

T.R.
GEBZE TECHNICAL UNIVERSITY
INSTITUTE OF ENERGY TECHNOLOGIES

**TRANSIENT BLADE TIP CLEARANCE CALCULATION,
OPTIMISATION AND VALIDATION OF CENTRIFUGAL
COMPRESSOR IN TURBOSHAFT ENGINES**



BEDİR İNCİ

**A THESIS SUBMITTED FOR THE DEGREE OF
MASTER OF SCIENCE
DEPARTMENT OF APPLIED PROPULSION SYSTEM
DESIGN & ENGINEERING FOR AEROSPACE TECHNOLOGIES**

GEBZE
2022

T.R.
GEBZE TECHNICAL UNIVERSITY
INSTITUTE OF ENERGY TECHNOLOGIES

**TRANSIENT BLADE TIP CLEARANCE
CALCULATION, OPTIMISATION AND
VALIDATION OF CENTRIFUGAL
COMPRESSOR IN TURBOSHAFT ENGINES**

BEDİR İNCİ

**A THESIS SUBMITTED FOR THE DEGREE OF
MASTER OF SCIENCE
DEPARTMENT OF APPLIED PROPULSION SYSTEM
DESIGN & ENGINEERING FOR AEROSPACE
TECHNOLOGIES**

**THESIS SUPERVISOR
PROF. DR. İLYAS KANDEMİR**

GEBZE

2022



YÜKSEK LİSANS JÜRİ ONAY FORMU

GTÜ Enerji Teknolojileri Enstitüsü Yönetim Kurulu'nun 30/05/2022 tarih ve 2022/07 sayılı kararıyla oluşturulan jüri tarafından 08/06/2022 tarihinde tez savunma sınavı yapılan BEDİR İNCİ'nin tez çalışması Enerji Teknolojileri Anabilim Dalında YÜKSEK LİSANS tezi olarak kabul edilmiştir.

JÜRİ

ÜYE

(TEZ DANIŞMANI) : Prof. Dr. İLYAS KANDEMİR

ÜYE

: Prof. Dr. MAHMUT FARUK AKŞİT

ÜYE

: Dr. Öğr. Üyesi SEDAT TOKGÖZ

ONAY

Gebze Teknik Üniversitesi Enerji Teknolojileri Enstitüsü Yönetim Kurulu'nun
...../...../..... tarih ve/..... sayılı kararı.

İMZA/MÜHÜR

SUMMARY

Blade tip clearance has significant effect on compressor performance, operability and rotor-stator integrity. Excessive tip clearance allows a portion of air to flow over blade tip without creating useful work. On the other hand, tight clearances arise rubbing risk between blade and casing. Therefore tip clearances have to be designed carefully during the design phases. Generally, checking only steady state tip clearances is not enough to understand surge margin clearances and rub risk during operation. Transient tip clearance assessment which includes acceleration, deceleration and shutdown part of mission are calculated in order to understand how tip clearances are changing during flight. In this thesis, two stage centrifugal compressor transient blade tip clearances are calculated and optimized to avoid any rub and surge margin reduction. Analytical results show that stage 1 tip clearances are affected from operability vortexes which change metal temperatures at different power level. That's why shroud displacements are not predicted accurately for some portion of the test mission. In order to fix this issue, inducer bleed port is added on shroud as a design improvement. Predictions show that stage 2 tip clearances are sensitive to mechanical loads. It is obtained that return channel component moving too much during the acceleration and deceleration conditions. Therefore, design changes are implemented to reduce sensitivity and improve tip clearances.

Analytical results compared with real time tip clearance measurement from the test and results show that analytical data has good agreement with the experimental measurements. Clearances stay within the magnitude of measurement uncertainty.

Key Words: Blade tip clearance, surge margin, rub, steady state, transient, performance, efficiency, optimization, verification, compressor

ÖZET

Kanat uç açıklıklarının kompresör performansı, çalışabilirliği ve rotor-stator bütünlüğü üzerinde önemli etkisi vardır. Büyük uç açıklıkları, faydalı bir iş yaratmadan havanın bir kısmının kanat ucu üzerinden akmasına izin verdiğinden çalışma sırasında verimlilik düşüşüne neden olur. Öte yandan, dar açıklıklar, kanat ve kılıf arasında sürtünme riski doğuracaktır. Bu nedenle, tasarım aşamalarında uç açıklıkları dikkatli bir şekilde tasarlanmalıdır. Genel olarak, yalnızca kararlı haldeki uç açıklıklarını kontrol etmek, çalışma sırasında dalgalanma marjı boşluklarını ve sürtünme riskini anlamak için yeterli değildir. Uçuş sırasında uç açıklıklarının nasıl değiştiğini anlamak için hızlanma, yavaşlama ve kapatma kısımlarını içeren geçici uç açıklık değerlendirmesi yapılmalıdır. Bu tezde, sürtmeyi engellemek ve dalgalanma marjının azalmasını önlemek için iki aşamalı santrifüj kompresördeki geçici kanat uç açıklıkları hesaplanmıştır. Analizlerde kademe 1 uç açıklıkları, farklı güç seviyelerinde metal sıcaklığını değiştiren çalışabilirlik girdaplarından etkilendiği görülmüştür. Bu nedenle kılıf deplasmanları doğru şekilde tahmin edilememiştir. Bu durumu düzeltmek amacıyla kılıf üzerine pasif indükleyici çıkış portu eklenerek tasarım iyileştirmesi sağlanmıştır. Açıklık hesaplamaları kademe 2 uç açıklıklarının mekanik yüklere duyarlı olduğunu göstermiştir. Dönüş kanalı parçasının, motorun hızlanması veya yavaşlaması sırasında çok fazla hareket ettiği görülmüştür. Bu durumu iyileştirmek ve hassasiyeti azaltmak amacıyla tasarım değişiklikleri yapılmıştır.

Açıklık hesaplarını ve metotlarını doğrulamak amacıyla gerçek zamanlı kanat uç açıklıkları ölçümü gerçekleştirilmiş ve analitik sonuçlarla karşılaştırılmıştır. Elde edilen veriler, analitik sonuçların deneysel ölçümlerle iyi bir uyum içinde olduğunu ve açıklıkların ölçüm belirsizliğinin içinde kaldığını göstermektedir.

Anahtar Kelimeler: Kanat uç açıklığı, Akış ayrılması marjı, sürtme, zamana bağlı, performans, verim, optimizasyon, doğrulama, kompresör

ACKNOWLEDGMENTS

I would like to express my deep and sincere gratitude to my supervisor Prof. Dr. İlyas Kandemir who guides and supports me on this thesis as well as Prof. Dr. Osman Saim Dinç who initiates this Master of Science program to collaborate between TEI and GTU.

I want to express my warm and sincere thanks to my co-advisor Sinan Şal for his vision and technical guidance during this journey.

Lastly, I thank TUSAŞ Engine Industry Inc. management for permission to publish this thesis.

TABLE OF CONTENTS

	<u>Page</u>
ÖZET	v
SUMMARY	iv
ACKNOWLEDMENTS	vi
TABLE of CONTENTS	vii
LIST of ABBREVIATIONS and ACRONYMS	ix
LIST of FIGURES	xi
1. INTRODUCTION	1
1.1 Scope of Thesis	2
2. EFFECTS OF TIP CLEARANCE ON COMPRESSOR	3
2.1 Tip Clearance Leakages	3
2.2 Tip Clearance Effect on Operability	3
2.3 Tip Clearance Effect on Efficiency	4
3. BLADE TIP CLEARANCES	6
3.1 Importance of Transient Tip Clearance Analysis	7
3.1.1 Hot Rotor Reburst	11
3.1.2 Hot Rotor Restart	11
3.1.3 Rotor Bow	12
3.2 Transient Clearance Trends	12
3.3 Abradable coatings	13
3.4 Inputs for Blade Tip Clearance Calculation	15
3.4.1 Performance Cycle	16
3.4.2 2D Whole Engine Thermo-Mechanical Analysis	16
3.4.3 Assembly tolerances (1.5 RSS tolerances)	17
3.4.4 Rotor Vibration:	19

3.4.5	3D stator Distortion	20
3.4.6	3D Blade Displacements	22
3.4.7	Squeeze Film Damping	24
3.4.8	Blade Tip Clearance Calculation Methodology	24
3.4.9	Blade Tip Clearance Results	27
4.	EXPERIMENTAL VERIFICATION OF TIP CLEARANCE	33
4.1	Capacitive Measurement Technique	33
4.2	Tip Clearance Instrumentation	34
4.3	Pre-test Measurements	36
4.4	Uncertainty Analysis of Measurements	37
4.5	Tip Clearance Measurements	38
4.5.1	Cold Crank Measurements	38
4.5.2	Tip Clearance Measurement Results	39
5.	CONCLUSIONS	41
	REFERENCES	42
	BIOGRAPHY	44

LIST OF ABBREVIATIONS AND ACRONYMS

<u>Abbreviations And Acronyms</u>	<u>Explanations</u>
Avg	: Average
2D	: Two Dimensional
3D	: Three Dimensional
b	: Blade Height
BC	: Boundary Condition
BTC	: Blade Tip Clearance
CBC	: Cold Build Clearance
C _c	Cold clearance
CFD	: Computational Fluid Dynamics
C _h	Hot Clearance
Cl	: Clearance
COG	: Center of Gravity
disp	: Displacement
e	: Eccentricity
FEM	: Finite Element Modeling
g	: Gravity
GGT	: Gas Generator Turbine
ISA	: The International Standard Atmosphere
LE	: Leading Edge
Max	: Maximum
Min	: Minimum
MTO	: Maximum Take-Off
N	: Shaft speed
NG	: Gas Generator Shaft Speed
PH	: High Pressure
PL	: Low Pressure
R _b	: 3D Blade displacement
Rbearing	: Bearing axial play
R _d	: Disc displacement
Ref	: Reference

ref	: Reference
r_g	: Radius of Gravity
R_o	: 3D Distortion/Ovalization
R_s	: Stator displacement
RSS	: Root Sum Square
R_{stall}	: Stall displacement
R_v	: Rotor vibration
SFC	: Specific Fuel Consumption
SFD	: Squeeze Film Damping
SLS	: Sea Level Standard
SSTO	: Steady State Take-Off
Stg	: Stage
T3	: Temperature compressor exit
TE	: Trailing Edge
TO	: Take-Off
TOT_c	: Total Closure
TR	: Transient
WE	: Whole Engine

LIST OF FIGURES

<u>Figure No</u>	<u>Page</u>
1-1: Two stage centrifugal compressor.	2
2-1: Tip leakage flow and secondary flows.	3
2-2: Tip clearance effect on operability and pressure ratio.	4
2-3: Efficiency drop due to tip clearance changes.	5
2-4: Effect of tip clearance on pressure ratio.	5
3-1: Tip clearance locations.	6
3-2: Tip clearance ratio.	7
3-3: Blade tip rub.	8
3-4: Dynamic blade cracking.	8
3-5: Transient blade tip clearance changes.	9
3-6: Typical transient square cycle mission.	10
3-7: Tip clearance changes during hot restart.	11
3-8: Rotor bow due to temperature gradient.	12
3-9: Tip clearance trends.	13
3-10: Hard and soft abradable effect on tip clearance.	14
3-11: Different form clearance gaps due to rubbing.	15
3-12: Typical square cycle.	17
3-13: 2D whole engine analysis	17
3-14: Typical tolerance stack up sheet.	18
3-15: Rotor-stator eccentricity.	18
3-16: Rotor orbit due to vibration.	19
3-17: Normalized vibration displacement.	20
3-18: 3D stator ovalization.	21
3-19: Rotor-stator centerline shift.	21
3-20: Definition of 3d case distortion.	21
3-21: Points of 3d airfoil displacements.	23
3-22: Configuration squeeze film damper.	24
3-23: Blade tip clearance locations.	25
3-24: Displacement locations of impeller.	27
3-25: Transient mission profile.	28
3-26: Stage 1 LE tip clearance.	28

3-27: Stage 1 TE blade tip clearance.	29
3-28: Stage 1 deformation and temperature plot	30
3-29: Stage 2 LE tip clearances.	31
3-30: Stage 2 TE tip clearances.	31
3-31: Deformed shape of return channel and stage 2 shroud.	32
3-32: New and old design of two stage compressor.	32
4-1: Schematic view of clearance probe.	33
4-2: Various type of capacitive probe according to installation.	34
4-3: Instrumentation scheme of tip clearance probe.	34
4-4: Picture of instrumented stage 1 shroud	36
4-5: Recess measurement.	37
4-6: Stage 1 LE tip clearance comparison.	39
4-7: Stage 1 TE tip clearance comparison.	40

LIST OF TABLES

<u>Table No:</u>	<u>Page</u>
3-1: Tip Clearance Targets.	7
3-2: Displacement delta between 2D and 3D impeller.	27
4-1: Clearance measurement uncertainties.	38
4-2: Deviation between CBC and probe measurements	38



1. INTRODUCTION

Blade tip clearance is the radial distance between blade and casing. It is one of the significant parameter as it directly affects operability, efficiency, SFC and deterioration of compressor. From the rotor integrity point of view, if tip clearance is too tight in operation, rubbing risk arises between casing and blade. Excessive rubs may cause failure of blades which creates safety issues on the engine. On the other hand, too large tip clearance causes tip leakage which results in surge margin and efficiency deficit in compressor. Therefore, optimum tip clearance is essential for efficiency and safe operation of the compressor.

Since steady state tip clearance calculation is not enough to find minimum clearances during the operation, transient tip clearance assessment has to be performed to understand operability and performance of compressor whilst avoiding any blade rubbing during the operation.

There are a few publications regarding the transient tip clearance calculation of centrifugal compressor in the literature. Arkhipov et al [18] calculated transient tip clearances using 2D FE model including 3D effects such as airfoil twisting and case bending. Analytical results were compared with real engine testing clearances and good agreement obtained between them. Giovannetti et al [13] evaluated tip clearance reduction using abradable material at inner shroud surface. All inputs obtained during operation such as vibration, thermal-mechanical expansion and assembly tolerance stacks have been considered in clearance assessment. Study shows that tip clearance reduction at hot condition using abradable material is greater than 90% with respect to the hot design point of without abradable coating. Luca et al. [23] calculated radial and axial tip clearances using 2D FE model. 3D effect of airfoil displacements due to twisting and bending were considered. 3D casing FE model was also created to calculate casing distortions due to temperature differences. Proposed procedure was verified with test data using clearance probe and thermocouples. Measurements had good agreement with predictions in all transient operation conditions.

1.1 Scope of Thesis

The main goal of this thesis is to calculate and optimize transient blade tip clearance of centrifugal compressor and compare the analytical results with experimental data using a real time tip clearance measurements.

In scope of this thesis, analytical calculations are performed for two stage centrifugal compressor as shown in Figure 1-1 using 2D thermo-mechanical models by including also vibration, assembly tolerances and ovalization effects. Centrifugal compressor has abradable coating on the shroud component at both stages. In case of any rub, no blade or shroud base material deformation is expected. All clearance calculations are done according to no rub at worst case condition.

