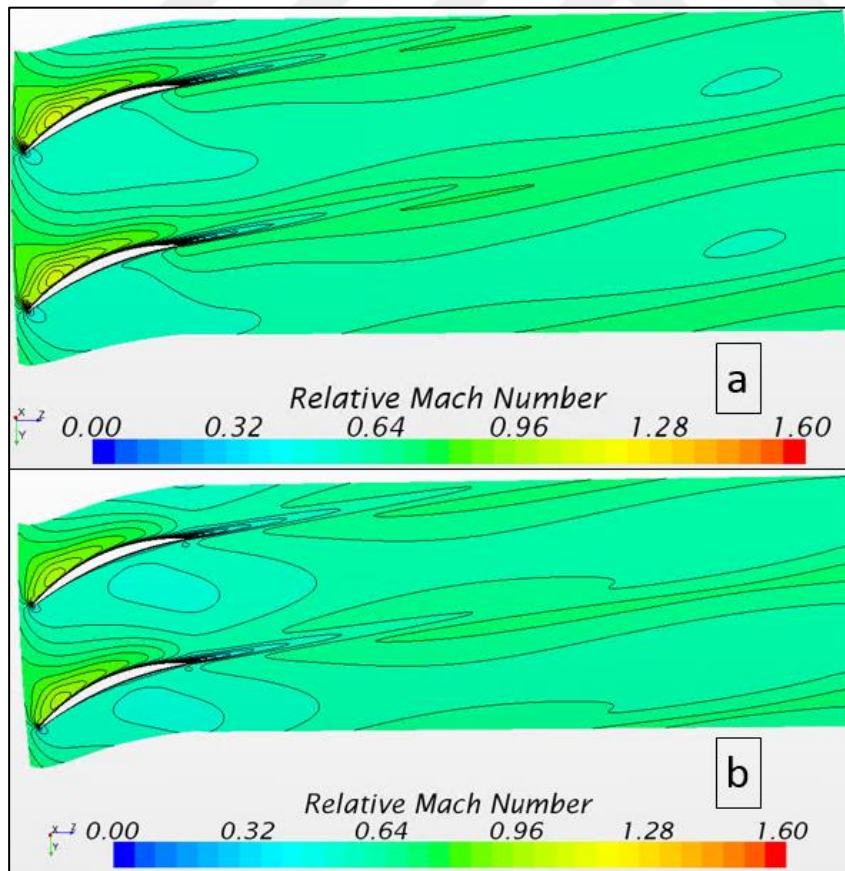


Figure 5.3: Stator domain relative mach number, a) 36 vaned stator row, b) 46 vaned stator row.

Figure 5.4 illustrates the static absolute pressure distribution on the stator domain. Although 36 vaned stator case has slightly higher pressure gradients on the suction side and pressure side of the stator vane than the 46 vaned stator case.

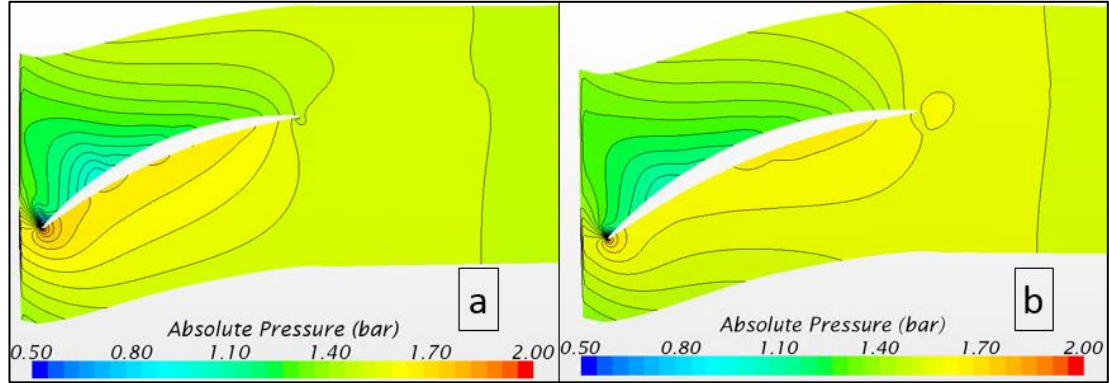


Figure 5.4: Stator domain absolute pressure, a) 36 vaned stator row, b) 46 vaned stator row.

Overall, in order to save a large amount of modeling work and simulation time, a rotor-stator blade row comprising one rotor blade and one stator vane was modeled with periodic boundary conditions. That means, the rotor blade and stator vane sweeps 10° in tangential direction since the full rotor wheel consists of 36 blades and stator row consists of 36 vanes.

Another modeling assumption is removing the tip clearance of the rotor blade since, the flow near the tip clearance is quite complex and modeling this flow requires many elements (Du et al, 2013). By removing tip clearance, considerable number of element and accordingly simulation time was reduced.

In the aeroacoustic simulations, eliminating the spurious acoustic waves reflected from the inlet and outlet boundary conditions is significant. In order to reduce the effect of the reflected acoustic waves inlet and outlet domains were extended five chord lengths of the rotor blade.

As seen in Figure 5.5 each aeroacoustic simulation geometry consists of an inlet, rotor, stator and outlet domains. As mentioned, inlet and outlet domains were intentionally extended. There is no rotor tip clearance in the models and rigid body motion was used to simulate the rotation.

Figure 5.6 illustrates the boundaries of the simulation model. As seen, the side surfaces were assigned as rotational periodic interface. Since the rotor domain rotates and the inlet and stator domains are stationary, the interface selection between rotor-inlet and

rotor-stator is important. In Star CCM+, these two interfaces were assigned as internal interface with repeating topology. As the rotor domain completes one blade passing period it returns to beginning position thanks to this topology setting. Although the stator and outlet domains are both stationary, in order to meshing them independently they were separated with an interface.

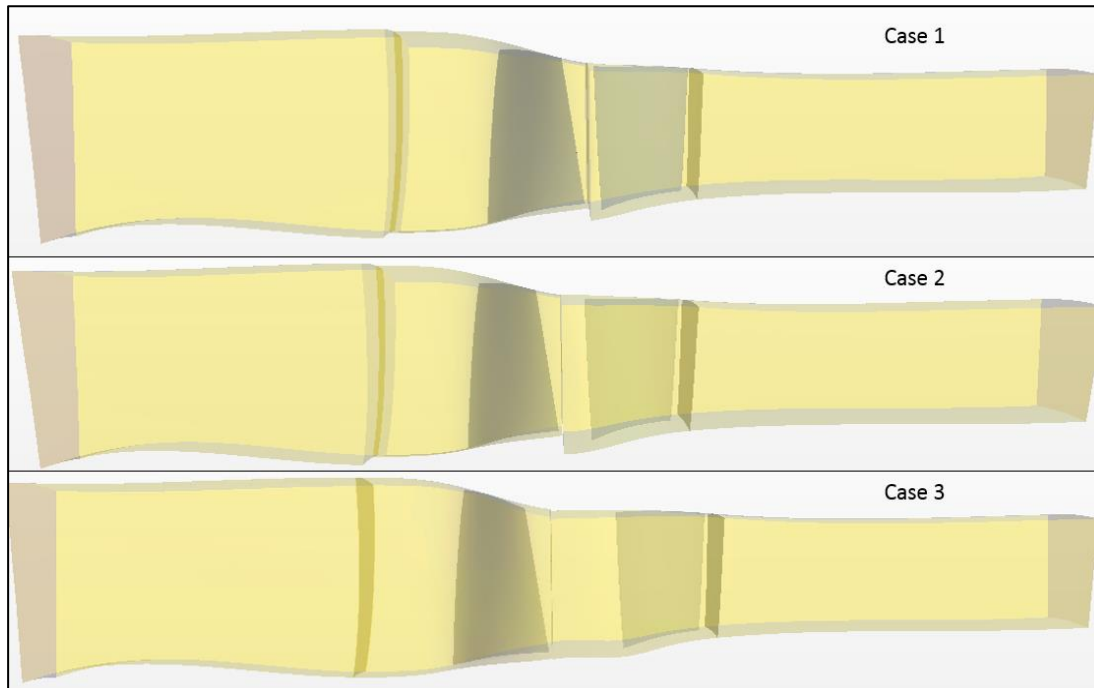


Figure 5.5: Geometries of aeroacoustic simulation cases.

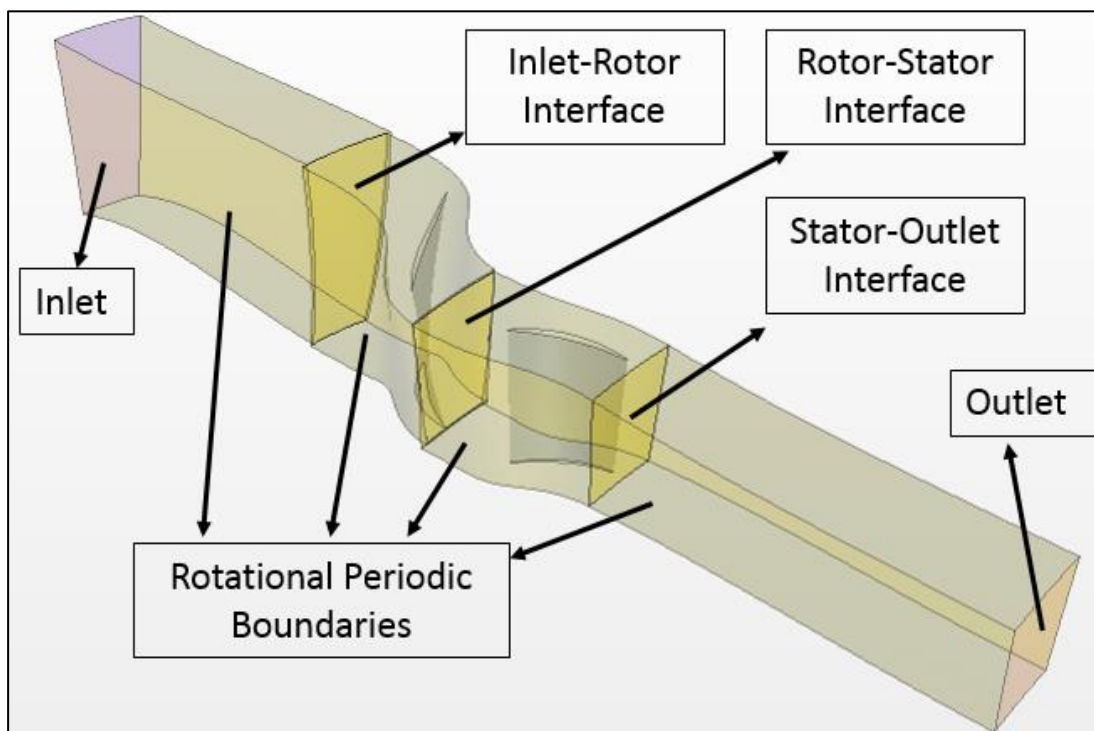


Figure 5.6: Boundaries on the simulation geometry.

Meshing strategy of the geometry is completely based on resolving the related aeroacoustics features from the flow field. In general, there are two main considerations when doing meshing for aeracoustics simulations: the mesh must resolve the sound generating turbulence scales and propagating acoustic waves. In this study, since the turbulence generated broadband noise is not the research topic, the mesh structure is not able to resolve turbulent eddies. Moreover, even if the mesh structure had been fine enough, URANS turbulence model could not have captured the demanded turbulence scales, since it models the turbulence instead of resolving scales like LES & DES. Thereby, the main consideration of the meshing is to capture the propagating acoustic waves in the near-field region, since acoustic analogy model was used for far-field noise propagation. Moreover, the mesh structure must be able to resolve the rotor-stator interaction. The stator potential field on the rotor and the rotor wake interaction on stator vanes are the two key consideration for the interaction noise. The mesh structure must capture the small unsteady pressure fluctuations on the rotor surface and the rotor wake. If the rotor wake undergoes dissipation due to the coarse mesh before it hits the stator, the interaction cannot be captured properly.

The first step for capturing acoustic wave propagation to determine the upper frequency limit in the flow domain that the simulation resolves. The acoustic wave with the upper frequency has the minimum wavelength. The best practices for the second order numerics proposes at least 20 elements per wavelength. Thus, dividing the minimum wavelength into 20 gives the minimum element size in the domain. For the near-field acoustics, the upper frequency limit was determined as the fourth harmonics of the BPF. This means that the tonal noise up to the fourth harmonics of the BPF can be captured by the simulation, whilst the higher harmonics may undergo a numerical dissipation.

$$\lambda = \frac{c_0}{f} \quad (5.3)$$

$$\Delta x = \frac{\lambda}{20} \quad (5.4)$$

The first formula above shows the wavelength calculation and the second represents the element size for resolving the desired wavelength.

Sound waves propagating upstream towards the mean flow have shorter wavelengths than sound waves of equal frequency propagating downstream with the mean flow.

Since the upstream wavelengths are shorter, the upstream part mesh must be finer than the downstream.

$$\lambda_{downstream} = \frac{c_0 + V_{mean}}{f} \quad (5.5)$$

$$\lambda_{upstream} = \frac{c_0 - V_{mean}}{f} \quad (5.6)$$

Beyond the near-field acoustic region the sound waves were intentionally dissipated before they reach the inlet and outlet boundaries. As mentioned, the reflected sound waves from the inlet and outlet boundaries may add some spurious waves in the flow domain. In order to eliminate this effect, the inlet and outlet domains were extended and the element size gradually increased towards these boundaries.

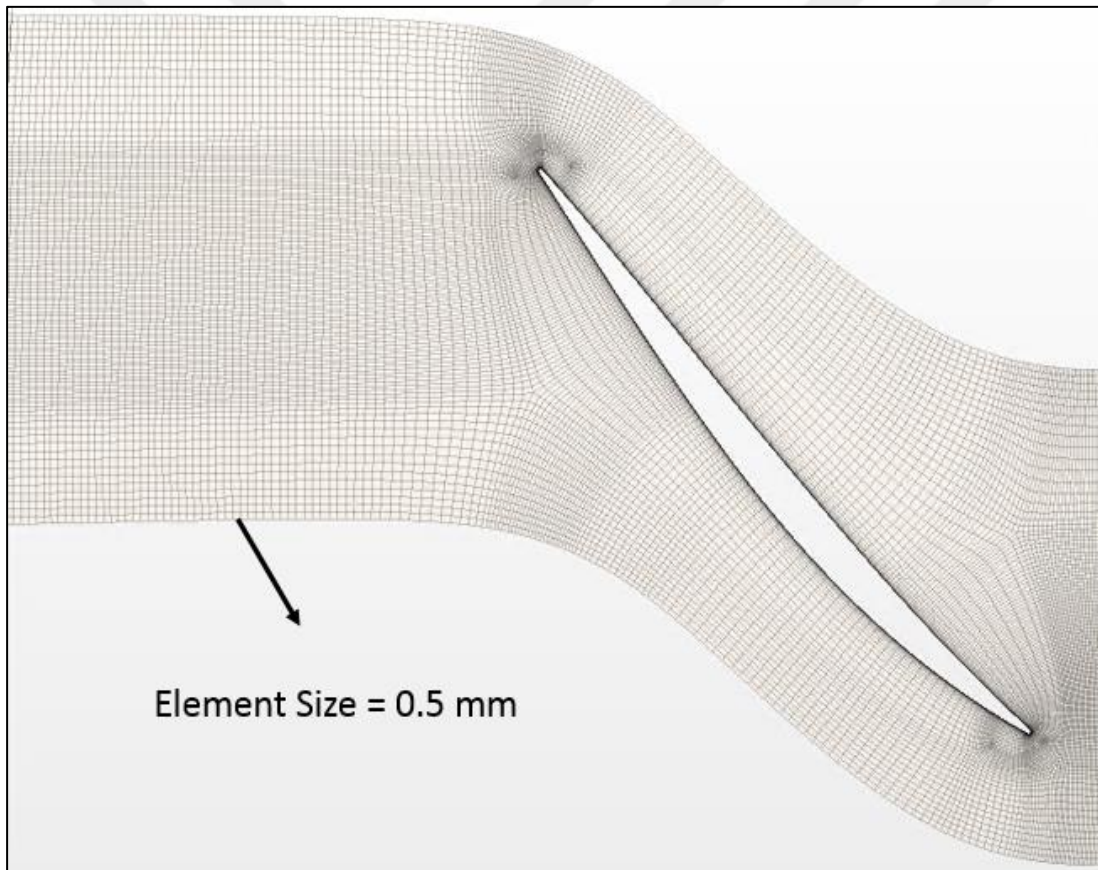


Figure 5.7: Rotor domain 50% span mesh.

As seen in Figure 5.7 and Figure 5.8 the mesh size was kept as small as possible to resolve the near field acoustic wavelengths. Figure 5.9 shows how the mesh is coarsened for avoiding the spurious wave reflections from inlet and outlet domains as mentioned before.

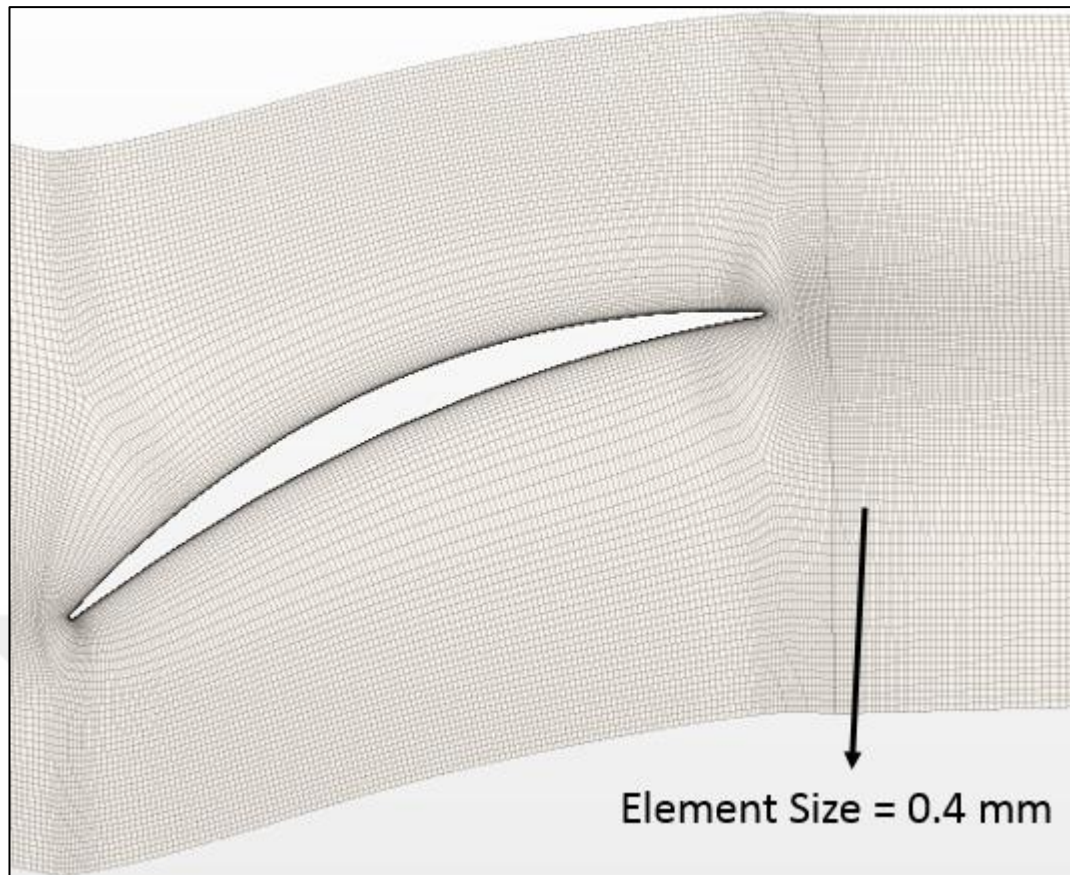


Figure 5.8: Case 1 stator domain 50% span mesh.

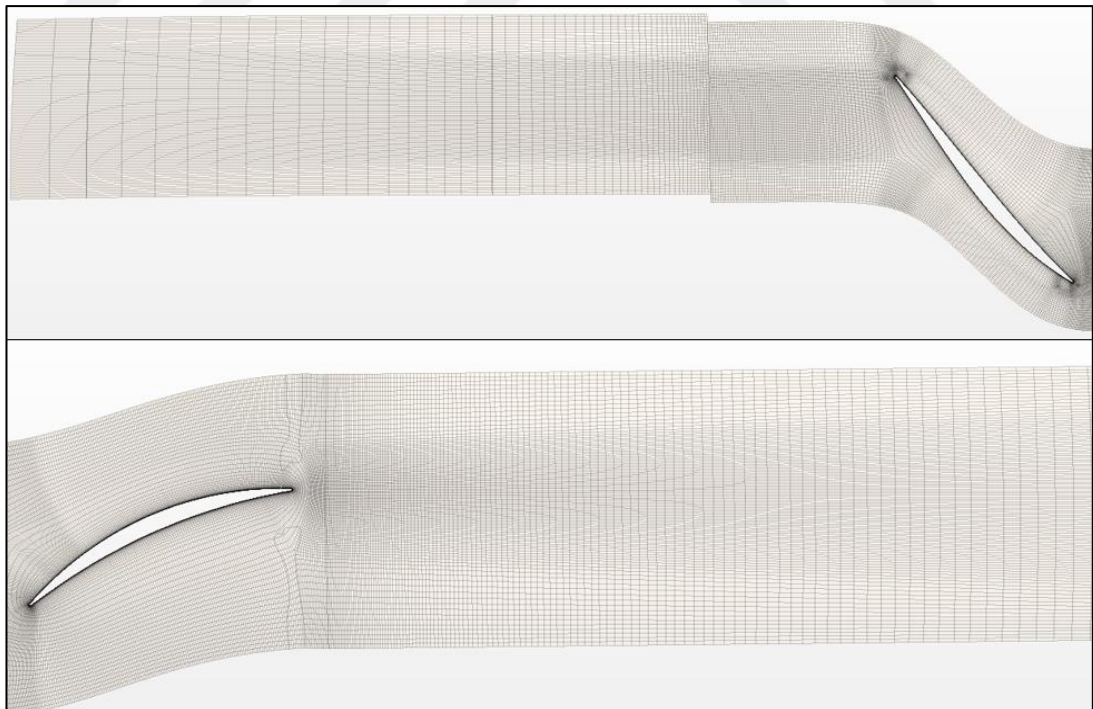


Figure 5.9: Inlet and outlet domain.

3D mesh structure of three cases were constructed in ANSYS TurboGrid. Afterwards, each mesh structure was imported in Star CCM+ to conduct the aeroacoustics

simulations. As shown in Table 5.6, since the axial gap is modified by changing the stator geometry, inlet and rotor mesh were not altered. However, the number of elements in the stator region increased as the axial gap became larger.

Table 5.6: Number of elements on each region

Case	Inlet Region	Rotor Region	Stator Region	Outlet Region	Total
Case 1	265,000	1,216,562	1,434,328	903,000	3,818,890
Case 2	265,000	1,216,562	1,519,677	839,040	3,840,279
Case 3	265,000	1,216,562	1,888,621	804,224	4,174,407

Boundary layer resolution on rotor blade and stator vane surfaces was perfectly captured by keeping the y^+ value below one. As seen in Figure 5.10, y^+ increases on the suction surface of rotor blade due to accelerating expansion waves and it sharply decreases across the shock wave.

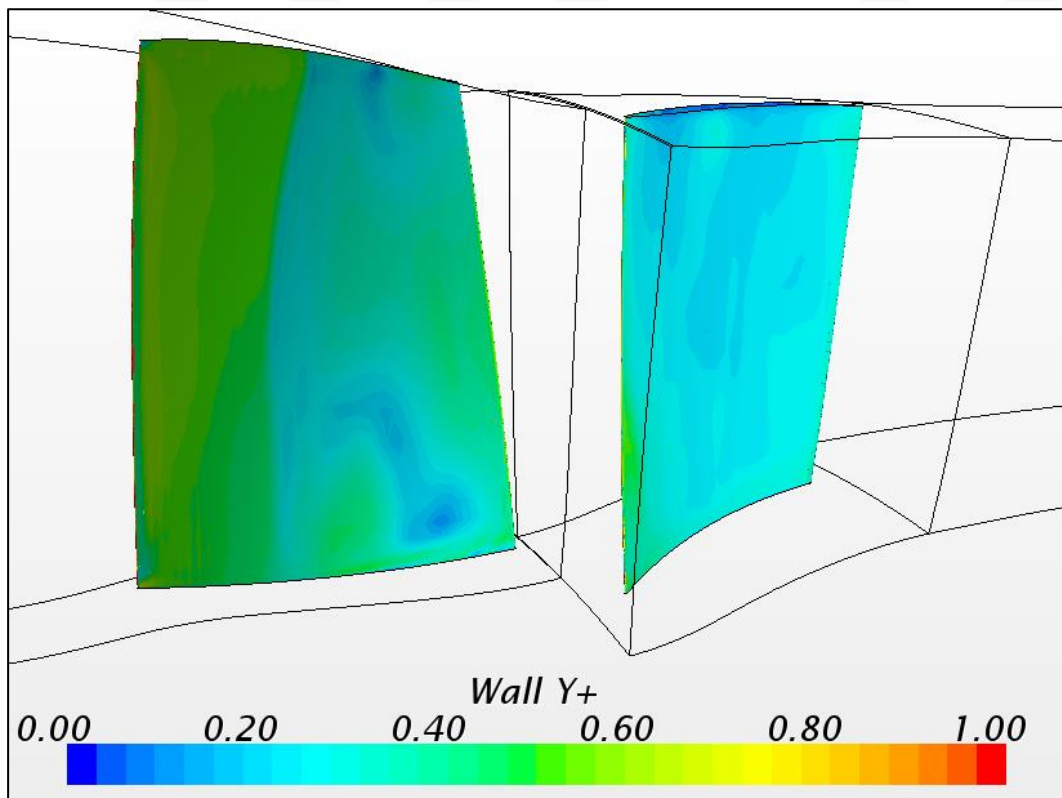


Figure 5.10: y^+ distribution on rotor blade and stator vane surfaces.

5.3 Aerodynamic Investigation of the Flow

The effects of the axial gap on the aerodynamics of the transonic compressor stage was systematically investigated by comparing the flow field data of each case. The flow field in the compressor stage, the unsteady rotor blade forces and the unsteady stator vane forces.

5.3.1 Flow Field In Compressor Stage

The aerodynamic flow features of the three cases which effects the aerodynamic outputs and, accordingly, the generated noise were compared each other. The main flow features in a transonic compressor stage are Mach number, velocity, pressure, temperature, entropy, turbulence viscosity ratio, speed of sound distributions. Each of these parameters has an influence of generated noise. Apart from the noise generation, aerodynamic outputs such as total pressure ratio, total temperature ratio and the isentropic efficiency must be also considered because the main design consideration is to obtain lowest noise level with the highest stage efficiency. Time averaged aerodynamic outputs of the transient simulation of three cases are compared as in the Table 5.7 below. It can be observed that, as the axial gap increases aerodynamic outputs of total pressure ratio, total temperature ratio and isentropic efficiency decreases. Hünecke (1997) stated that, relatively large spacing for noise reduction generally degrades fan efficiency by 1 percent. As mentioned, noise reductions on turbomachinery noise is a trade-off between efficiency and noise generation.

Table 5.7: Time averaged aerodynamic outputs

Case	Total Pressure Ratio	Total Temperature Ratio	Isentropic Efficiency
Case 1	1.953	1.257	0.832
Case 2	1.939	1.254	0.820
Case 3	1.915	1.245	0.800

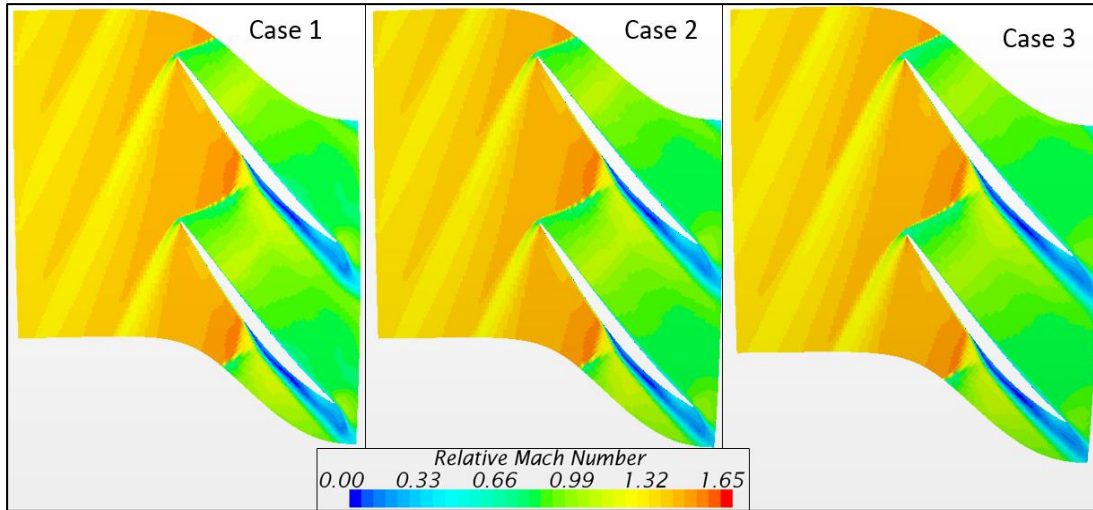


Figure 5.11: Rotor midspan mach number distribution.

As seen in Figure 5.11, relative Mach number distributions are almost similar for three cases. However, it is observable that the rotor wake is affected by the stator potential field. The rotor wake of first case is largely deflected in the opposite direction of rotation. There is a slight turn in the wake of case 2 as well, whilst the wake of the case 3 is not deflected since the stator is located sufficiently away from the rotor. On the other hand, the strength of the shock wave, shock boundary layer interactions, suction side and pressure side Mach number distributions are almost same.

The Mach number distribution through the stator and outlet domains are illustrated in Figure 5.12. As seen, for each case, due to the angle of attack of the incoming flow, on the suction side of the stator, the flow accelerates and exceeds speed of sound. In the case 1 and case 2 after the acceleration flow detaches from the stator surface and a separation bubble forms. Afterwards, flow reattaches again until it reaches the trailing edge and separates again. The separation bubble is larger in case 1. This means, as the blade passes the stator the angle of attack of the incoming flow changes with a higher amplitude than the case 2. This results in larger unsteady pressure fluctuations on case 1, consequently higher tonal noise generation. The angle of attack is lower in the case 3 since the rotor wake degrades until it arrives stator leading edge. That is why lower amplitude unsteady pressure arises on case 3 than the case of case 1 and case 2. However, the stator wake structure is larger than it is for case 1 and case 2. This introduces higher turbulence content to the main flow in case 3 and effects the broadband noise generation.

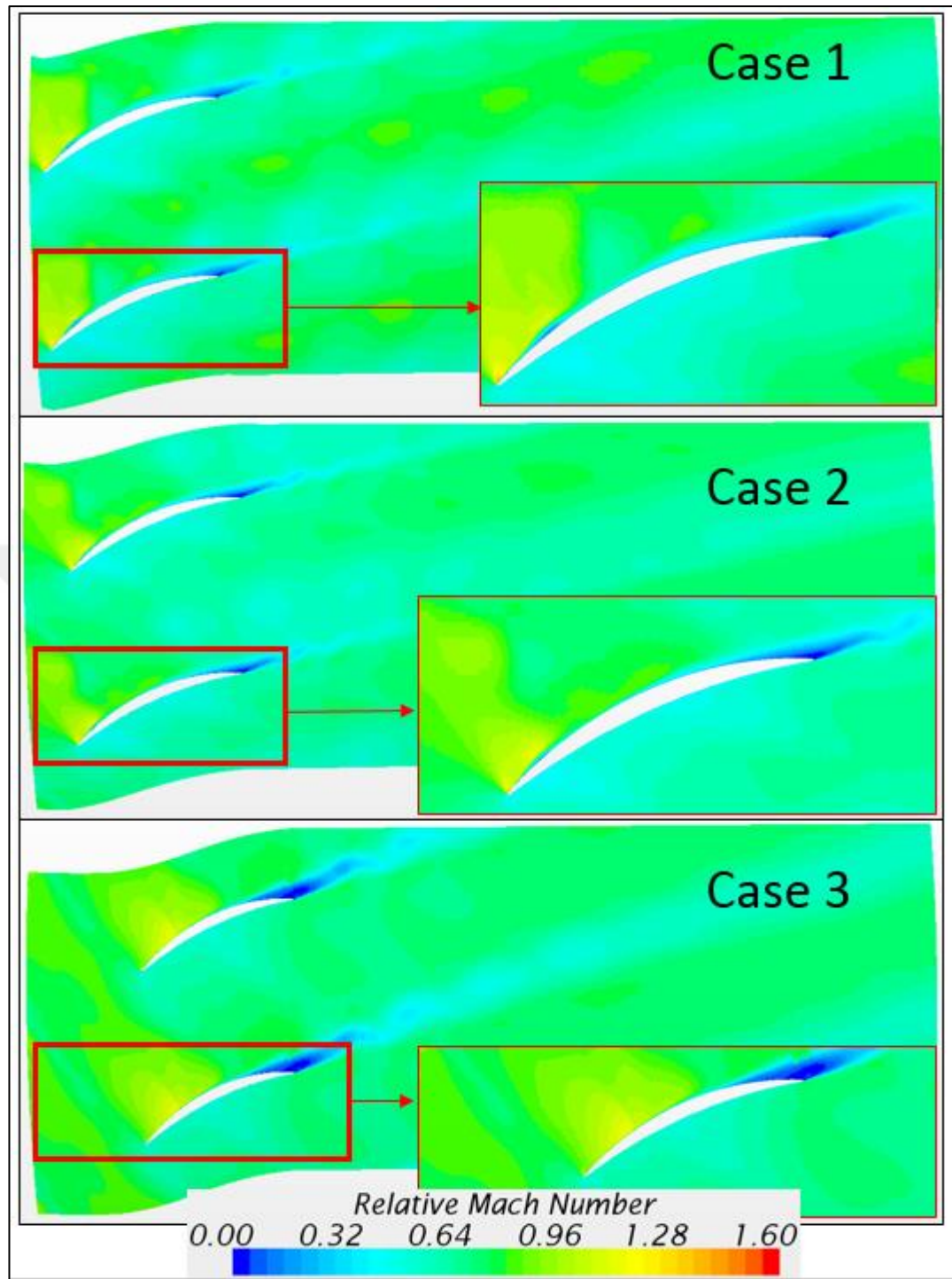


Figure 5.12: Stator midspan mach number distribution.

