

CRANFIELD UNIVERSITY

VASILEIOS KATSAPIS

TECHNO-ECONOMIC ANALYSIS AND PREDICTIVE MODEL FOR
HEAVY-DUTY GAS TURBINE, IN RELATION TO COOLING AIR
SYSTEM DEGRADATION

SCHOOL OF AEROSPACE, TRANSPORT AND
MANUFACTURING

DOCTOR OF PHILOSOPHY

Academic Year: 2018 - 2019

Supervisor: Dr Panos Laskaridis
Associate Supervisor: Dr David McManus
October 2018

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ABSTRACT

Modern gas turbines are featured with cooling air systems that allow them to operate at temperatures much higher than the melting points of most of their hot gas path components. When these systems degrade, creep life consumption is accelerated. If the risk is not assessed and managed in time, the engine could operate with high lifing risk. In the world of power industry, the operators are in most cases presented with two options when a lifing risk in their engine is identified. They can undertake a forced opening of the engine, against their existing planning of maintenance events, aiming to re-instate cooling air system's functionality. Alternatively, they can derate the engine, by decreasing firing temperature, to a level that will allow continuous operation until the next planned overhaul event.

The model presented in this work allows the operator to follow a third path. The operator can still manage effectively the lifing risk but also, critically, to do it in a commercially optimised way by taking into consideration electricity market conditions and the incentives that the market may present. This is important as this model fills a gap in the relevant research where the majority of the works are focused in managing lifing risks through a binary time-interval-reduction or firing- temperature-decrease approach.

The developed in this work tool uses a combination of performance modelling, analytical and computational methods, as well as Monte Carlo simulations. The cooling air system degradation is detected and assessed and the effect on cooling air temperature is determined with use of CFD. The impact on blades metal temperature is calculated with a combination of analytical model and Monte Carlo simulations. In a similar combination of analytical method and Monte Carlo, stress is calculated and together with OEM's life expectancy, a

creep life estimation model then is derived. The lifing risk is quantified and appropriate mitigation is proposed. The mitigation is in the form of a novel method, which uses market triggered factors to adjust dynamically engine's firing temperature and define by this way engine's performance. This hybrid approach results is an optimised operation in which the lifing risk is effectively managed, the original maintenance planning is respected and, critically, the commercial losses due to engine derate are significantly trimmed. The tool, with its risk-versus-value function, allows the operator to exercise a more or less conservative operation, following a case-by-case approach.

KEYWORDS

pre-swirl nozzles, CFD, blade metal temperature, blade stress, creep-induced lifing risk, Spark Spread value, loading-strength diagram, Monte Carlo

“...τους φόβους που σε θρέφουν
να τους κάνεις κάτι άλλο,
κάνε τους ποιήση ή μηχανήματα σπουδαία,
μην γυρνάς την πλάτη, μην ζεις στο ψέμα..”

Κωνσταντίνος Βήτα

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Now that I have finally reached my Ithaca, I would like to think back and refer all those who, one way or the other, helped me to keep my ship sailing to the “right” direction, especially in those days and nights when the sea was rough and the compass was broken.

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LIST OF ABBREVIATIONS

AR	Asymptotic Range
BCH	Blade Cooling Hole
BCP	Blade Cooling Passage
BL	Base Load
BR	Blade Row
BS	British Standard
CAS	Cooling Air System
CAD	Computer Aided Design
CAPEX	Capital Expenditure
CE	Cyclic Event
CC	Convective Cooling
CC	Combined/Closed Cycle
CCGT	Combined Cycle Gas Turbine
CCPP	Combined Cycle Power Plant
CD	Constant Derate
CFD	Computational Fluid Dynamics
CFL	Courant Number
CFX	Commercial name of ANSYS CFD Solver
CG	Center of Gravity
CMAS	Cooling Mantle Air System
CMSX4	Commercial name for Nickel-based super-alloy
CSS	Clean Spark Spread
CT	Cooling Tower
DC	Direct Cooling
DCS	Distributed Control System
DD	Dynamic Derate
DDD	Deep Dynamic Derate
DLN	Dry Low NO _x
DM	Design Modeller, commercial name for ANSYS CAD software
DNS	Direct Numerical Simulations
DOD	Domestic Object Damage
DS	Dark Spread
DT	Destructive Tests
EGS	Exhaust Gas System
ELU	Engine Life Usage