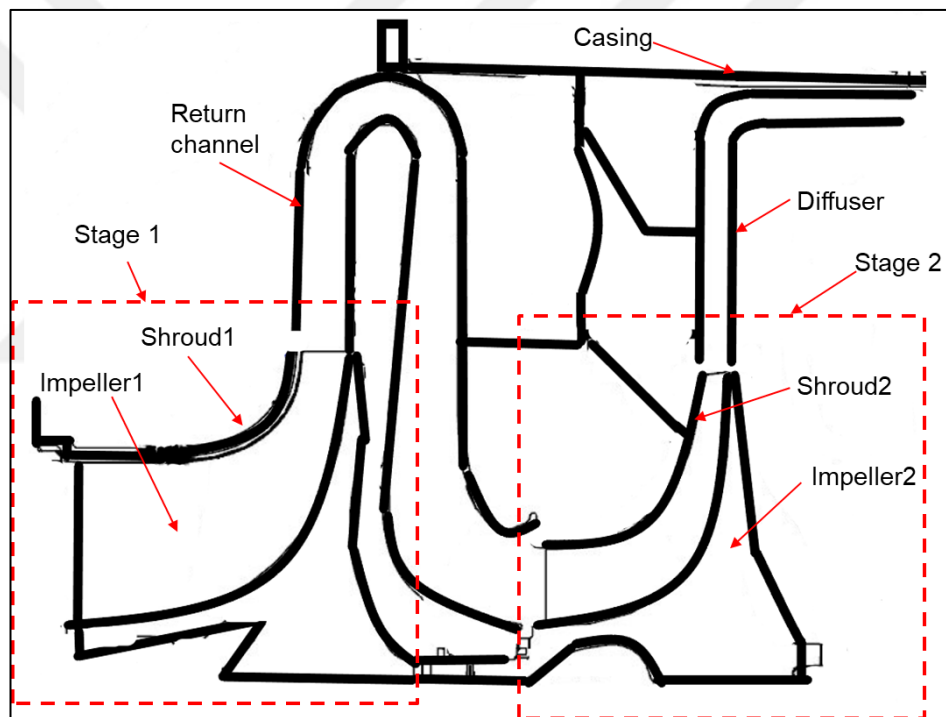


Figure 1-1: Two Stage Centrifugal Compressor.

Non-contact real time tip clearance probes are installed on stage 1 to verify analytical results with test data. Two probes are installed on LE and two ones on TE to understand average tip clearances during the test. Instrumentation details are shown in Section 4.2

2. EFFECTS OF TIP CLEARANCE ON COMPRESSOR

2.1 Tip Clearance Leakages

One of the significant sources of loss is tip clearance at gas turbine compressors. Since tip clearances cannot be removed completely, leakage flows over blade tips occurs and affects internal flow structure, stability margin and energy transport. [1] Figure 2-1 shows secondary losses and tip clearance losses. When tip clearances get larger, air leaks over blade tip from pressure side to suction side which disturbs air flow around the airfoil. Since the turbulent air comes from the front stages, effect of tip loss mechanism gets bigger through the aft stages.

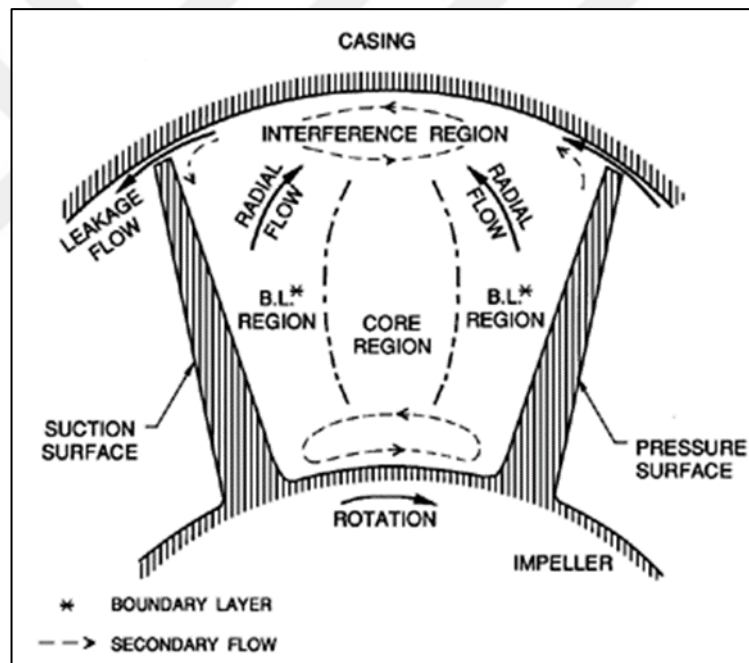


Figure 2-1: Tip leakage flow and secondary flows.

2.2 Tip Clearance Effect on Operability

Compressor stability is one of the most important issues for the operability and performance. It is affected by tip clearances too much. Figure 2-2 shows that when tip clearances get larger speed lines moves down and left side, which causes less pressure ratio at a flow and less capacity at constant speed. Additionally, surge

margin line is getting closer to the operating line and it must therefore be prevented if possible.

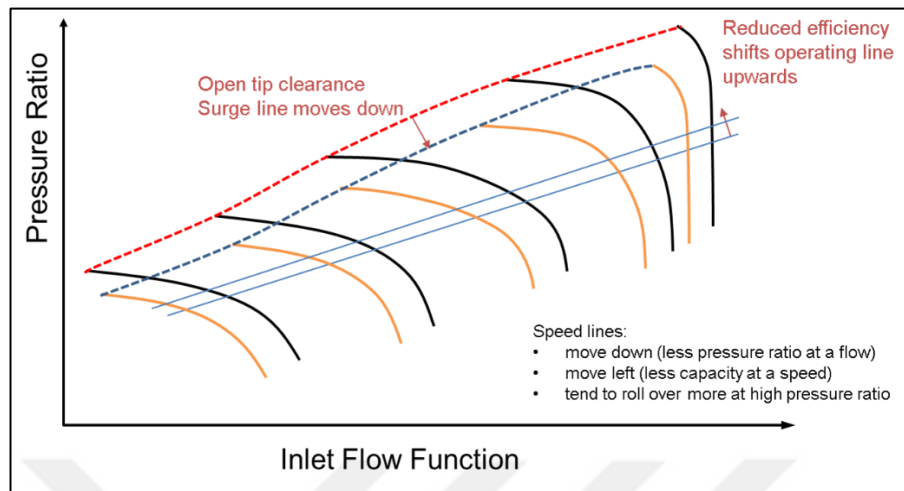


Figure 2-2: Tip clearance effect on operability and pressure ratio.

2.3 Tip Clearance Effect on Efficiency

Researchers have investigated the tip clearance has an effect on the centrifugal compressor flow field for a long time. Results show that a larger tip clearance causes larger efficiency drop in compressor. [2] Figure 2-3 shows efficiency drop due to tip clearance changes in centrifugal compressor in the literature. Different peripheral Mach number speeds either with vaneless or vaned diffusers are tested in centrifugal compressor. Conclusion is when the relative tip clearance (Clearance / rotor exit blade height) increased by 10%, efficiency drops by 3 %.

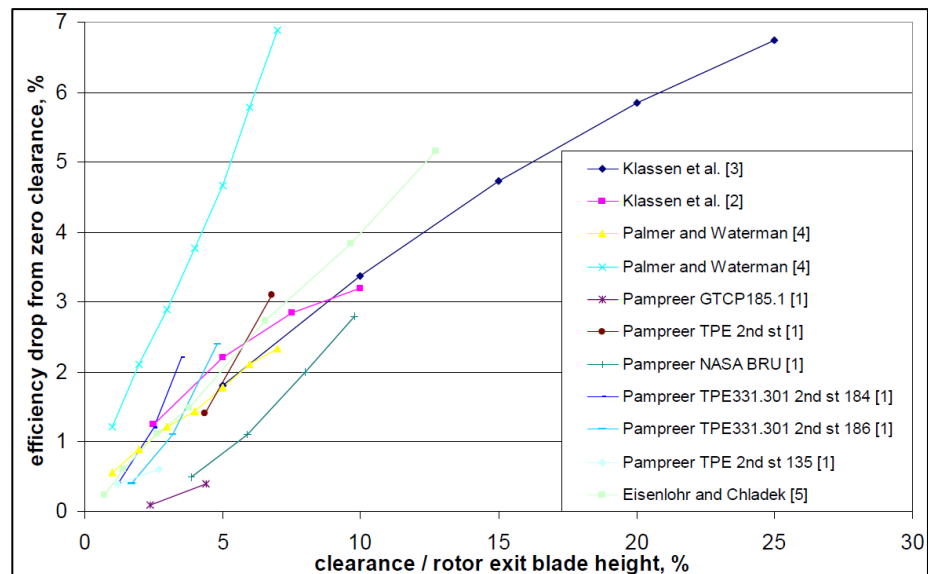


Figure 2-3: Efficiency drop due to tip clearance changes.

Denmark et al [3], investigated the effect of tip clearance variations on performance with analytical as well as experimental method. Experimental results were as close to each other as analysis. In the study, 9 different tip openings (0, 0.1mm, 0.2mm, 0.3mm, 0.4mm, 0.5mm, 0.6mm, 0.8mm and 1mm) were analyzed by programmed fluid dynamics method. As shown in Figure 2-4, if the tip clearance increases, pressure ratio decreases at the same flow rate. For instance at 0.35 kg/s flow rate, it decreased by approximately 15% when the tip clearance was increased from 0 to 1 mm.

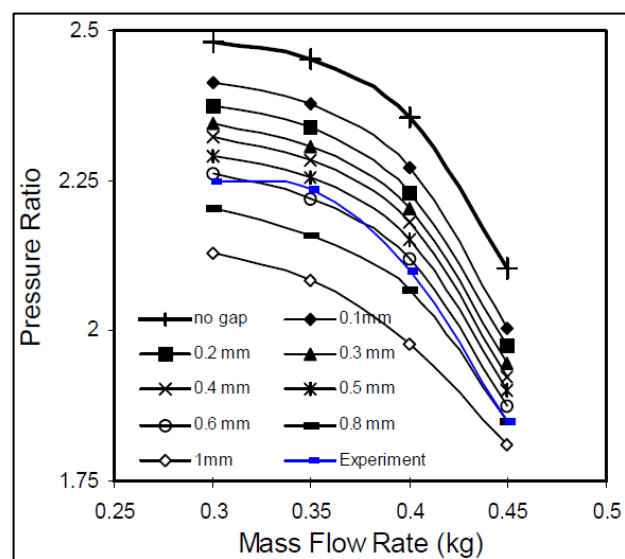


Figure 2-4: Effect of tip clearance on pressure ratio.

3. BLADE TIP CLEARANCES

In gas turbine engines, clearance is the gap between rotor and stator. Tip clearance is the radial closure between end of rotating and the static components. Figure 3-1 shows centrifugal compressor tip clearance locations calculated in the scope of the thesis.

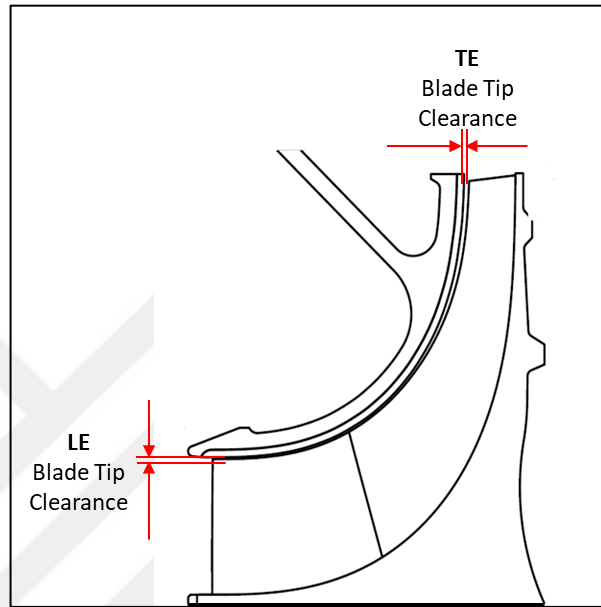


Figure 3-1: Tip Clearance Locations.

Generally, blade tip clearances are specified relative to blade height in the literature as shown in Figure 3-2 [4] and it is calculated according to equation (3-1). In this thesis, target blade tip clearances that should be caught are shown in Table 3-1. Proposed target tip clearance is usually 1-2% of blade height in the literature. It is hard to catch less than 2% (Cl/b) target clearances for the short blades. Because target clearances are getting tighter with short blades according to literature however, some adders such as assembly tolerance stacks, maneuver loads still exist and can be as same as big engines. Thus, tip clearance target can be bigger than 2 % for small engines.

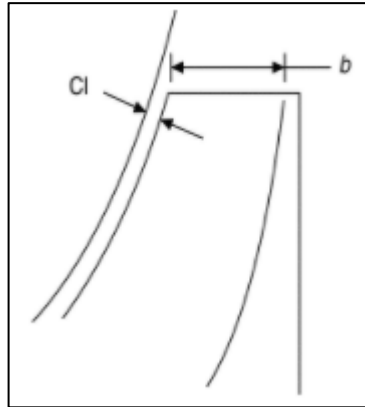


Figure 3-2: Tip clearance ratio.

$$\left(\frac{Cl}{b}\right) \% = \frac{Cl}{b} \times 100\% \quad (3-1)$$

Table 3-1: Tip clearance targets.

Cl/b	LE target	TE target
Stage 1	0,7%	2,2%
Stage 2	1,3%	3,6%

3.1 Importance of Transient Tip Clearance Analysis

As mentioned in Section 2, larger tip clearance affects performance and operability of compressor in a negative way and tighter tip clearance increase risk of rubbing which may cause failure of components. Figure 3-3 [5] shows consequence of tip rub due to insufficient built gap. Material rolls on blade surface and this situation causes flow separation from airfoil surface and opens clearance that results in inefficient compressor. Further tip rubs may cause blade tip crops or liberation as shown in Figure 3-4 [6].

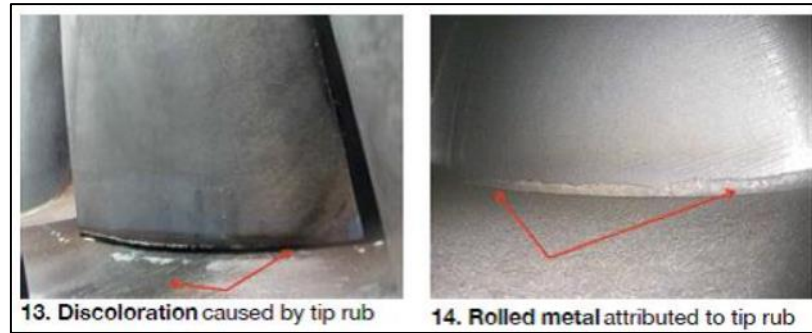


Figure 3-3: Blade tip rub.