The potential field of the stator directly affects the flow field of the rotor blade. As seen in Figure 5.13, the pressure gradients on the rotor blade is denser in case 1. Moreover, the higher gradients are located between the rotor downstream and stator upstream in case 1 as well. As the axial gap increases the potential of the stator blade decreases and the pressure field between the rotor and stator becomes smoother.

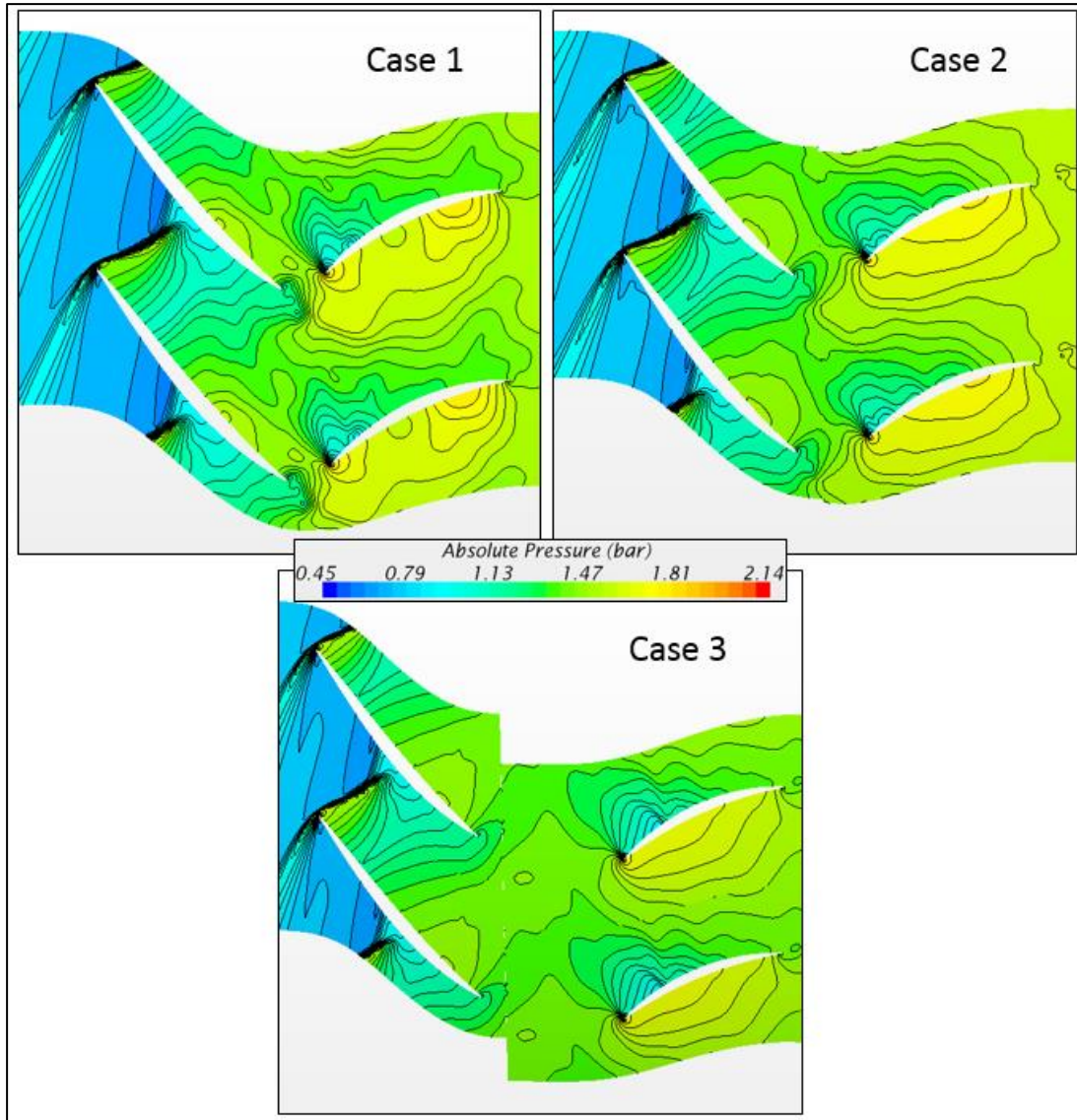


Figure 5.13: Pressure field in at 50% span section.

In order to see the effect of the relative position of the rotor blade to stator vane on the pressure gradients the case 1 and case 3 was compared at different time steps as seen in Figure 5.14. The rotor and stator are positioned face to face in the pictures on the left. This is the position when the distance between rotor and stator is shortest and densest interaction is expected. On the other hand, the rotor blade is positioned between the two stator vanes, when the interaction is expected to be lowest level. As seen, the pressure contour lines on the rotor blade and the stator blade change significantly in case 1. Especially, on the pressure side of the stator vane a remarkable change takes place. In contrast, the pressure gradients on the rotor and stator blades in case 3 are almost similar. This illustrates that unsteady pressure fluctuations on rotor and stator blade increase as the axial gap decreases.

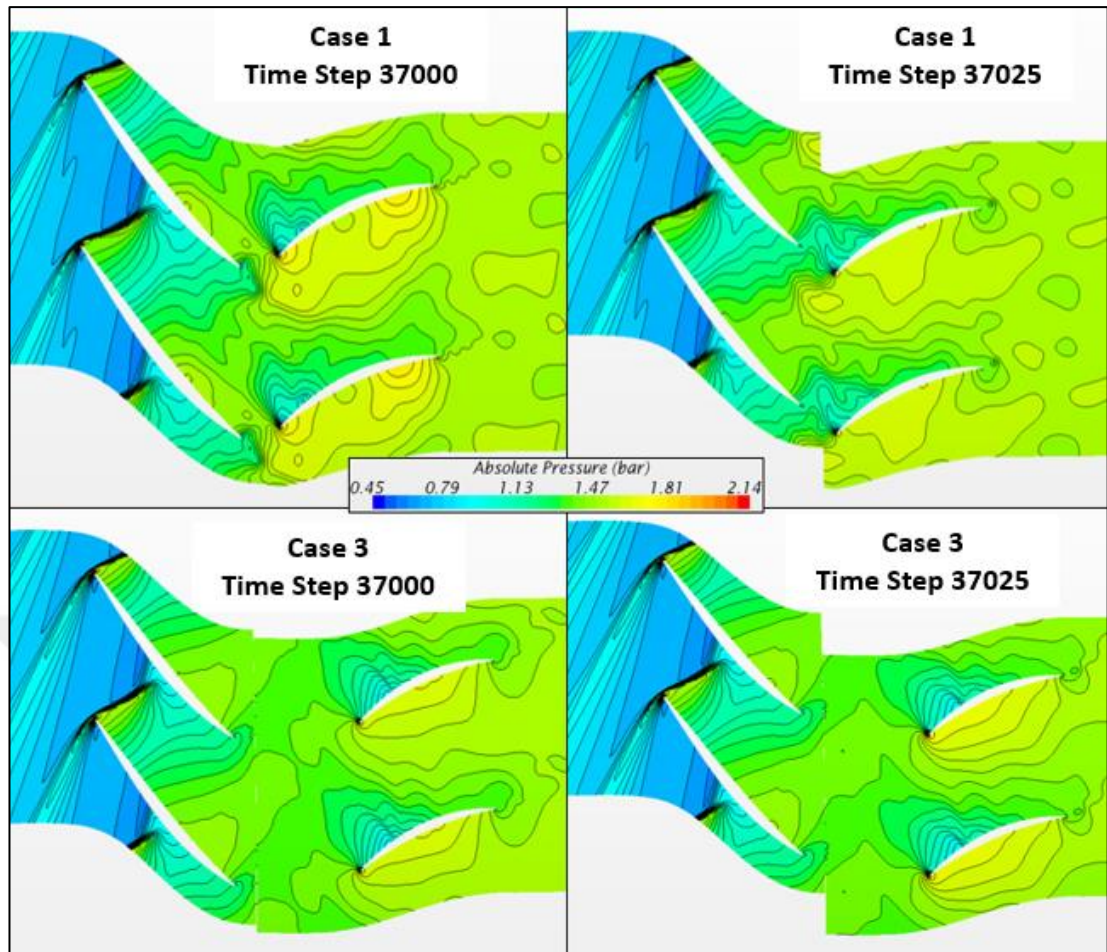


Figure 5.14: Pressure field of case 1 and case 3 at different time steps.

The temperature field in the compressor stage is shown in Figure 5.15. Incoming flow has the temperature of 15 °C. This increases across the shock wave. On the other hand, it reaches its minimum on the suction side of the rotor blade by accelerating due to the expansion waves. As seen, static temperature reaches its maximum value in the shock-boundary layer interaction- generated flow separation on the rotor suction surface. This highly turbulent region undergoes the dissipation of eddies and the temperature in this field increases. The temperature field is a good visualization tool for rotor wake. As seen, high-temperature rotor wake is in a denser interaction with the stator vane in case 1. There are three rotor blade wakes are observable on the pressure side of the stator vane in case 1 and 2. However, the rotor wake of case 1 is stronger since it does not degrades quickly due to smaller axial gap. Beacuse of the larger axial gap the rotor wake of case 3 loses its strength until it reaches the stator leading edge. Only two rotor blade wakes are observable on pressure side of the stator vane for case 3. This also proves that the densests interaction between the rotor wake and stator vane takes place in case 1.

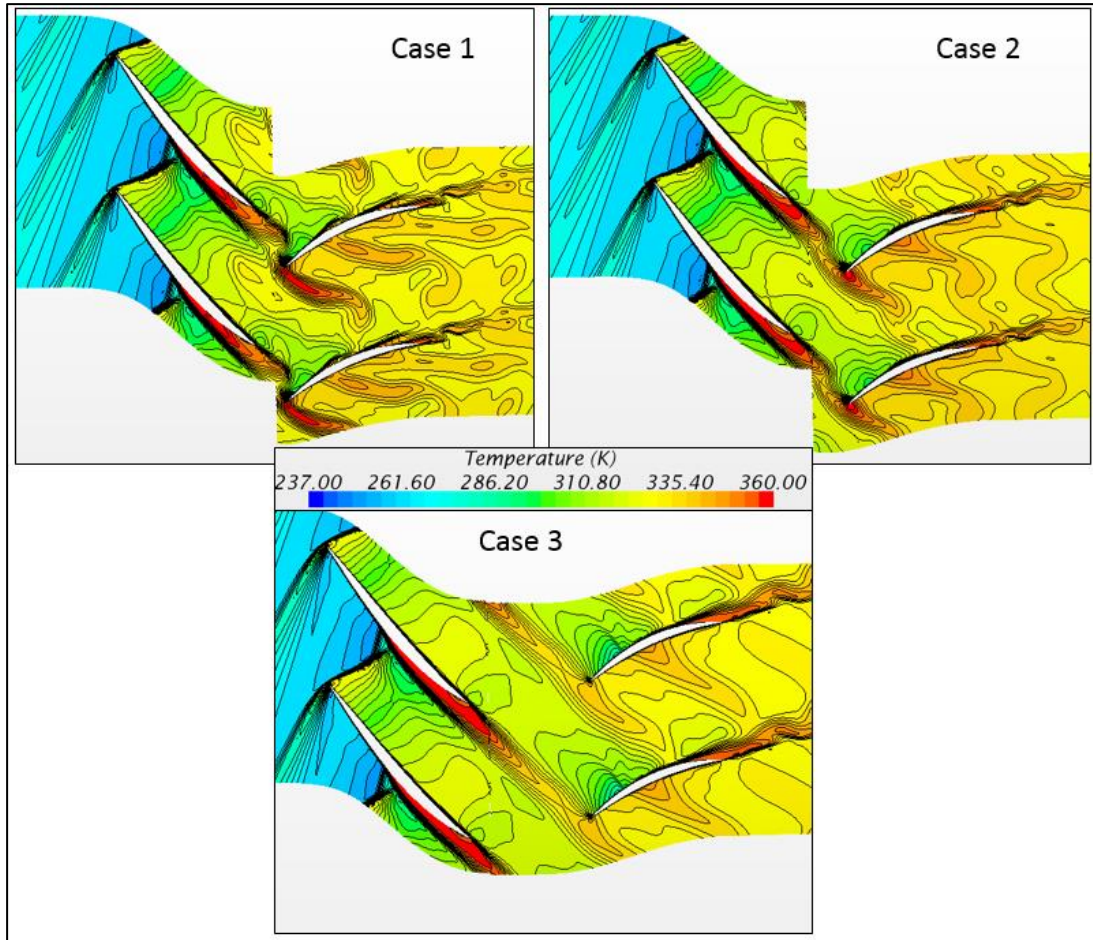


Figure 5.15: Temperature field at 50% span section.

Entropy field within the transonic compressor stage is illustrated in Figure 5.16. The main entropy generation mechanisms in Figure 5.16 are shock wave boundary layer interaction on the suction side of the rotor blade. In addition, that shock wave produces entropy itself as can be seen with a slight increase between the rotor blades. As seen, due to the smaller axial gap between the rotor and stator, the entropy wakes are deflected by the stator leading edge in case 1 and case 2. Whilst, the entropy wake approaches the stator vane smoothly in case 3. The entropy wakes are deflected between two stator vanes since suction side and pressure side of the stator vanes has different flow velocity. Because of the smaller axial gap entropy wakes stayed on the stator vane pressure side longer time on case 1 and eventually more entropy wake are located on stator at the same time than the other cases. This leads to a stronger unsteadiness of flow property change on stator vane and as a result higher tonal noise generation. Moreover, the strength of the wake is maximum in case 1. This increases the tangential velocity of the flow field and flow hits the stator vane with a higher incidence angle in case 1.

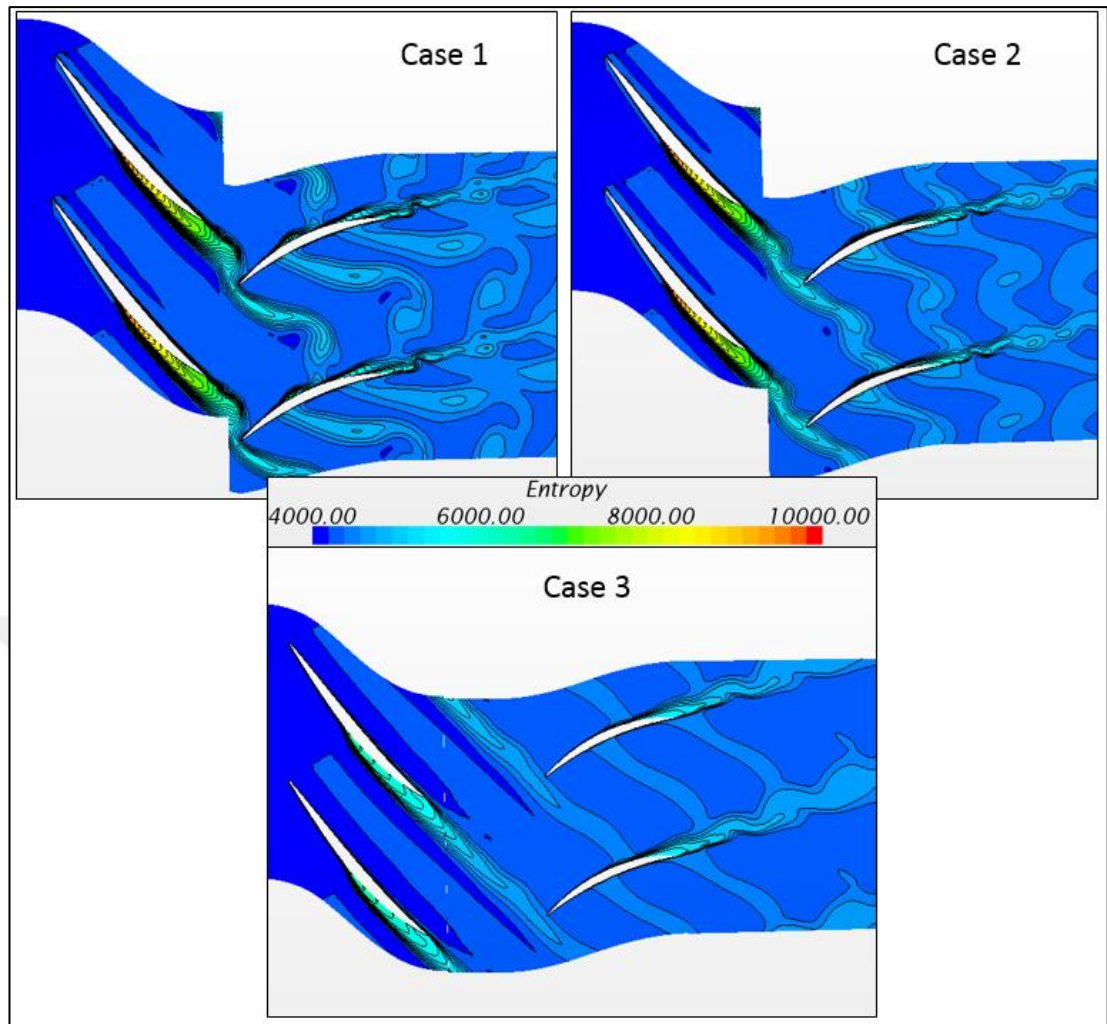


Figure 5.16: Entropy field at 50% span section.

As seen in Figure 5.17, field of turbulence viscosity ratio shows the same trend as the temperature and entropy fields for three cases. For case 1 and case 2, TVR reaches its maximum value in the rotor wake. However, TVR has its maximum value in the stator vane wake in case 3. This occurs because of the large separation on stator vane trailing edge as seen in Figure 5.12. However, this turbulence content in the downstream of stator may only contribute to broadband noise spectrum rather than tonal one.

Another criteria for investigating the rotor wake-stator vane interaction is upwash velocity as detailed explained in 1.4.2.3. Since the rotor wake has a velocity deficit compared to free stream, the angle of attack of the rotor wake that impinges the leading edge of the stator vane is higher than the attack angle of the free stream. The angle of attack on the stator vane surface changes at the BPF and this leads to unsteady pressure fluctuations on stator vane. As seen in Figure 5.16 the rotor wake loses its strength as it travels further from the rotor trailing edge. This means that, the rotor wake in case 1

which is the strongest one would have the lowest wake relative and the highest upwash velocity.

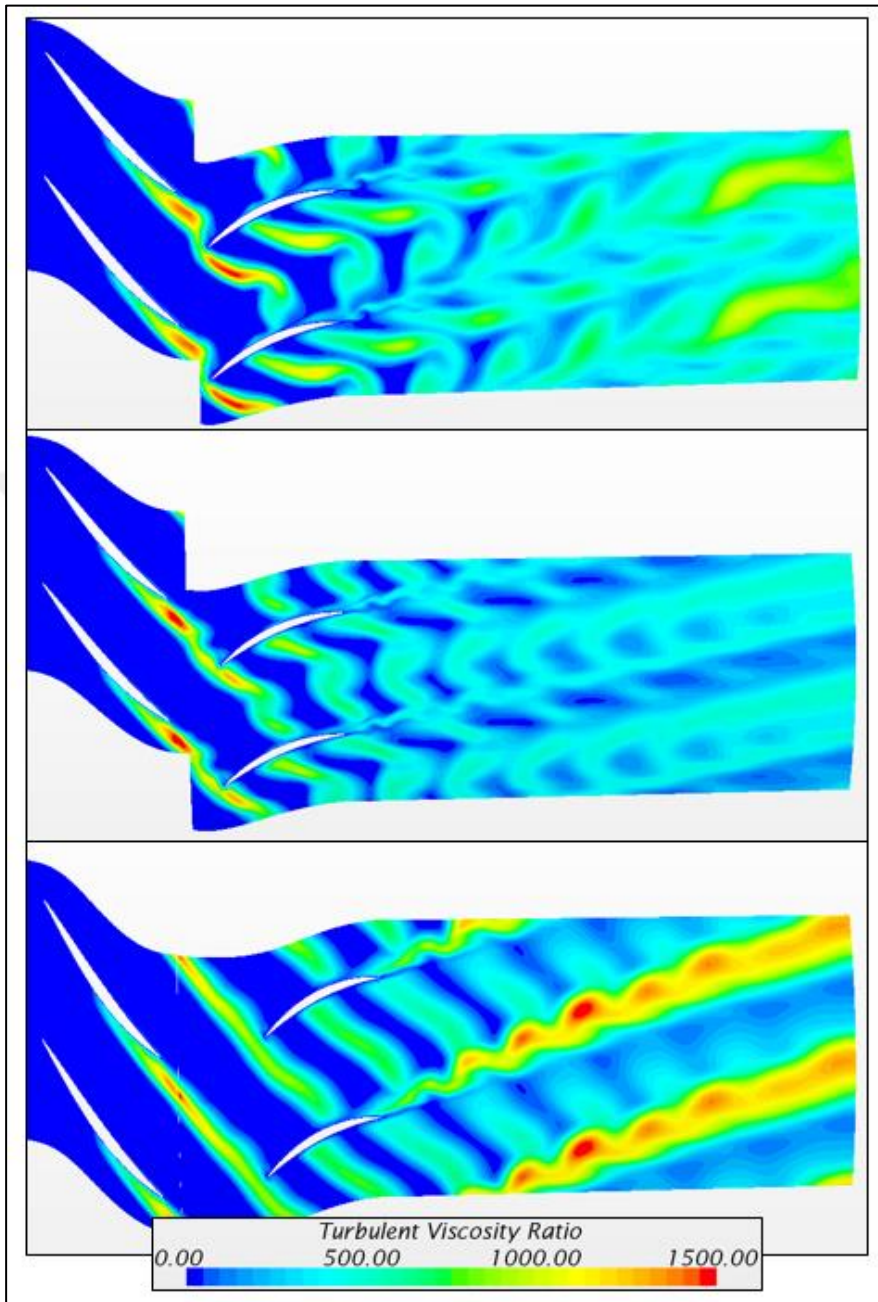


Figure 5.17: Turbulence viscosity ratio field at 50% span section.

In order to calculate the upwash velocity for three cases, two probes are located in front of the leading edge of stator vane at midspan and 75%span as seen in Figure 5.18. These probes measure the required absolute velocity and the relative velocity at these locations for upwash velocity calculations as seen in Figure 5.19. The free stream velocity triangle is represented with black color, whilst the wake centerline's velocity triangle is in red.

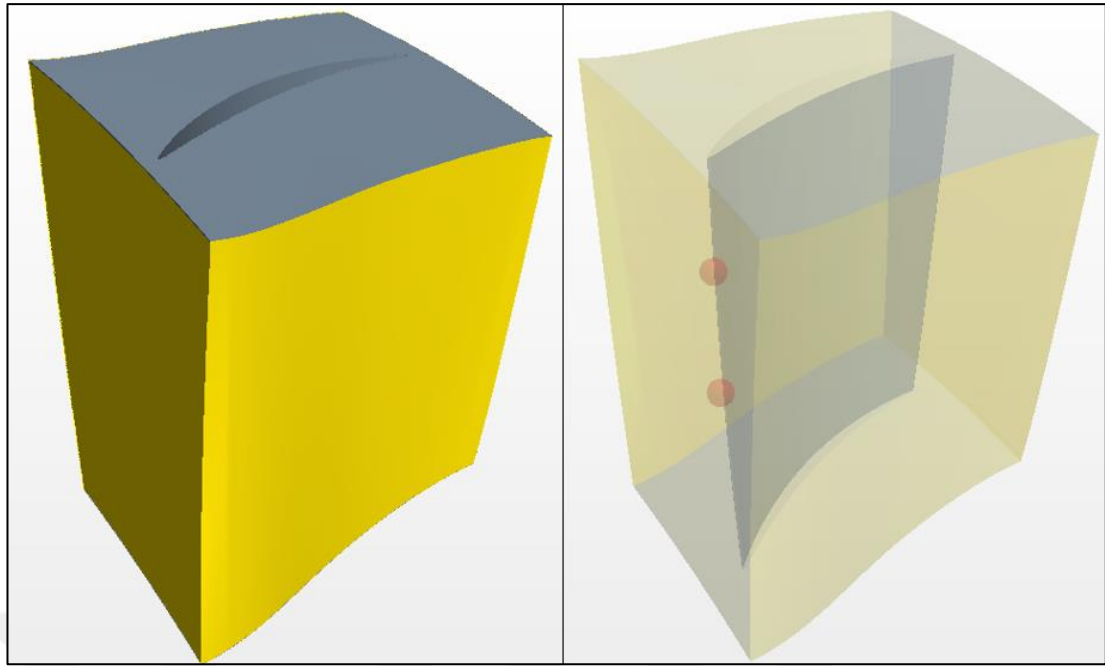


Figure 5.18: Velocity probes in upstream of stator vane.

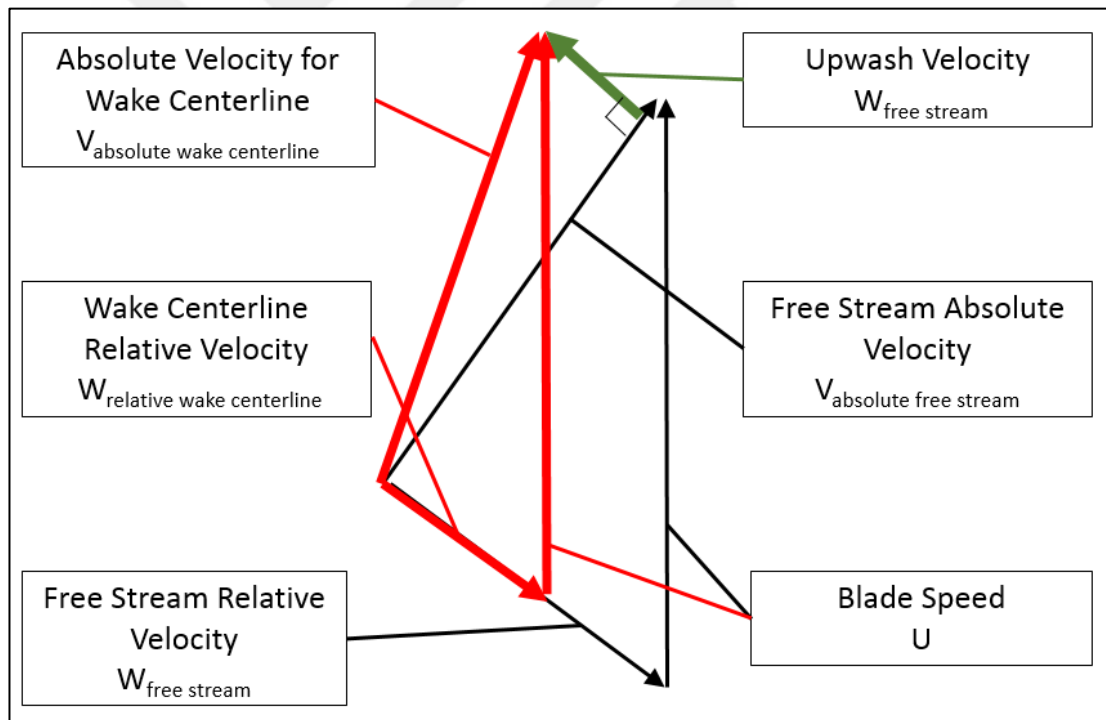


Figure 5.19: Upwash velocity components.

The upwash velocity calculations on midspan and 75% span probes for case 1, case 2 and case 3 is tabulated in Table 5.8 and Table 5.9. As seen, the calculated upwash velocity is larger for smaller axial gap. This results in higher amplitude of pressure fluctuations on stator vane surface. Since these values were extracted from the centerline of the wake, the resulted upwash velocity has its maximum value at the

probe. At blade passing frequency the upwash velocity will change between its maximum value and zero.

Table 5.8: Upwash velocity calculation for 50% span probe.

Case	Blade Speed [m/s]	Free Stream Absolute Velocity [m/s]	Free Stream Relative Velocity [m/s]	Absolute Velocity for Wake Centerline [m/s]	Wake Centerline Relative Velocity [m/s]	Upwash Velocity [m/s]
Case 1	387	195	247	213	177	147.9
Case 2	387	195	250	201	194	119.7
Case 3	387	213	262	212	230	49.4

Table 5.9: Upwash velocity calculation for 75% span probe.

Case	Blade Speed [m/s]	Free Stream Absolute Velocity [m/s]	Free Stream Relative Velocity [m/s]	Absolute Velocity for Wake Centerline [m/s]	Wake Centerline Relative Velocity [m/s]	Upwash Velocity [m/s]
Case 1	414	167	301	200	210	146.2
Case 2	414	201	272	233	191	128.6
Case 3	414	245	290	247	225	92

Apart from the stator upstream probes, a probe was located in rotor blade downstream at 50% span, as seen in Figure 5.20, in order to understand the stator potential field on downstream velocity field of rotor blade. Figure 5.21 shows that the absolute velocity profiles of each case at the downstream probe of the rotor blade. As seen, the absolute

velocity decreases with the smaller axial gap between rotor and stator because the rotor downstream flow is affected by the stator potential field denser and the flow starts to decelerate till reaching the stagnation point on stator leading edge.

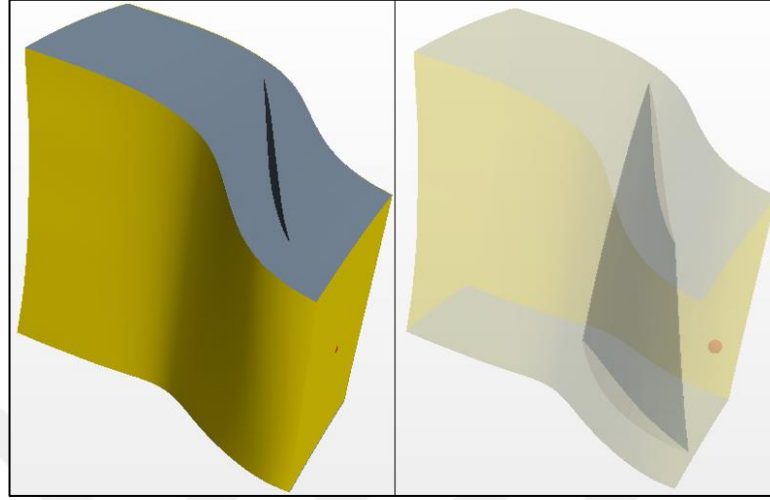


Figure 5.20: Rotor blade downstream velocity probe.