EOH	Equivalent Operating Hours
EUETS	European Union Emissions Trading System
EV	EnViromental Burner
EVM	Eddy Viscosity Models
E ³	Energy Efficient Engine
FC	Film Cooling
FEA	Finite Element Analysis
FH	Factored/Fired Hours
FMEA	Failure Mode Effects Analysis
FOD	Foreign Object Damage
FR	Frozen Rotor
FT	Firing Temperature
GCI	Grid Convergence Index
GCI	Generalised Grid Interface
GSP	Goldhoof-Sherby Parameter
GT	Gas Turbine
GT26	GE (Alstom) Frame 9 Heavy Duty Gas Turbine
HCF	High Cycle Fatigue
HGP	Hot Gas Path
HP	High Pressure
HPT	High Pressure Turbine
HR	Heat Rate
HRSG	Heat Recovery Steam Generator
IC	Impingement Cooling
ICEM	Commercial name for ANSYS Mesh generation software
IGV	Inlet Guide Vanes
IRR	Internal Rate of Return
ISO	International Standards Organisation
KKS	Kraftwerks Kennzeichen System (Power Plant Nomenclature System)
L- S	Loading - Strength
LCF	Low Cycle Fatigue
LE	Leading Edge
LES	Large Eddy Simulations
LMP	Larson Miller Parameter
LP	Low Pressure
LPT	Low Pressure Turbine
LPT1	Low Pressure Turbine 1 st Stage (Vane and/or Blade)
MBH	KKS code for gas turbine cooling system

MC	Monte Carlo
MCMC	Monte Carlo Markov Chain
MCrAlY	Commercial name for protective coating on HGP parts
MHP	Mason-Haferd Parameter
MSP	Mason-Succop Parameter
MTBF	Mean Time Between Failures
MTTR	Mean Time To Repair
NDT	Non-destructive Test
NG	Natural Gas
NGV	Nozzle Guide Vane
NO _x	Nitrogen Oxides
NPV	Net Present Value
NS	Navier-Stokes (equations)
NX	Commercial name for Siemens CAD software
O&M	Operation and Maintenance
OC	Open Cycle
OEM	Original Equipment Manufacturer
OGV	Outlet Guide Vane
OH	Operating Hour
OPEX	Operational Expenses
OQ	Orthogonal Quality
OSDP	Orr-Sherby-Dorn Parameter
OTC	Once Through Cooler
PAT	Plant Acceptance Test
PBP(T)	Pay Back Period (Time)
PDE	Partial Differential Equations
PR	Pressure Ratio
PSN	Pre-swirl Nozzle
R&D	Research and Development
RAM	Reliability, Availability and Maintainability
RANS	Reynolds Averaged Navier Stokes (equations)
RDS	Root Datum Section
RMS	Root Mean Square
RNG	Random Number Generator
RSM	Reynolds Stress Models
SDD	Shallow Dynamic Derate
SEV	Sequential EnViromental (Combustor)
SFC	Specific Fuel Consumption
SO _x	Sulphur Oxides

SS	Spark Spread
SSC	Start-Stop Cycle
SSS	Synchro-Self-Shifting Clutch
SST	Shear-Stress Transport
ST	Steam Turbine
TAT1	Temperature After (High Pressure) Turbine
TAT2	Temperature After (Low Pressure) Turbine
TBC	Thermal Barrier Coating
TC	Thermo-Couple
TE	Trailing Edge
TERA	Techno-economic and Environmental Risk Analysis
TIT1	(High Pressure) Turbine Inlet Temperature
TIT2	(Low Pressure) Turbine Inlet Temperature
URANS	Unsteady Reynolds Averaged Navier Stokes (equations)
VBA	Visual Basic for Applications
VIGV	Variable Inlet Guide Vane
VR	Vane Row
WAR	Water-to-Air Ratio
WCE	Weighted Cyclic Event
WI	Wobbe Index
WOH	Weighed Operating Hour

LIST OF SYMBOLS

A	annulus area (m ²)
A	blade section area (m ²)
A,B,C	types of GT26 inspections
Bi	Biot number
C	sound velocity (m/sec)
C	creep constant (-)
C	friction factor (-)
ch	chord (m)
c _p	specific heat over constant pressure (kJ/kg K)
c _v	specific heat over constant volume (kJ/kg K)
c _w	non-dimensional mass flow rate (-)
D	diameter (m, mm)
d	diameter (m, mm)
d	distance (m, mm)
F	force (N)
g	gravitational acceleration (m/sec ²)
H	height (mm)
Hz	Hertz
h	heat transfer coefficient (W/m ² K)
k	thermal conductivity (W/mK)
L	length (mm)
M	moment (Nm)
Ma	Mach number
m	mass (kgr)
$\dot{m}$	mass flow rate (kgr/sec)
m*	dimensionless mass flow rate (-)
MWh	Megawatt-hour
N	number (-)
N	number of cycles (-)
Nu	Nusselt number (-)
P	Pressure (Pa, bar, bara)
p/th	pennies per therm
Pr	Prandtl number
Q	heat (J, kJ)
Q	flow function [kg K ^{0.5} (m ² kPa sec)]