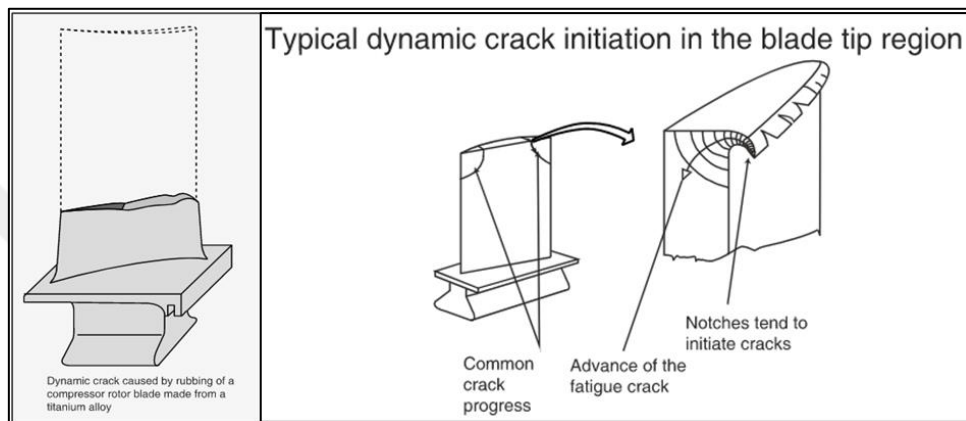


Figure 3-4: Dynamic blade cracking.

Most of tip clearance assessments comprise aero design point condition in the literature and transient condition analysis is less compared to steady state condition. And also checking clearances under steady state condition is easy because all flow and metal temperatures are thermally stabilized. However, pinch point which is defined as minimum clearance can occur during start-up, acceleration, deceleration, and shutdown or hot re-starts. Therefore, transient clearance assessments are necessary to optimize clearances for all operation conditions.

Cheng et al [7] analyzed compressor tip clearance transiently using FEM model. Results compared with real time tip clearance measurements. Measurements show good agreement with the designed tip clearance value excluding the startup operation. In this paper, shutdown clearances are also investigated and both calculated and measurement data show that minimum clearances occur at during the shutdown. Different shutdown times are investigated to optimize clearances and avoid for blade rubbing.

Figure 3-5 [8] shows potential rub conditions during flight. When the engine accelerates, tip clearances get tighter due to centrifugal forces acting on rotor. If there is not enough gap for this operation, potential rub may occur. Similarly, after deceleration if the engine re-accelerates, hot rotor expands radially faster than stator due to centrifugal forces and new pinch point may occur at this time. Therefore when the cold gap is set, all these operating conditions have to be taken into the consideration for clearance assessments.

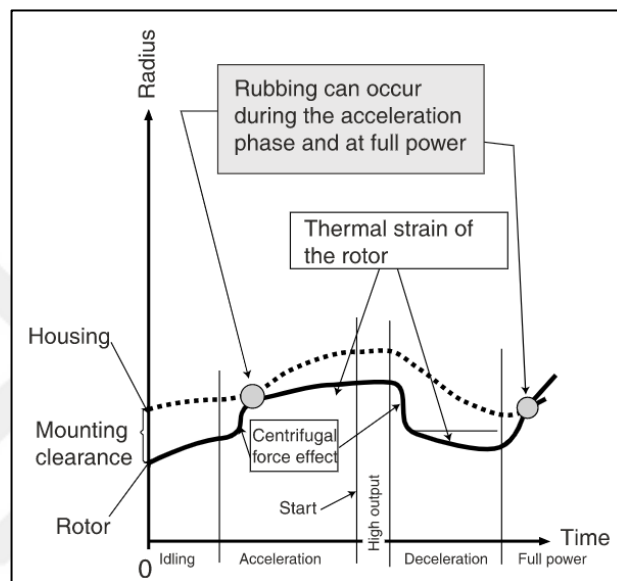


Figure 3-5: Transient blade tip clearance changes.

Transient blade tip clearance of a centrifugal compressor changes during start-up, acceleration and deceleration. In order to maximize the efficiency, tightest clearances should be at Maximum Take-off (MTO) condition where aero design point is set. To achieve this condition thermal and mechanical effect on rotor – stator components have to be understood well. It means that, during acceleration and deceleration, mechanical loads act on components faster than thermal loads and rotor thermally stabilizes slower than stator part. Therefore, stabilization time and magnitude of each load must be separated from each other to understand how to optimize clearances. Basically the aim of the optimization is to match the expansion and transient responses of the rotor and casing within the design limitations. To do that, a square cycle from idle-MTO-idle is used to consider the response of the system to step changes.

Figure 3-6 [9] shows typical transient square cycle mission which contains fast acceleration and deceleration between idle and MTO conditions. The bottom plot also shows tip clearance closure relative to square cycle mission as a function of time. It can be clearly seen that effect of thermal and mechanical response of rotor – stator components during the mission. When the engine starts (1) and accelerates to idle (2) tip clearance is rapidly diminished due to centrifugal force on the rotor. Then rotor-stator components are thermally stabilized and clearances get constant (3). When engine accelerates from idle to MTO, clearances are getting tighter rapidly due to centrifugal force (4) then stator expands due to heating and clearances are getting larger (5). After that point, temperature of stator stabilized however, rotor continues to heating because mass of rotor much more than stator and thereby heats slower than stator (6). Clearances are getting constant when rotor – stator thermally stabilized (7). When the engine decelerates to idle, tip clearances are increasing rapidly due to removal of centrifugal force on rotor (8). Then stator cools down faster than rotor and clearances are getting tighter again until temperature of stator stabilized (9). However, rotor continues to cooling down and clearances starts to open until rotor is also thermally stabilized (11).

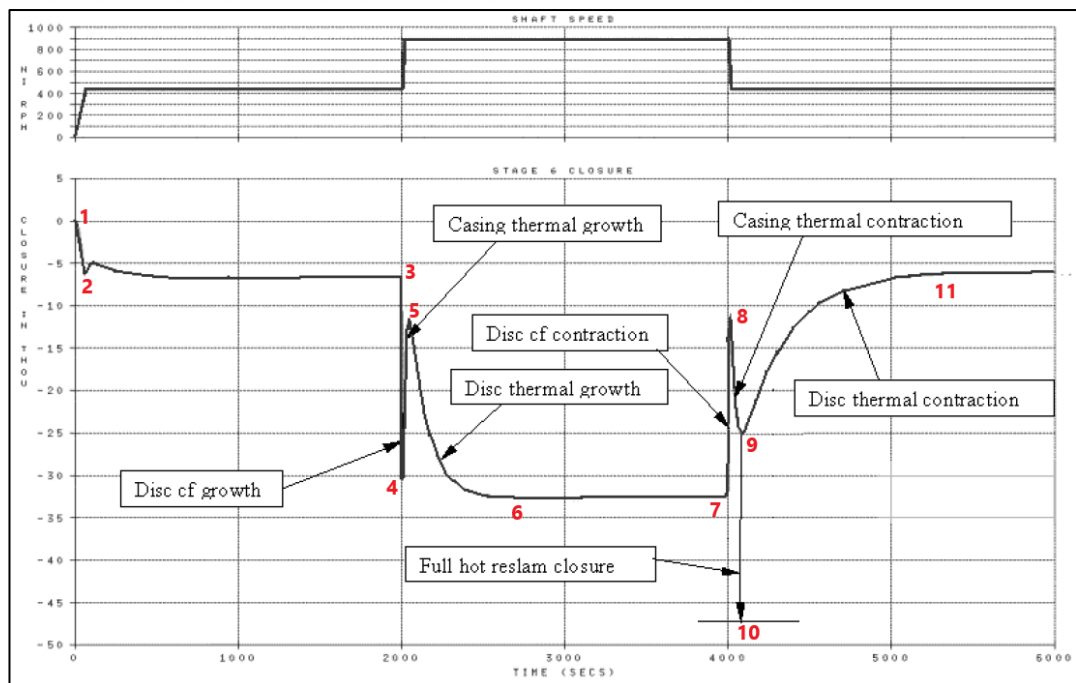


Figure 3-6: Typical transient square cycle mission.

3.1.1 Hot Rotor Reburst

Another important operation that must be considered during clearance assessment is hot rotor reburst or hot reslam. If the engine is re-accelerated to MTO condition from point 9 in Figure 3-6, tighter clearance is obtained than the initial condition because rotor is still warm and on top of that, displacements due to centrifugal load is added (10). Cold clearances should be adjusted according to this operation in order to prevent rubbing.

3.1.2 Hot Rotor Restart

Another important condition that has to be checked for tip clearance is hot restart condition. When the engine shutdowns, stator component cools down faster than the rotor which makes tip clearances tighter. If the engine is restarted at a time where tip clearance pinched and since the rotor is still warm, more closure can be obtained than the cold start as shown in Figure 3-7. Closure coming due to hot restart must be considered in clearance assessments to avoid any rub and maintain the engine integrity.

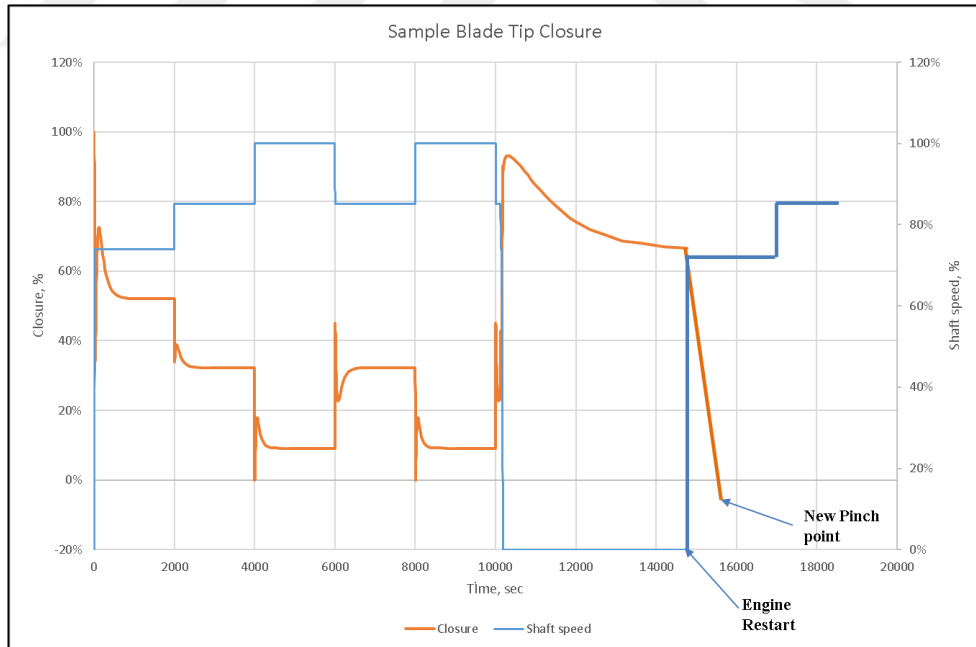


Figure 3-7: Tip clearance changes during hot restart.

3.1.3 Rotor Bow

During the operation, rotor temperatures increase or decrease axisymmetric due to continues gas flow. After the shutdown, there is no forced flow anymore and asymmetric cooling occurs along the circumference of rotor. Rotor is still warm and exchanges heat with the surrounding gas through the natural convection. After a while, top side of rotor becomes warmer than the bottom side and this causes rotor to bow as shown in Figure 3-8 [10]. Rotor bow needs to be considered for clearance assessments because it creates large imbalances on rotor and during the hot restart, high vibrations can occur which may cause blade tip rubbing if there is not enough gap. Impact of rotor bow is excluded in this thesis. Since shaft of this engine is short compared to turbo fan engines, it is assumed that bow rotor adder is less.

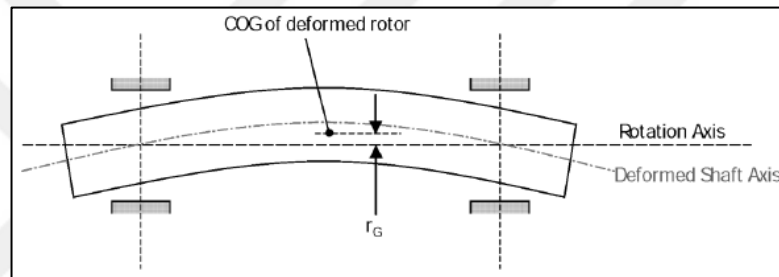


Figure 3-8: Rotor bow due to temperature gradient.

3.2 Transient Clearance Trends

In order to maximize the compressor efficiency from the clearance standpoint of view, tip clearances have to be the tightest at the aero design point. Three different possible clearance trends can be obtained due to relative radial displacements of rotor and stator components as shown in Figure 3-9 [11]. Generally, thermal loads generate different clearance trends. Ideally, when the rotor and stator components thermally match, pinch point occurs at maximum steady state condition which is the desired one for maximum efficiency. If thermal response of casing is faster than rotor, tip clearance gets larger after acceleration and thereby surge margin is affected in a negative way. If thermal response of rotor is faster than casing then pinch point occurs at acceleration and cold clearance needs to be opened to prevent any rub between rotor and casing. Thus steady state hot clearance value increases. In order to increase efficiency, as mentioned in section 3.1 rotor – stator components must be

thermally matched. To achieve that condition, following properties have to be considered.

- Large volumes heat slowly
- Large surface areas heat quickly
- Area/Volume ratio is critical
- Higher heat transfer coefficient areas will heat quicker
- Material with large capacitance heats slower

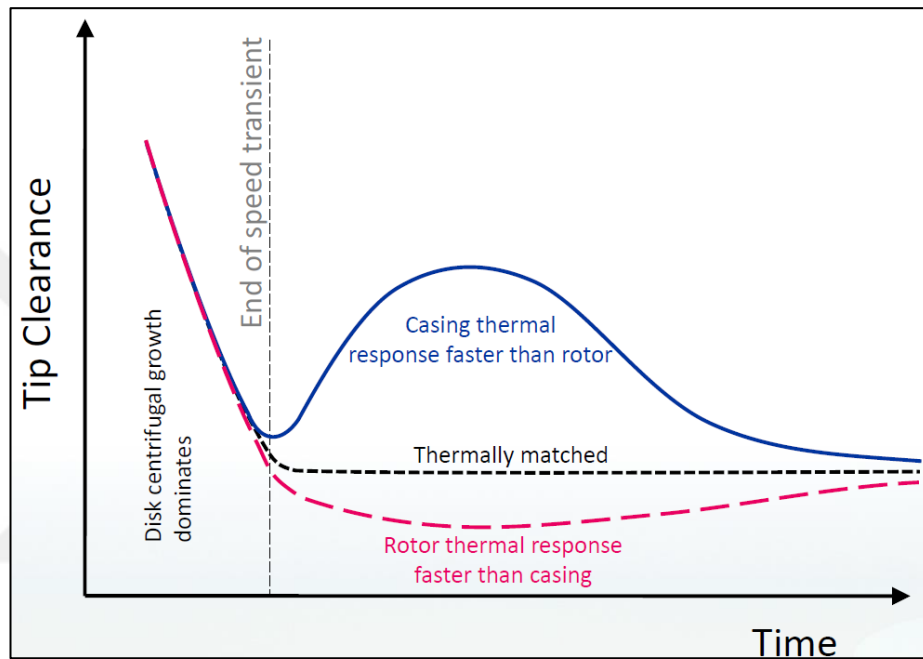


Figure 3-9: Tip clearance trends.