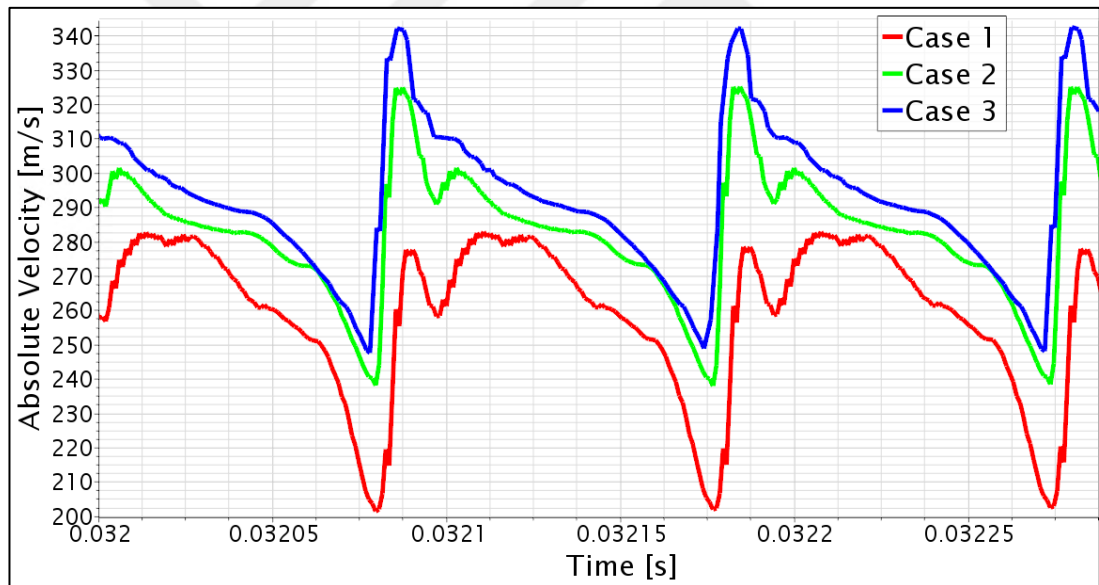


Figure 5.21: Absolute velocity plot of rotor downstream probe.

5.3.2 Unsteadiness on rotor blade surface

Unsteady pressure fluctuations on the rotor blade is vital to investigate the stator potential field on rotor blade. As the effect of stator potential on rotor blade increases the unsteady pressure fluctuations increase and this leads to higher amplitude noise generation, radiated from rotor blade surface.

Unsteady rotor surface forces are integrated in axial, radial and tangential direction. The amplitude of the unsteady forces on rotor blade surface increases strongly as the

axial gap becomes smaller as seen in Figure 5.22 and Figure 5.23. Highest amplitudes of unsteady forces are observed in case 1. Since there was no precise difference in the forces of radial direction, it was not plotted.

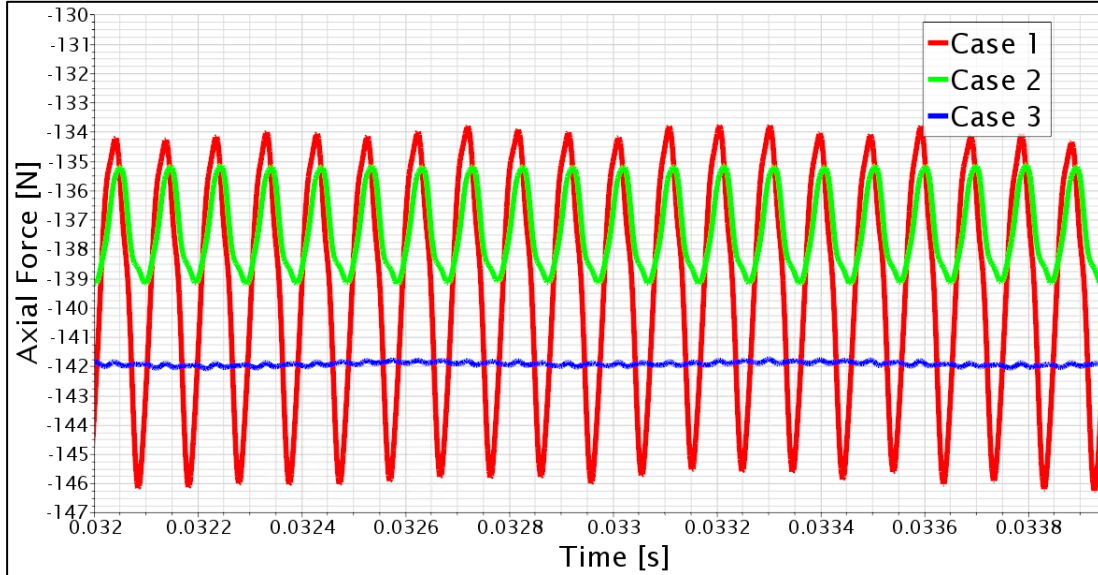


Figure 5.22: Axial force exerted on rotor blade surface.

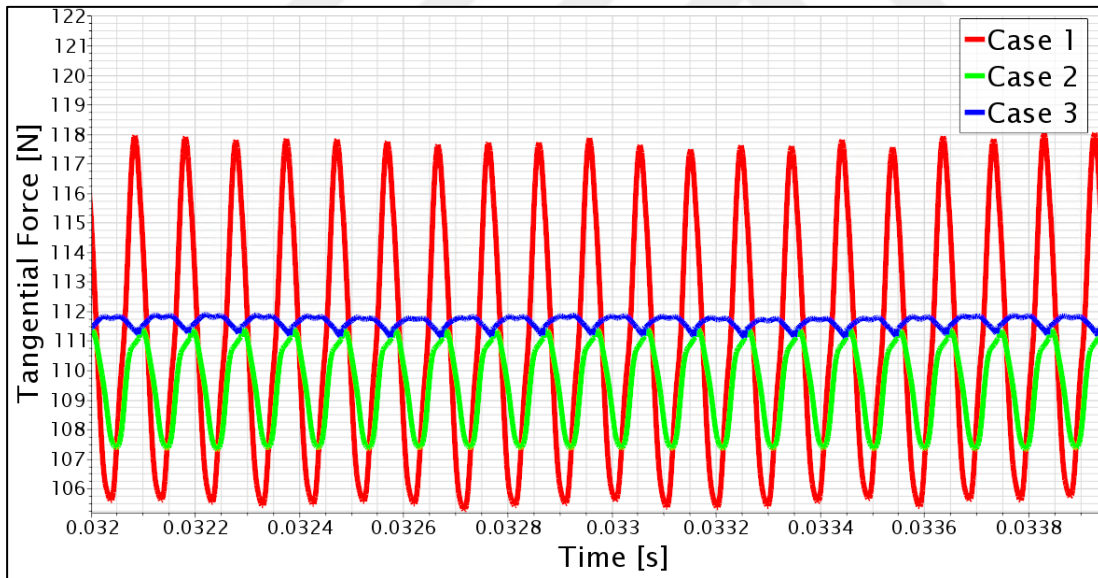


Figure 5.23: Tangential force exerted on rotor blade surface.

Unsteady pressure fluctuations were also calculated locally on the rotor blade surface by assigning six pressure probes on suction and pressure surface of the rotor blade. Three probes were located at 50, 75 and 80% chord length of the rotor blade at suction side surface. The other three probes are located in pressure side in the same way as seen in Figure 5.24. These probes collect the transient pressure data on their locations.

After collecting the desired pressure data, averaged normalized unsteady pressure at these points are calculated as shown below.

$$\bar{p}_{normalized} = \frac{\sum_{i=1}^{i=n} \frac{p(t)_i - \bar{p}}{p_{0,ref}}}{n} \quad (5.7)$$

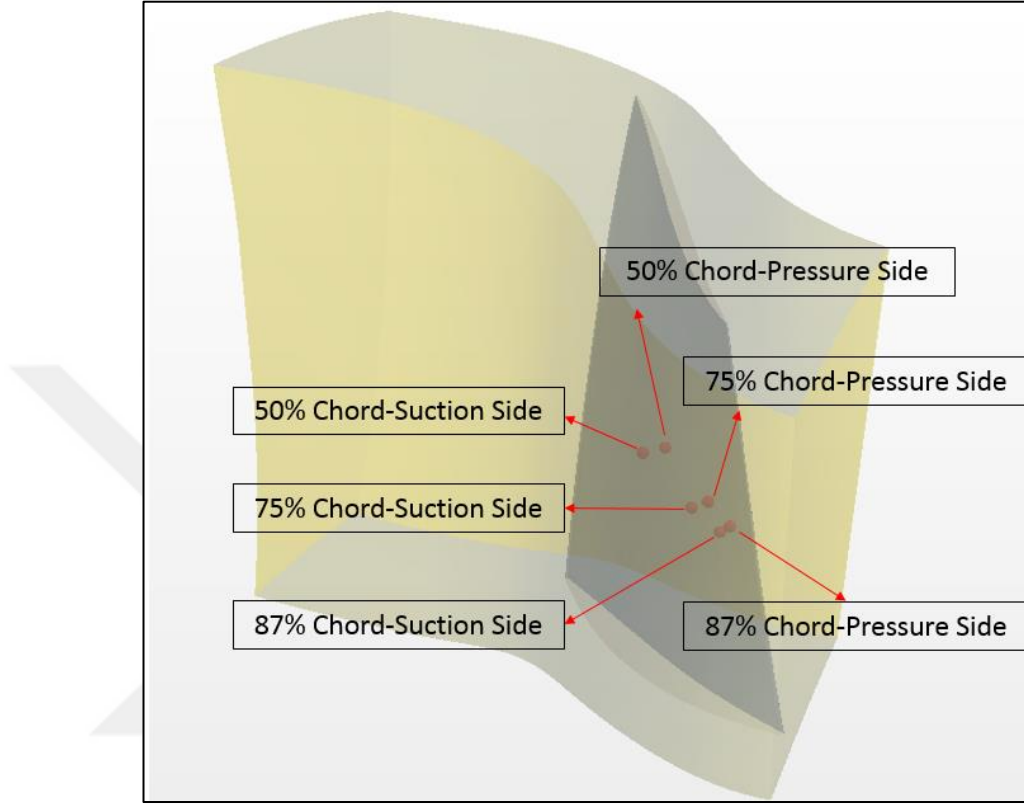


Figure 5.24: Pressure probes on rotor blade surface.

Pressure probes at pressure side of the rotor blades are plotted in Figure 5.25, Figure 5.26 and Figure 5.27. As seen, as the axial gap increases the amplitude of the unsteady pressure decreases. From 50% to 87% chord length the amplitude of the unsteady pressure increases since the stator potential field is more effective in the trailing edge of the rotor blade. Table 5.10 shows the averaged normalized unsteady pressure values at the pressure probes on the pressure side of the rotor blade. This value is directly related to amplitude of the pressure signal.

Table 5.10: Pressure side averaged normalized unsteady pressure.

Case	50% Chord	75% Chord	87% Chord
Case 1	0.0244	0.0477	0.0393
Case 2	0.0048	0.0159	0.0246
Case 3	0.0011	0.0050	0.0051

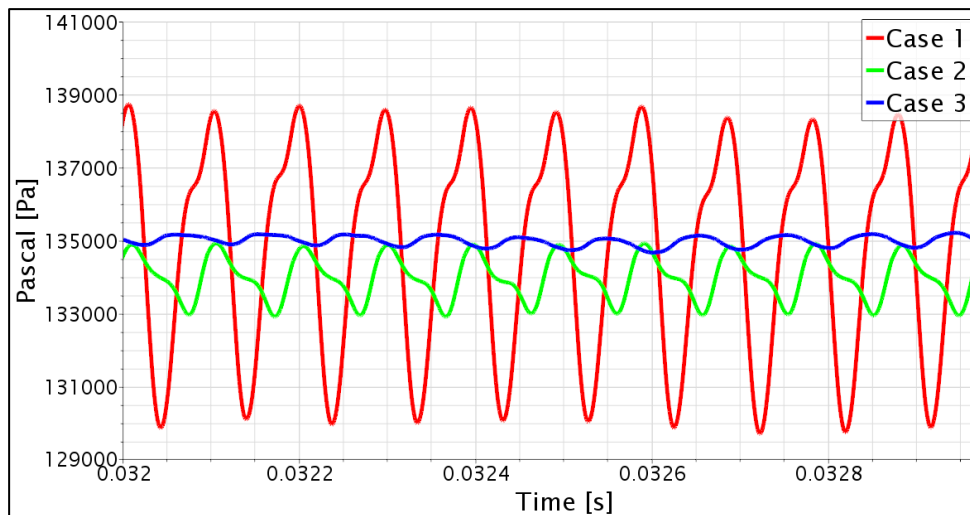


Figure 5.25: Pressure side 50% chord probe.

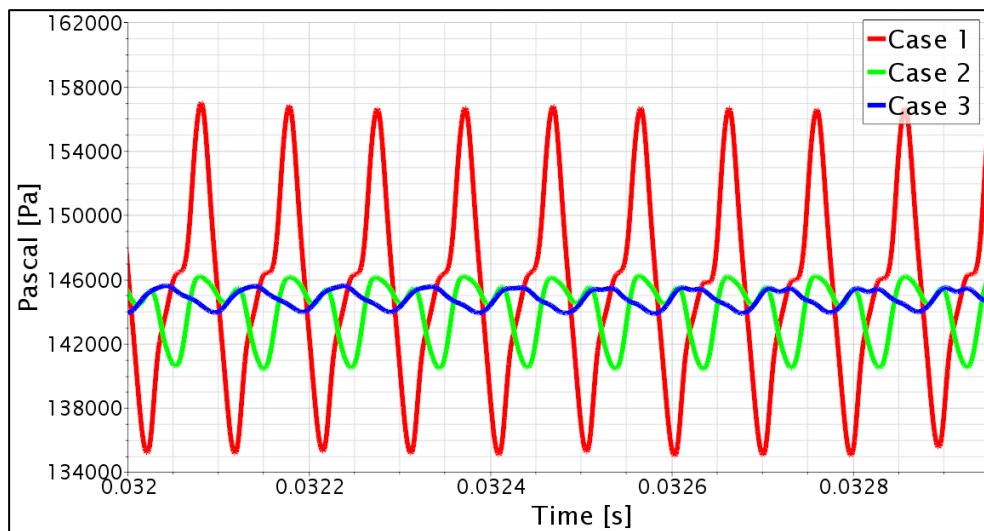


Figure 5.26: Pressure side 75% chord probe.

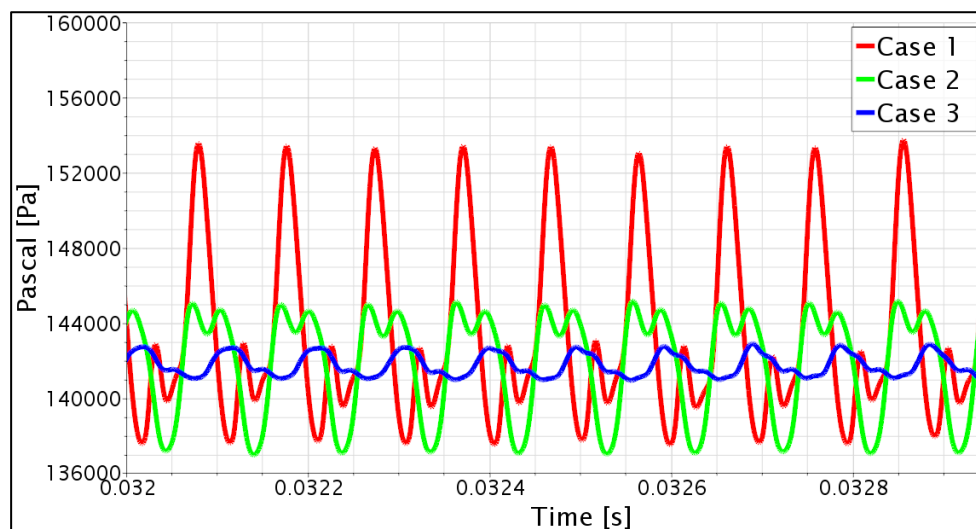


Figure 5.27: Pressure side 87% chord probe.

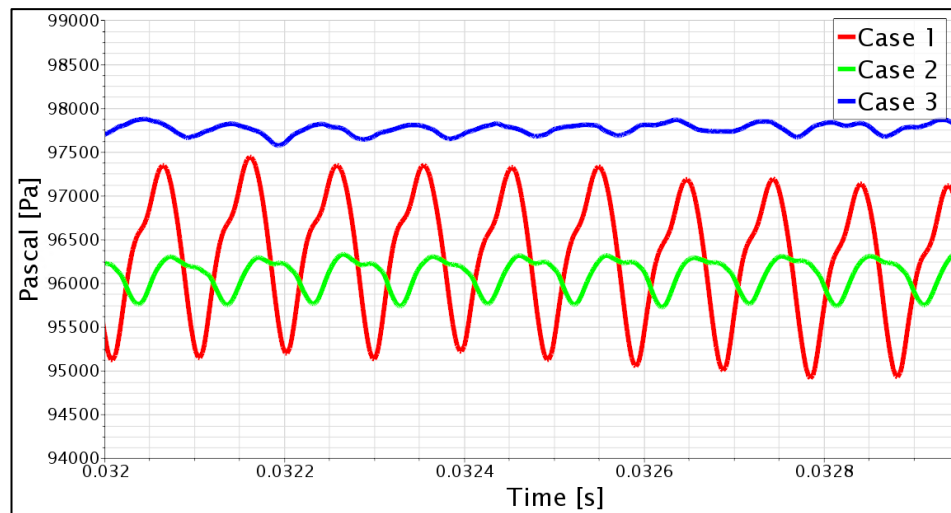


Figure 5.28: Suction side 50% chord probe.

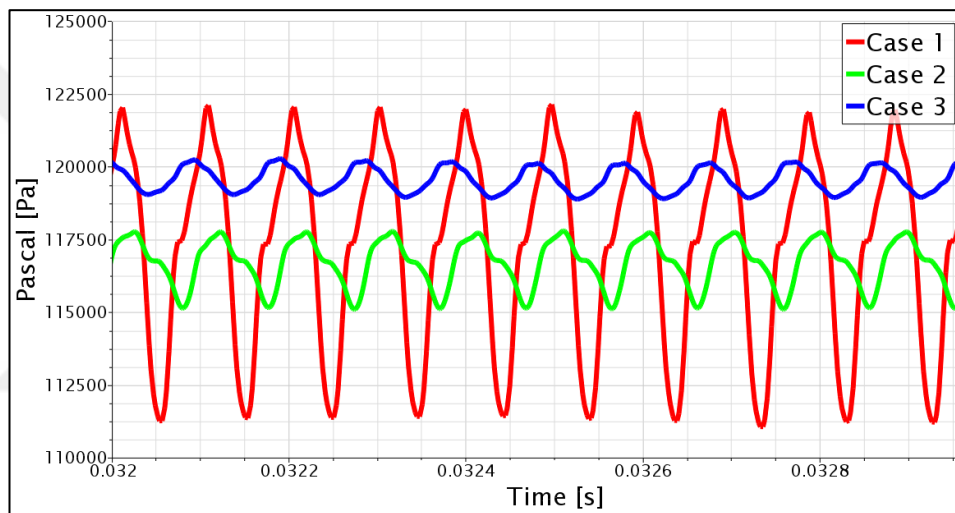


Figure 5.29: Suction side 75% chord probe.

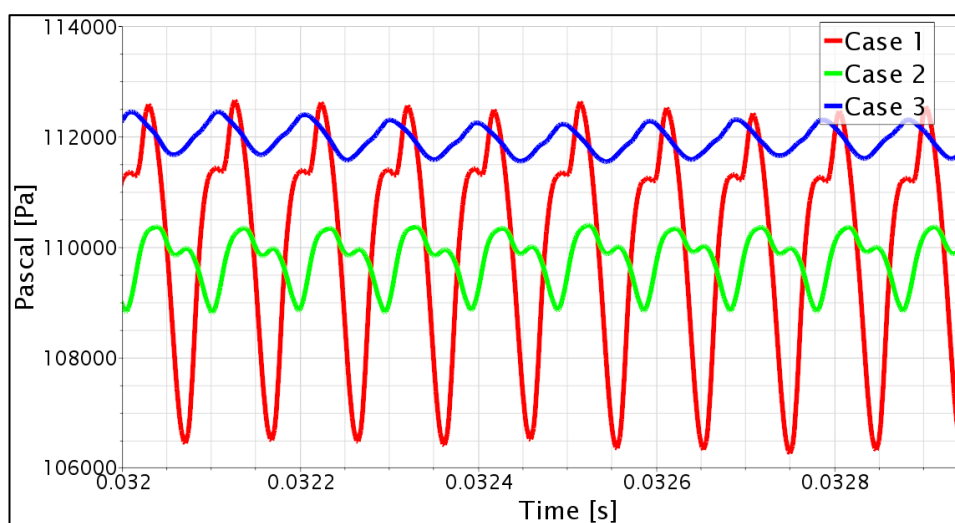


Figure 5.30: Suction side 87% chord probe.

Suction side pressure plots are shown in Figure 5.28, Figure 5.29 and Figure 5.30. Same trend is observed in these plots as done in pressure side probes. The averaged normalized unsteady pressure values are tabulated in Table 5.11.

Table 5.11: Suction side averaged normalized unsteady pressure.

Case	50% Chord	75% Chord	87% Chord
Case 1	0.0060	0.0282	0.0166
Case 2	0.0015	0.0066	0.0037
Case 3	0.0005	0.0037	0.0021

Star CCM+ enables user to use Fourier transform method on transient pressure data. User can process the time history pressure data to obtain different amplitude functions at different frequencies. The amplitude functions available in Star CCM+ are power spectral density and sound pressure level. The power spectral density is the frequency-domain description of the signal power. The sound pressure level (SPL) is a logarithmic measure of the effective sound pressure relative to a reference value and measured in decibels. It requires the reference sound pressure which is 2.0E-5 for air.

$$SPL = 10 \log \left(\frac{p_{rms}^2}{p_{ref}^2} \right) \quad (5.8)$$

Calculating these values on the rotor blade surface at different blade passing frequencies is a good demonstration how the stator potential field is effective on the rotor blade and at what level noise is radiated from the rotor surface. In this study, these parameters were calculated for rotor suction and pressure surface at first, second and third blade passing frequencies.

Power spectral density shows the energy of the pressure fluctuations as a function of the blade passing frequency. From Figure 5.31 to Figure 5.36 the power spectral density on rotor pressure side and suction side are illustrated. As seen in figures, the energy content of the pressure fluctuations degrades as the axial gap increases, since the effect of the stator vane potential field on the rotor blade decreases. Moreover, the influence of the stator vane field is denser on the trailing edge of the rotor and it loses its strength towards rotor leading edge. Energy content of the pressure fluctuations are apparently higher on the pressure side of the blades compared to suction side. This may stem from the alignment of the rotor blades and accordingly the rotation direction.

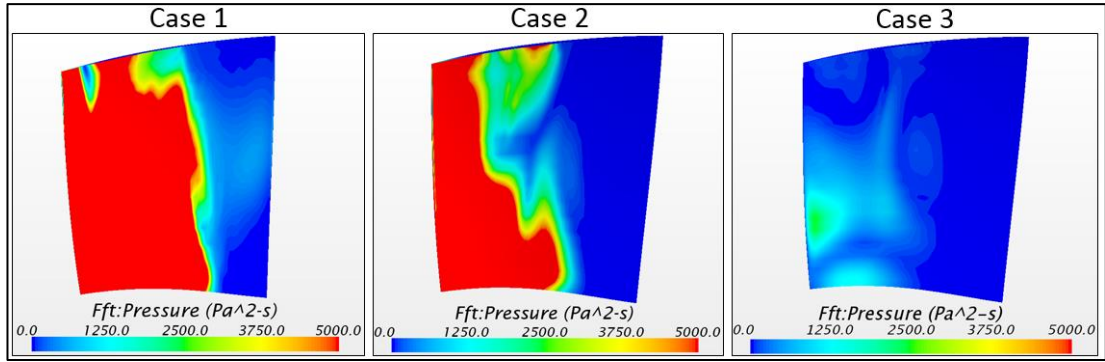


Figure 5.31: Power spectral density on pressure side at 1st BPF.

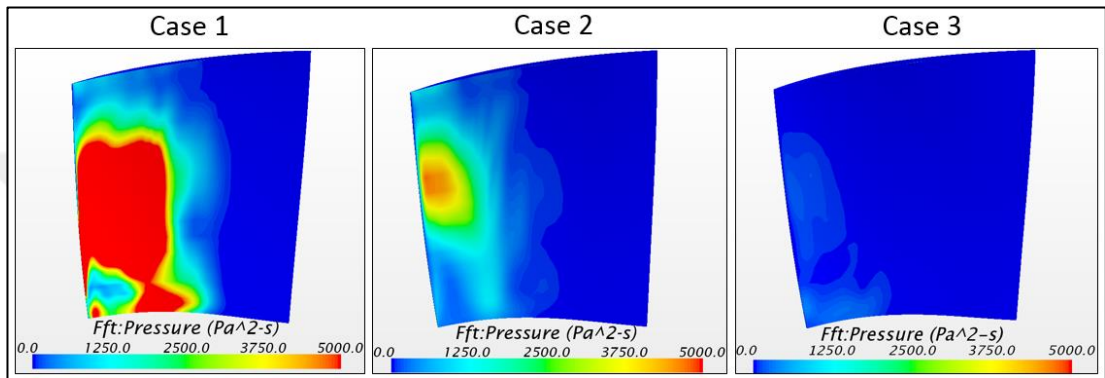


Figure 5.32: Power spectral density on pressure side at 2nd BPF.

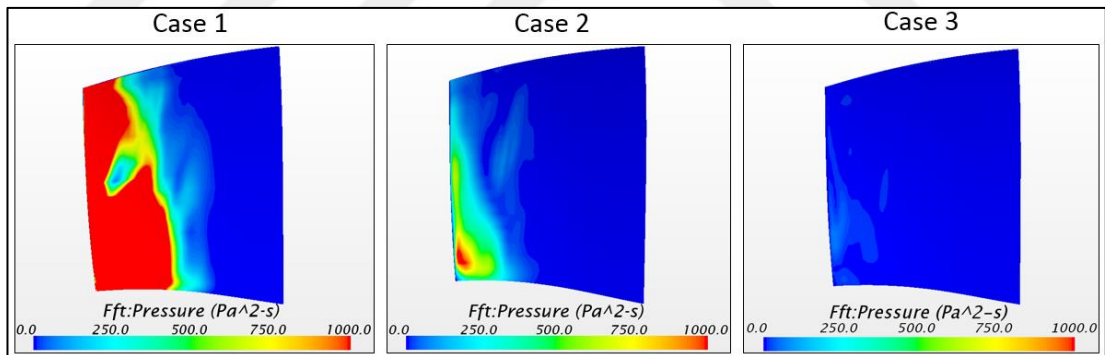


Figure 5.33: Power spectral density on pressure side at 3rd BPF.

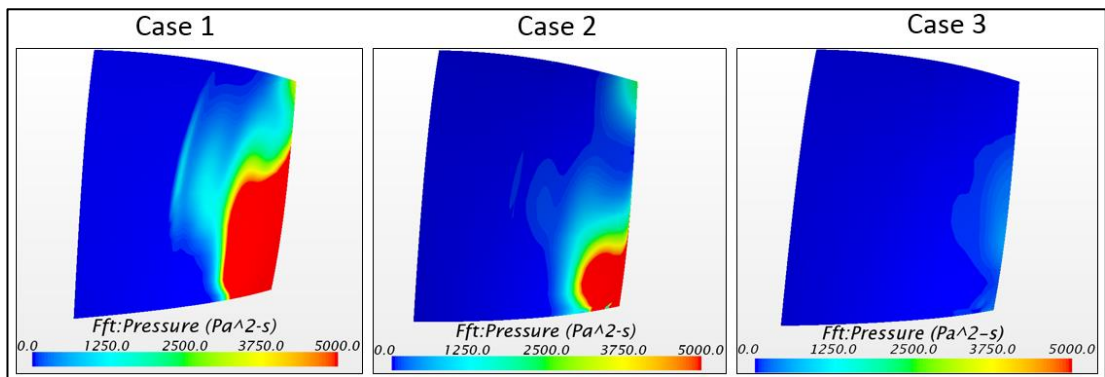


Figure 5.34: Power spectral density on suction side at 1st BPF.

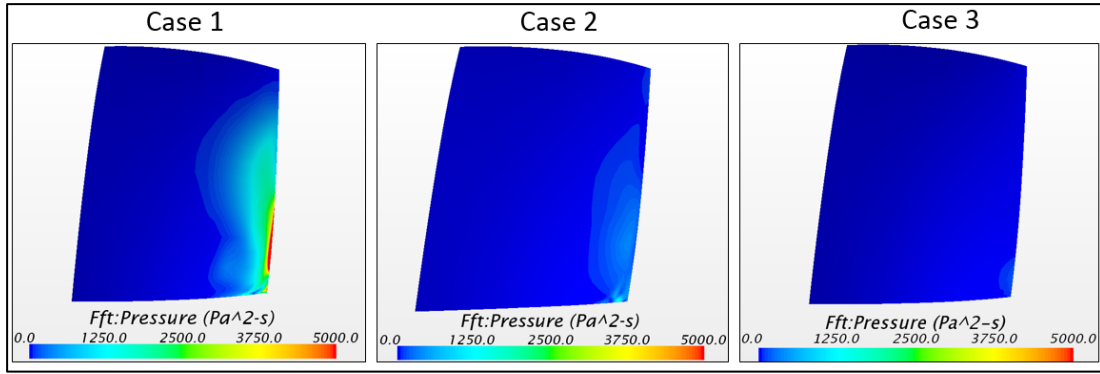


Figure 5.35: Power spectral density on suction side at 2nd BPF.

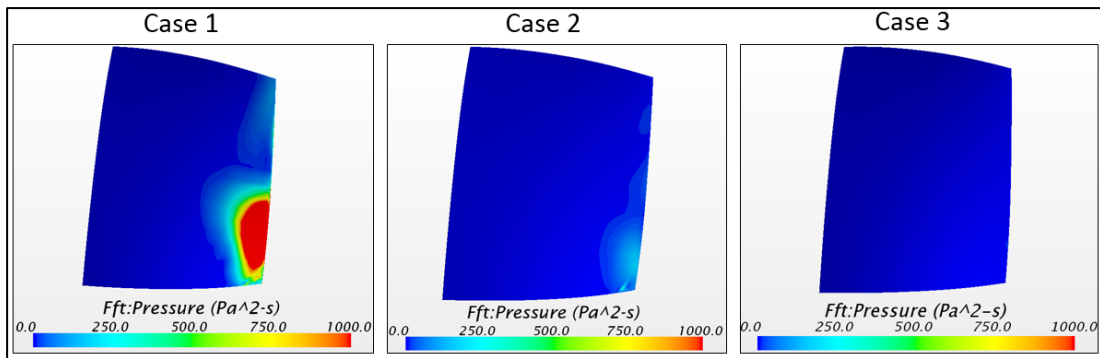


Figure 5.36: Power spectral density on suction side at 3rd BPF.

Sound pressure level values at first three BPF on rotor pressure and suction surfaces are visualized from Figure 5.37 to Figure 5.42. Sound pressure level on the rotor surfaces gives a preliminary idea of the amplitude of the radiated tonal noise at certain frequencies. As seen, the radiated noise amplitude decreases as the axial gap increases. Moreover, as in power spectral density figures, more noise is radiated from the pressure side of the blade since the pressure fluctuations are denser compared to suction side. On the suction side, there is a significant difference in the value of the sound pressure before and after the shock wave.

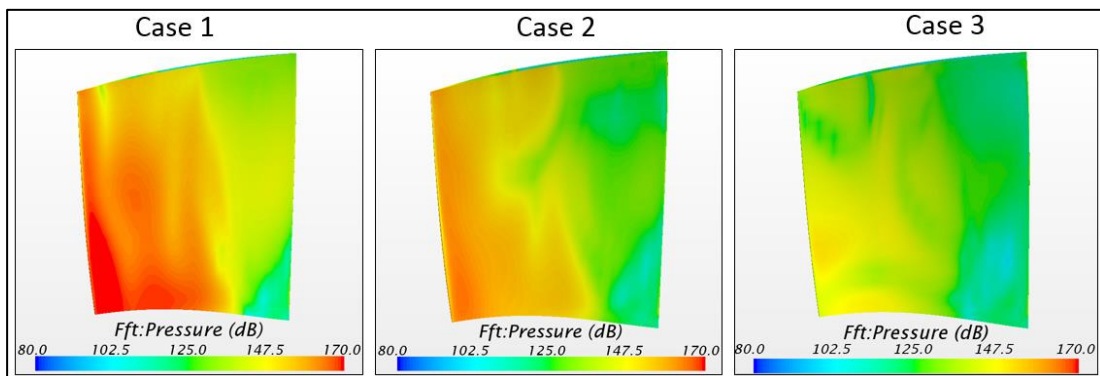


Figure 5.37: Sound pressure level on pressure side at 1st BPF.