3.3 Abradable coatings

Various technologies such as honeycomb, labyrinth seals, carbon seals are developed to reduce clearances. However these technologies are used in hot locations, generally abradable coatings are used in cold modules for clearance reduction. If there is a contact between rotor and stator components due to tight clearance adjustments, there can be material loss at tip location which causes efficiency loss and damage on blades and stator components or large cold gaps are set for blade tips to prevent any rub condition. In order to reduce clearances and prevent any material loss in case of contact, abradable coatings maybe used at stator side. As shown in Figure 3-10 the abradable is soft material and in case of any contact, no material loss is obtained at the blade side and clearances can still be kept

tight. Just local material removal is obtained at stator side which does not affect efficiency so much [12].

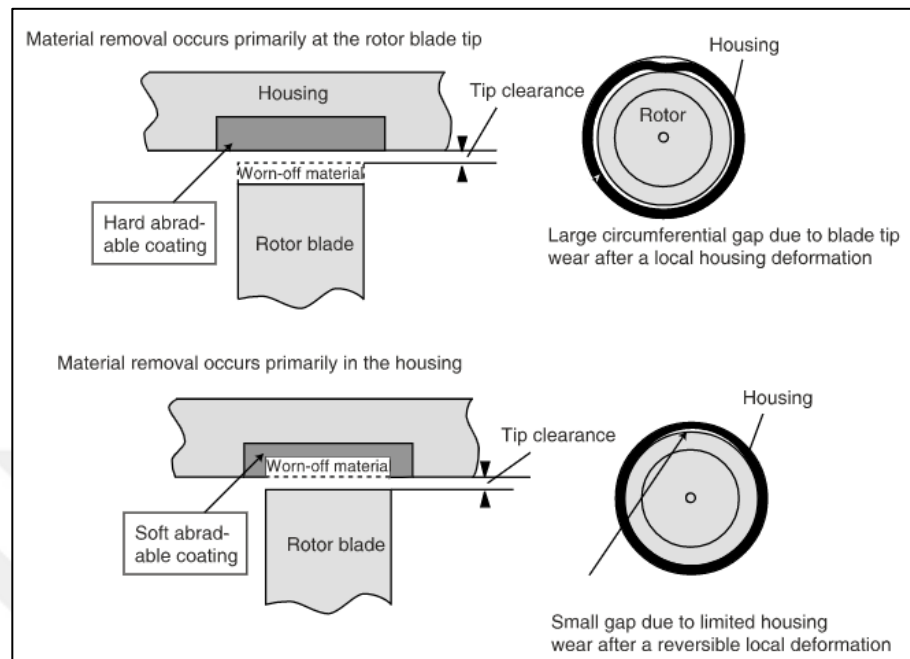


Figure 3-10: Hard and soft abrasible effect on tip clearance.

Rubbing creates different form of clearance gaps on stator component. [14] Figure 3-11 shows that how rubbing changes clearance gaps under different effects when soft or hard abrasibles are used. At first column, maneuver loads cause radial rotor shift relative to stator and in abrasible systems, this condition causes local rubs on the stator, whereas in hard or no abrasible systems, rubbing causes the tips of the entire blading to be worn down. Second column shows rubbing form due to imbalances. Deflected rotors rub entire circumference of stator in abrasible systems, whereas in hard or no abrasible systems, only small blade tips of rotor section are worn down. Generally, a combination of these effects causes more complex shapes of stator.

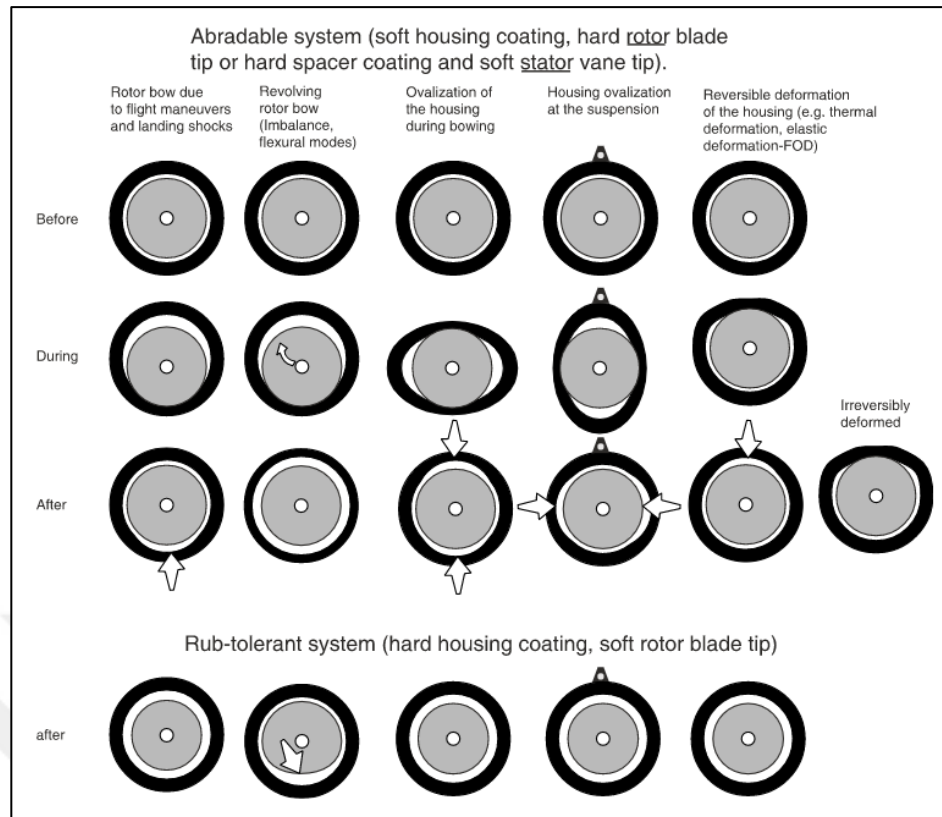


Figure 3-11: Different form clearance gaps due to rubbing.

3.4 Inputs for Blade Tip Clearance Calculation

Evaluation of blade tip clearances is one of the important topics in turbomachinery. During the operation, engine components are exposed to thermal loads and mechanical forces caused by centrifugal force, aerodynamic loads, flight maneuvers, gusts, or landing shocks, gas forces, vibrations, gyroscopic forces [15]. All these loads affect clearances between engine components and have to be evaluated in blade tip clearance assessments. Main inputs for the calculation are as follows:

- Performance Cycle
- 2D Whole Engine thermo- mechanical analysis
- Assembly tolerances (1.5 RSS tolerances)
- Rotor vibration
- 3D stator distortion (Maneuver, gyroscopic load)
- 3D blade deflections
- SFD
- Stall displacements of impeller (Excluded in scope of this thesis)

3.4.1 Performance Cycle:

Gas turbine engine components are designed according to thermal and mechanical loads that are exposed to during the operation. Performance cycle includes operating conditions such as shaft speed, inlet - outlet temperatures and pressure, mass flow rates, fuel etc. and engine components are analyzed according to these operating performance cycles so that deflections can be used in clearance assessments. Additionally, target clearances are evaluated according to performance design point which is SLS ISA MTO condition for centrifugal compressor in this thesis.

3.4.2 2D Whole Engine Thermo-Mechanical Analysis

Rotor and stator displacements used in clearance assessments are taken from transient 2D axisymmetric whole engine FEM analysis. Non-axisymmetric components are also modeled with appropriate techniques. 2D model includes following inputs:

- Thermal loads
- Mechanical loads
 - Speed
 - Gas pressure
 - Aero loads
- Transient performance cycle
- BCs: interference, preload, contacts

For transient displacement analysis generally square cycle mission profile that comprises fast acceleration and deceleration from idle to take-off condition is used. Figure 3-12 shows variation of T3 as a function of time for a typical square cycle. For detailed transient analysis low power or high power points can be added in mission profile [9]. In this thesis, 2D thermo-mechanical analysis is performed in a FE program that provides shorter running time compared to other software.

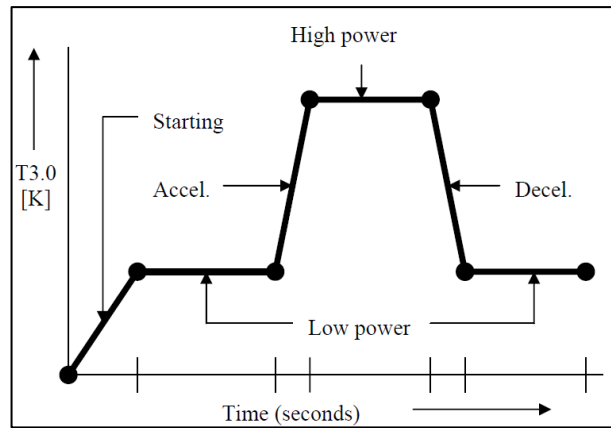


Figure 3-12: Typical square cycle.

Figure 3-13 shows typical 2D whole engine thermos mechanical analysis for commercial turbofan engine [16]. Generally, total displacements (thermal + mechanical) are used in clearance calculations however, in order to understand which load (thermal or mechanical) drives the rotor-stator displacement, thermal only displacements are also analyzed.

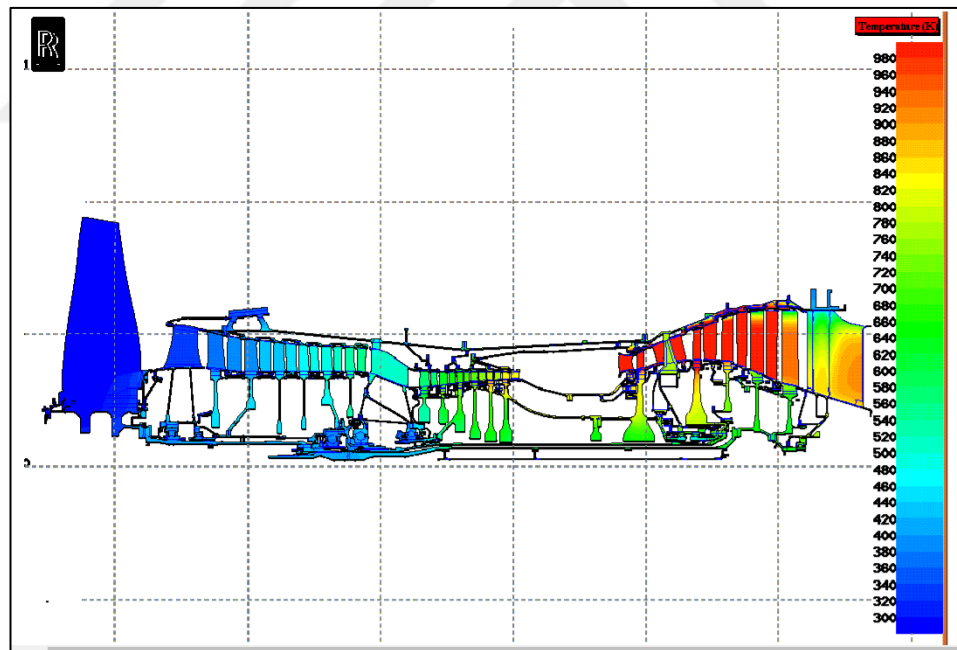


Figure 3-13: 2D Whole engine analysis

3.4.3 Assembly tolerances (1.5 RSS tolerances):

Engine components are not always manufactured in nominal dimensions as they are designed. They have minimum and maximum tolerances and dimension of the component has to stay within the specified tolerances in drawing. After the assembly,

these tolerances are added on top of each other and final assembly tolerances are defined. Generally, RSS methodology is used for tolerance stack calculation and multiplied by 1.5 to increase percentage of probability. Because it is considered that probability of all dimensions to be manufactured at minimum or maximum tolerances is very low. [17] Figure 3-14: shows a sample tolerance stack calculation spreadsheet that includes minimum, maximum and nominal dimensions after assembly.

Program:						Stackup Information			
Product: Part Number Rev Description						Stackup No:			
Problem:						Date:			
Objective:						Revision:			
						Units:			
						Direction:			
						Author:			
Description of Component / Assy	Part Number	Rev	Item	Description	+ Dims	- Dims	Tol	Worst-Case % Contrib	Dim / Tol Source & Calcs
A			1			9.1000	± 0.1000	10.0%	
B			2	POSITION			± 0.1500	15.0%	
C			3	BONUS			± 0.1000	10.0%	
D			4	SHIFT			± 0.0750	7.5%	
E			5		5.2250		± 0.0750	7.5%	
F			6			5.4500	± 0.0500	5.0%	
G			7	POSITION			± 0.2000	20.0%	
H			8	BONUS			± 0.1000	10.0%	
I			9	SHIFT			± 0.0500	5.0%	
J			10		10.4000		± 0.1000	10.0%	
Dimension Totals:					15.6250	14.5500			
Nominal Distance: Pos Dims - Neg Dims =					1.0750				
RESULTS:					Norm	Tol	Min	Max	
					Arithmetic Stack (Worst-Case)	1.0750 ± 1.0000	0.0750	2.0750	
					Statistical Stack (RSS)	1.0750 ± 0.4345	0.0000	1.0000	
Multiplier: 1.5					Adjusted Statistical Multiplier RSS	1.0750 ± 0.6517	0.4233	1.7267	

Figure 3-14: Typical tolerance stack up sheet.

Figure 3-15 shows rotor-stator eccentricity due to assembly tolerances. This adder must be included in clearance assessments to avoid any potential rub. It is clearly seen that due to assembly tolerances one side of clearance gets tighter than the opposite side. This condition affects the minimum and maximum tip clearances of compressor which is very important for surge margin stack up.

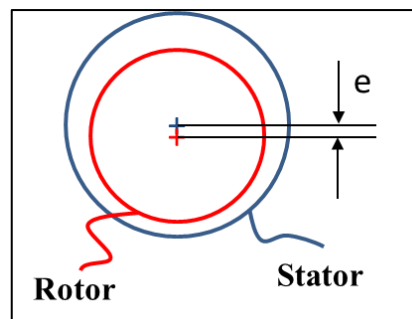


Figure 3-15: Rotor-stator eccentricity.

3.4.4 Rotor Vibration:

After the rotor assembly, balance operation is performed on rotor assembly to reduce imbalances until an acceptable level. When the engine is started, rotor starts to vibrate and creates an orbit around the rotor. It is significant to run a rotor dynamic analysis to evaluate radial displacements of rotor due to vibration over the entire engine operating cycle. In order to avoid any tip rub, rotor vibration effect has to be included in clearance assessments. Figure 3-16 shows rotor orbit due to vibration at steady state condition [13].