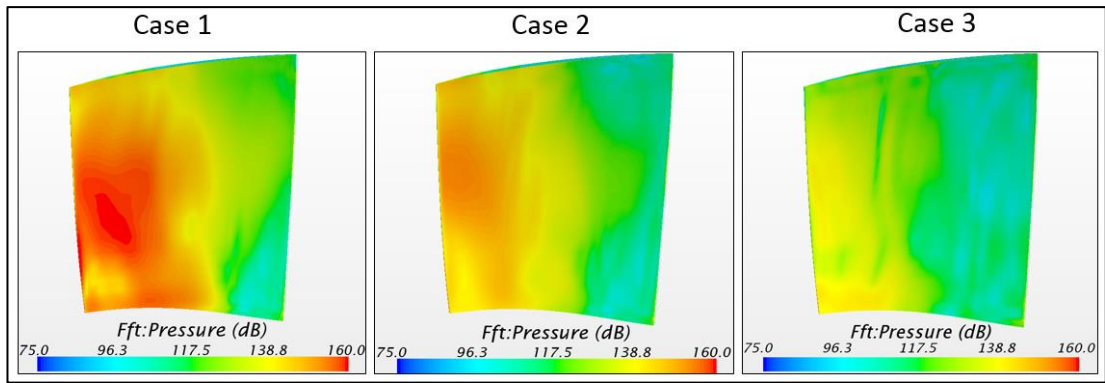


Figure 5.38: Sound pressure level on pressure side at 2nd BPF.

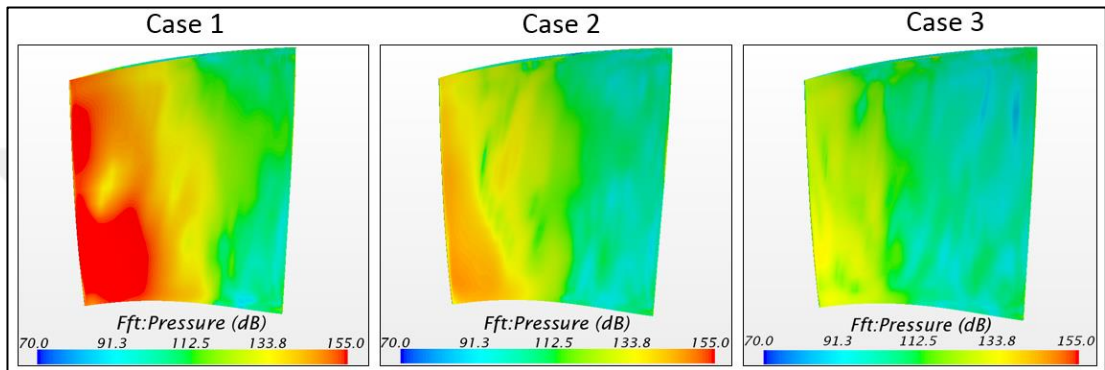


Figure 5.39: Sound pressure level on pressure side at 3rd BPF.

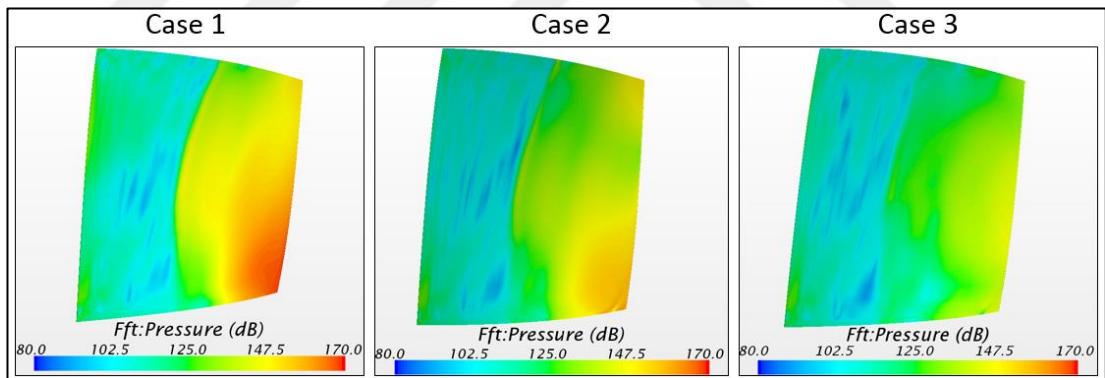


Figure 5.40: Sound pressure level on suction side at 1st BPF.

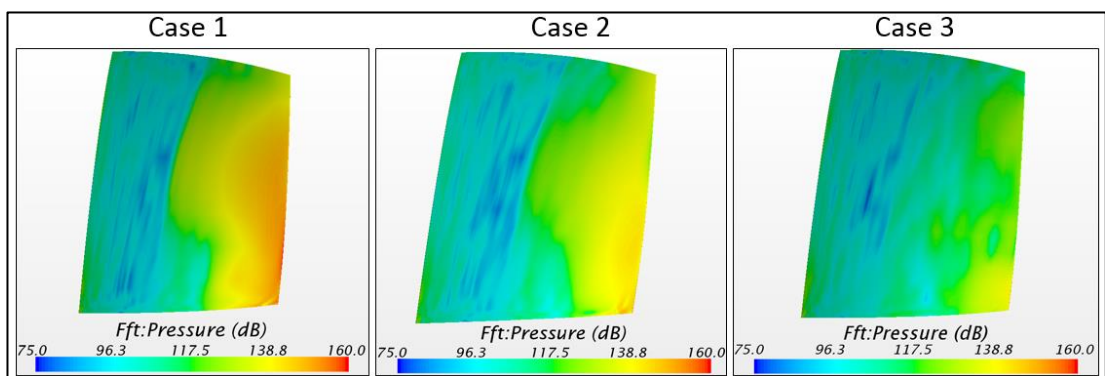


Figure 5.41: Sound pressure level on suction side at 2nd BPF.

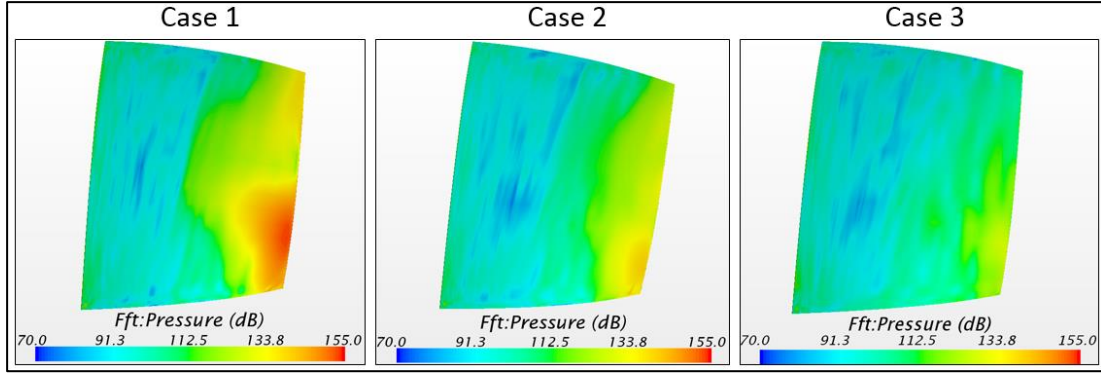


Figure 5.42: Sound pressure level on suction side at 3rd BPF.

5.3.3 Unsteadiness on stator vane surface

Main mechanism of the tonal noise generation by rotor-stator interaction is the interaction of the stator surface with the rotor wake. Due to the rotor wake, the incoming fluid impinge the stator surface with periodically fluctuating velocities and causes lift fluctuations on stator vanes. This generates tonal noise at blade passing frequency of the rotor and its higher harmonics.

Unsteady force components exerted on the stator vane surface are axial, tangential and radial forces. The behaviour of each forces under different axial gap configurations were investigated in this study. Results shows that the magnitude of the unsteady forces on stator surfaces decreases as the axial gap increases, since the rotor wake loses its strength and the upwash velocity fluctuations become smaller. Case 1 is largely affected by the rotor wake interactions, which leads to generate higher tonal noise than the other cases do.

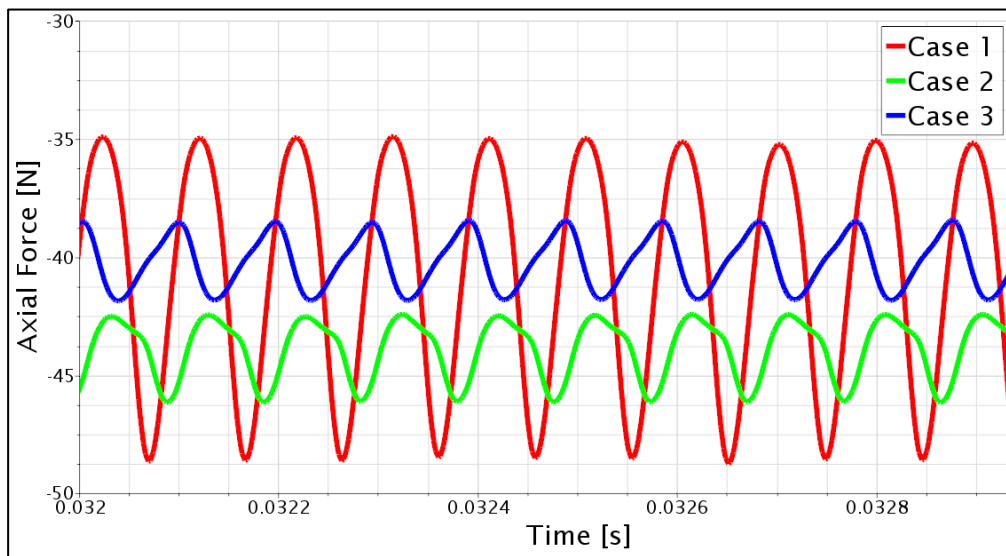


Figure 5.43: Axial force exerted on stator vane surface.

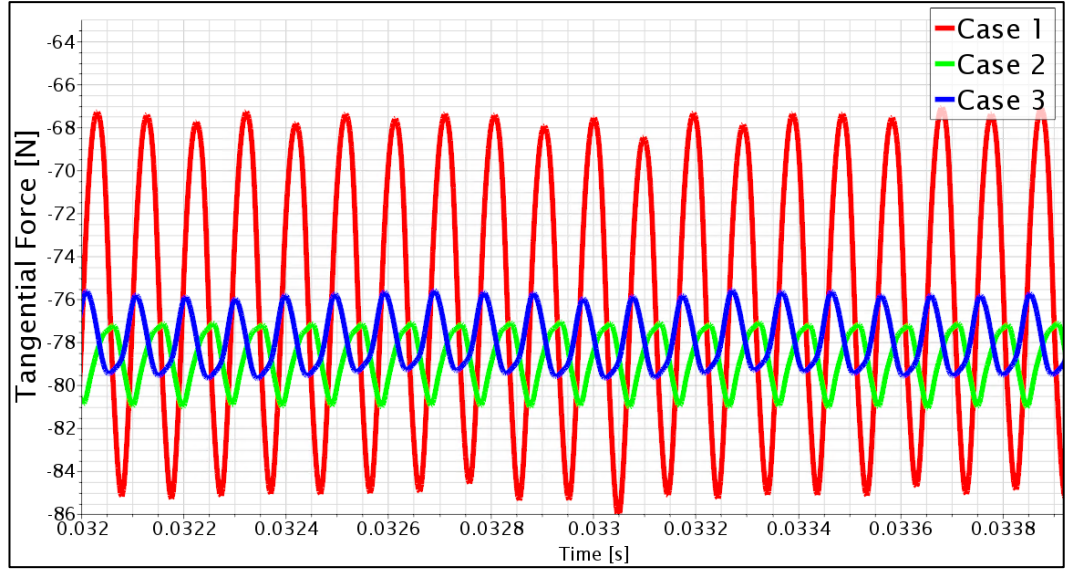


Figure 5.44: Tangential force exerted on stator vane surface.

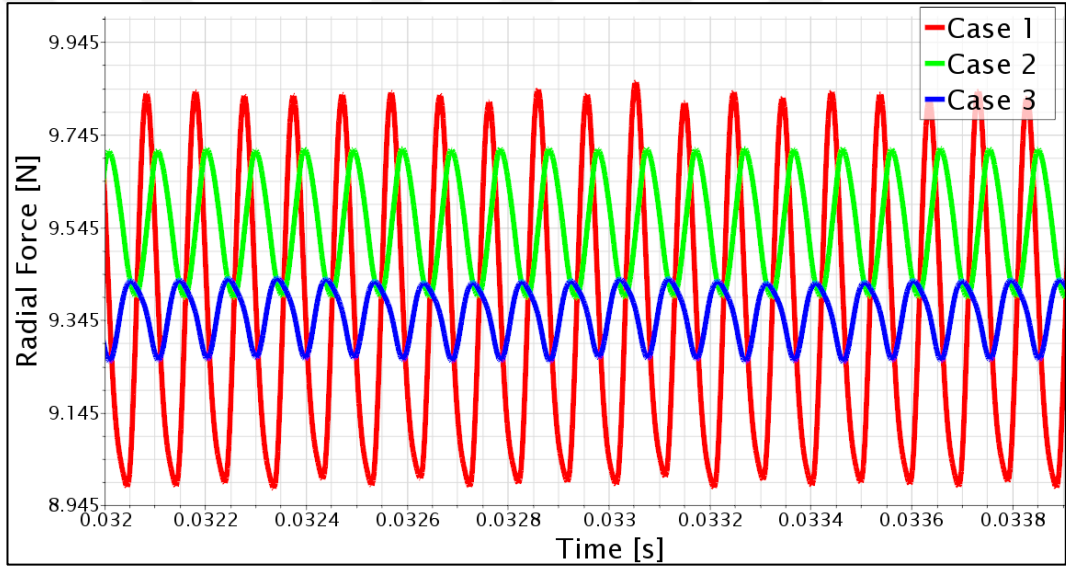


Figure 5.45: Radial force exerted on stator vane surface.

Space-time plots were utilized to examine the unsteady pressure fluctuations on the stator vane surface. Line probes were located from the leading edge to trailing edge of the stator on suction and pressure side of the stator vanes at 10, 25, 50, 75 and 90% span of the stator vane as seen in Figure 5.46. Each of the lines comprises at least 200 pressure probes from leading edge to trailing edge that records the pressure signal at 500 time steps, which corresponds to five blade passing period. Averaged unsteady normalized pressure at each pressure probe was calculated by employing equation 5.7. Figure 5.47 shows the layout of the space-time plot diagram. The vertical axis of the plot is space and starts from the leading edge and ends with the trailing edge. Between the leading edge and trailing edge pressure probes were located with a spacing of Δx .

Horizontal axis represents the time and consists of 500 time steps of Δt which corresponds to 5 periods of blade passing.

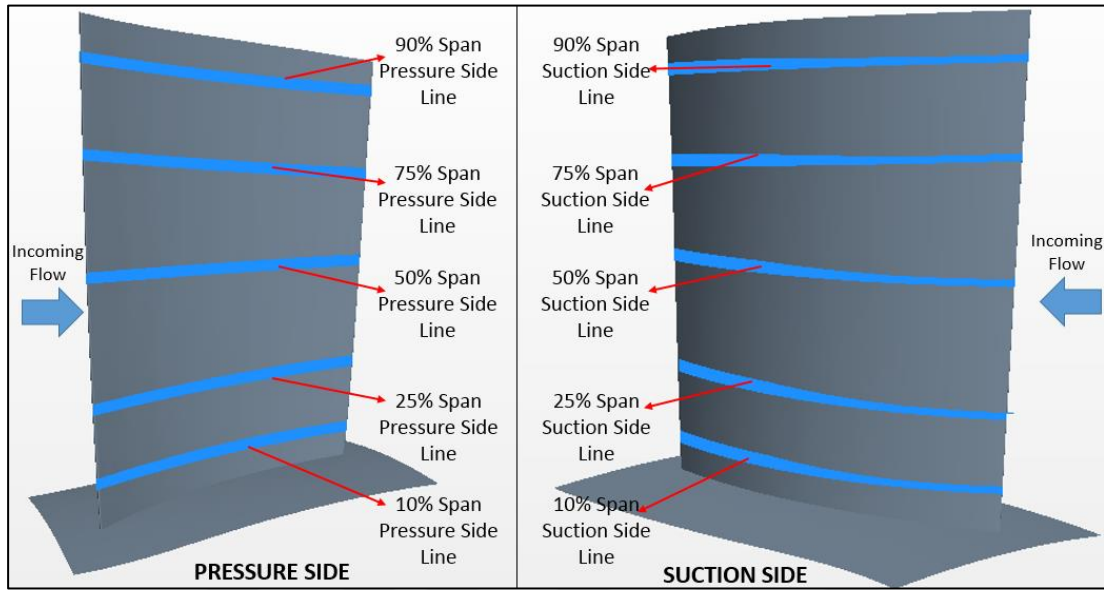


Figure 5.46: Line probes on stator pressure and suction sides.

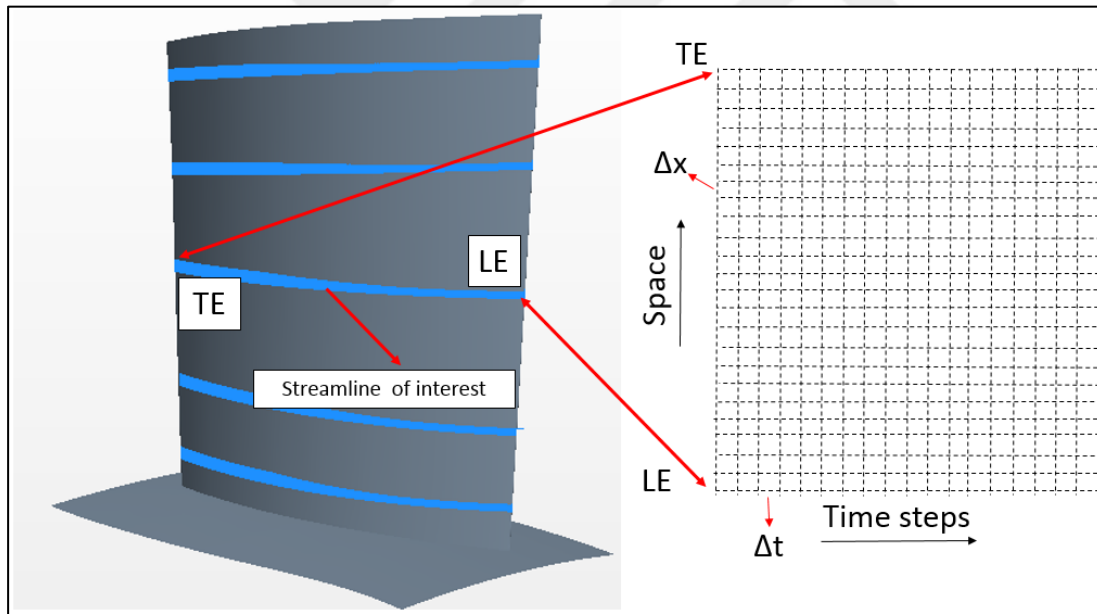


Figure 5.47: Space-time plot diagram.

Space-time plots on pressure side and suction side at 50% span of the stator vane of three cases are illustrated from Figure 5.48 to Figure 5.53. The other plots are given in Appendix B. Apparently, unsteady pressure fluctuations in the case 1 is denser than the two other cases both for pressure side and suction side. As seen, unsteadiness on the pressure side extends to trailing edge of the stator, while this effect is restricted in the leading edge side of the stator vane and it vanishes up to the trailing edge. This may

stem from the wake structure on the each side of the stator vane. The rotor wake is slower on the pressure side compared to suction side, thus more rotor wake are located on the pressure side at the same time from leading edge to trailing edge. This way, the rotor wakes changes the pressure on the pressure side of the stator more than they do on the suction side and this results in higher pressure fluctuations on the pressure side. Moreover, rotational direction of the rotor blade makes the pressure side of the stator vane faces the rotor blade. The pressure value on the pressure side of the stator vane is highly dependent on the position of the rotor blade.

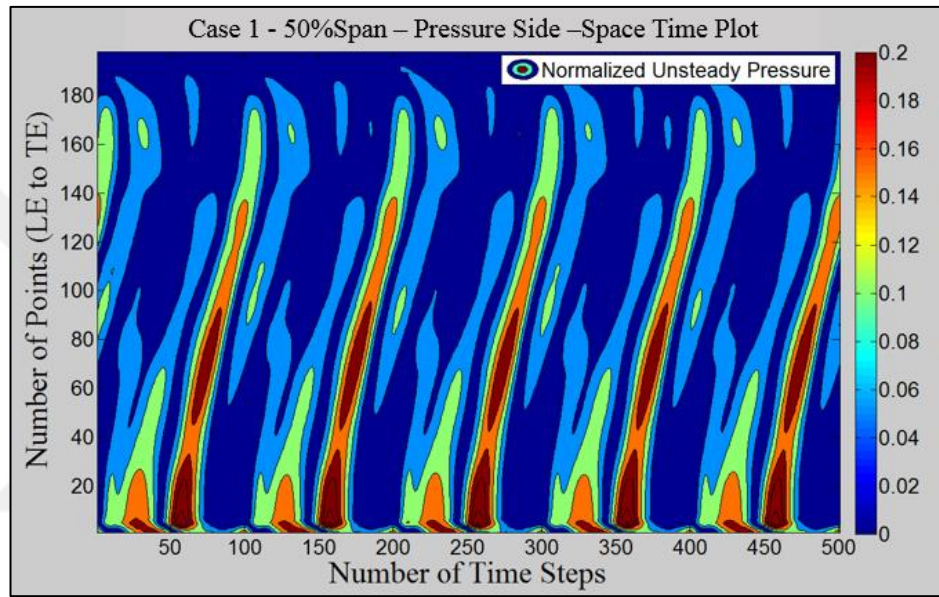


Figure 5.48: Case 1 pressure side space-time plot at 50%span.

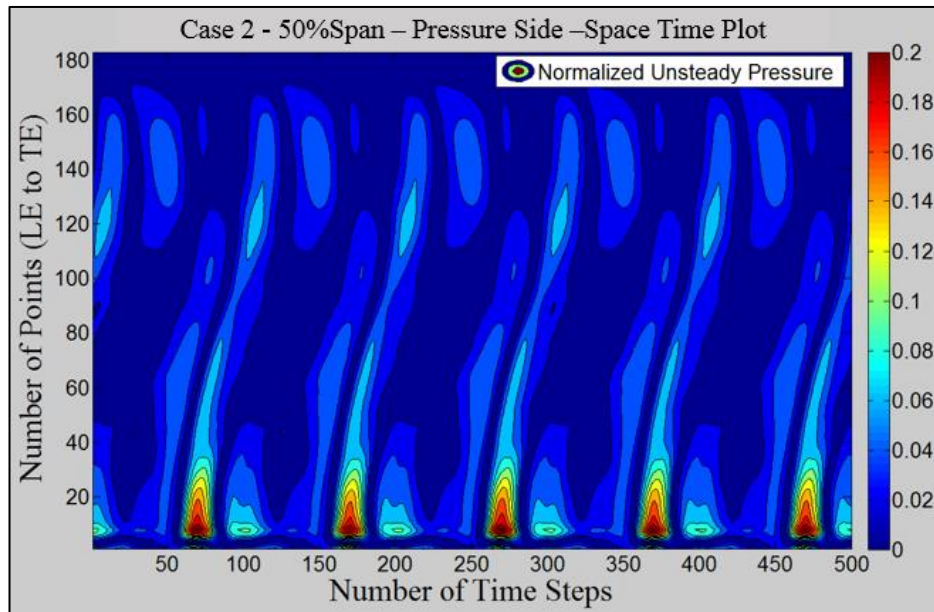


Figure 5.49: Case 2 pressure side space-time plot at 50%span.

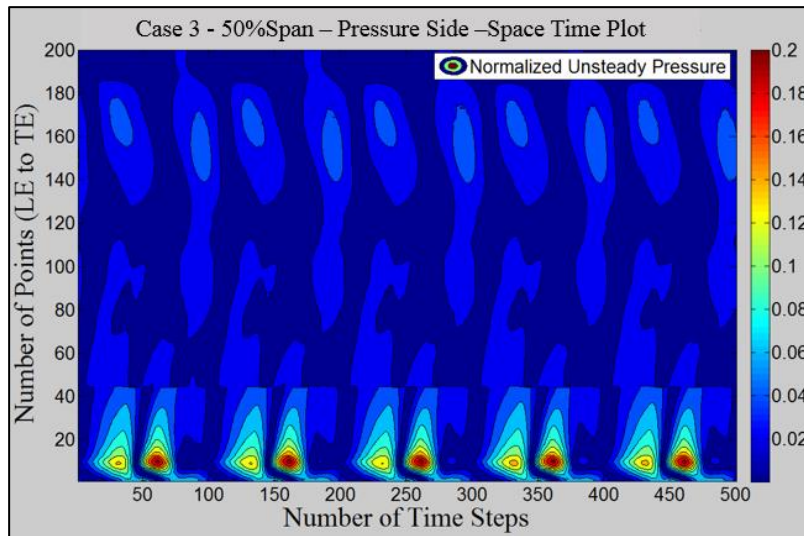


Figure 5.50: Case 3 pressure side space-time plot at 50%span.

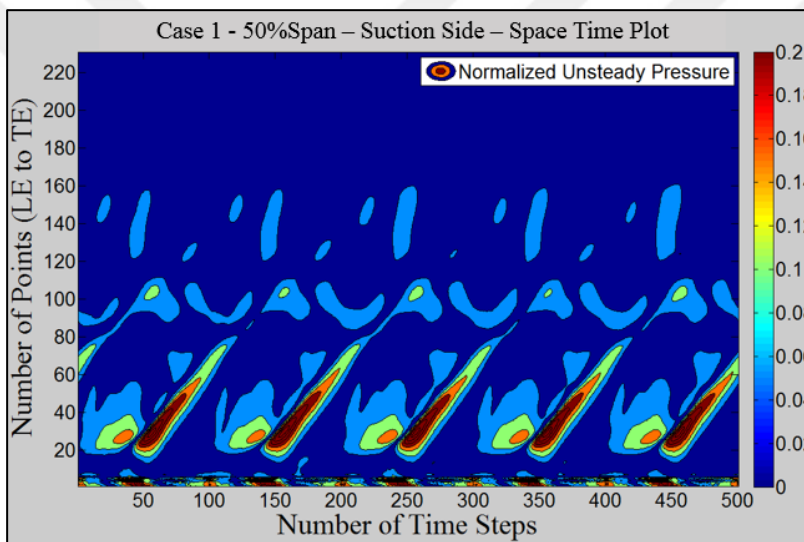


Figure 5.51: Case 1 suction side space-time plot at 50%span.

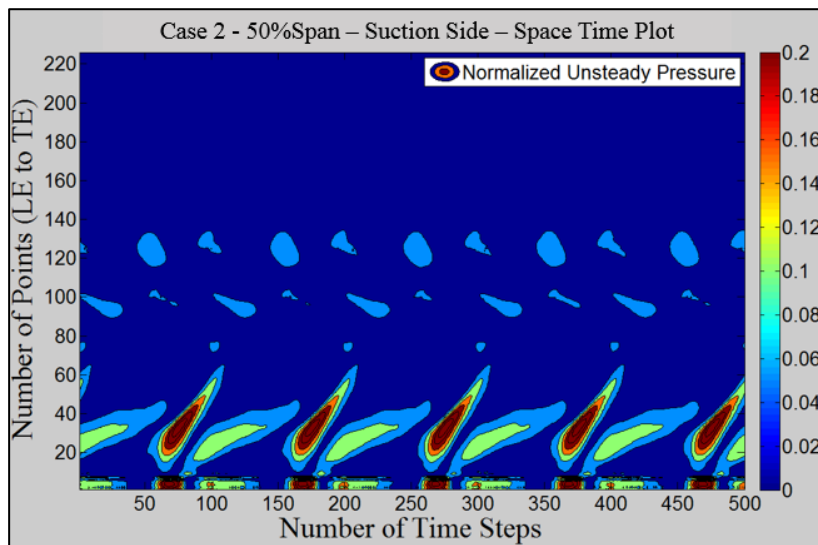


Figure 5.52: Case 2 suction side space-time plot at 50%span.

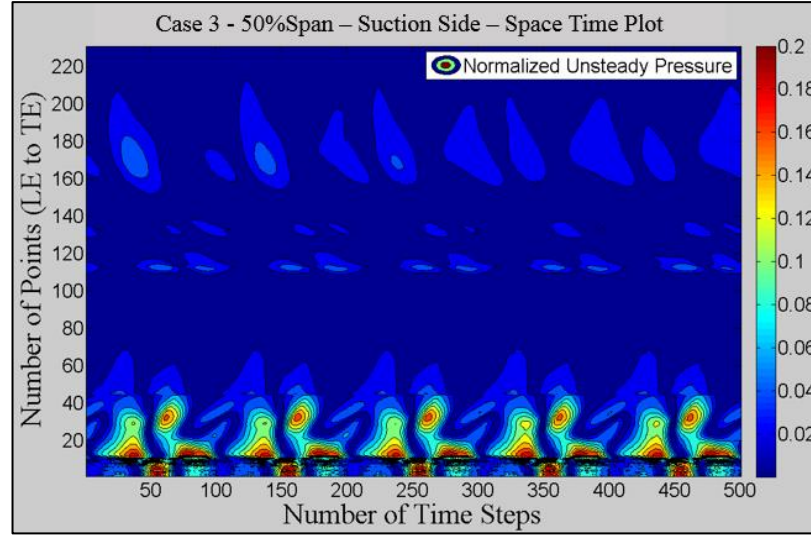


Figure 5.53: Case 3 suction side space-time plot at 50%span.

As resorted in the investigation of the unsteadiness on rotor blade surface, for the stator vane, surface Fourier transform calculations were conducted. Power spectral density and sound pressure levels on the pressure and suction side of the stator vane were calculated at first three blade passing frequency. The pressure data on the pressure side and suction side was recorded during 72-rotor blade passing duration. This pressure data was processed with Fourier transform in order to investigate the unsteadiness on both side of the stator vane.

Power spectral density on the pressure side of the stator vane are compatible with the pressure side calculations of the space-time plots. As the axial gap increases the power of the unsteadiness decreases on stator surface. Moreover, the unsteady pressure fluctuations are densely populated near the shroud region rather than the hub. This is also captured in the space-time plots as seen in Appendix B. This stems from the stronger interaction near the shroud because of the larger wake coming from the rotor.

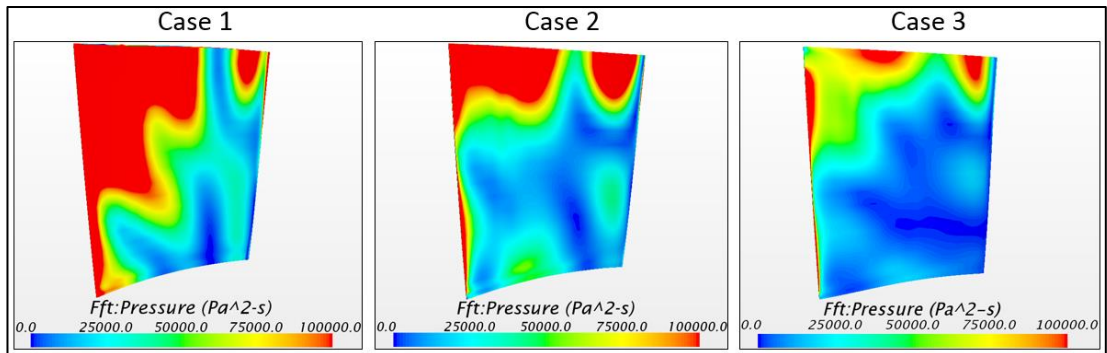


Figure 5.54: Power spectral density on pressure side at 1st BPF.

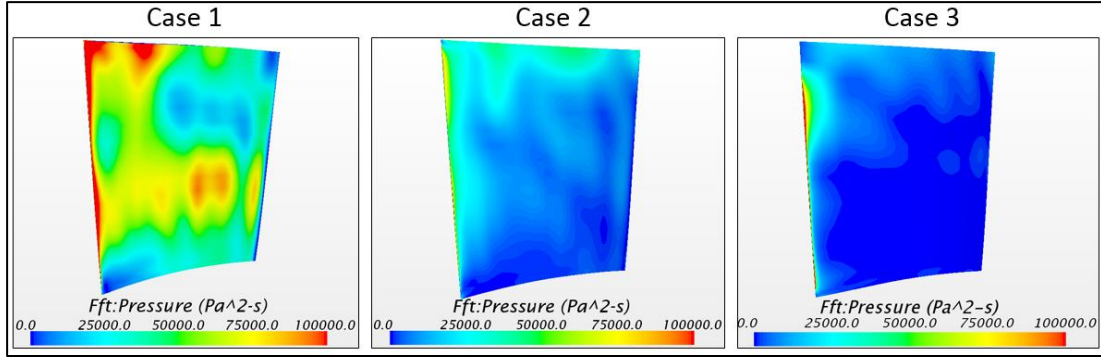


Figure 5.55: Power spectral density on pressure side at 2nd BPF.