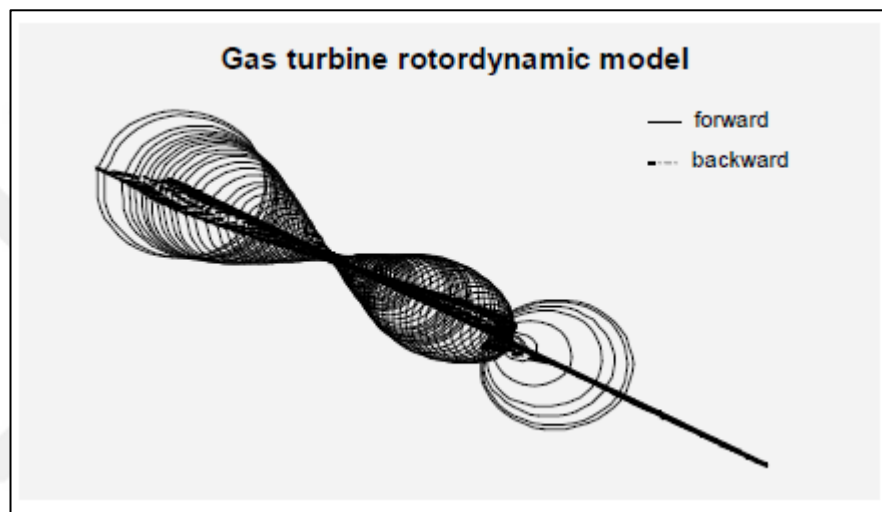


Figure 3-16: Rotor orbit due to vibration.

[13] Figure 3-17 shows normalized sample vibration displacements as a function of normalized speed. Maximum displacement obtained at 60% shaft speed and decreases as the speed reaches to 100% (nearly 3 times lower than maximum value). These displacements must be added transiently on top of displacements because it can change pinch point of clearances.

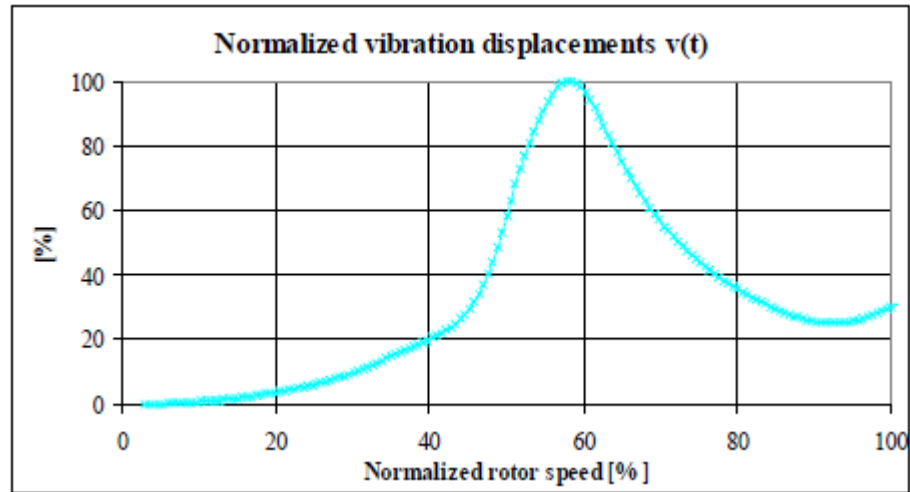


Figure 3-17: Normalized vibration displacement.

3.4.5 3D Stator Distortion

During the operation, rotor and structures of the engine are subjected to deformation due to thermal and mechanical loading. Casings do not expand perfectly round due to uneven circumferential temperatures and maneuver loads. Therefore, shape of casing can change during the operation. Basically, source 3D stator distortions are as follows:

- Engine mounts cause case bending under Thrust
- Maneuver loads
- Nacelle aerodynamic under Take-off rotation
- Gyroscopic moments on rotor under pitch and yaw maneuver
- Non-axisymmetric thermal (bleed port, uneven ventilation)
- Split flanges

These loads are applicable for large turbofan engines. Maneuver load, gyroscopic moment, non-axisymmetric temperatures are only considered as source of case distortions in this thesis.

Asymmetric effects can be split into two groups:

- Those which cause the stator not to be a perfect circle as shown in Figure 3-18
- Those which cause a centerline shift between stator and rotor as shown in Figure 3-19

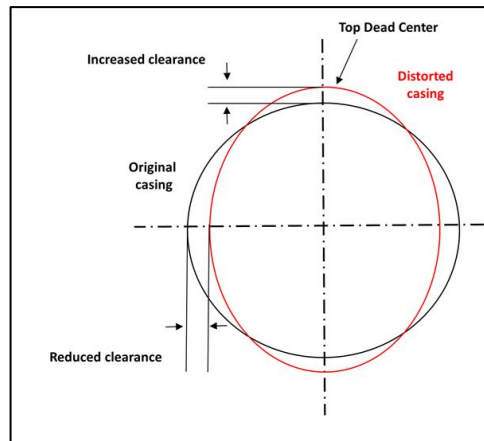


Figure 3-18: 3D Stator ovalization.

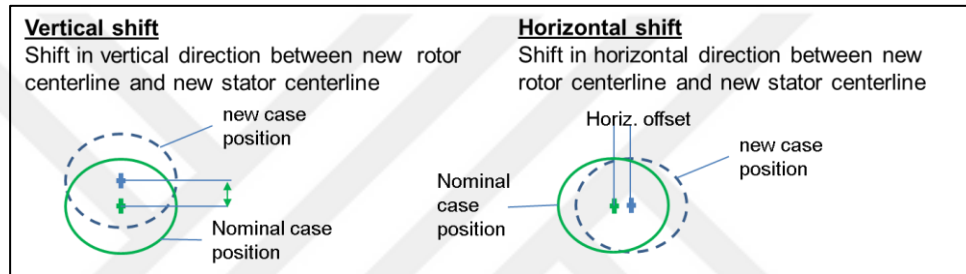


Figure 3-19: Rotor-stator centerline shift.

3D case distortions are important for clearance assessments for two reasons. Firstly, it is required to calculate minimum clearances to avoid any potential rub, secondly, local maximum tip clearances are important for surge margin stack up. Figure 3-20 shows circumferential stator deflections with respect to rotor and local clearances that affect the blade tip rub and surge margin.

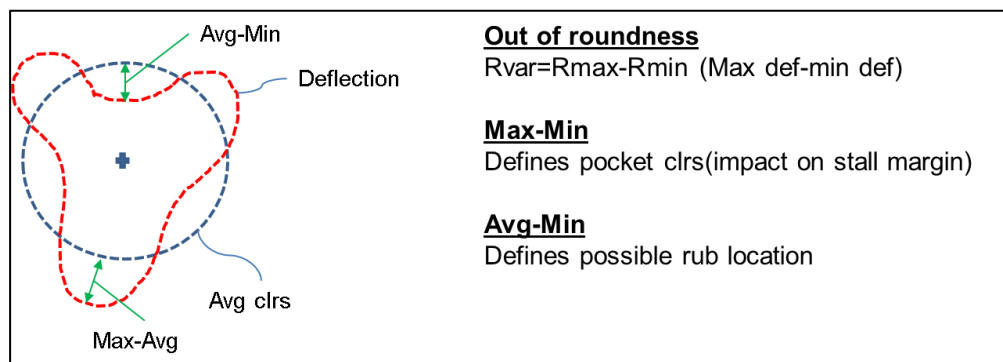


Figure 3-20: Definition of 3D case distortion.

3.4.6 3D Blade Displacements:

One of the inputs used in clearance assessment is rotor blade displacement. Since it has twisted geometry and cannot be used directly from 2D model, 3D transient blade/airfoil analysis has to be run for clearance calculations. Running a transient 3D blade analysis takes too much time and expensive methodology. Thus, transient scaling methodology is used as an alternative to the 3D transient analysis approach.

During the engine run, airfoils are exposed to thermal, aero, and centrifugal loads. In order to scale 3D airfoil displacements transiently, steady state displacements of each type of loadings are needed as mentioned Arkhipov A. N et al [18].

Inputs needed for scaling are as follows:

- 2D WE Thermal only & Total displacements
- Transient cycle parameters
- 3D Blade Steady state take off displacements with following conditions separately.
 - Thermal only displacements
 - Speed only displacements (Analysis has to include temperature but thermal expansion coefficient has to be 0)
 - Pressure only displacements
- 3D Blade stress analysis cycle parameters

3D Airfoil displacements are extracted from points shown in Figure 3-21. Radial displacements for LE & TE side are calculated by taking delta between points 1-3 and 2-4 respectively.

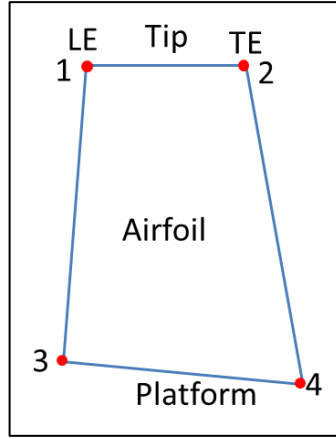


Figure 3-21: Points of 3D airfoil displacements.

3.4.6.1 Scaling methodology:

Scaling of thermal displacements: It is assumed that transient thermal expansion trend of 2D airfoil is similar to 3D transient airfoil. However, 2D thermal airfoil displacements cannot be used directly and still need correction according to 3D thermal displacements. They are corrected by multiplying with ratio of 3D SSTO thermal and 2D thermal SSTO displacements. This ratio is considered constant for all transient operation as stated in Equation (3-2)

$$3D_Thermal_{TR} = 2D_Thermal_{TR} \times \frac{3D_Thermal\ disp_{SSTO_ref}}{2D_Thermal\ disp_{SSTO}} \quad (3-2)$$

Limitation for thermal scaling is 2D WE model has to contain airfoils to reflect transient thermal trend.

Scaling of speed only displacements: Displacements due to centrifugal load is proportional to square of rotational speed and it can be scaled transiently as stated in equation (3-4) [18]

$$3D_speed_{TR} = 3D_speed_{ref} \times \frac{N_{TR}^2}{N_{ref}^2} \quad (3-3)$$

Scaling of pressure only displacements: Pressure only displacements occur due to gas load and it is proportional to pressure difference of blade stage of front and aft end.

$$3D_pressure\ disp_{TR} = 3D_pressure\ disp_{ref} \times \frac{PH_{TR} - PL_{TR}}{PH_{ref} - PL_{ref}} \quad (3-4)$$

Total 3D airfoil displacement is sum of 3D thermal, 3D speed and 3D pressure only displacements as stated in equation

$$Total\ Airfoil\ Disp. = Thermal\ disp. + Speed\ disp. + Pressure\ disp. \quad (3-5)$$

3.4.7 Squeeze Film Damping

“Squeeze film bearing dampers are lubricated elements providing viscous damping in mechanical systems. Squeeze film dampers in rotating machinery provide structural isolation, reduce the amplitudes of rotor response to imbalance, and in some instances, assist to suppress rotor dynamic instability.” [19]. Figure 3-24 shows configuration of Squeeze film damper and since the rotor moves in oil film in the bearing, its effect has to be included in clearance calculations.

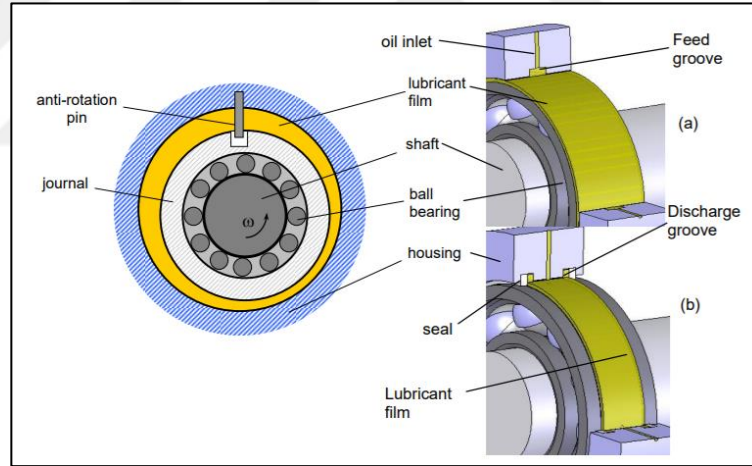


Figure 3-22: Configuration squeeze film damper.

3.4.8 Blade Tip Clearance Calculation Methodology

Compressor tip clearance is distance between stator and rotor components as shown Figure 3-23. It is calculated based on no rub requirement in this thesis using the methodology stated in equations (3-6), (3-7) and (3-8), (3-9) for LE and TE respectively. In order to find hot tip clearances first of all, cold clearance has to be calculated by including all adders that cause closure during the transient mission. Both stator and rotor displacements comprise effects of centrifugal, pressure and

thermal loads in it. Radial component of blade stall displacements can also be included in LE clearance calculation to avoid metal-metal contact. But if there is an abradable coating on casing side, hot clearances may not be designed according to stall condition, only metal-metal contact checking can be enough.

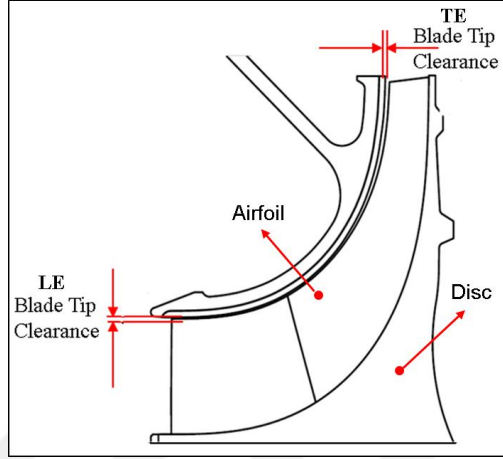


Figure 3-23: Blade tip clearance locations.

$$TOT_{c_LE} = R_s - (R_d + R_b) - (R_v + R_o + 1.5 \times RSS + SFD + R_{stall}) \quad (3-6)$$

Absolute magnitude of total closure value must be equal to cold clearance of blade tip in order to avoid tip rub as stated in equation (3-7).

$$|TOT_{c_LE}| = C_{c_LE} \quad (3-7)$$

LE tip clearance can be calculated as stated in Equations (3-6) and (3-7) however, TE tip clearance must be calculated according to axial clearance calculation methodology as stated in equation (3-8), (3-9) and in that case, both stator and impeller displacement must be calculated relative to thrust bearing, since bearing carries axial load on the rotor and all displacements are taken relative to it. Additionally, bearings have radial gaps between inner and outer races which results in axial movement of shaft when the rotor is loaded. Therefore, this bearing axial movement is included in TE tip clearance calculations. There might be fire risk for titanium blades in case of tip rubs. In order to avoid this condition, axial component of blade stall displacements can be added on total closure formula. It is excluded for hot clearance calculations but metal-metal contact is checked at stall conditions.

$$|TOT_c_{TE}| = (R_d + R_b + R_{bearing}) - R_s - (R_v + R_o + 1.5 RSS + R_{stall}) \quad (3-8)$$

$$|TOT_c_{TE}| = C_c_{TE} \quad (3-9)$$

Once the cold gap is calculated based on no rub requirement then, hot clearances can be estimated by adding hot closures on cold gaps.

$$C_h_{LE} = C_c_{LE} + R_s - (R_d + R_b) \quad (3-10)$$

$$C_h_{TE} = C_c_{TE} + R_s - (R_d + R_b) \quad (3-11)$$

Vibration, 3D distortion, 1.5RSS and SFD values can be considered as safety adders and have to be minimized to get tight clearances at aero design point.