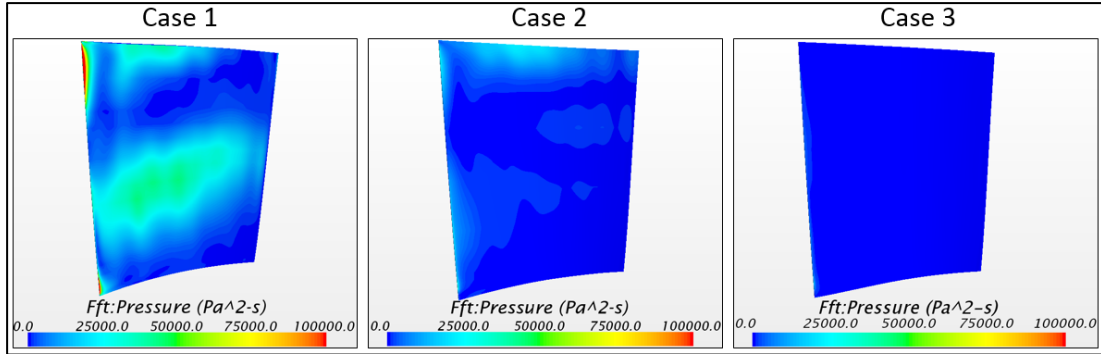


Figure 5.56: Power spectral density on pressure side at 3rd BPF.

On the suction side, the main fluctuation energy content is located near the leading edge for each case. This stems from the expansion waves and separations developed in the boundary layer near the leading edge. As seen in Figure 5.14, separations on the leading edge is stronger for the case 1 and case 2. At higher spans towards the trailing edge denser pressure fluctuation energy regions are present in case 1. This can be seen also in suction side space-time plots at 90% and 75% span in Appendix B. Moreover, the power density of the pressure fluctuations becomes smaller at higher frequencies.

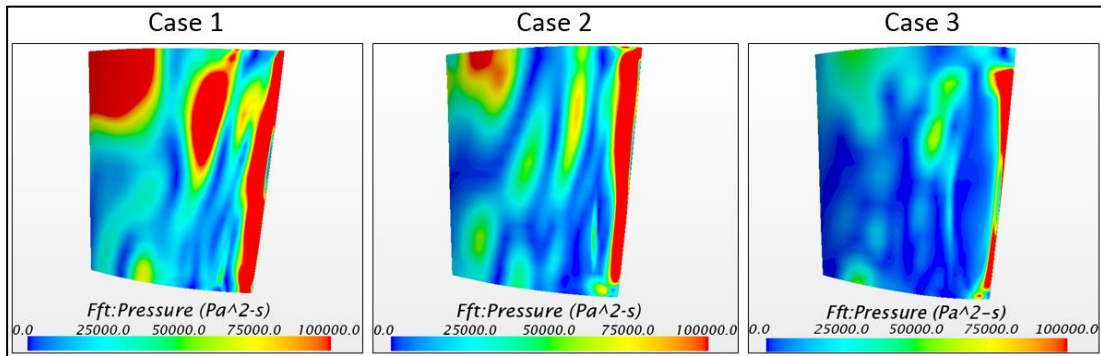


Figure 5.57: Power spectral density on suction side at 1st BPF.

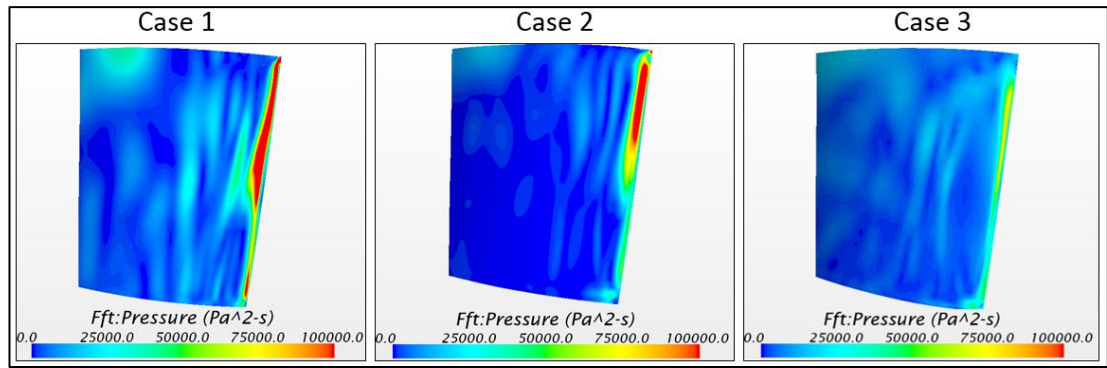


Figure 5.58: Power spectral density on suction side at 2nd BPF.

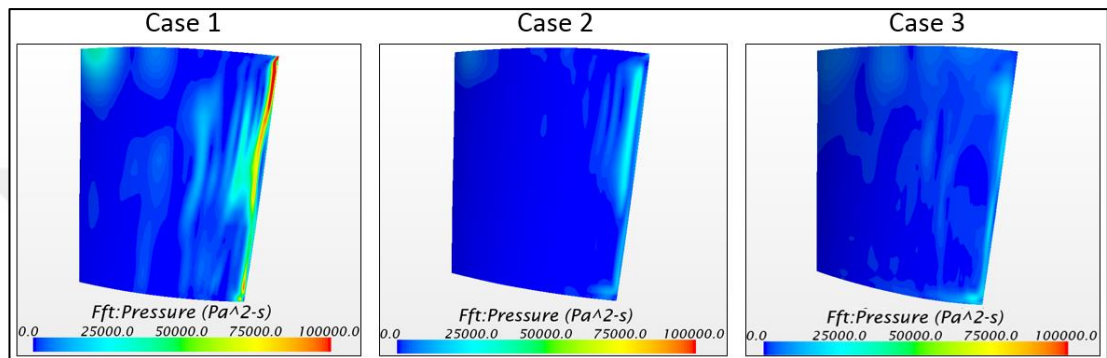


Figure 5.59: Power spectral density on suction side at 3rd BPF.

Sound pressure level results calculated on pressure and suction side of the stator vane are compatible with the power spectral density contours. Sound pressure level reaches its maximum value where the densest power spectral density is present on stator surface. As seen, larger area on the pressure side is effective for radiating acoustic waves with higher noise level content than the suction side. The noise content within the radiated acoustic wave decreases as the axial gap and the frequency increases. Moreover, compared to the rotor blade SPL plots, the stator vane contributes higher content on the noise level of the compressor stage.

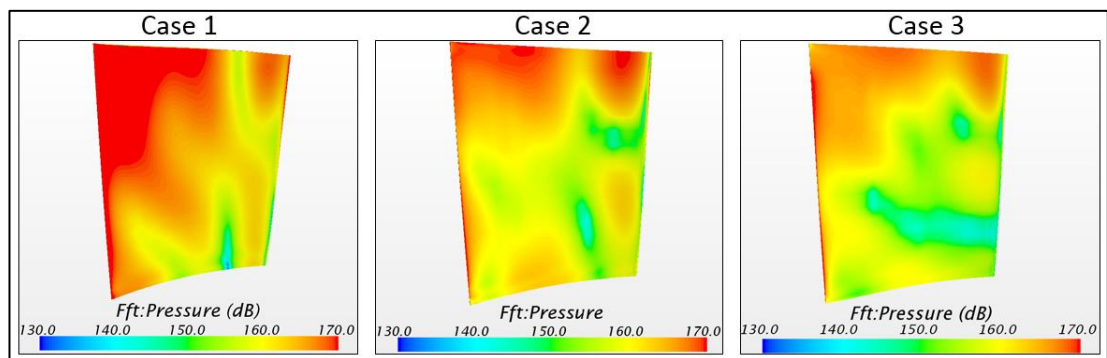


Figure 5.60: Sound pressure level on pressure side at 1st BPF.

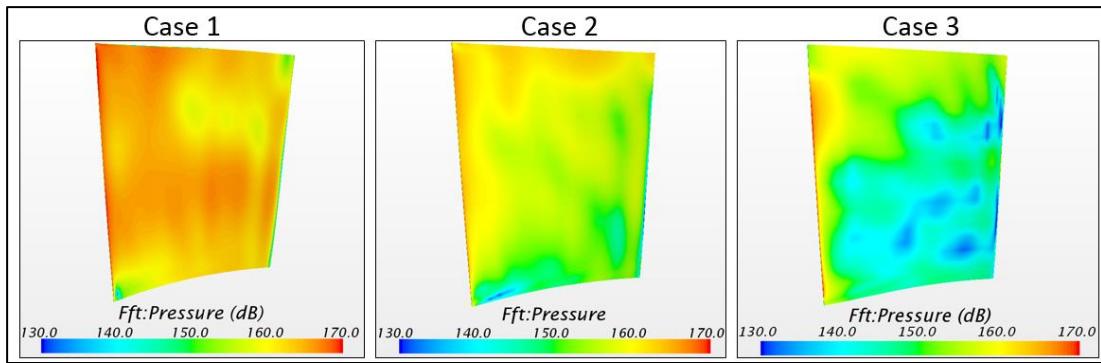


Figure 5.61: Sound pressure level on pressure side at 2nd BPF.

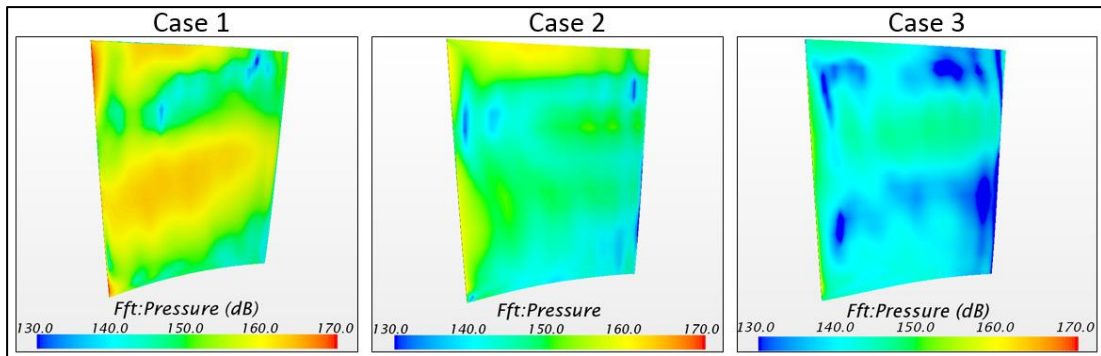


Figure 5.62: Sound pressure level on pressure side at 3rd BPF.

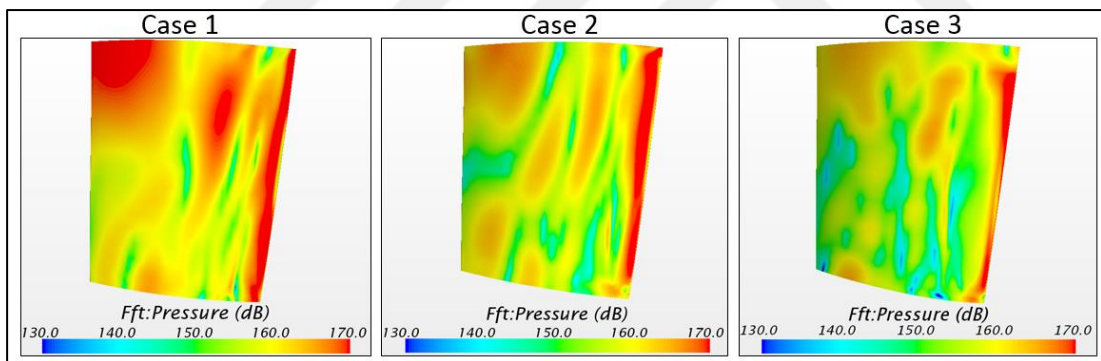


Figure 5.63: Sound pressure level on suction side at 1st BPF.

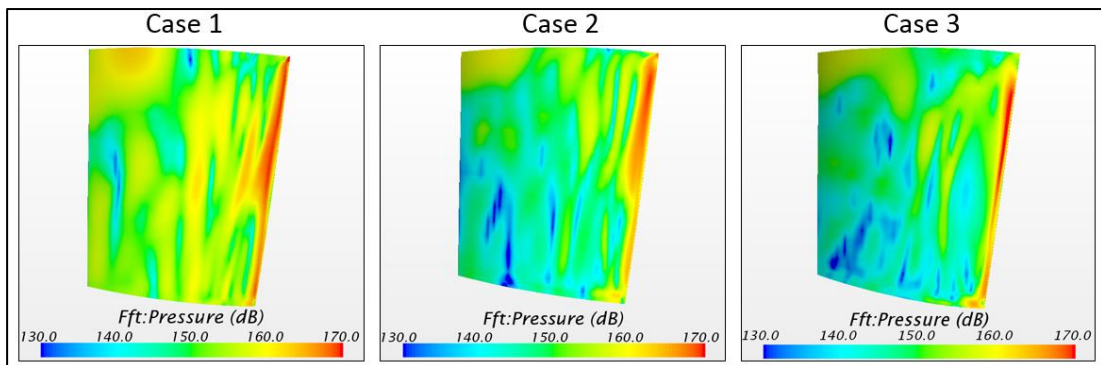


Figure 5.64: Sound pressure level on suction side at 2nd BPF.

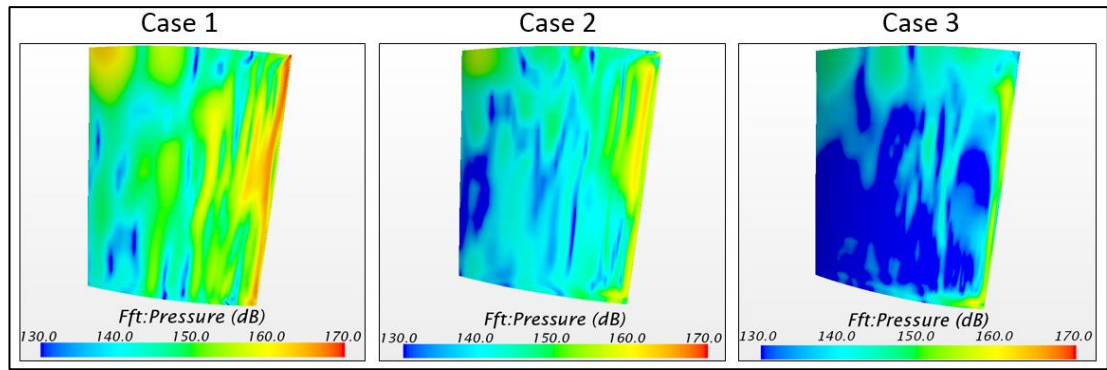


Figure 5.65: Sound pressure level on suction side at 3rd BPF.

5.4 Noise Investigation

In this study, the tonal noise generation within the transonic compressor stage was investigated in two steps. The first one is the near-field noise investigation, which is conducted in the fluid domain adjacent to the dipole noise sources on rotor blade and stator vane surface. Near-field noise calculation is directly extracted from the CFD solution. The second one is the far-field noise study, which is related to propagation of the acoustic waves from source to the observer location in the far-field. This requires using an additional acoustic solver, which uses the CFD data as an input.

5.4.1 Near-field noise investigation

The near-field noise investigation is based on processing the pressure data obtained by CFD solution near the source region. Since the mesh structure is only able to capture the propagated acoustic waves near the sound sources, a couple of pressure probes were located adjacent to the stator vane in order to calculate the noise level on these probes in terms of sound pressure level.

Totally nine pressure probes were located at midspan in each case as they surrounds the stator vane. The distance of the closest probe was 10 mm from the trailing edge of the stator as seen in Figure 5.66. Recorded pressure data on these probes comprises the radiated acoustic waves from sources including stator vane surface, rotor blade surface, hub and shroud surfaces and the turbulent noise. Dominant noise sources are the dipole noise sources on the stator and rotor surfaces. These are tonal and appear in the noise spectrum with peaks. The mesh grid is able to capture the tonal noise up to the fourth harmonics of the blade passing frequency. At higher frequencies, acoustic waves may undergo damping due to the insufficient element size to resolve the corresponding acoustic wavelength at these frequencies.

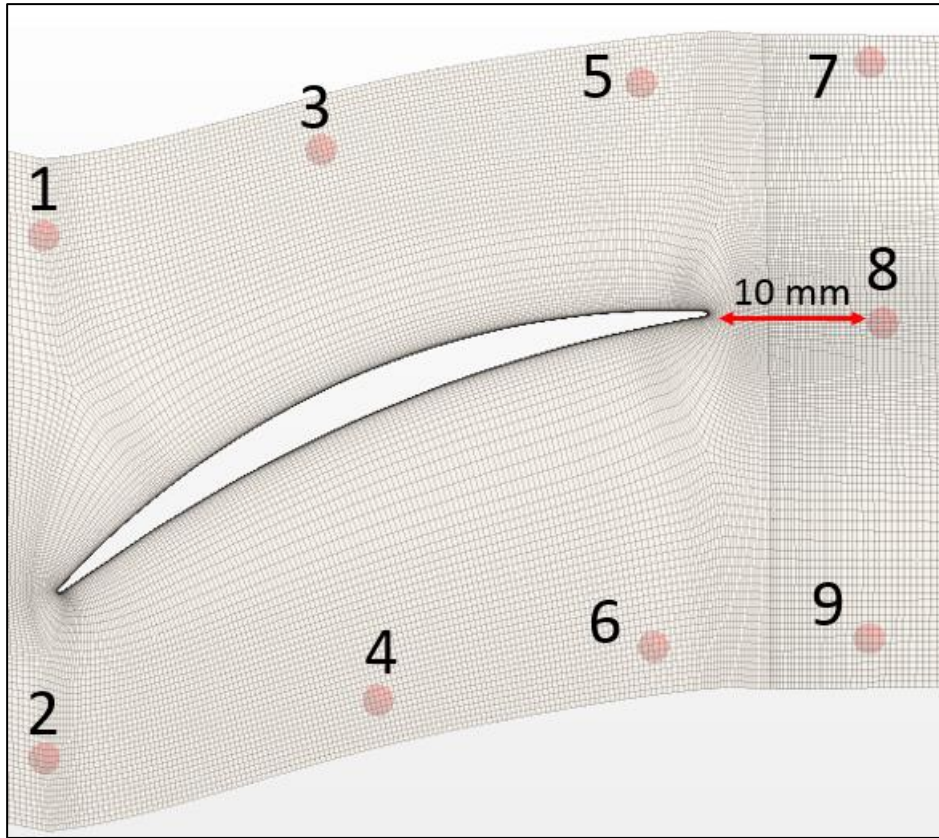


Figure 5.66: Near-field pressure probes at midspan.

The probes record the time history of the absolute pressure. Afterwards, these pressure data are processed with the Fourier transform in order to extract the noise spectrum at these locations. The noise spectrum contains tonal noise and broadband noise in terms of SPL.

Recorded pressure data on probe 1 is shown in Figure 5.67. As seen, the frequency of the unsteady pressure fluctuations is equal to blade passing frequency. The amplitude of the unsteady pressure is larger for the case 1 that has the shortest axial gap between rotor and stator. As the axial gap increases the amplitude of the pressure signal decreases. This proves that the interaction between the rotor and stator significantly increases with the smaller axial gap. Denser interaction results in larger unsteady pressure fluctuations on rotor and stator surface and eventually radiation of the higher amplitude acoustic waves into the flow domain.

Noise spectrum of probe 1 is illustrated in Figure 5.68. As seen, the URANS simulation has successfully captured the discrete noise at blade passing frequencies. Apart from that, it gives an idea about the level of the broadband noise. The tonal noise is plotted up to the 9th harmonics of the blade passing frequency. As mentioned, up to

the fourth harmonics of the blade passing frequency, which corresponds to 40492 Hz, the model is able to capture the tonal noise precisely. However, for the higher frequencies the solution might be damped.

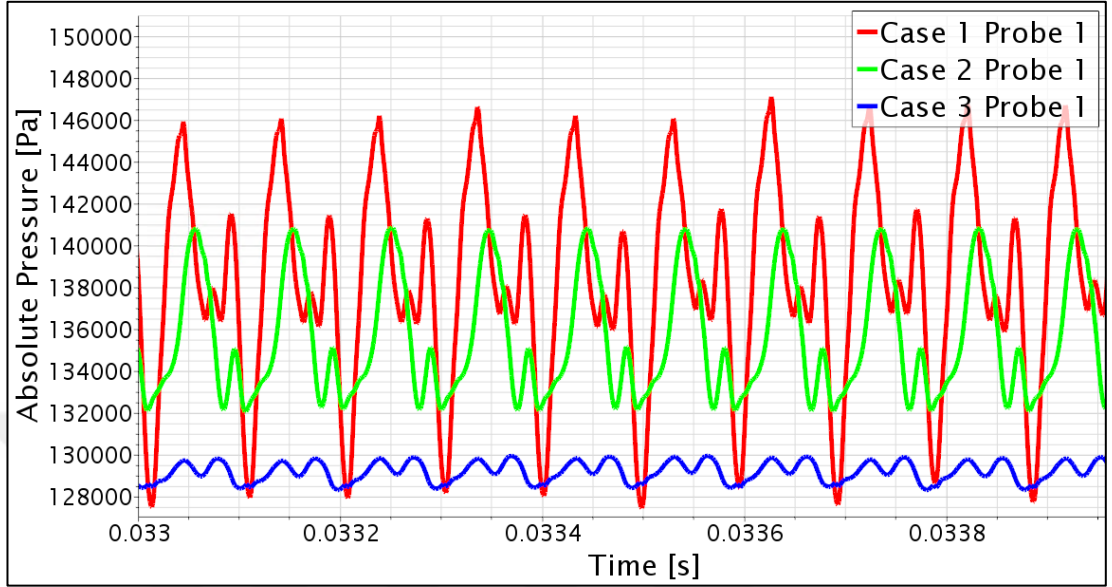


Figure 5.67: Pressure data on probe 1.

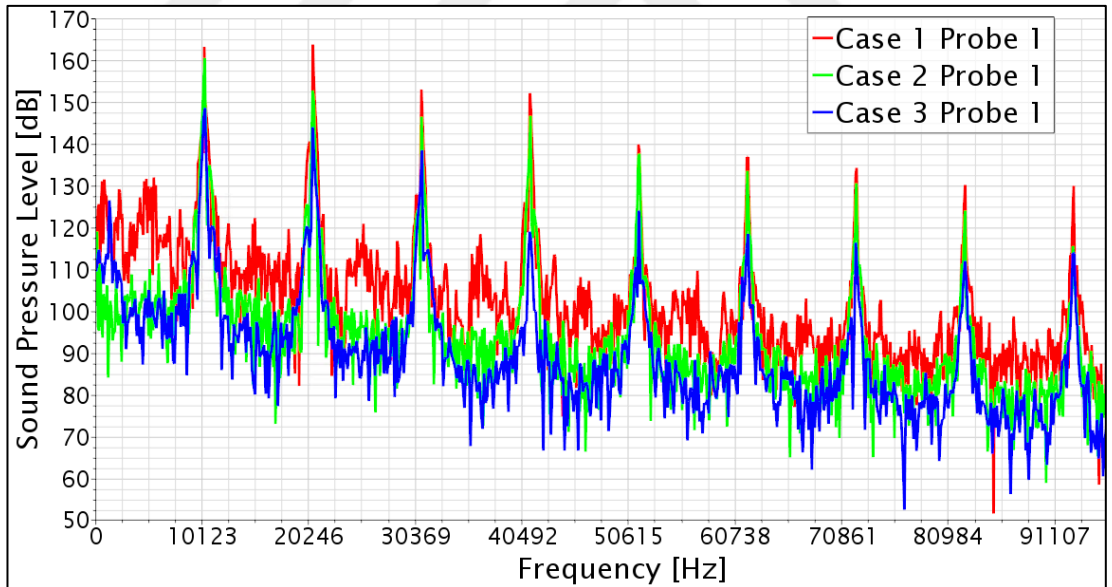


Figure 5.68: Noise spectrum of probe 1.

The noise spectrum of probe 1 shows that the tonal noise and the broadband noise of the case 1 is dominant over the other cases. At first two blade passing frequency the tonal noise reaches its maximum value, for the higher frequencies it decreases gradually. This proves that the smaller axial gap causes higher content of the tonal noise in noise spectrum. The tonal noise values at first three harmonics are also compatible with the SPL investigations on the rotor and stator surfaces. As shown in

previous sections, radiated SPL value from rotor and stator surface is between 150-170 dB for the first three harmonics of BPF.

The broadband noise is shown as continuous distribution on noise spectrum. The broadband noise content also increases as the axial gap decreases. Since the numerical model is not able to capture the turbulent quadrupole noise, which is the main source of the broadband noise, the higher broadband noise content of the case 1 might stem from the incident turbulence of the incoming flow in the upstream of the stator vane. The wake of the rotor blade in case 1 has the highest content of turbulence as seen in Figure 5.16. High velocity fluctuations due to the turbulence content in the wake causes higher broadband noise in the noise spectrum.

The pressure data and the noise spectrum of the probe 2 is illustrated in Figure 5.69 and Figure 5.70 respectively. As recorded in pressure probe 1, case 1 has the highest amplitude of the unsteady pressure, which varies between 140000 Pa, and 160000 Pa. Case 2 follows the case 1 with an unsteady pressure amplitude of 15000 Pa and the minimum amplitude belongs to the pressure data of case 3. Recorded pressure values prove that the unsteadiness of the flow field is the strongest in case 1. This yields in a noise spectrum, which is dominated by the case 1 in terms of tonal and broadband noise.

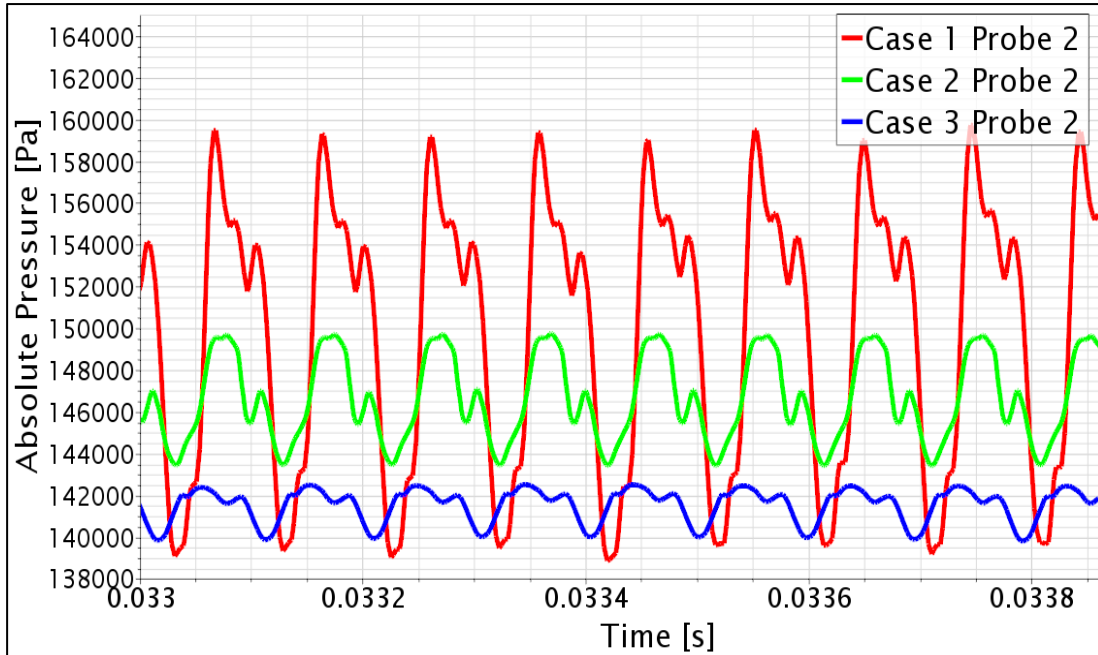


Figure 5.69: Pressure data on probe 2.

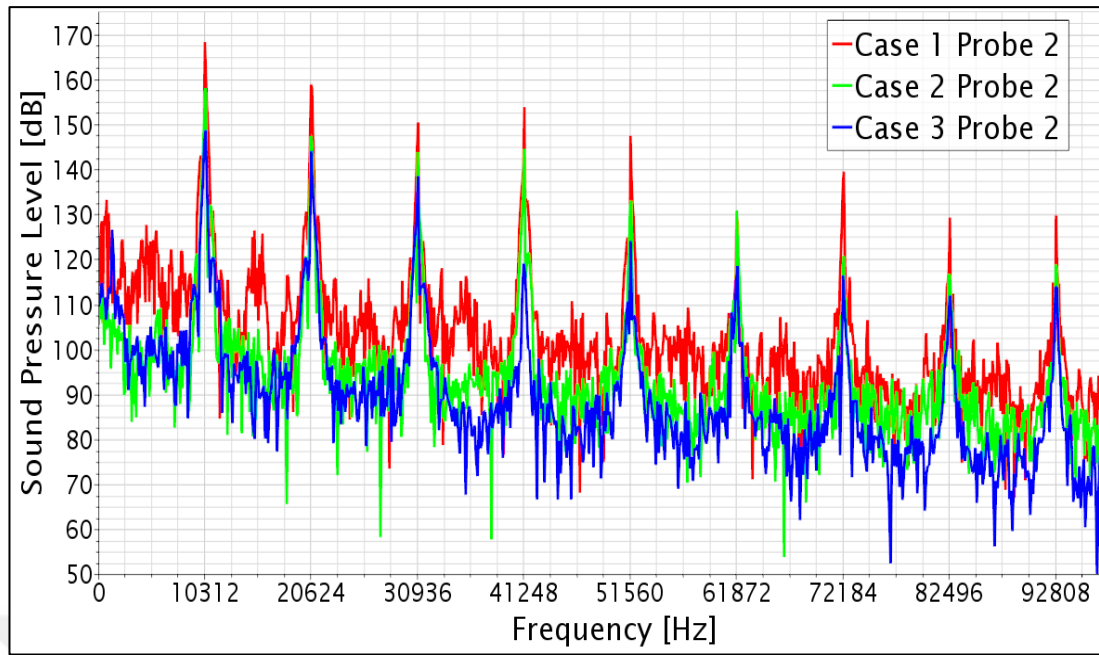


Figure 5.70: Noise spectrum of probe 2.

Tonal noise calculation results at pressure probe locations are tabulated in Table 5.12 and Table 5.13. As seen, the unit of the tonal noise values is decibel and up to the 9th harmonics of the BPF the results are shown. As mentioned, model is able to capture the first four harmonics and at the further harmonics, tonal noise value might be disrupted due to the numerical dissipation of the acoustic waves.

According to the results, a substantial reduction of tonal noise was achieved by increasing the axial gap between the rotor and stator. This is most distinctive at first four harmonics of BPF. At first BPF, doubling the axial gap reduced to tonal noise up to 9 dB and quadrupling the axial gap yielded in a noise reduction of 20 dB.

Overall, increasing the axial gap between the rotor blade and stator vane yielded in a considerable tonal noise reduction at blade passing frequency and its higher harmonics. This was achieved by the reducing the unsteady pressure fluctuations on stator vane and rotor blade surfaces. Increasing the axial gap degrades the rotor wake-stator vane interaction and the effect of the stator potential field on the rotor blade surface. In addition to that, broadband noise level of the compressor stage lost its effectiveness as the axial gap increases. This might be achieved by the decrease in the turbulence content of the rotor wake until it hits the stator vane which is located further downstream of the rotor blade.

Table 5.12: Tonal noise values of probe 1 to probe 5 in dB.

Probe	Case	1 st BPF	2 nd BPF	3 rd BPF	4 th BPF	5 th BPF	6 th BPF	7 th BPF	8 th BPF	9 th BPF
Probe 1	Case 1	163.3	163.8	153.0	152.2	139.9	136.9	134.3	130.2	130.0
	Case 2	160.6	152.8	146.6	146.7	137.7	133.5	130.6	124.1	115.5
	Case 3	148.7	144.0	138.5	119.0	124.5	118.5	116.4	112.2	113.9
Probe 2	Case 1	168.2	158.8	150.4	153.8	147.4	130.5	139.5	129.3	129.7
	Case 2	158.0	147.4	143.8	144.5	131.9	130.9	120.7	116.8	118.8
	Case 3	148.7	144.0	138.5	119.0	124.1	118.5	116.4	112.2	113.9
Probe 3	Case 1	170.3	151.6	150.2	148.2	140.5	140.9	129.8	124.8	121.2
	Case 2	161.3	146.0	136.4	135.2	136.8	132.6	124.8	117.4	98.8
	Case 3	157.3	145.8	140.8	127.1	117.7	117.8	106.1	108.8	114.5
Probe 4	Case 1	165.6	166.0	154.3	144.1	135.1	143.8	117.3	130.3	113.4
	Case 2	160.8	150.8	144.7	136.8	126.3	127.8	123.1	112.1	106.1
	Case 3	151.5	137.5	135.1	119.0	118.1	112.8	112.1	100.6	107.3
Probe 5	Case 1	161.7	161.0	143.4	136.4	131.6	129.7	124.8	115.5	98.0
	Case 2	156.8	155.8	145.3	134.3	113.5	121.3	124.1	100.9	103.0
	Case 3	153.6	142.5	110.3	116.3	103.6	109.3	107.6	88.4	101.4

Table 5.13: Tonal noise values of probe 6 to probe 9 in dB.

Probe	Case	1 st BPF	2 nd BPF	3 rd BPF	4 th BPF	5 th BPF	6 th BPF	7 th BPF	8 th BPF	9 th BPF
Probe 6	Case 1	165.0	154.1	151.6	139.1	143.6	125.4	119.3	114.2	108.0
	Case 2	156.3	148.0	145.2	135.7	132.9	113.0	109.8	110.9	95.2
	Case 3	153.8	143.9	130.5	124.7	111.6	99.6	97.7	102.7	94.3
Probe 7	Case 1	171.4	164.3	152.7	149.7	128.5	126.3	123.0	117.0	105.5
	Case 2	166.0	161.2	139.4	142.9	129.1	125.0	113.3	107.8	99.5
	Case 3	163.3	150.1	135.0	120.8	113.1	100.6	91.2	93.7	90.3
Probe 8	Case 1	162.0	157.5	141.9	136.8	137.1	129.6	123.4	115.6	115.1
	Case 2	161.7	138.7	132.7	129.0	124.1	117.6	114.0	111.6	93.1
	Case 3	149.8	117.4	121.2	118.9	110.3	105.2	103.8	98.0	99.5
Probe 9	Case 1	159.6	160.2	144.2	116.3	132.2	105.9	121.1	107.2	103.0
	Case 2	156.5	148.9	133.7	128.3	119.0	110.3	114.2	104.9	99.3
	Case 3	152.0	142.2	125.4	121.8	114.8	110.9	110.8	109.9	96.1

5.4.2 Far-field noise investigation

In addition to the near-field noise of the compressor stage, the acoustic waves propagate from the noise source and proceed in the fluid medium to the far-field locations. For instance, an observer on the ground might be strongly irritated by the generated noise of an airplane, which cruises at an altitude of thousands of feet. That is why the noise propagation in the downstream should be carefully investigated.

As mentioned in the chapter aeroacoustics, meshing the CFD domain until the far-field observer to calculate the sound pressure at this location is not a reasonable way for far-field noise investigation. This significantly increases the mesh size and the computational time of the solution. As an alternative, hybrid methods are used for the far-field noise applications. This method requires an acoustic solver addition to the CFD solution. First, the transient CFD simulation is run for capturing the noise sources in the flow domain. Second, an acoustic solver is utilized to propagate the acoustic waves from source to the observer location in the far-field.

In this study, the mesh structure is sufficient only for the resolving the tonal noise sources and the near-field acoustics. As a hybrid method URANS simulation and FW-H acoustic analogy was utilized to calculate the far-field noise of the compressor stage. In the first step URANS simulation was run to collect unsteady pressure data on the noise source locations. In order to collect sufficient data for processing with the FW-H solver, the CFD solution was run for two complete rotation of the rotor wheel which corresponds to the duration of 72 blade passing. Rotor blade surface and stator vane surface are utilized as the tonal noise sources for far-field noise propagation solver. During the CFD simulation the pressure data was recorded on the noise sources. Since a blade passing was modeled with 100 time steps, pressure was recorded during 7200 time steps in order to provide a sufficient data for the FW-H acoustic analogy solver.

The workflow of the hybrid method utilized in this study for the far-field noise calculation is shown in Figure 5.71.

Totally six FW-H far-field observer locations were set in the cases. All far-field probes are in the downstream of the stator vane. Three of them are inside the CFD domain at midspan, the rest are located outside of the CFD domain.

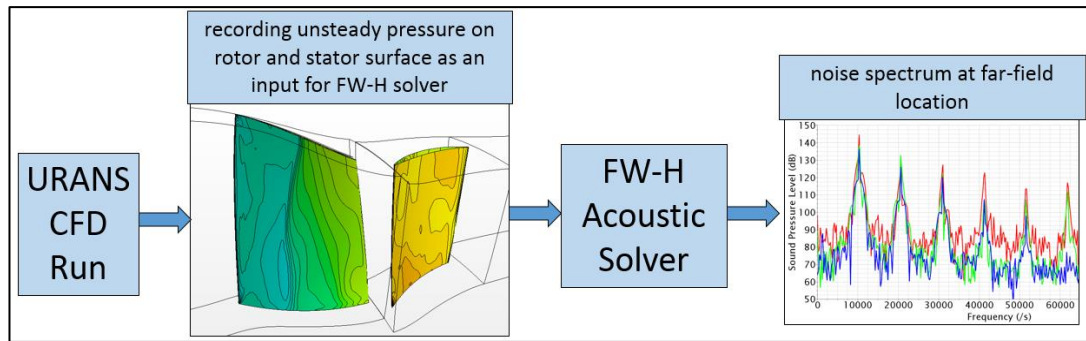


Figure 5.71: Hybrid method workflow.

As seen in Figure 5.72, three of far-field observer probes were located inside the CFD domain. Since the model is not able to capture the acoustic waves due to the numerical dissipation which stems from the gradually coarsing mesh towards outlet, these 3 probes were intentionally located inside the domain to evaluate the noise spectrum further downstream of the near-field probes. As seen, the closest probe was located in 65 mm downstream of the stator vane.

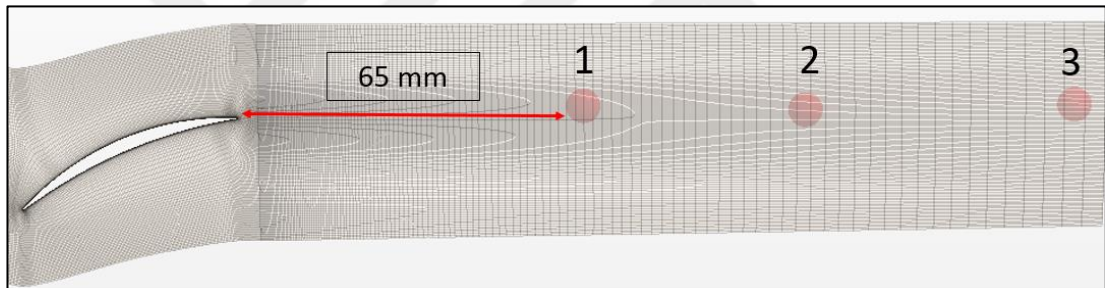


Figure 5.72: FW-H observer locations inside CFD domain.

The distance between the trailing edge of the stator vane and the FW-H observer locations are tabulated in Table 5.14.

Table 5.14: FW-H observer distance from stator trailing edge.

Probe	Distance [mm]
Probe 1	65
Probe 2	105
Probe 3	155
Probe 4	270
Probe 5	420
Probe 6	670

Noise spectrum of the FW-H solution at first three far-field observer locations are illustrated in Figure 5.73, Figure 5.74 and Figure 5.75. The rest of the FW-H plots are given in Appendix C.

The noise spectrums were plotted up to the 9th harmonics of the blade passing frequency. As seen, after the 4th harmonics insufficient data sampling causes damping of the solution especially for the case 3. Tonal noise at first BPF is dominated by the case 1, whereas the tonal noise at second BPF is slightly higher in case 2. Further frequencies are dominated by case 1 as well, apart from the fifth BPF. The general trend of the tonal noise is decrease as the frequency value increases. Moreover, comparing the each plot of the FW-H observer probes proves that the noise level decreases as the acoustic waves move away from the sound sources.

As seen the broadband noise could not be captured by FW-H solver, since no input was supplied for the volumetric integral term which is the turbulence quadrupole noise. Nevertheless, the broadband noise component of the case 1 appears to be higher than the other two cases. The broadband noise level of case 2 and case 3 are almost superimposed which means proves the inability of the FW-H solver without supplied any broadband noise source term.

Compared to the near-field noise spectrums, until the acoustic waves reach to far-field locations, the compressor noise level substantially diminished.

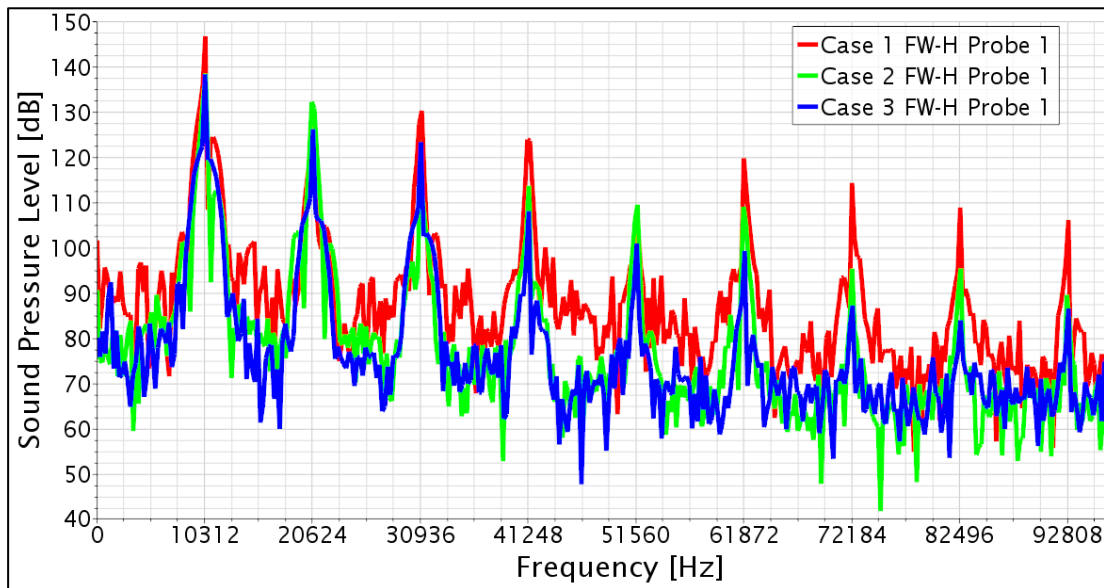


Figure 5.73: Noise spectrum of FW-H probe 1.

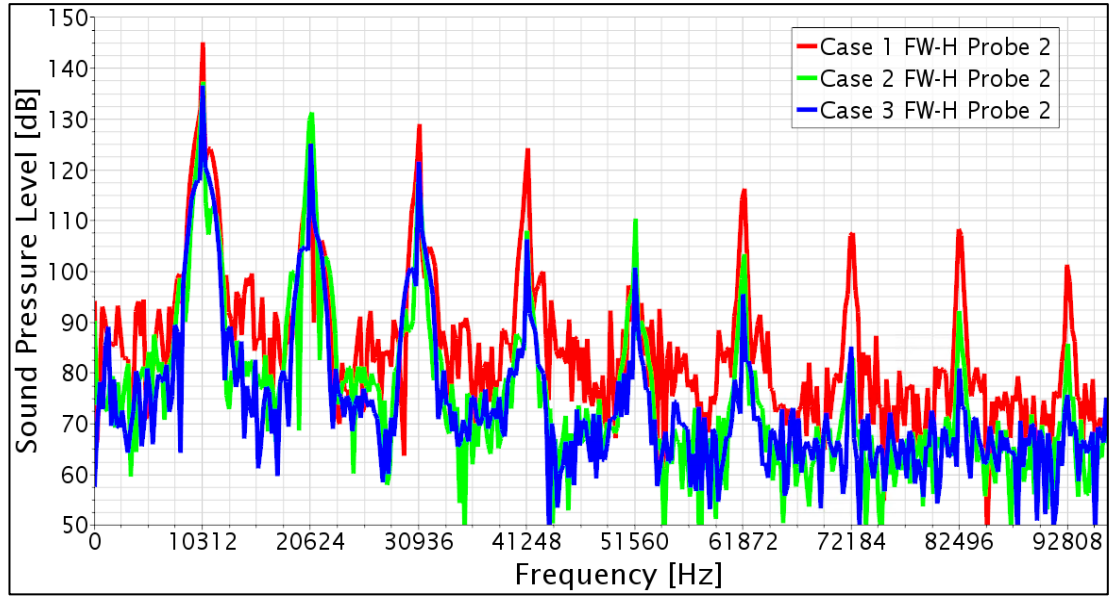


Figure 5.74: Noise spectrum of FW-H probe 2.

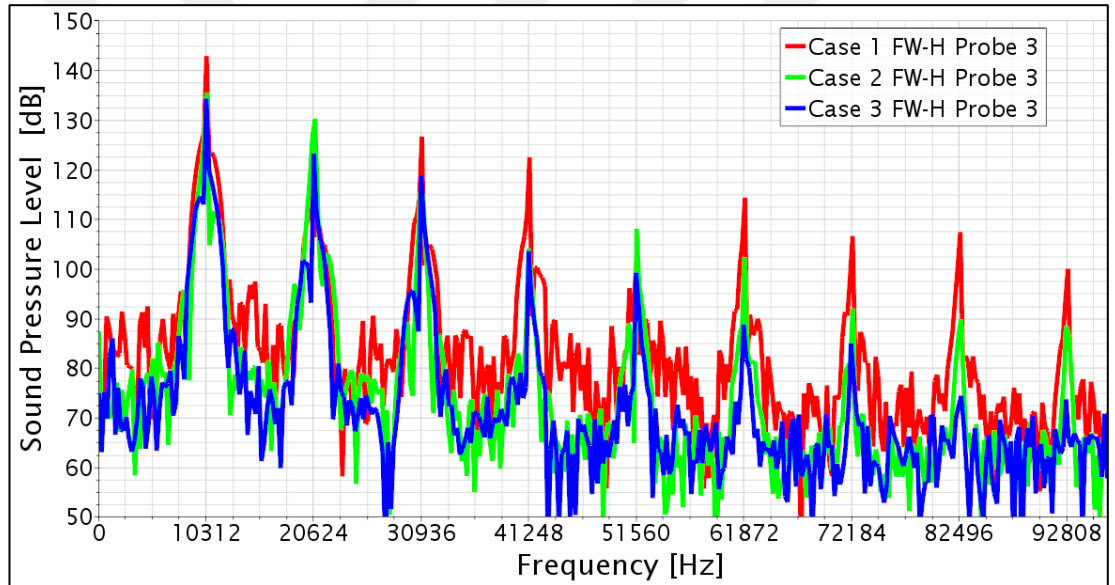


Figure 5.75: Noise spectrum of FW-H probe 3.

Tonal noise values up to the 4th harmonics of the BPF are tabulated in Table 5.15. Apart from the second harmonics, the tonal noise values are dominated by the case 1 as expected. Tonal noise value of case 2 is slightly higher than case 1 at the second BPF.

Far-field noise investigation study also proves that the tonal noise due to rotor-stator interaction decreases as the axial gap increases.

Table 5.15: Tonal noise values of FW-H probes in dB.

FW-H Probe	Case	1st BPF	2nd BPF	3rd BPF	4th BPF
Probe 1	Case 1	146.7	130.6	130.3	124.0
	Case 2	138.5	132.3	118.6	113.7
	Case 3	138.4	126.2	123.4	108.1
Probe 2	Case 1	145.0	129.9	129.0	124.3
	Case 2	137.3	131.3	117.6	108.0
	Case 3	136.5	125.0	121.5	106.3
Probe 3	Case 1	142.9	127.9	126.6	122.5
	Case 2	135.4	130.2	117.4	104.0
	Case 3	134.3	123.2	118.8	103.6
Probe 4	Case 1	140.3	125.5	123.3	119.6
	Case 2	132.9	128.6	114.5	104.8
	Case 3	129.6	118.3	111.8	99.2
Probe 5	Case 1	136.7	121.8	119.5	115.8
	Case 2	129.6	125.5	111.8	101.0
	Case 3	126.7	115.9	109.0	94.0
Probe 6	Case 1	132.8	117.6	115.6	111.3
	Case 2	125.9	121.9	109.0	98.7
	Case 3	114.6	109.7	100.1	83.4

6. CONCLUSION

In this study tonal noise characteristics of a transonic axial compressor stage were numerically investigated under different axial gap length configurations. The effect of the axial gap length between the rotor blade and stator vane on noise spectrum of the compressor stage was measured by doubling and quadrupling the original state of the axial gap. The overall work completed, important concluding remarks of this study and the further work plan are given below.

- Noise generation mechanisms, especially the rotor-stator interaction generated noise, in turbomachinery applications were understood detailed.
- Reduction methodologies of the rotor-stator interaction noise were investigated.
- NASA Stage 37 test case was selected for the tonal noise investigation.
- Numerical model of the NASA Rotor 37 and NASA Stage 37 were constructed.
- Validation of the aerodynamic performance of the numerical models were conducted by comparing the CFD solutions of NASA Rotor 37 and NASA Stage 37.
- Fundamentals of aerouacoustics theory were deeply understood.
- Computational aeroacoustics methods which are direct noise calculations and hybrid methods were properly comprehended.
- Three different CFD case of compressor stage, each having different axial gap length, were constructed for aeroacoustics simulations. The axial gap length is 8, 16 and 32 mm in case 1, case 2 and case 3 respectively.
- Transient CFD analyses were carried out for each case.
- Direct noise calculation and FW-H acoustic analogy solver were utilized for near-field and far-field acoustic investigations respectively.
- Results show that the efficiency of the stage decreases as the axial gap increases.
- Flow field investigation of the three cases shows that the unsteadiness increases with the decreasing axial gap.

- Amplitude of the unsteady pressure fluctuations and the magnitude of the radiated noise from the surface on rotor blade decreases as the axial gap between the rotor and stator increases. This stems from the decrease of the effect of the potential field of the stator on rotor blade.
- Reduction in the unsteady pressure fluctuations and the radiated noise magnitude were also observed on the stator vane surface. This is caused by the degradation of the rotor wake-stator vane interaction with the increasing axial gap.
- Near-field noise investigation results show that the tonal noise decreased substantially at each rotor blade passing frequencies as the axial gap increases. Moreover, considerable decrease in broadband noise was observed.
- Far-field noise calculations also prove that increasing the axial spacing between rotor and stator is an effective method for noise reduction.
- Overall, increasing the axial gap between the rotor and stator in an axial compressor has a substantial effect on reduction of the tonal noise generation of the machine. However, it must be considered that the aerodynamic efficiency of the compressor stage decreases as the axial gap increases. Thus, it a trade-off between the efficiency and noise generation.

Since, in this study, URANS turbulence model was used, due to the modeling approach of the URANS, acoustic waves could not be captured properly in these models. In order to resolve the acoustic features in the flow domain elaborately, LES or DES based turbulence models must be utilized in further studies. This way, apart from the tonal noise sources, turbulence generated broadband noise sources also can be captured.

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APPENDICES

APPENDIX A: Space time plot of pressure side and suction side of the stator vane at 10, 25, 75 and 90% span.

APPENDIX B: Noise spectrum of near-field probes.

APPENDIX C: Noise spectrum of far-field probes.



APPENDIX A

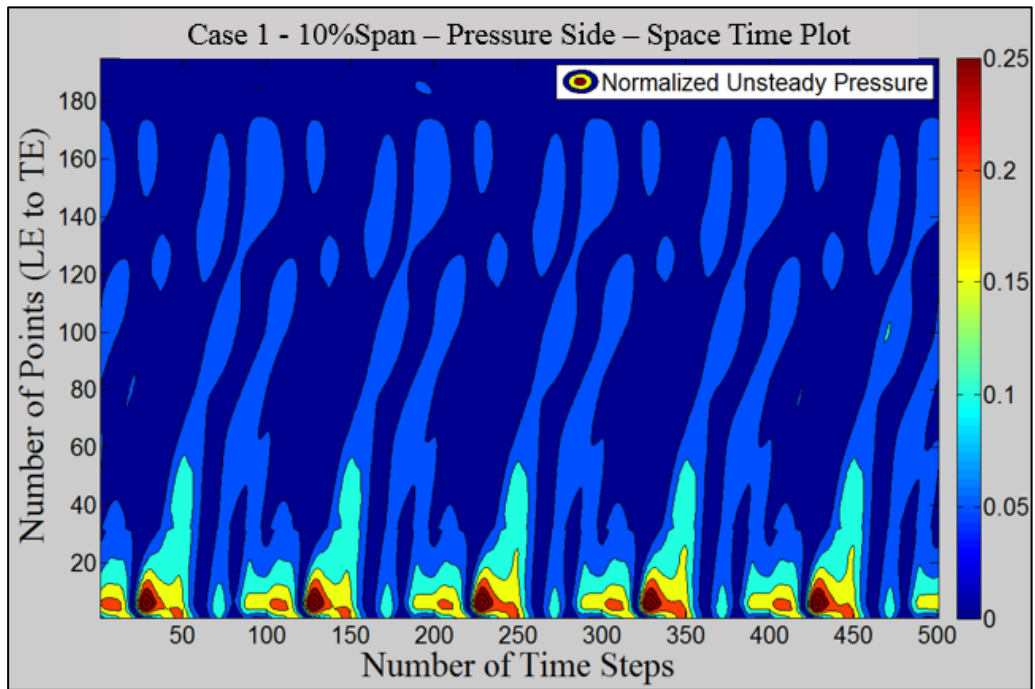


Figure A.1: Case 1 pressure side space-time plot at 10% span.

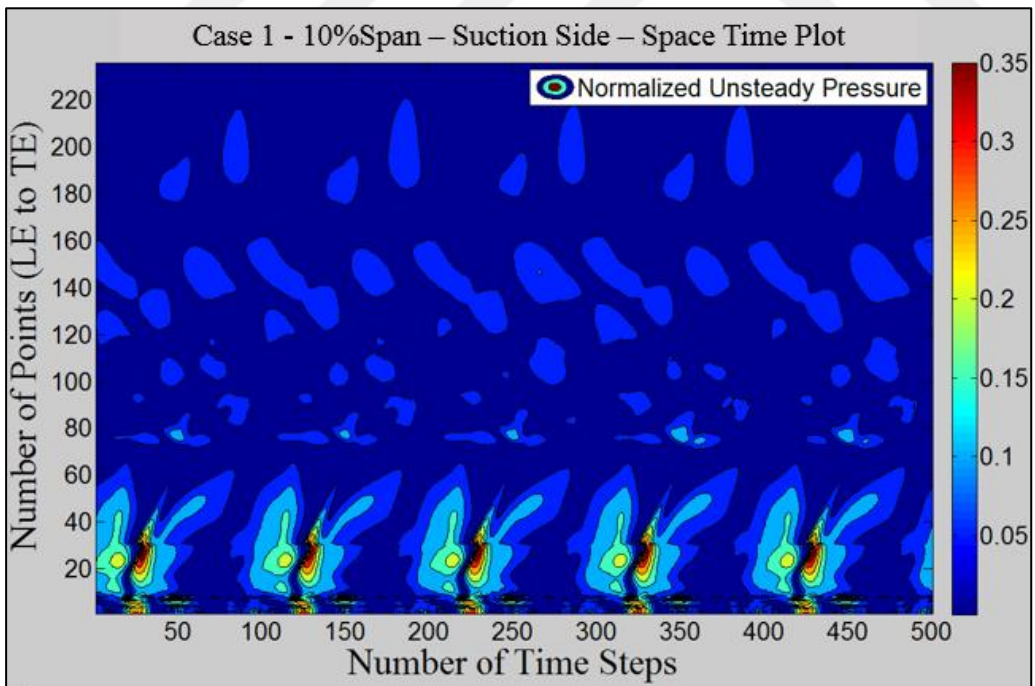


Figure A.2: Case 1 suction side space-time plot at 10% span.

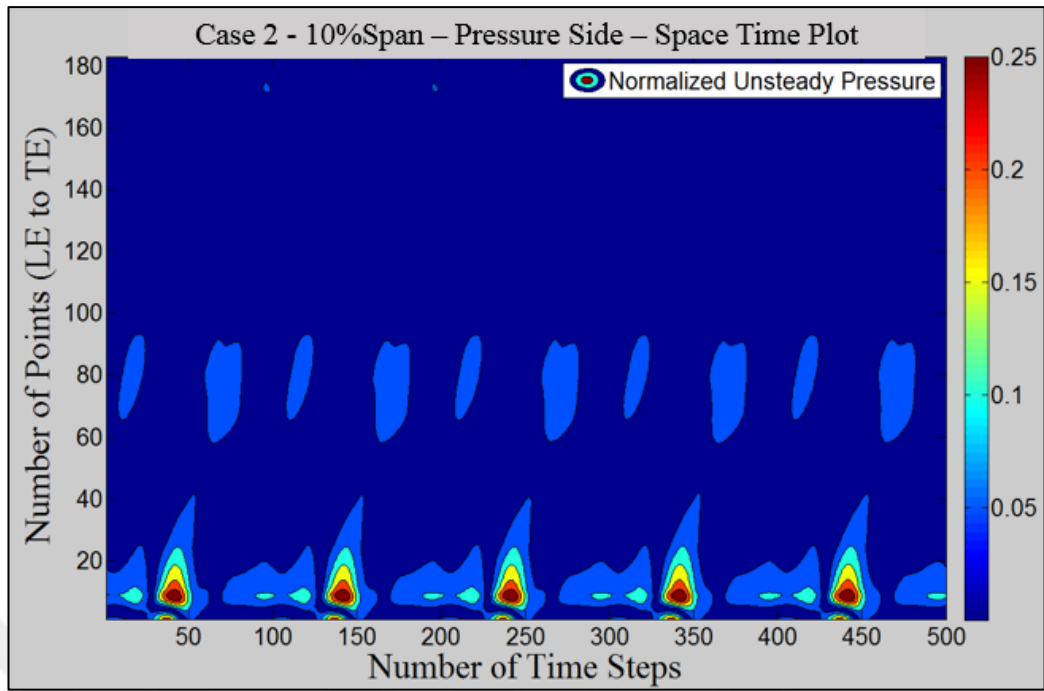


Figure A.3: Case 2 pressure side space-time plot at 10% span.

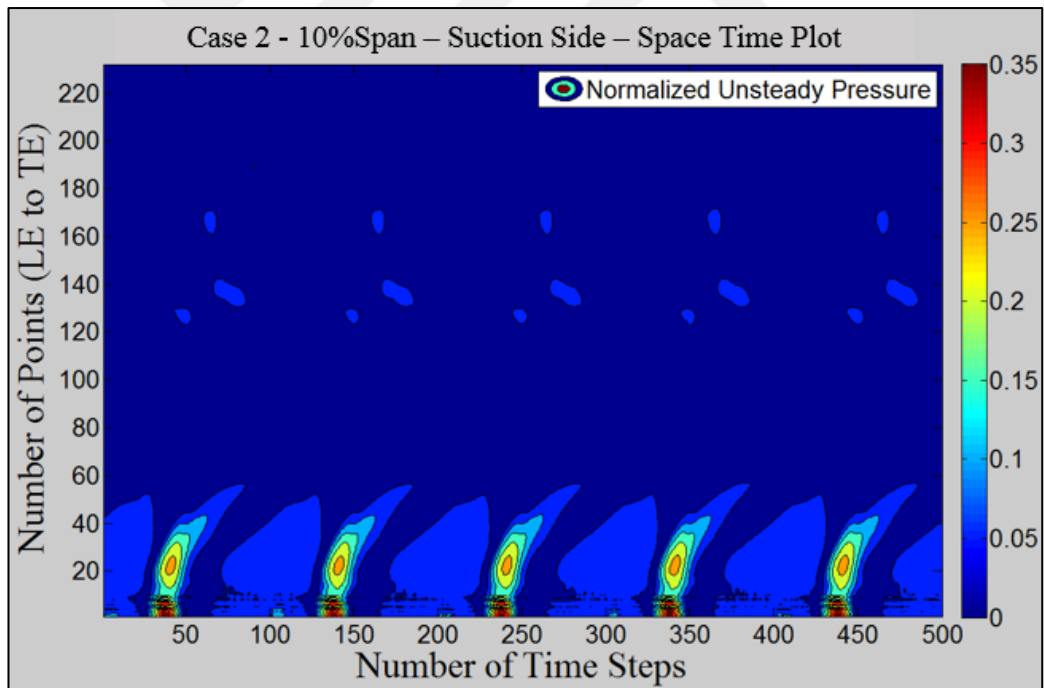


Figure A.4: Case 2 suction side space-time plot at 10% span.

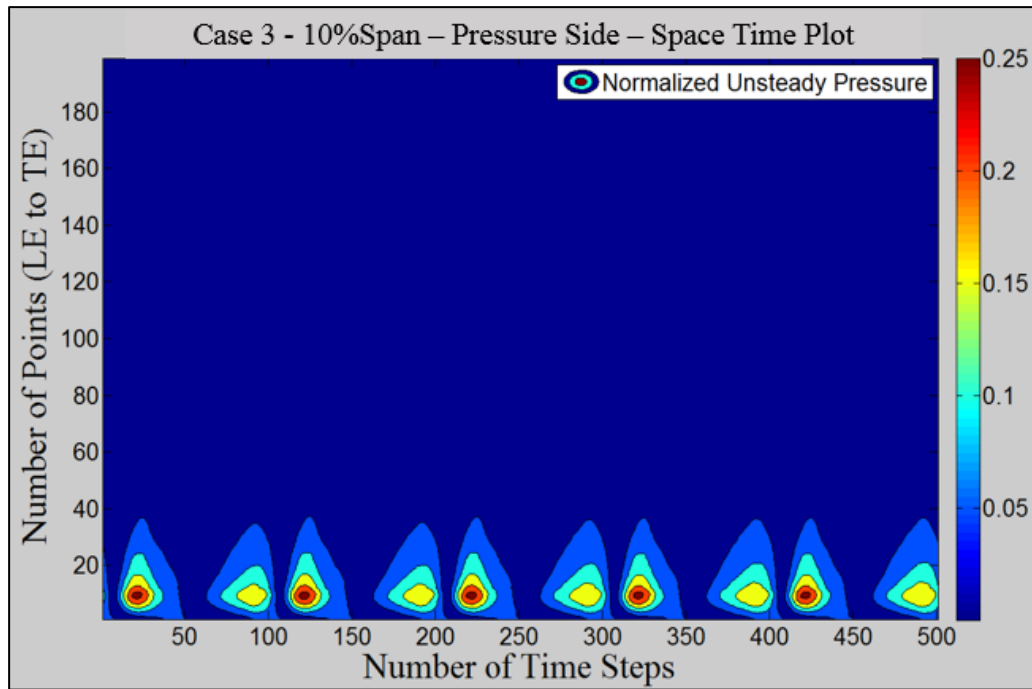


Figure A.5: Case 3 pressure side space-time plot at 10% span.

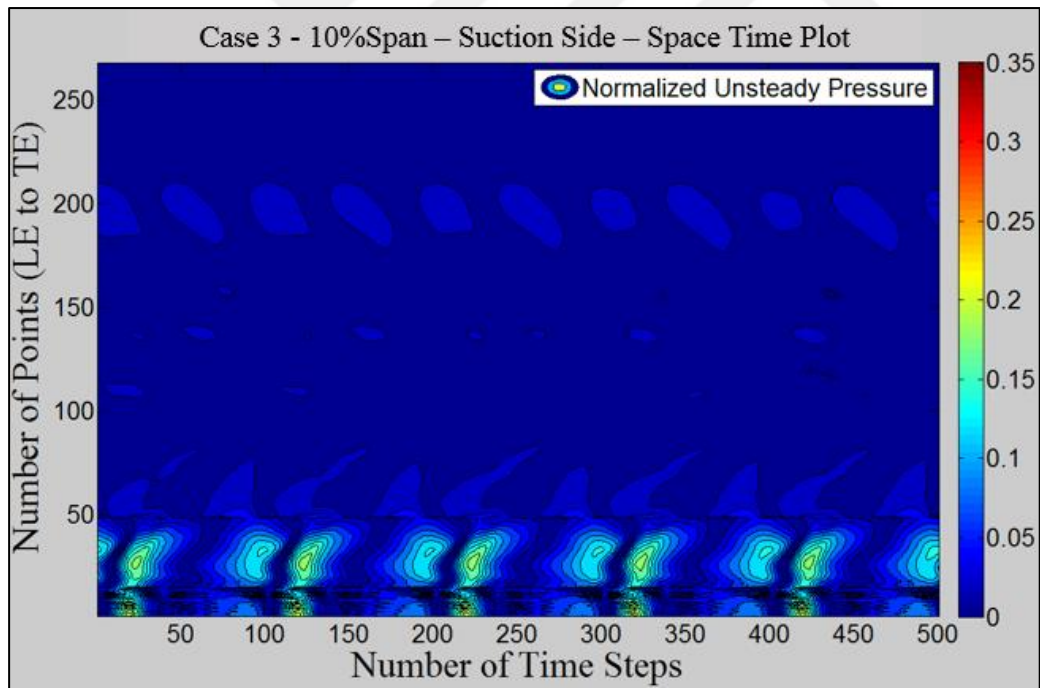


Figure A.6: Case 3 suction side space-time plot at 10% span.