Disc and 3D blade displacements stated in equations (3-6) and (3-8) can be considered as impeller displacements for centrifugal compressors. Since the disc feature of impeller is axisymmetric, displacements can be taken from 2D thermo-mechanical model. Airfoils deflections of impeller can be calculated according to equations specified in section 3.4.

During displacement calculation process of the impeller, it is obtained that 2D displacements of disc feature are not matching well with 3D results. Figure 3-24 shows locations that the displacements are extracted from analysis and Table 3-2 shows axial displacement delta between 2D and 3D analysis results. Delta increases at points close to the TE side. (Negative sign indicates that 3D impeller deflects more than 2D)

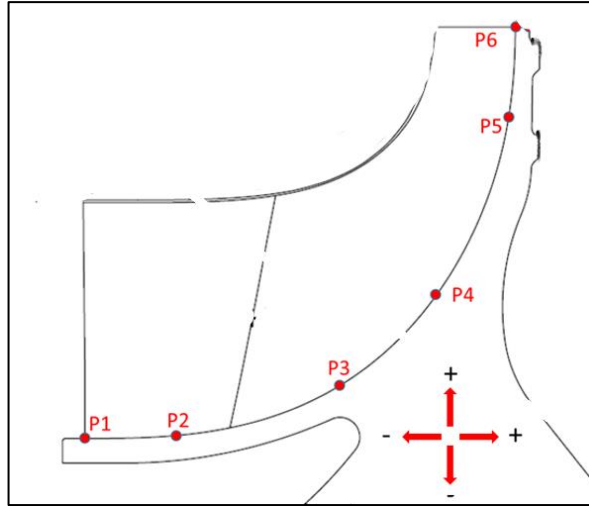


Figure 3-24: Displacement locations of impeller.

Table 3-2: Displacement delta between 2D and 3D impeller.

Delta(2D-3D) of Axial Total Displacements at Aero Design Point						
Unit, mm	P1	P2	P3	P4	P5	P6
Stage 1	-0,025	-0,014	0,001	0,015	0,109	0,140
Stage 2	0,026	0,032	0,039	0,059	-0,013	-0,070

Detailed investigations are done to understand reason of this delta and it is obtained that 2D modeling methodology of impeller creates additional stiffness on it. TE side of impeller in 2D model cannot move forward or aft side due to this stiffness and deflects less than 3D model. In order to eliminate this error, it is decided to run a transient 3D impeller analysis and use displacements directly from it in clearance calculations.

3.4.9 Blade Tip Clearance Results

Compressor tip clearances are calculated according to transient mission shown in Figure 3-25. It is a typical square cycle that comprises ground idle, Flight idle and Take off conditions. Acceleration and deceleration are kept as 3 seconds for all transient conditions.

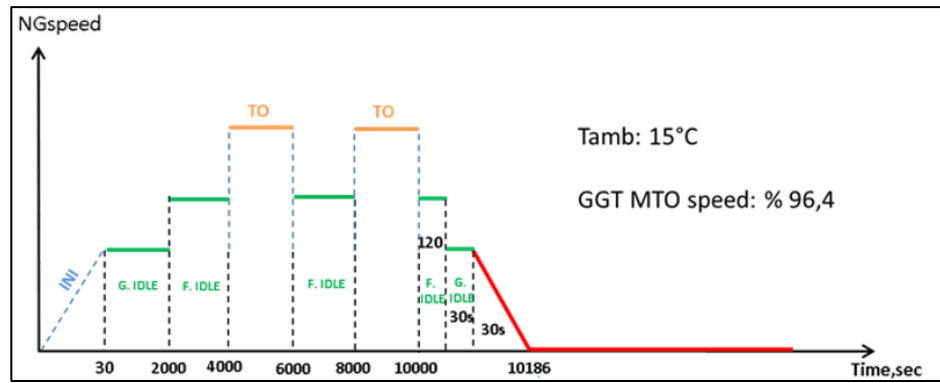


Figure 3-25: Transient mission profile.

All Clearances and speed are normalized according to blade tip height and shaft speed respectively. 3D distortion impact of stator due to maneuver is also included in calculations.

Figure 3-26 shows Stage 1 LE tip clearance results and calculated based on no rub requirement. Plot shows that tip clearances are getting tighter when the shaft speed is maximum. Hot clearance at LE is 0,77 % which is close to target (0,74%) and acceptable for this location. This plot also is typical example of clearance trend pinching at aero design point.

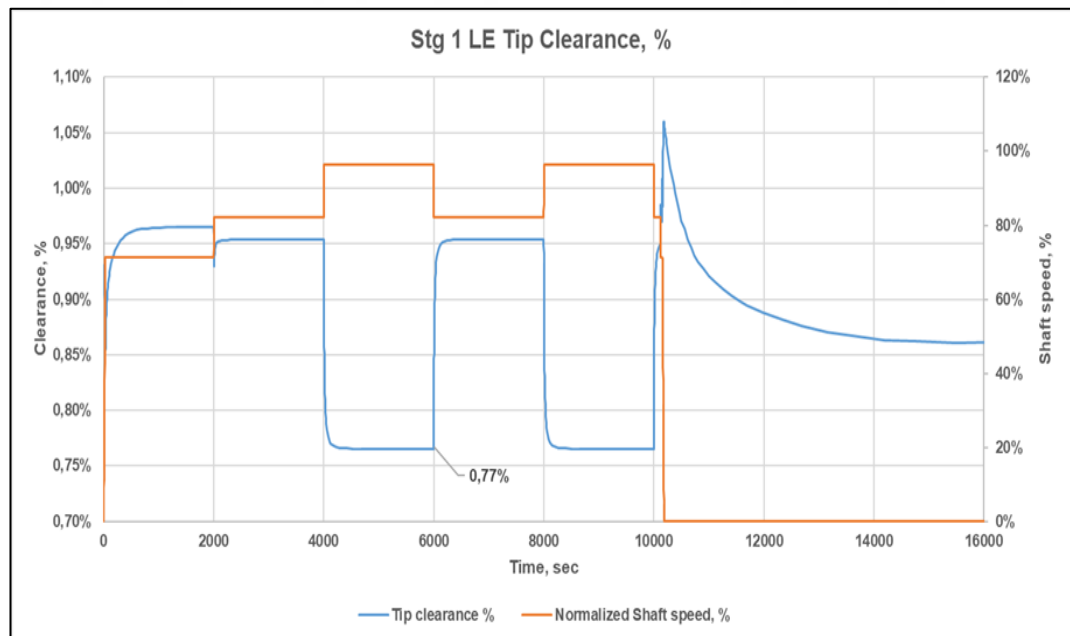


Figure 3-26: Stage 1 LE tip clearance.

Figure 3-27 shows stage 1 TE tip clearance variations along the transient mission. Hot clearance at TE is 2,99 % which is almost 0,8 % greater than the target

(2,2 %). Pinch point occurs at acceleration to TO. After that, clearances are opening about 1% when rotor-stator thermally stabilized. In order to achieve target, tightest clearances has to be at the TO level and plot shows there is a potential of 1 % improvement if the clearances are optimized well.

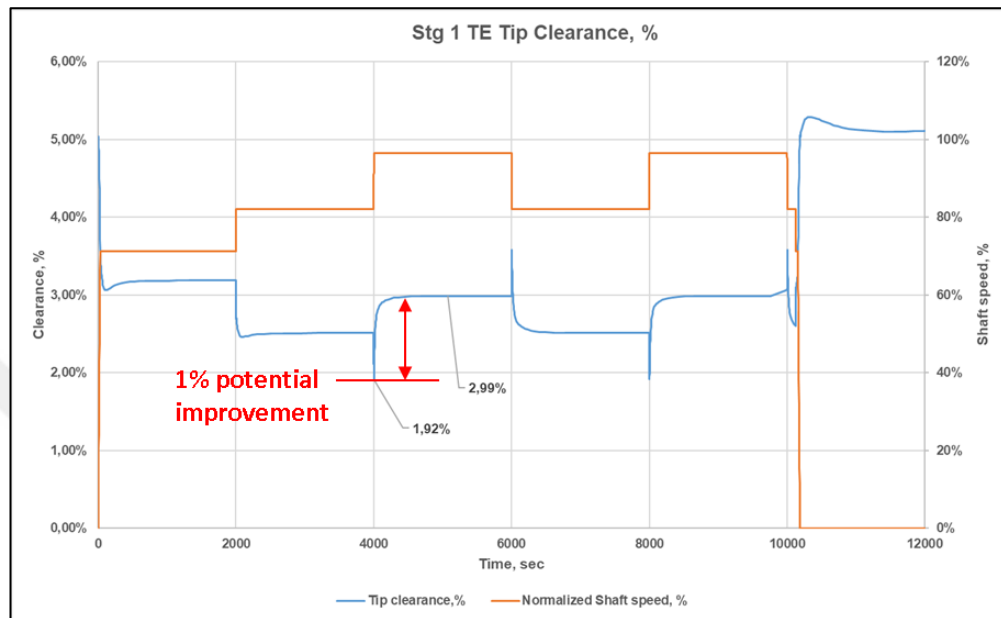


Figure 3-27: Stage 1 TE blade tip clearance.

TE clearances are getting tighter when the speed is increased from ground idle to flight idle level. Expectation is to have tightest clearance at TO level however, clearance are opening at TO level. Detail investigations are done to understand why the clearances opening at only TO conditions. It is obtained that shroud component deflects differently at TO level compared to idle level.

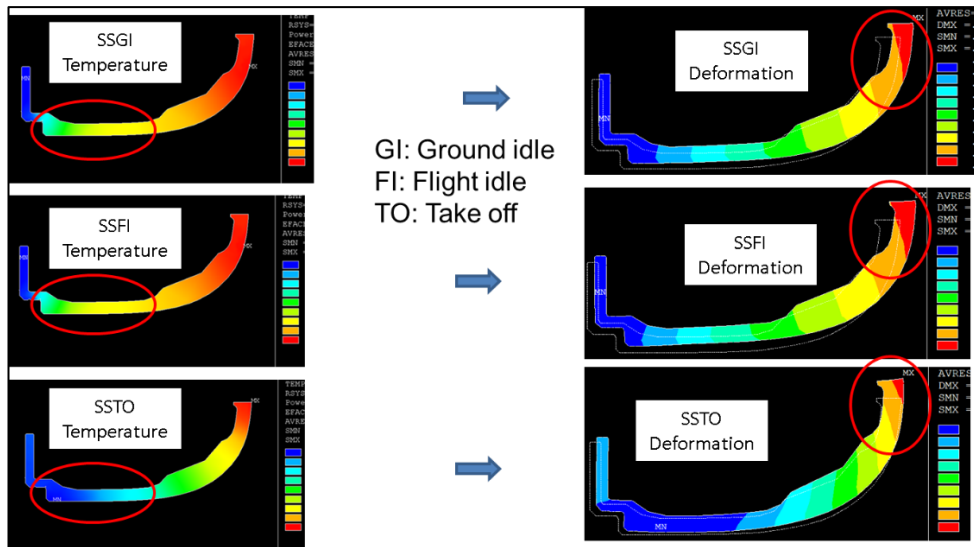


Figure 3-28: Stage 1 deformation and temperature plot

Figure 3-28 shows temperature and displacement plot of shroud at different power level. TE of stage 1 shroud moves to the aft side at ground and flight conditions. However, it moves forward side at TO due to the temperature gradient that exists at the LE side of shroud. When the temperatures of the shroud are checked, it is obtained that area in red circle is getting colder at TO level compared to flight and ground idle level. Detail investigations show that vortices occur at stage 1 due to operability issue. Hot air moves from TE to LE and this makes shroud warmer at flange side. At high speed, there is no more operability issue and shroud's LE cools down compared to idle levels which results in open clearance at TE of stage 1.

Figure 3-29 shows stage 2 LE tip clearance variations along the transient mission. Hot clearance at TE is 1,5 % and very close to target (1,3 %). Plot shows that rotor and stator displacements are almost the same at ground idle, flight idle and TO power levels which mean that they are expanding at the same rate at steady state conditions. 3D distortion and 1.5 RSS values are the main contributor for clearances. If these adders are optimized well then the target clearance can be achieved or further improved.

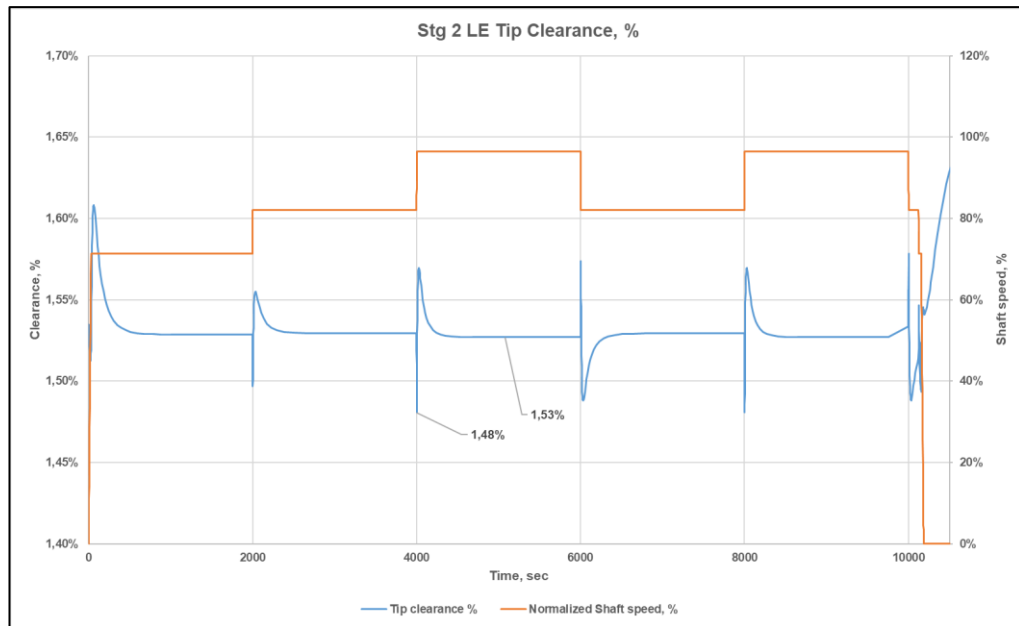


Figure 3-29: Stage 2 LE tip clearances.

Figure 3-30 shows stage 2 TE tip clearance variations along the transient mission. Hot clearance at TE is 4,91 % which is almost 1,3 % greater than the target (3,6 %). Pinch point occurs at deceleration to flight idle. In order to achieve target, tightest clearances has to be at the TO level and plot shows there is a potential of 1,5 % improvement if the clearances are optimized well.