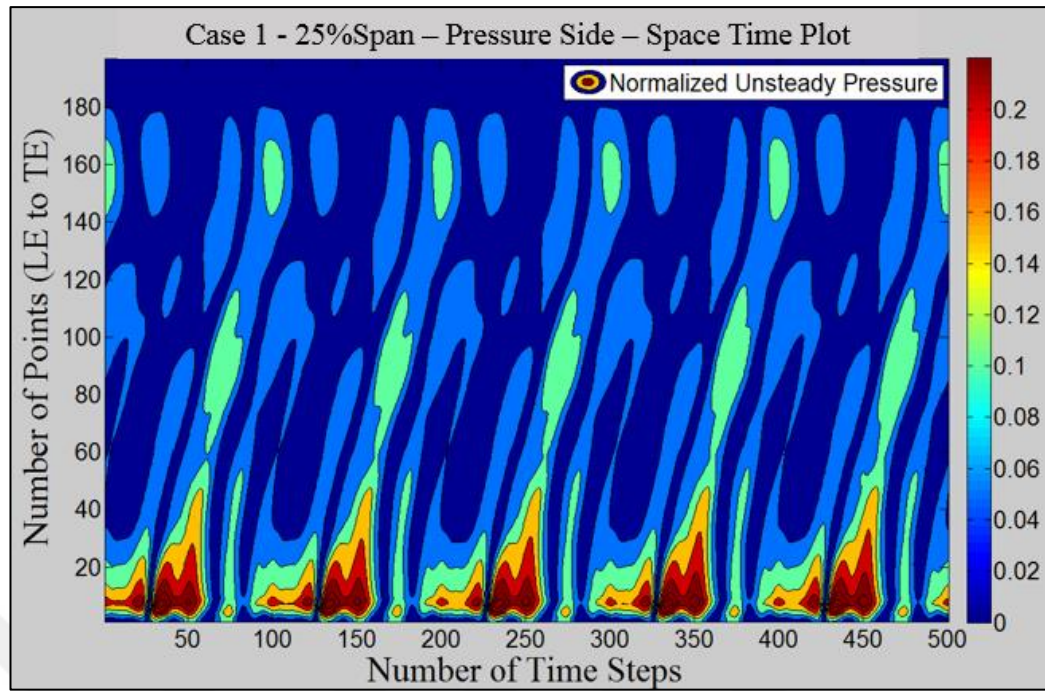


Figure A.7: Case 1 pressure side space-time plot at 25% span.

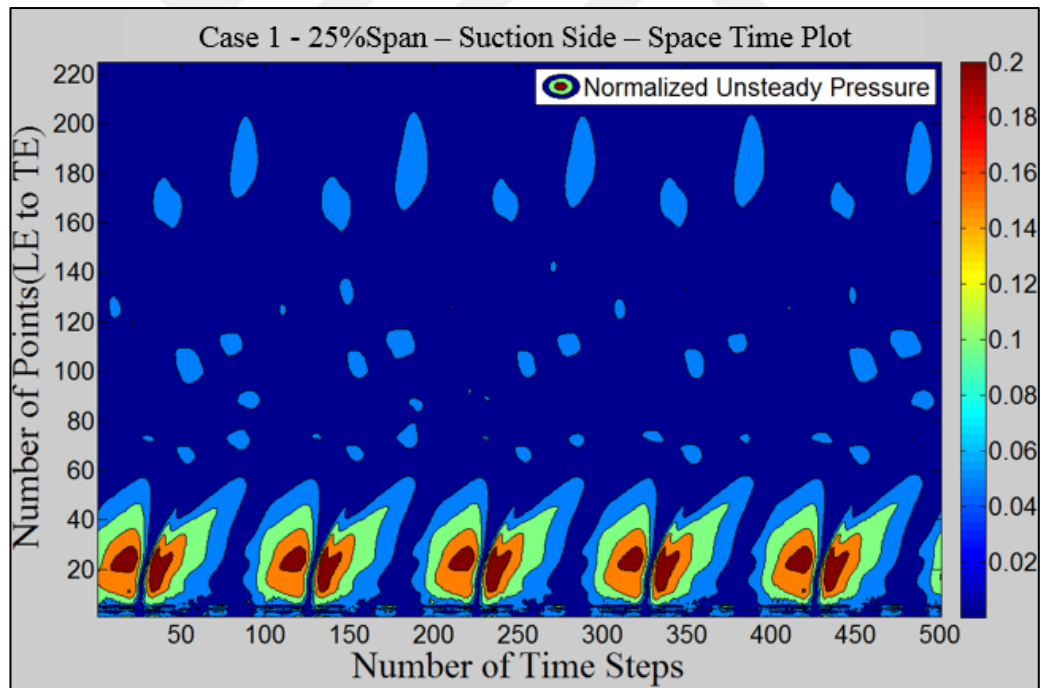


Figure A.8: Case 1 suction side space-time plot at 25% span.

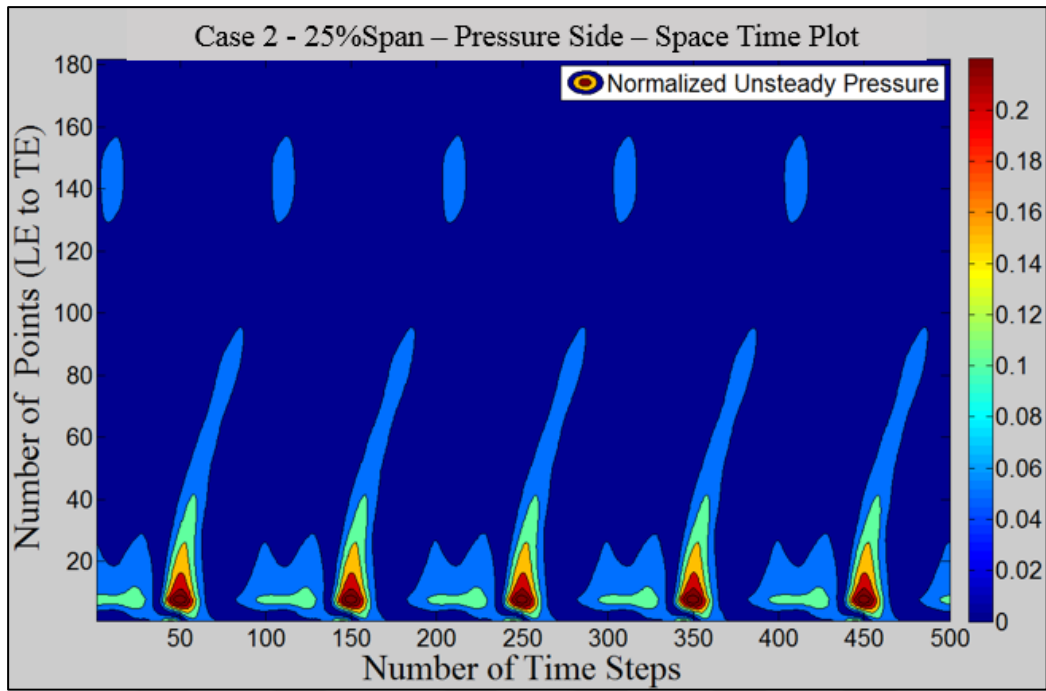


Figure A.9: Case 2 pressure side space-time plot at 25% span.

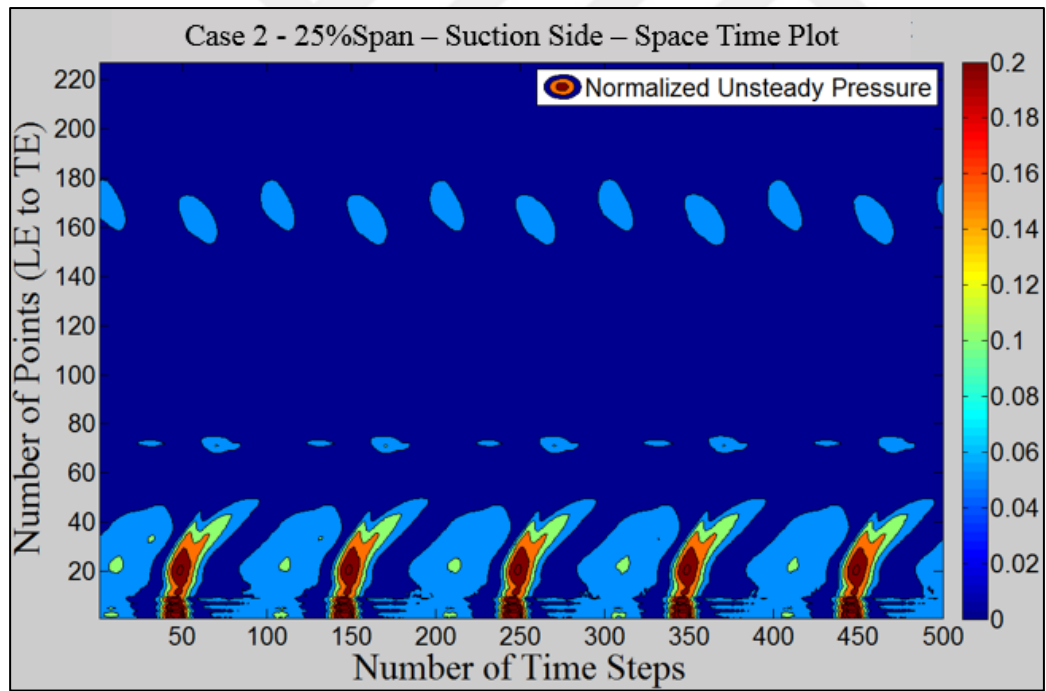


Figure A.10: Case 2 suction side space-time plot at 25% span.

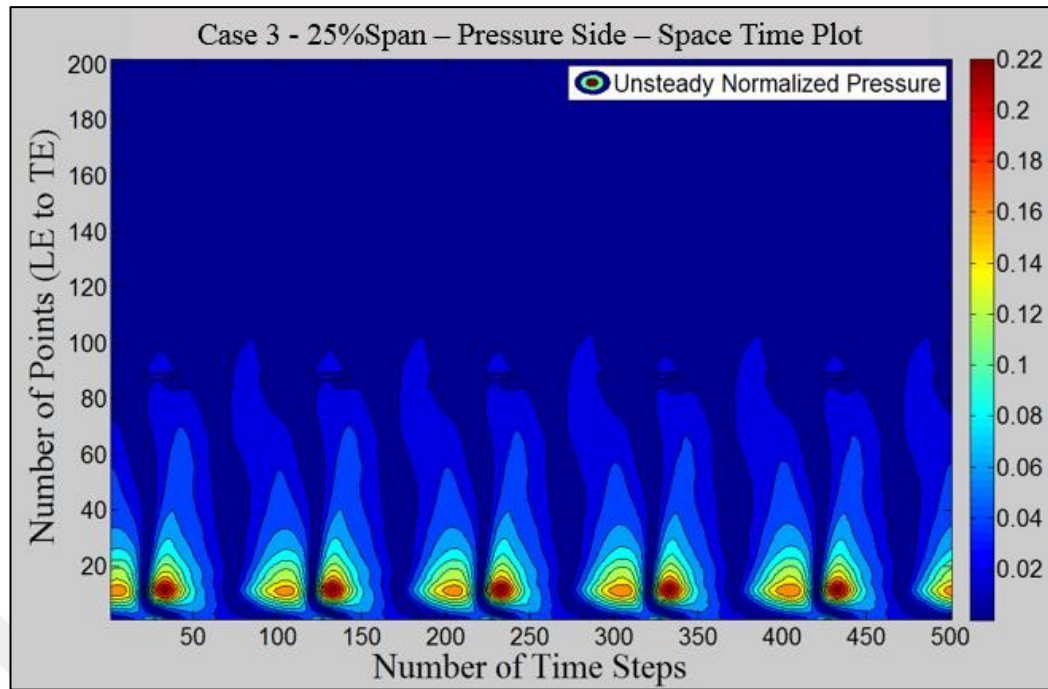


Figure A.11: Case 3 pressure side space-time plot at 25% span.

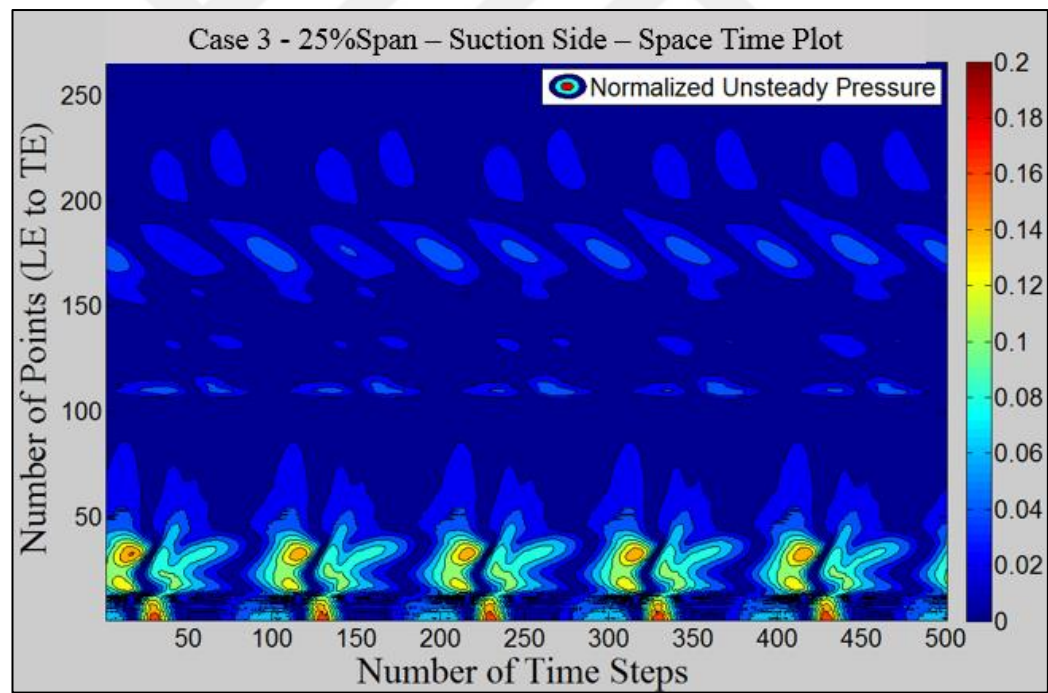


Figure A.12: Case 3 suction side space-time plot at 25% span.

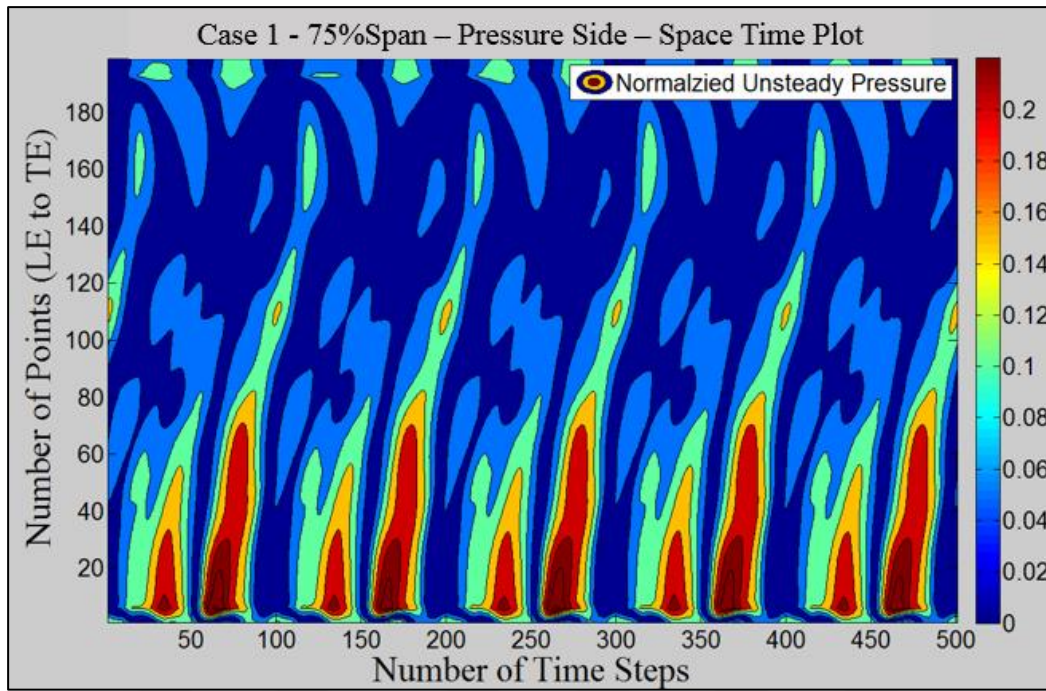


Figure A.13: Case 1 pressure side space-time plot at 75% span.

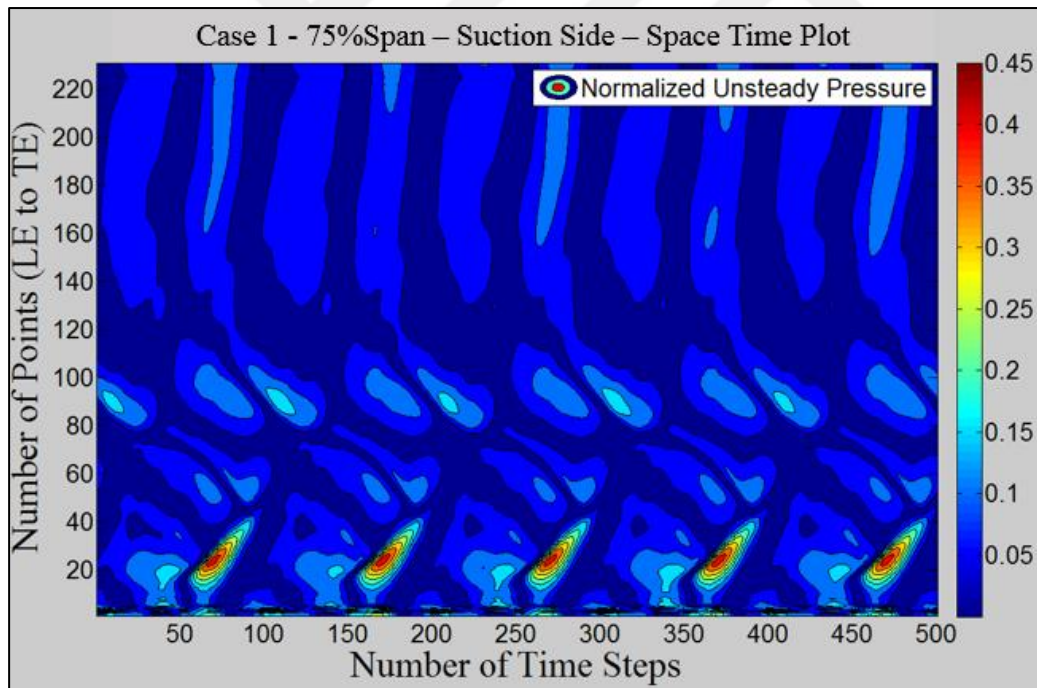


Figure A.14: Case 1 suction side space-time plot at 75% span.

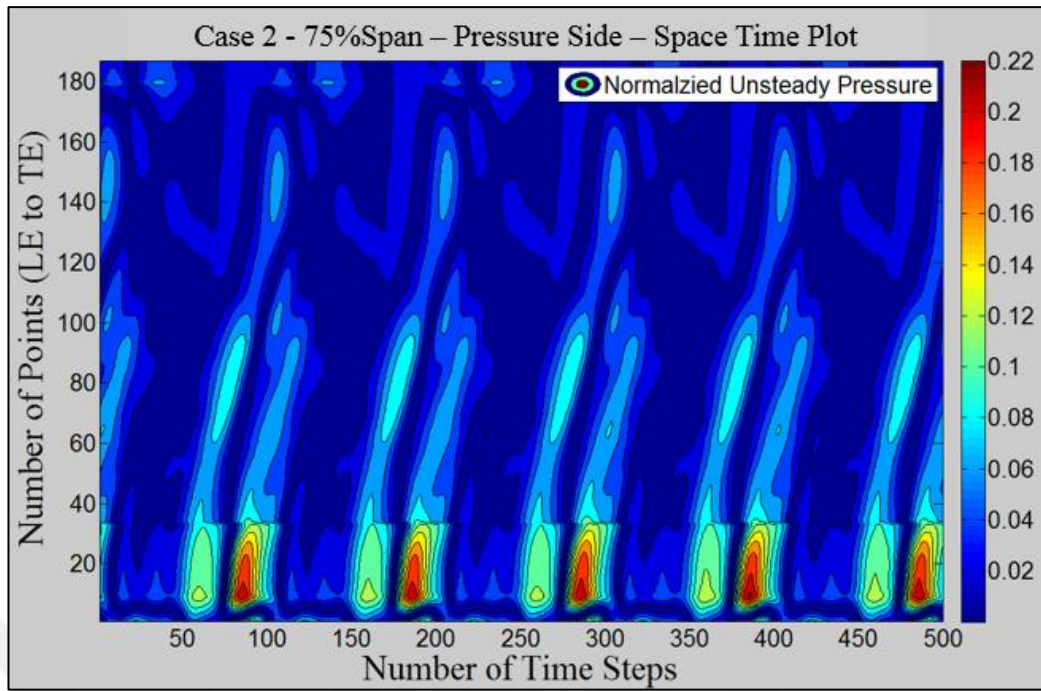


Figure A.15: Case 2 pressure side space-time plot at 75% span.

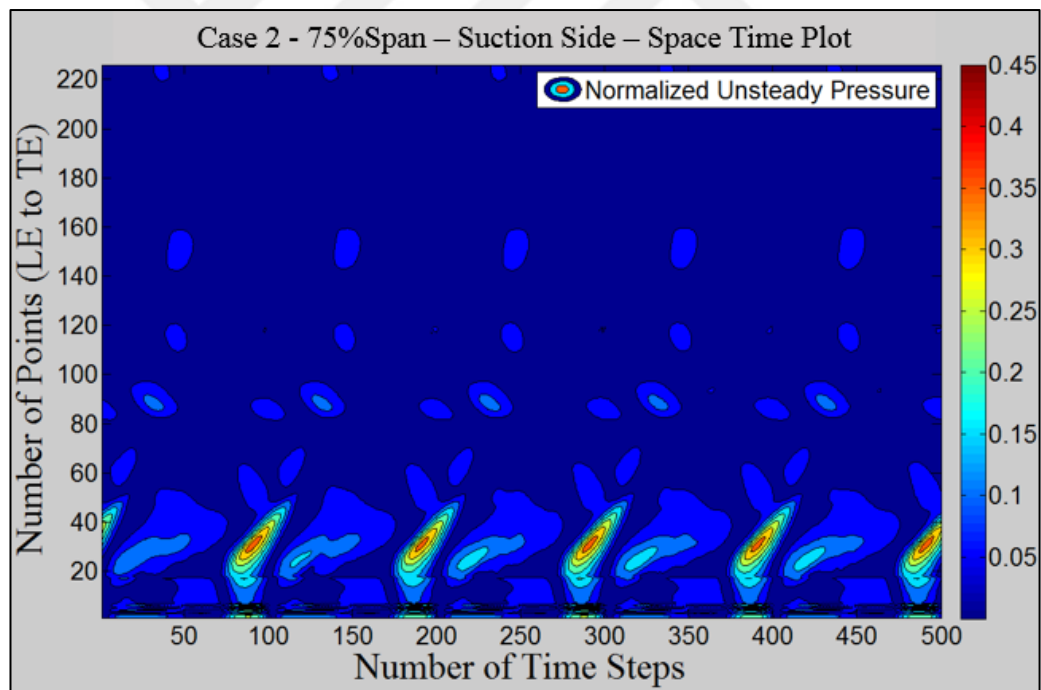


Figure A.16: Case 2 suction side space-time plot at 75% span.

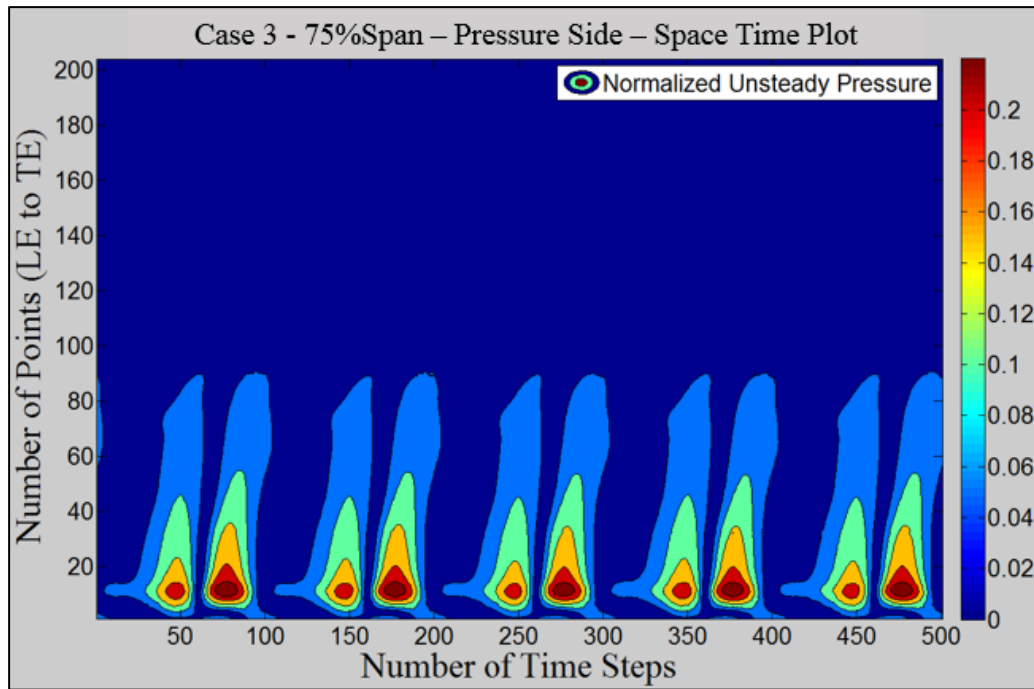


Figure A.17: Case 3 pressure side space-time plot at 75% span.

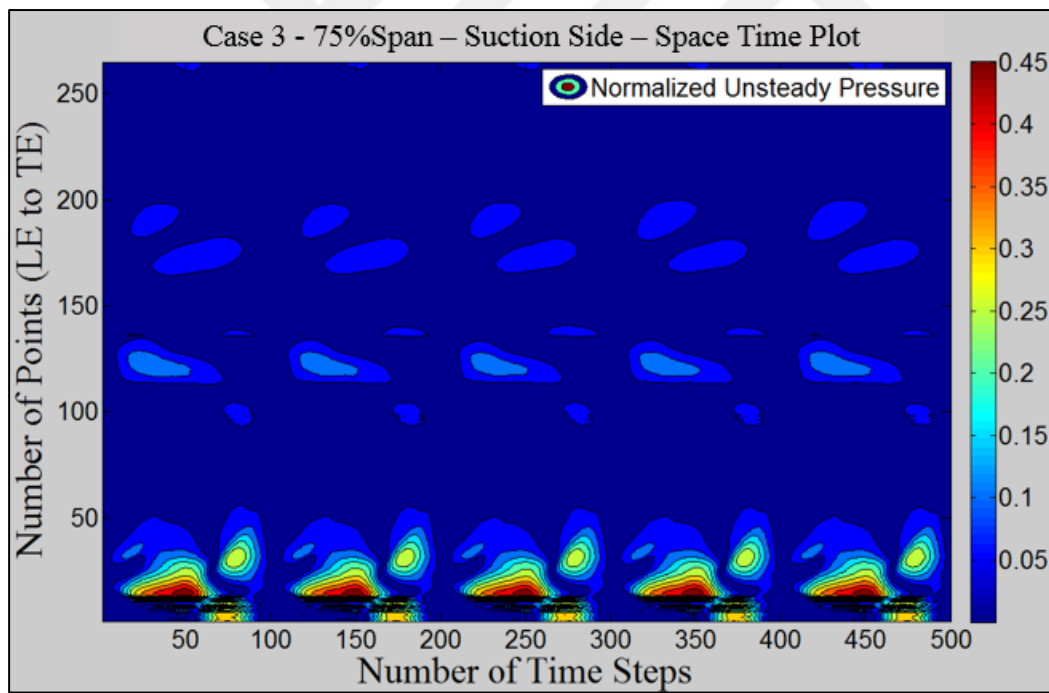


Figure A.18: Case 3 suction side space-time plot at 75% span.

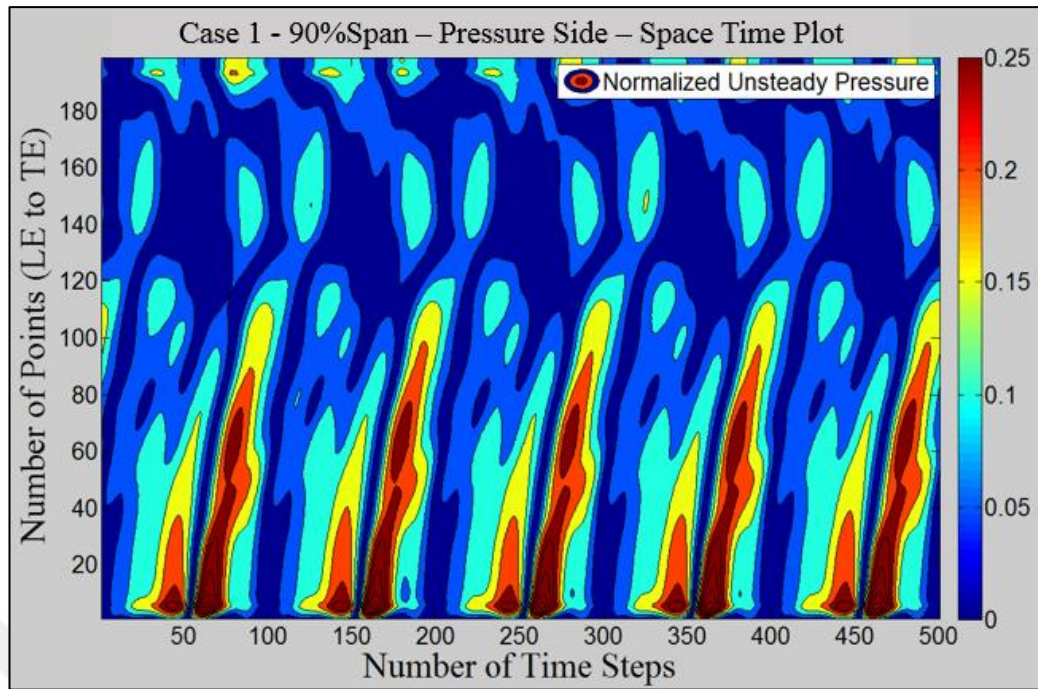


Figure A.19: Case 1 pressure side space-time plot at 90% span.

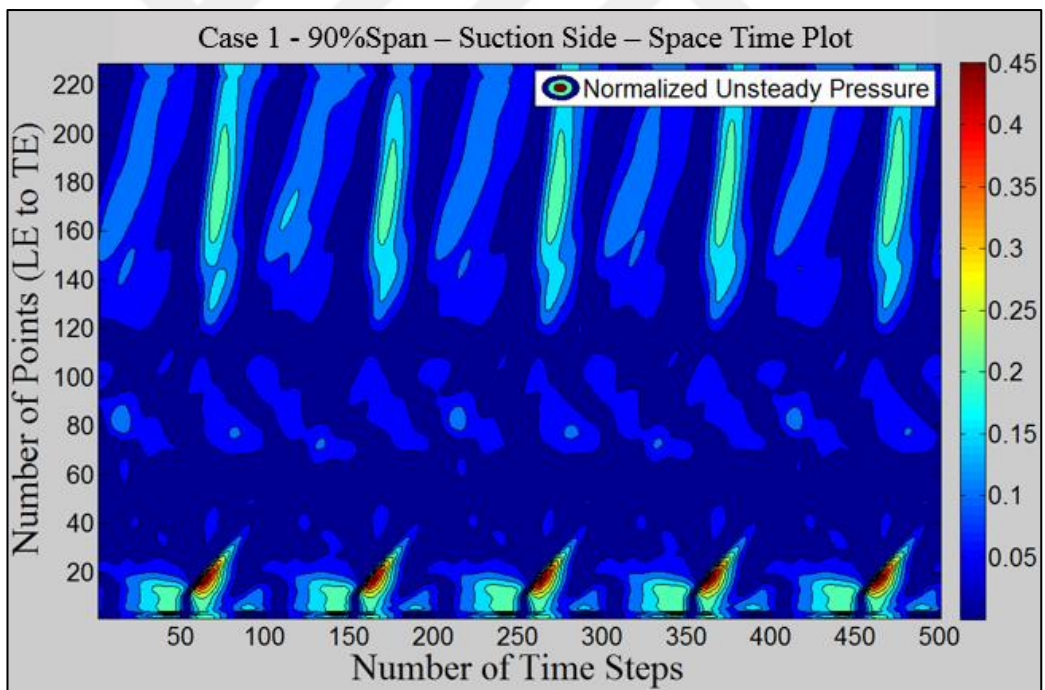


Figure A.20: Case 1 suction side space-time plot at 90% span.

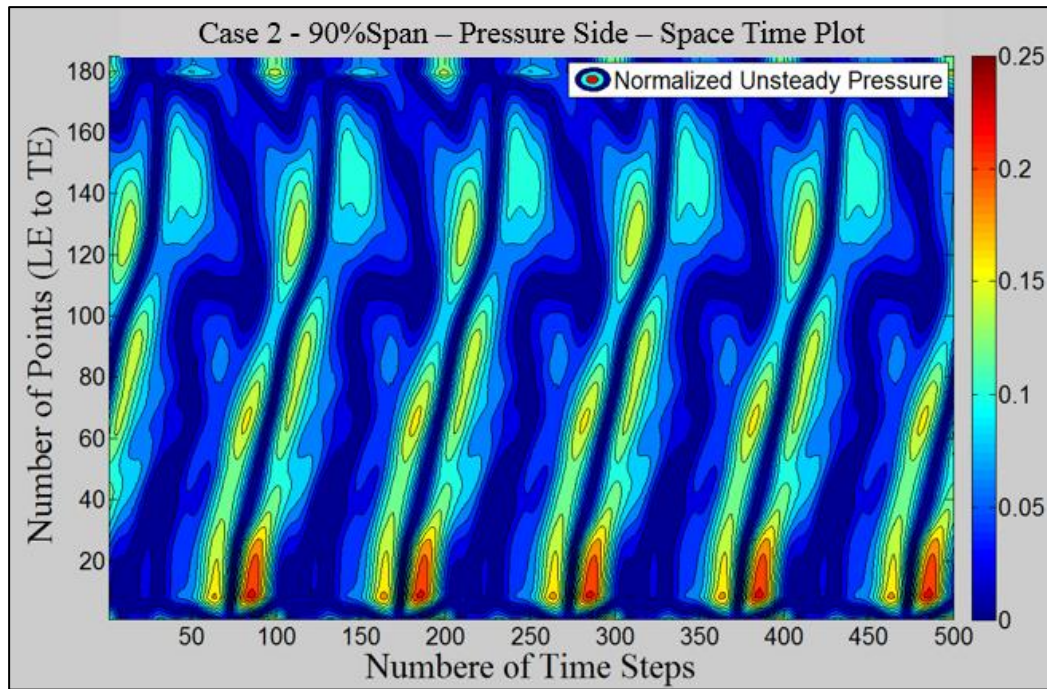


Figure A.21: Case 2 pressure side space-time plot at 90% span.

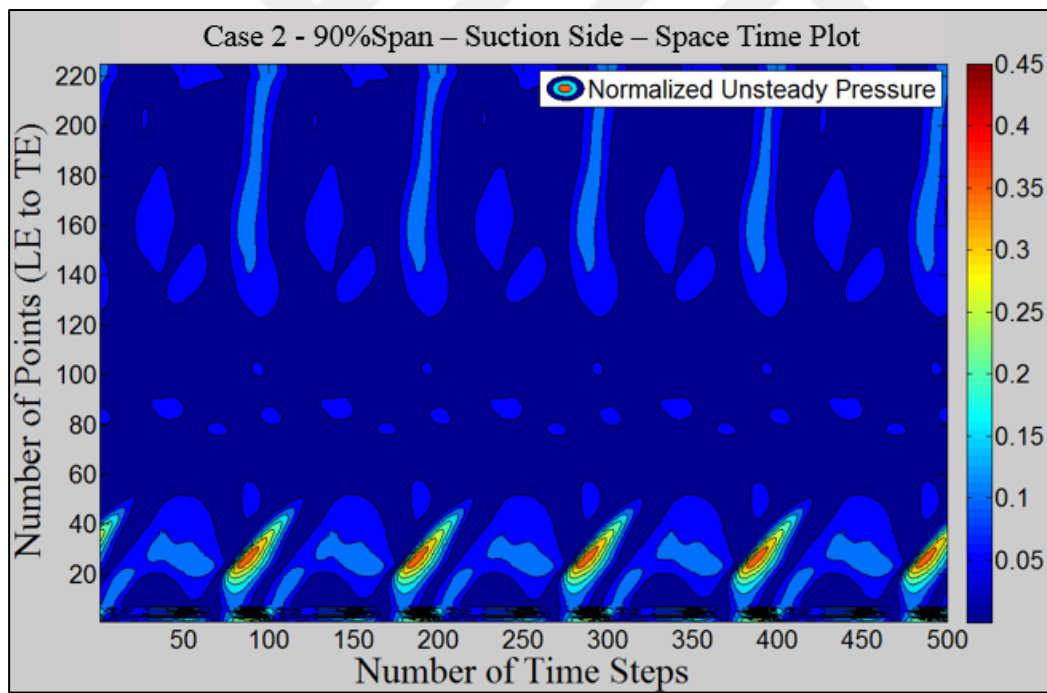


Figure A.22: Case 2 suction side space-time plot at 90% span.

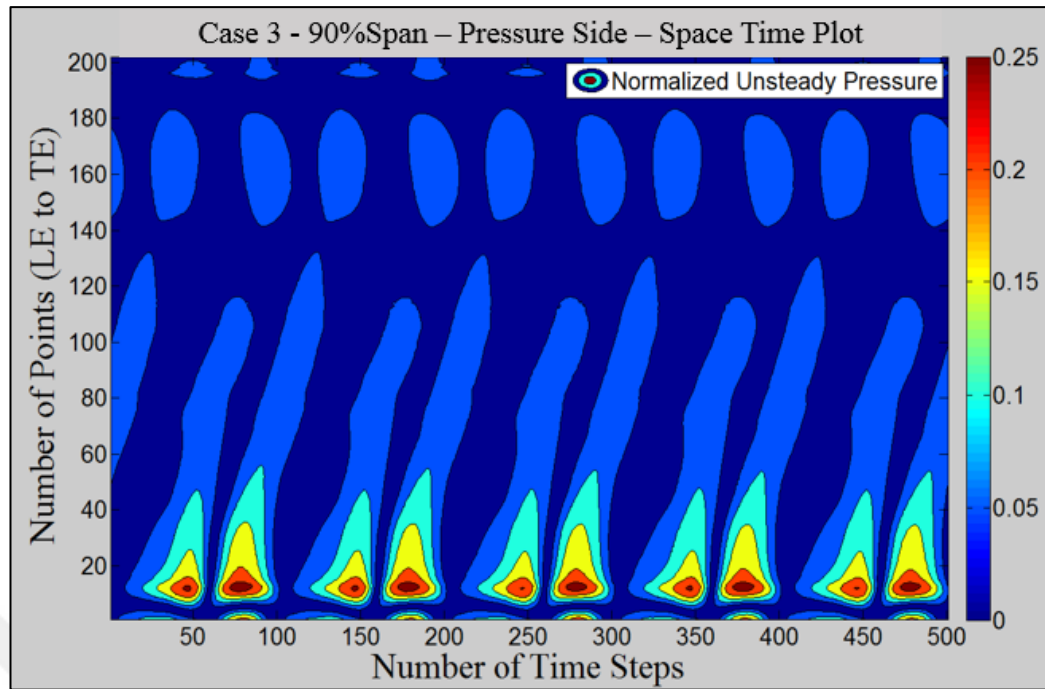


Figure A.23: Case 3 pressure side space-time plot at 90% span.

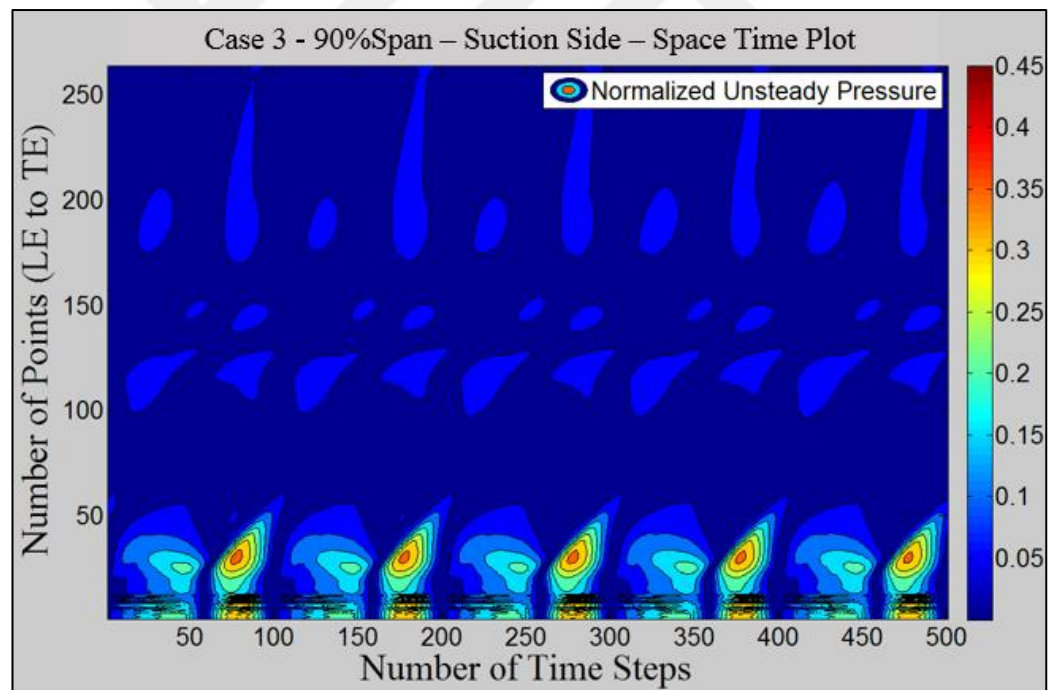


Figure A.24: Case 3 suction side space-time plot at 90% span.

APPENDIX B

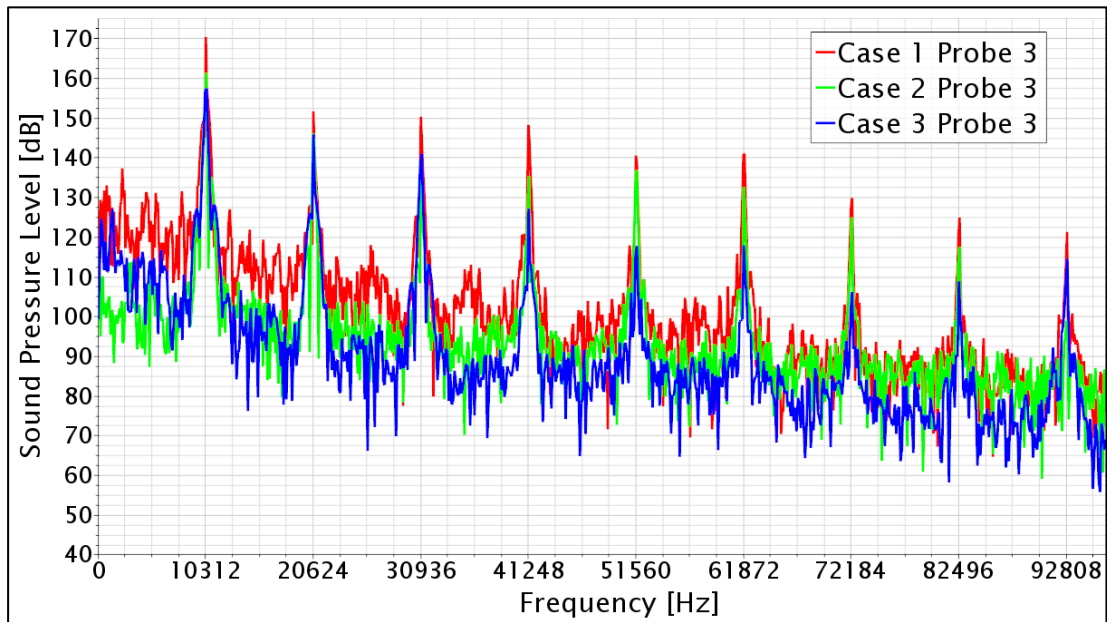


Figure B.1: Noise spectrum of near-field probe 3.

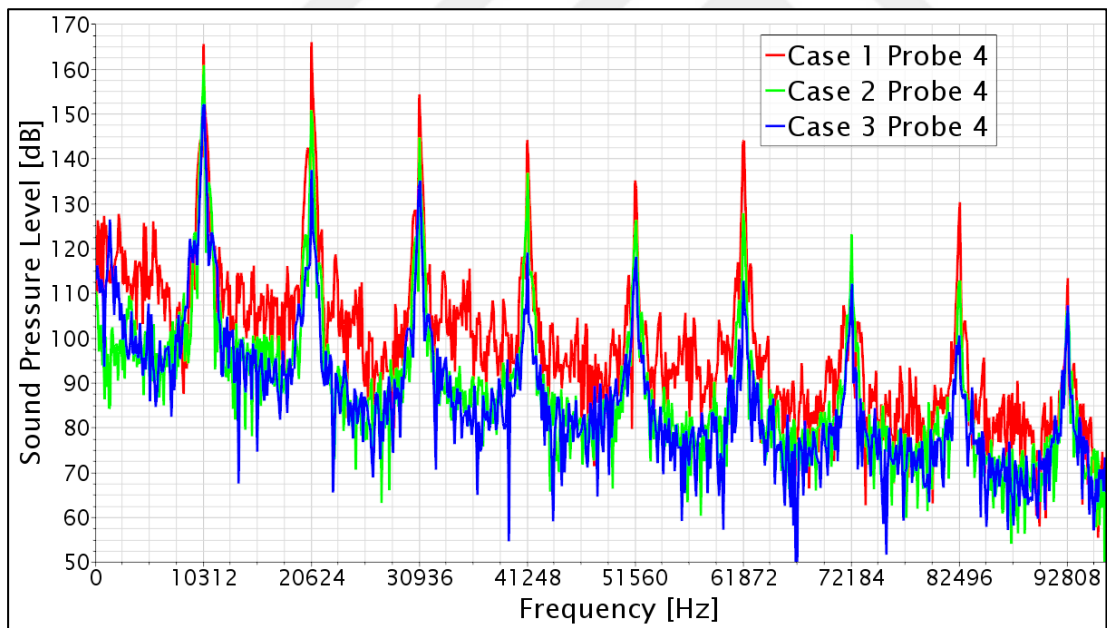


Figure B.2: Noise spectrum of near-field probe 4.

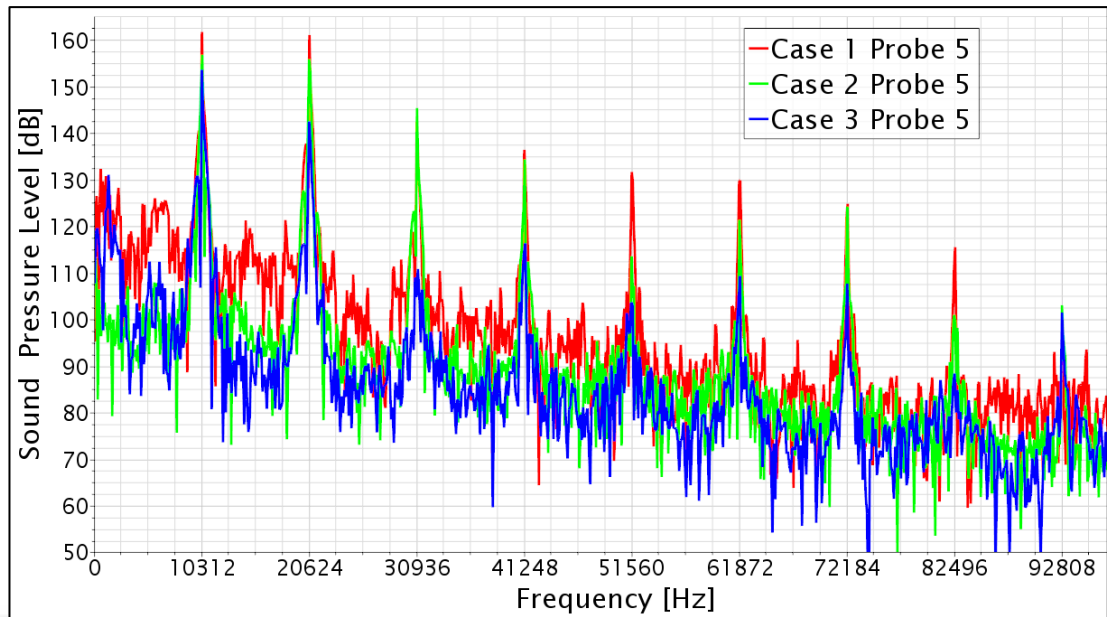


Figure B.3: Noise spectrum of near-field probe 5.

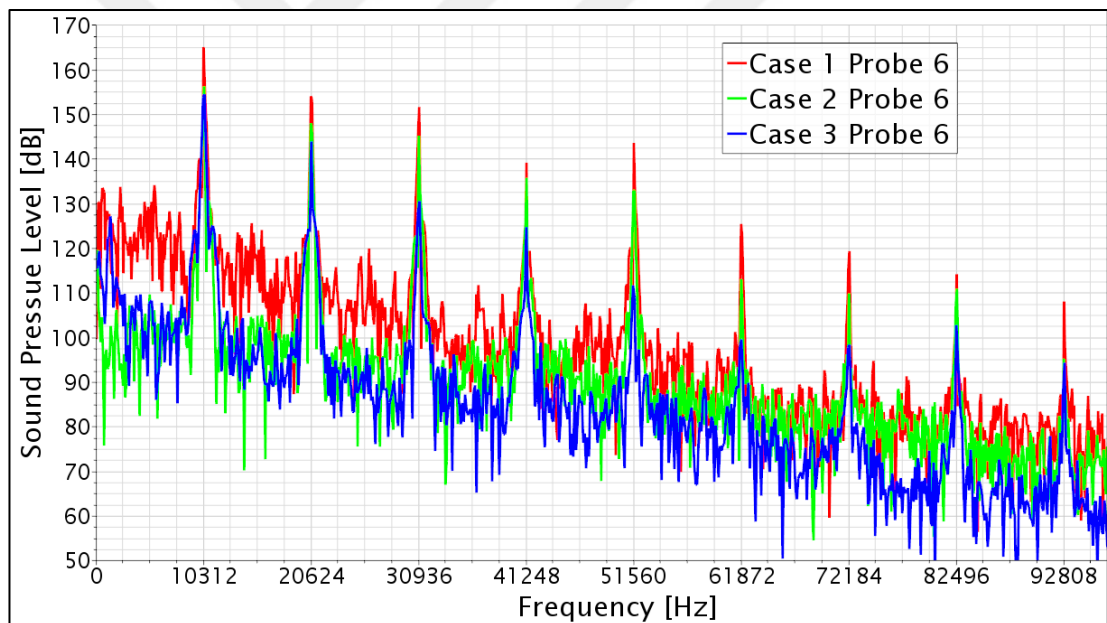


Figure B.4: Noise spectrum of near-field probe 6.

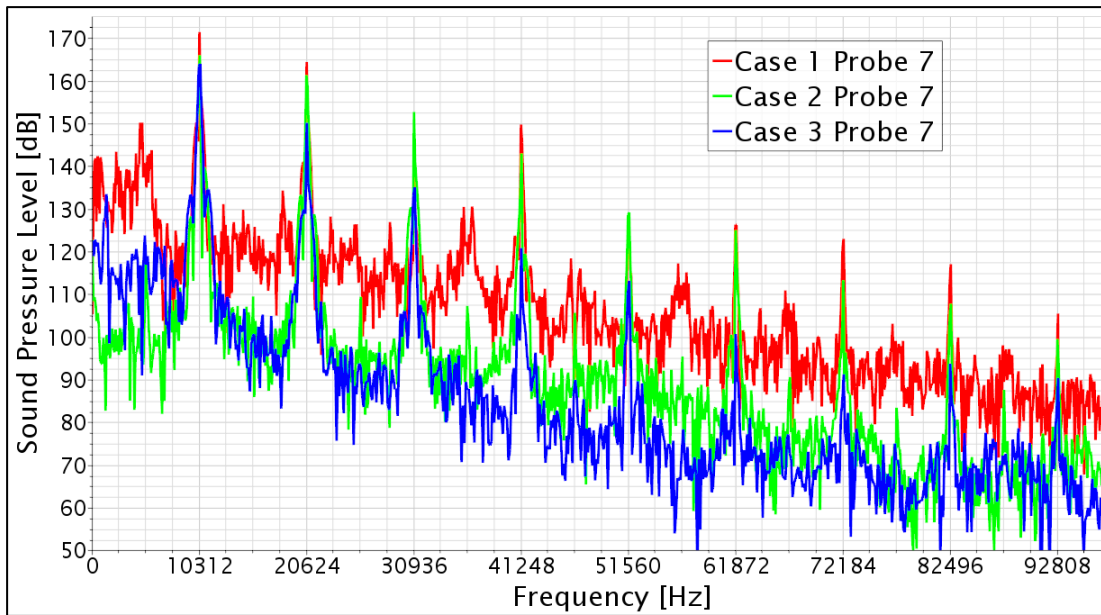


Figure B.5: Noise spectrum of near-field probe 7.

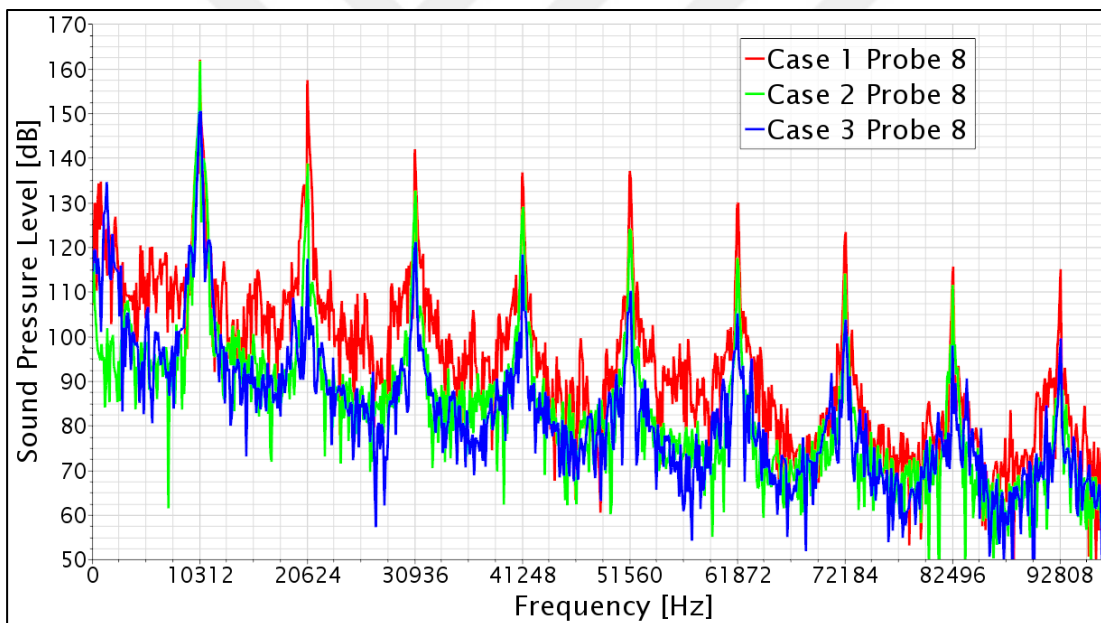


Figure B.6: Noise spectrum of near-field probe 8.

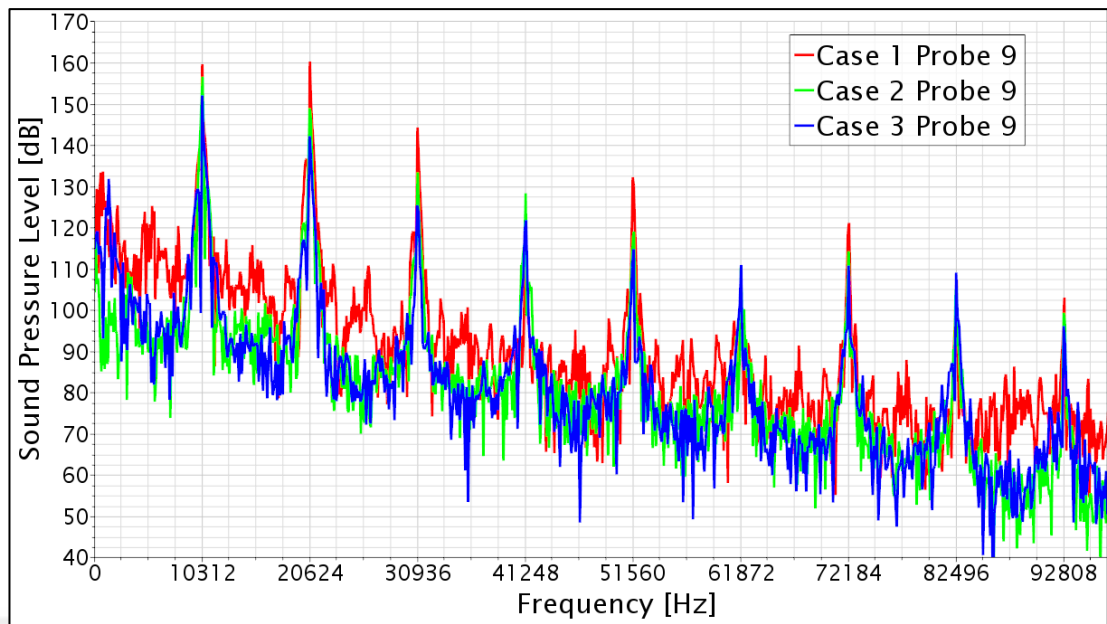


Figure B.7: Noise spectrum of near-field probe 9.

APPENDIX C

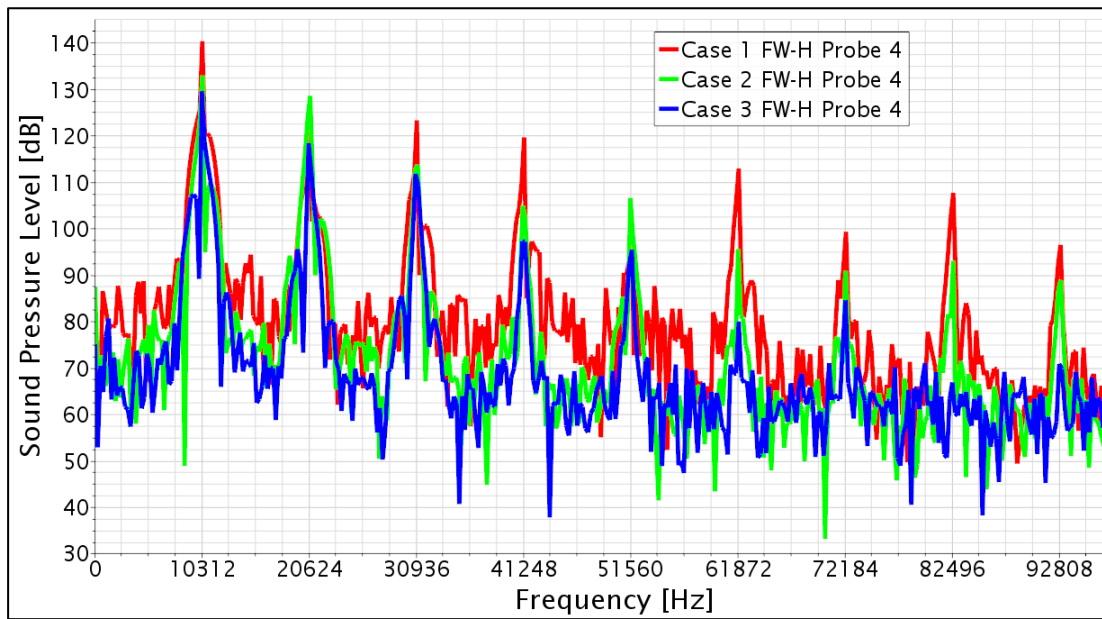


Figure C.1: Noise spectrum of far-field probe 4.

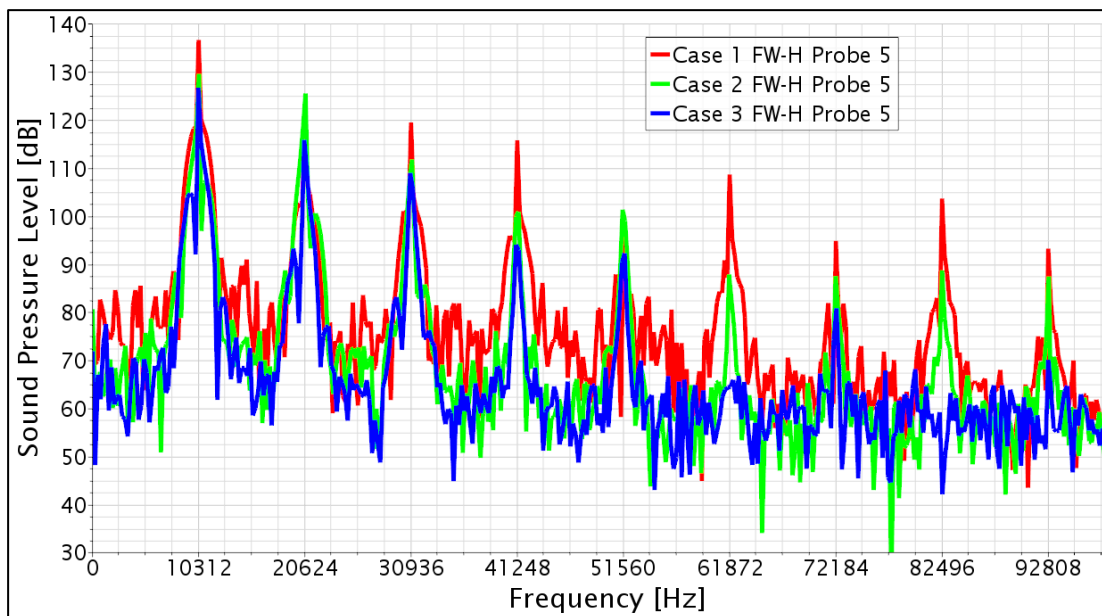


Figure C.2: Noise spectrum of far-field probe 5.

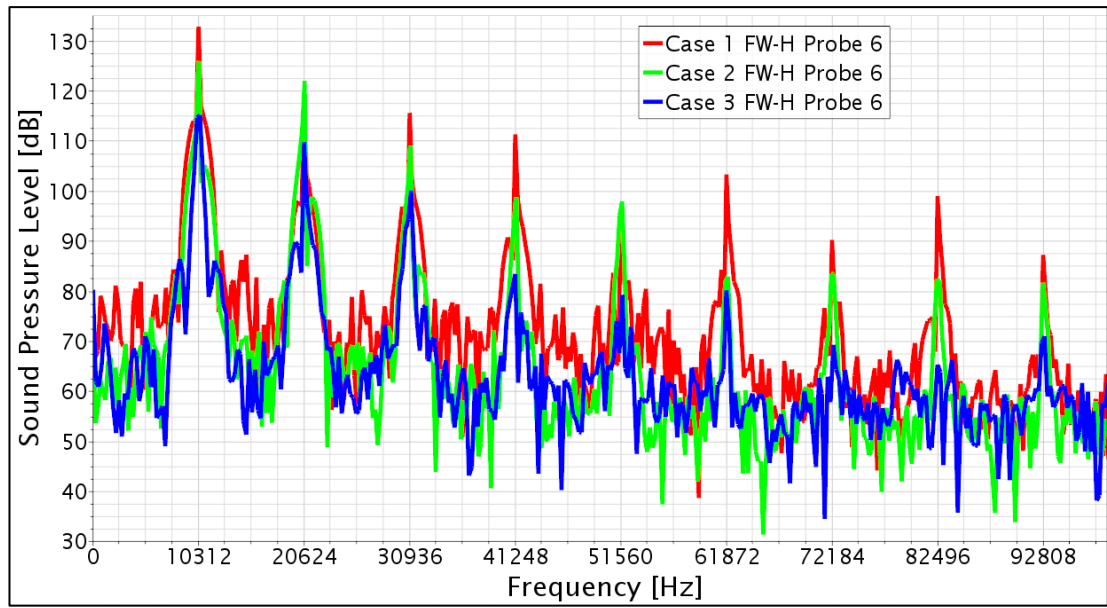


Figure C.3: Noise spectrum of far-field probe 6.



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