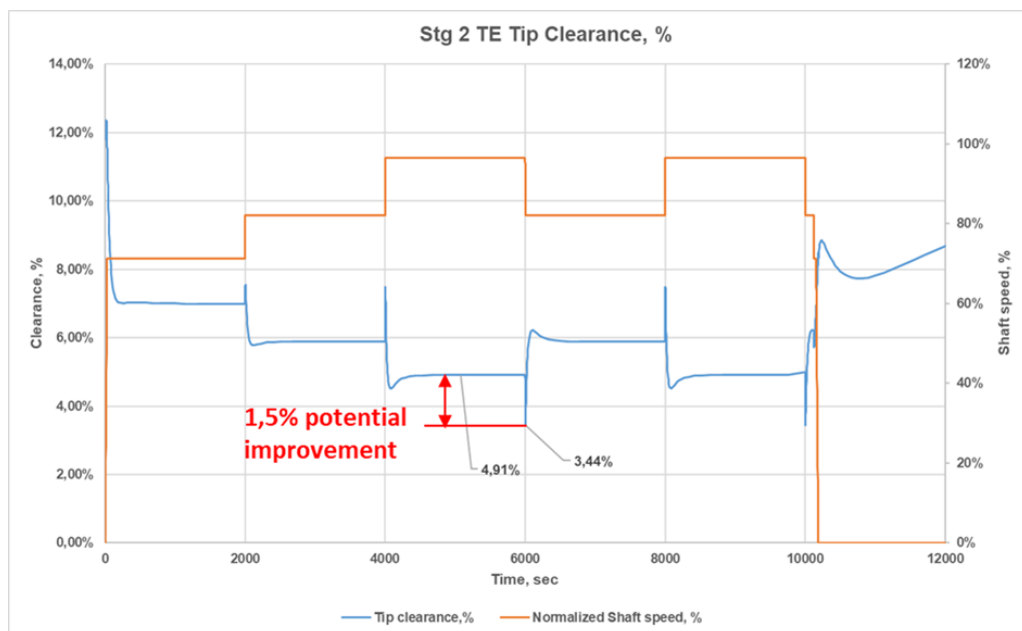


Figure 3-30: Stage 2 TE tip clearances.

TE clearance pinches immediately after deceleration which means that driver of this closure is mechanical load. Detail investigation is performed to understand if mechanical load of stator or rotor causes this closure. Figure 3-31 shows deformed shape of stage 2 shroud and return channel at different power levels. In this figure, black lines show original position and red lines show deformed position of components.

During the acceleration and deceleration, return channel moves forward and aft due to pressure changes. Since the shroud 2 is connected to return channel, its movement is also affected by return channel. When the pressure decreases, return channel moves to aft side immediately and this condition results in tighter clearances at TE.

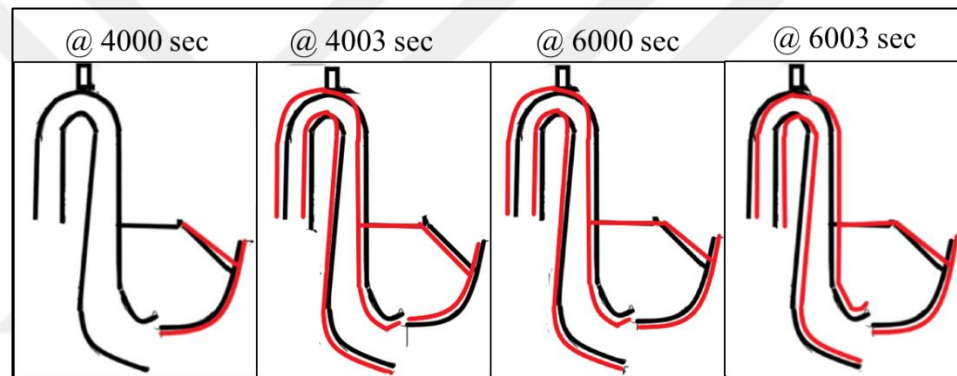


Figure 3-31: Deformed shape of return channel and stage 2 shroud.

In order to improve stage 1 and stage 2 TE clearances, design is changed as shown in Figure 3-32. Stage 1 shroud is made one piece and inducer bleed port added on it to increase surge margin during the operation. Additional supports are added between return channel and diffuser to minimize axial movement due to mechanical load for stage 2.

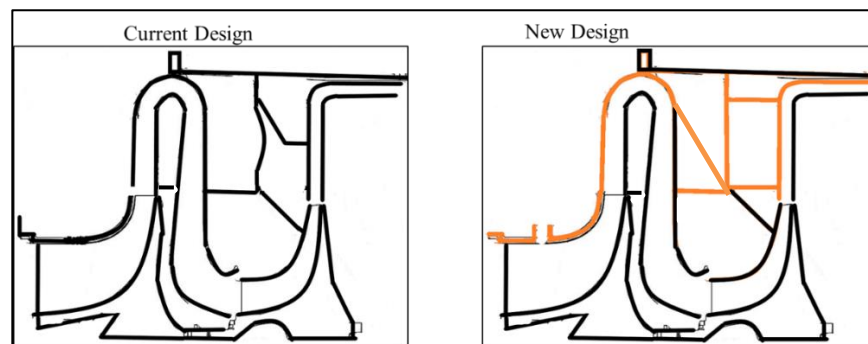


Figure 3-32: New and old design of two stage compressor.

4. EXPERIMENTAL VERIFICATION OF TIP CLEARANCE

CLEARANCE

Tip clearance measurement is important step for verification of analytical results. In the scope this thesis, stage 1 tip clearance is instrumented with non-contact probe to measure tip clearance and observe transient clearance variation during the test.

There are various non-contact tip clearance measurement techniques such as capacitance probe, optic probe, eddy current probe etc. and most widely used one is a capacitance type clearance probe which is inversely proportional to the distance.

4.1 Capacitive Measurement Technique

[7] Working principle of capacitance probe is that when blades pass the sensor, capacitance is formed between blade tip and electrode which is then converted to voltage-distance curve established in cold state. Figure 4-1 shows schematic view of capacitive probe whose capacitance can be expressed in Equation (4-1)

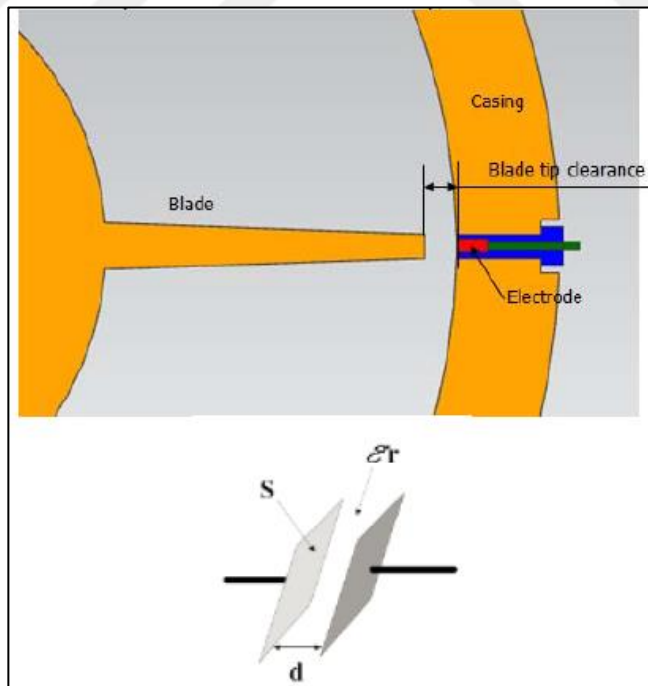


Figure 4-1: Schematic view of clearance probe.

$$C = \epsilon_r \times \epsilon_0 \times S/d \quad (4-1)$$

Where the C is capacitance,

ϵ_r is relative dielectric constant,

ϵ_0 is vacuum permittivity,

S is area of the plate,

d is distance between the plates.

Clearance is inversely proportional to distance between electrode and blade tip and direct proportional to target area. In order to increase capacitance, target area should be increased and distance between blade and electrode should be minimized.

Figure 4-2 shows capacitive sensors with various installation types. One of the probe can be selected depending on the installation space and range requirement for tip clearance measurements [20].

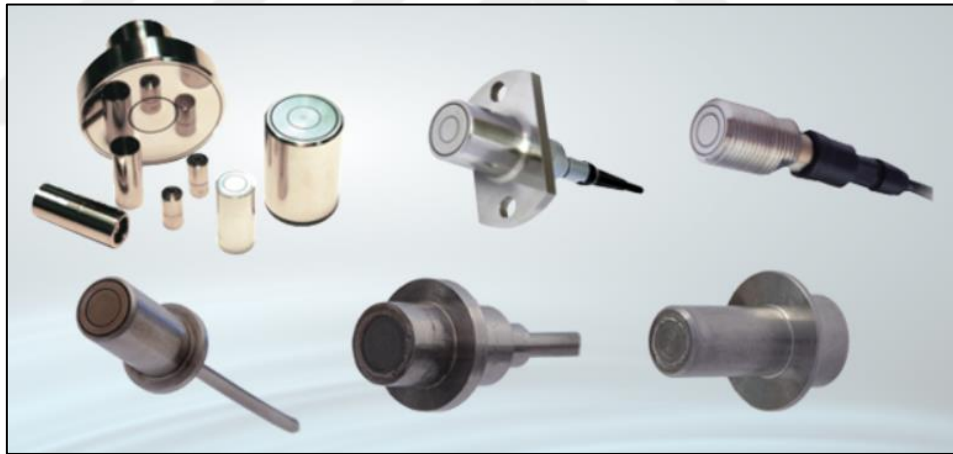


Figure 4-2: Various type of capacitive probe according to installation.

4.2 Tip Clearance Instrumentation

Figure 4-3 shows capacitive probe location and circumferential orientation on stage 1. Totally four probes are installed and two of them on LE and rest ones on TE.

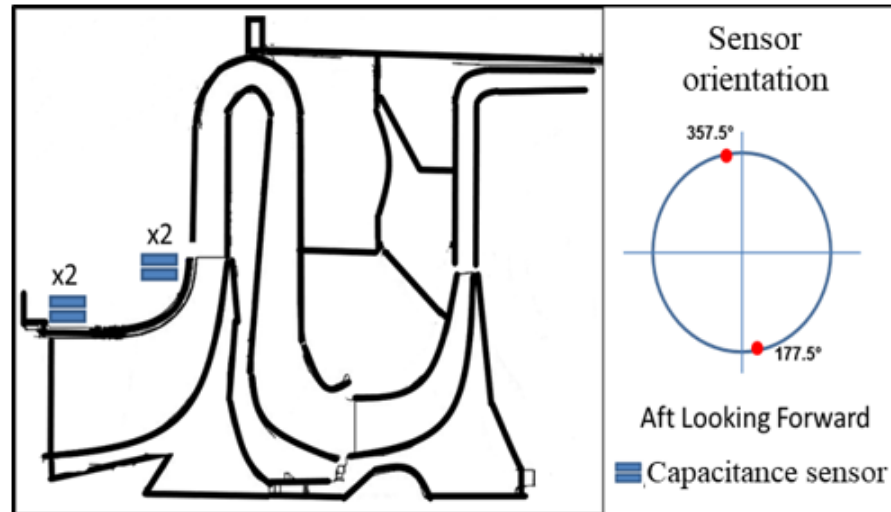


Figure 4-3: Instrumentation scheme of tip clearance probe.

Additional to capacitive probes, 4 metal thermocouples are installed on shroud at each LE & TE sides. In case of any incorrect measurements obtained by probe can be detected by checking metal temperatures. Therefore metal thermocouples should always be used next to the capacitive probe as a secondary check. Figure 4-4 shows instrumented stage 1 shroud that are used in the test. Apart from capacitive probe other instruments such as air temperature and pressure probes are also implemented on shroud for other verification process.

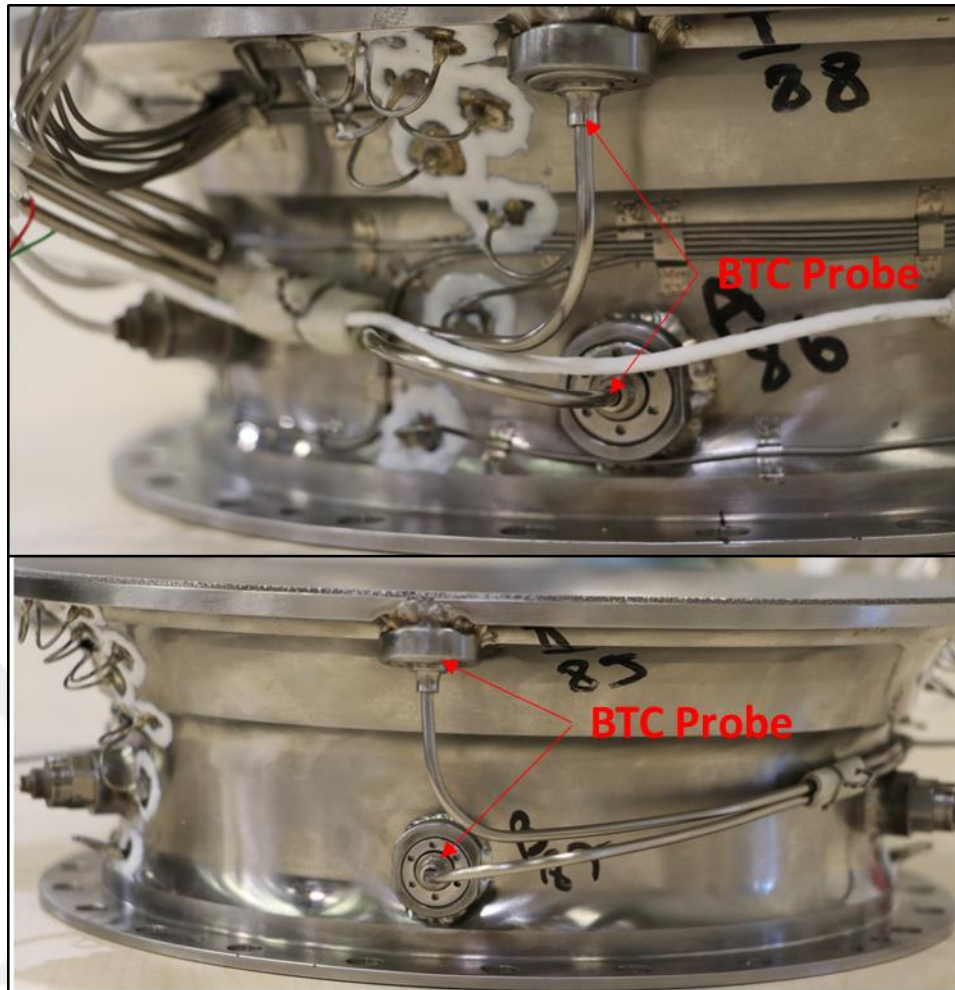


Figure 4-4: Picture of Instrumented stage 1 shroud.

4.3 Pre-test Measurements

Before the tip clearance measurement, several pre-test measurements such as recess, cold build gap have to be performed to have correct test data. Figure 4-5 shows recess measurement which is the value between shroud and probe surfaces. [21] Generally, probes are not mounted with the same surface of shroud profile to avoid damage to probe in case of any tip rub condition. After probe installation, recess value is measured via wax mold and recorded into data acquisition system. Then the distance between blade tip and probe is corrected by subtracting recess value. As a result, recess measurement is crucial step for tip clearance verification.

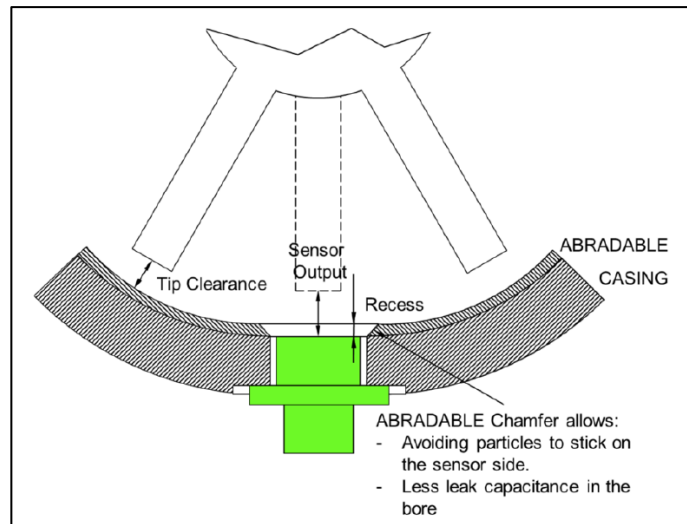


Figure 4-5: Recess measurement.

Another important measurement is cold build tip clearance measurement. During the compressor assembly, both LE & TE tip clearances are measured to use in analytical calculations and to compare with probe measurements. LE cold gap is calculated using manufacturing dimensions of each component while TE cold gap is measured by putting pin between blade and shroud during assembly.

4.4 Uncertainty Analysis of Measurements

Tip clearance measurement uncertainties can be classified into three broad categories. First category is system uncertainty which contains linearization, multiplicative and noise effect. Second category is calibration uncertainty which includes uncertainties during the probe calibration. And the third category is probe installation uncertainties [22].

In order to calculate total uncertainties from these three categories, RSS method is used and Table 4-1 shows total clearance measurement uncertainties for LE & TE. The difference between measurements and predictions has to be within the order of magnitude of measurement uncertainty.

Table 4-1: Clearance measurement uncertainties.

System Uncertainty (%)	LE	TE
Linearization Uncertainty	0.00%	0.01%
Noise	0.04%	0.12%
Multiplicative Uncertainty	0.03%	0.09%
Total System Uncertainty RSS	0.05%	0.15%
Calibration Uncertainty (%)	LE	TE
Axial Shift	0.00%	0.00%
Curve Fit	0.02%	0.06%
Clearance Measurement Uncertainty	0.04%	0.12%
Influence of Blade Thickness	0.02%	0.06%
Calibration of Demodulator	0.04%	0.12%
Total Calibration Uncertainty RSS	0.07%	0.20%
Installation Uncertainty (%)	LE	TE
Setback Uncertainty	0.06%	0.19%
Probe Head Length Measurement	0.01%	0.03%
Uncertainty of Installation of Probe Head	0.04%	0.12%
Total Installation Uncertainty RSS	0.08%	0.23%
Total RSS (%)	0.11%	0.34%

4.5 Tip Clearance Measurements

4.5.1 Cold Crank Measurements

Cold crank measurement is important step for cold gap verification between probe and cold build measurements. During crank test, blade tip clearance is measured and compared with cold build clearance measured during compressor assembly. Measurements show that cold crank tip clearance and cold build clearance matches very well. Table 4-2 shows deviation between probe and cold build clearances.

Table 4-2: Deviation between CBC and Probe measurements

	Deviation between CBC and Probe measurements
LE	0.02%
TE	0.01%

4.5.2 Tip Clearance Measurement Results

Blade Tip Clearance is measured at both LE and TE sides of stage 1 compressor. Analytical calculations are performed according to test cycle conditions and results compared with test data. Figure 4-6 shows comparison plot for stage 1 LE side.

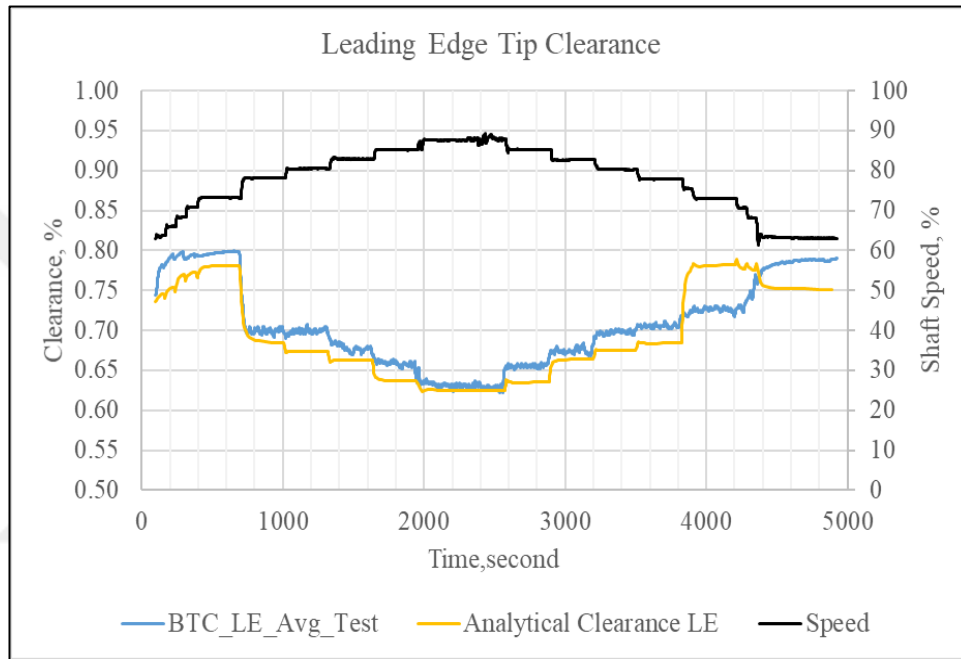


Figure 4-6: Stage 1 LE tip clearance comparison.

LE probe measurements show very good agreement with the analytical data especially for results above 70% shaft speed. Predictions stay within the measurement uncertainty for all time points. The biggest difference occurs at below 70% shaft speed. Detail investigations are performed to understand reason of this difference and it is obtained that due to operability issues, gas temperature cannot be estimated well with CFD analysis as mentioned in section 3.4.9. Therefore, metal temperature of shroud cannot be calculated well with thermo-mechanical analysis. Once this operability issue is resolved, good agreement can be obtained below 70% shaft speed as well.

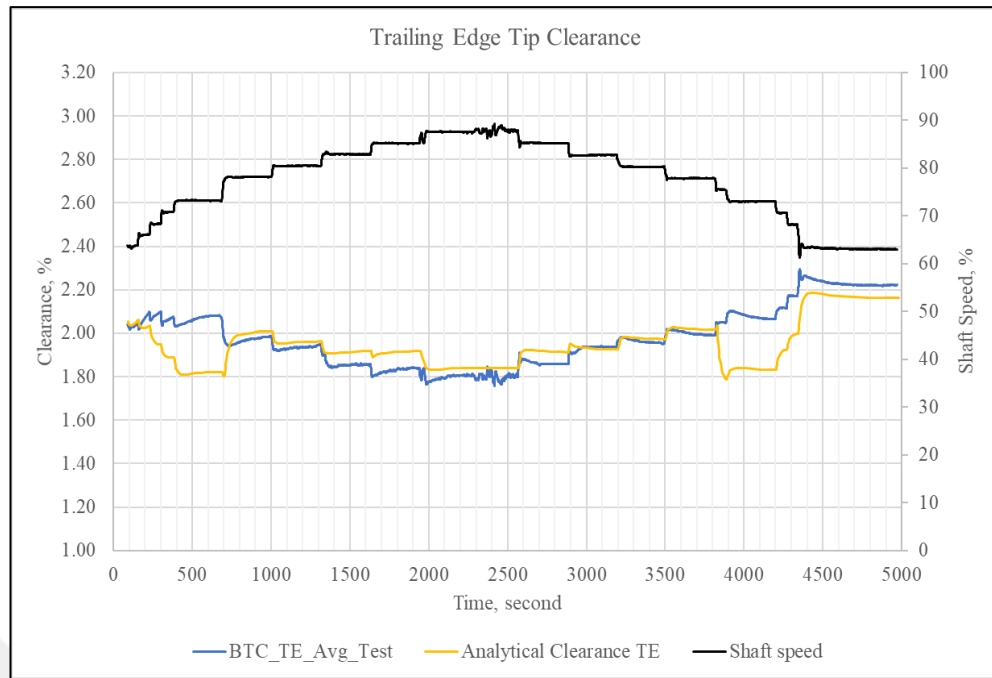


Figure 4-7: Stage 1 TE tip clearance comparison.

Figure 4-7 shows comparison plot for stage 1 TE side. Similar to LE side, probe measurements show very good agreement with the analytical data above 70% shaft speed. However below 70% shaft speed, there is a big difference. Similar to LE, due to operability issues, TE metal temperatures cannot be estimated very well with CFD. As mentioned in section 3.4.9, LE temperatures directly affect TE displacements. Therefore, big difference is obtained below 70% shaft speed. Once operability issue is resolved, good matching can be obtained at all transient time points.

5. CONCLUSIONS

In the scope of this thesis blade tip clearance calculation, optimization and validation methodology are introduced for a two stage centrifugal compressor.

Analytical results show that stage 1 shroud temperatures are affected by operability of compressor which results in unpredicted shroud displacements. In order to fix this operability issue, inducer bleed port is added on shroud as a design improvement. Stage 2 TE tip clearances are affected by mechanical load too much during the acceleration and deceleration conditions. During the analytical calculations, it is obtained that return channel component is very sensitive to pressure changes. Therefore, design is updated by adding supports on return channel to reduce this sensitivity which results in clearance improvement at TE.

Stage 1 compressor is instrumented with capacitive and temperature probes to measure real time tip clearance variations during the test and verify the results with analytical predictions. Test results show very good agreement with analytical data at both LE and TE. The biggest delta occurs at below 70% shaft speed. As predicted on CFD analysis, due to operability issue, temperature of shroud cannot be estimated very well. Therefore, analytical shroud displacements are separated from test displacements. After the implementation of inducer bleed port on shroud to improve the operability margin, thermal predictions with thermo-mechanical model will also be improved and better prediction of clearance will be obtained. Tip clearance probe will be installed on both stage 1 and 2 shrouds for clearance verification.

REFERENCES

- [1] Web 1, (2022), <https://authors.library.caltech.edu/25019/1/chap4.htm>, (Access Date: 04/02/2022).
- [2] Backman J. L. H., Reunanen A., Saari J., Saaresti T. T., Sallinen P., Esa H., (2007), Effects of impeller tip clearance on centrifugal compressor efficiency, ASME Turbo Expo 2007: Power for Land, Sea and Air, GT2007-28200, Montreal, Canada.
- [3] Danish S. N, Chaochen M., Yang C., (2006), “The Influence of Tip Clearance on Centrifugal Compressor Stage of a Turbocharger”, Proceedings of the 4th WSEAS International Conference on Fluid Mechanics and Aerodynamics, Elounda, Greece, August 21-23, 2006 (pp6-11).
- [4] Web 2, (2022), <https://www.sciencedirect.com/topics/engineering/blade-tip>, (Access Date: 04/02/2022).
- [5] Web 3, (2022), <https://www.ccj-online.com/3q-2012/gas-turbines/>, (Access Date: 04/02/2022).
- [6] Web 4, (2022), <https://aeroenginesafety.tugraz.at/doku.php?id=7:71:713:713> (Access Date: 04/02/2022).
- [7] Cheng L., Zhao S., Liu W., Ai S., Fan X., (2007), Axial flow compressor real-time tip clearance analysis and experimental verification, ASME 2017 Power Conference Joint With ICOPE-17 POWER2017-ICOPE-17, Charlotte, North Carolina, USA.
- [8] Web 5, (2022), <https://aeroenginesafety.tugraz.at/doku.php?id=7:71:712:712#prettyPhoto>, (Access Date: 04/02/2022).
- [9] Mills S. J., (2012), The Part Thermo-Mechanical Analysis Plays in the Design of Gas Turbines, EN-AVT-207-03, NATO STO, United Kindom.
- [10] Baldassarre L., Fontana M., (2010), Modelling of rotor bow during hot restart in centrifugal compressors, Proceedings of the thirty-ninth turbomachinery symposium.
- [11] Kurzke J., (2019), L18_TEI_TransientOperation1.pdf ,Gasturb seminar.
- [12] Web 6, (2022), <https://aeroenginesafety.tugraz.at/lib/exe/detail.php?id=7%3A71%3A712%3A712&media=7:71:712:leakage-losses-and-rub-wear-behaviour.svg>, (Access Date: 04/02/2022).

- [13] Giovannetti I., Bigi M., Giannozzi M., Sporer D. R., Cappuccini F., Romanelli M., (2008), Clearance reduction and performance gain using abradable material in gas turbines, ASME Turbo Expo 2008: Power for Land, Sea and Air, GT2008-50290, Berlin, Germany.
- [14] Web 7, (2022),
<https://aeroenginesafety.tugraz.at/lib/exe/detail.php?id=7%3A71%3A712%3A712&media=7:71:712:conclusions-from-clearance-shapes.svg>
 (Access Date: 04/02/2022).
- [15] Web 8, (2022),
<https://aeroenginesafety.tugraz.at/doku.php?id=7:71:712:712#prettyPhoto>
 (Access Date: 04/02/2022).
- [16] Chew J. W., Hills N., (2009), Computational fluid dynamics and virtual aeroengine modelling, Fluids Research Centre School of Engineering University of Surrey Guildford Surrey, GU2 7XH, UK.
- [17] Web 9, (2022), <https://www.eng-tips.com/viewthread.cfm?qid=450513>,
 (Access Date: 04/02/2022).
- [18] Arkhipov A. N., Karaban V. V., Putschkov I. V., Filkorn G., Kieninger A., Thewhole-engine model for clearance evaluation, ASME Turbo Expo 2009: Power for Land, Sea and Air, GT2009-59259, Orlando, Florida, USA.
- [19] Andrés L. S., (2010), Notes 13. Squeeze film dampers: operation, models & issues.
- [20] Web 10, (2022), <http://www.fogale.fr/turbomachinery/PHP/products-capacitive-sensors.php>,
 (Access Date: 04/02/2022).
- [21] Fogale Nanotech, (2016), High Temperature Capacitive Sensors Instruction Manual-Release B.
- [22] Müller D., Sheard A. G., Mozumdar S., Johann E., (1996), Capacitive measurement of compressor and turbine blade tip to casing running clearance, Asme 96-GT-349, Newyork, USA.
- [23] Luca B., Sivia B., Andrea P., Laura T.,(2012), Prediction of Clearance in Industrial Gas Turbine Validated by Field Operation Data, ASME Turbo Expo GT2012-69617, Copenhagen, Denmark.

BIOGRAPHY

Bedir İnci is a mechanical engineer. His professional career started as a Research and Development engineer at a white goods company. After that, he worked as mechanical design engineer at different aerospace companies.