R	universal gas constant (Kj/kg K)
r	radius (mm, m)
Re	Reynolds number (-)
Re _φ	rotational Reynolds number (-)
S	Blade external perimeter (m)
S _{AIR}	Sutherland Law constant
s	distance (m)
SM	Safety Margin (-)
St	Stanton number (-)
T	temperature (C°, K)
T _m	metal temperature (C°, K)
T _{ma}	metal temperature constant (K)
t	thickness (m)
t	time (sec)
t _a	time to failure constant (hours)
t _f	time to failure (hours)
U	rotor velocity (m/sec)
U _t	fluid velocity at first cell height (m/sec)
u	fluid velocity (m/sec)
W	width (m)
X	technology factor (-)
x	Chamfer distance from inlet (m)
y	first cell height (m)
Y ⁺	dimensionless distance from wall (-)
z	fuel factor (-)

LIST OF GREEK SYMBOLS

α	angle of pre-swirl nozzle to the plane of rotation (degrees°)
β	swirl ratio (-)
γ	ratio of specific heat capacity (-)
Δ	delta, difference
ε	blade cooling overall effectiveness (-)
ε	turbulence eddy dissipation (-)
θ	chamfer angle in pre-swirl nozzle
θ	blade stagger angle (degrees°)
λ_T	turbulent flow parameter (-)
μ	mean value
μ	dynamic viscosity (Pa s)
μ_0	dynamic viscosity (Pa s)
ν	kinematic viscosity (m ² /sec)
ρ	density (kgr/m ³)
Σ	sum of
σ	stress (Pa)
σ	typical variation
τ_w	wall shear stress (Pa)
Φ	tangential velocity (m/sec)
Ω	angular velocity (rpm, rps)
ω	angular velocity (rpm, rps)
ω	turbulence eddy frequency (-)

SUBSCRIPTS

act	actual
av	average
ax	axial
BCH	blade cooling hole (passage)
BL	Base Load
bl	blade
bw	blade wall
c	convective cooling
c,in	cooling air inlet
c,out	cooling air outlet
cc	cooling channel
ch	chord
coa	coarse
equi	equivalent
f	film cooling
fin	fine
g	hot gas
innerseal	inlet to the cavity seal
med	medium
outseal	outlet from the cavity seal
PSN(s)	pre-swirl nozzle(s)
sp	span
tn	tangential
w	wall

1 INTRODUCTION

1.1 Context

Modern gas turbines are featured with cooling air systems that allow them to operate at temperatures much higher than the melting points of most of their hot gas path components. When these systems degrade, creep life consumption is accelerated. Creep problems are common causes of mechanical failure in heavy-duty gas turbines and estimation of components' remaining creep life is of paramount importance for safety, engine integrity and economic reasons. Original Equipment Manufacturers, but also operators, constantly try to develop methods that will allow them to predict with confidence and accuracy the useful remaining life out of gas turbine components. If a creep-induced lifing risk is not assessed and managed in time, the engine could operate with unacceptably high lifing risk.

A number of different design characteristics and operating parameters define and affect creep life in gas turbine components; some of the design characteristics and operational parameters are used as input factors (constants or variables) to lifing models. In most cases, the design characteristics are in relation to the component's material properties, while the operational parameters are in relation to the hot gas path flow conditions, primarily hot gas temperature, and to the stress levels that the component is subjected to. In contrast, gas turbine cooling air systems, which allow engines to operate with firing temperatures close to or above the melting points of their hot gas path components, are in most cases missed out from being factored inputs to lifing models, despite their criticality for engine's integrity and longevity. Pre-swirl nozzles are at the heart of gas turbine's cooling air systems.

In the world of electricity generation industry, the operators are presented in most of the cases with two options when a lifing risk in their engine is identified.

Either to undertake a forced opening of the engine, against their existing planning of maintenance events, or to choose a costly derate of the engine's firing temperature as the only mean to allow continuous safe operation until the next planned maintenance event.

Furthermore, creep-induced lifing risks can rarely be quantified with confidence. Analytical models, when used in processes with high levels of uncertainty, are unavoidably constrained to produce largely qualitative rather than quantitative outcomes. Moreover, in most cases, prevailing electricity market conditions do not influence any decisions on how engine's remaining life has to be managed. The often-observed approach is that conservative and uncertain, by their nature, lifing models are used in isolation of the economic environment that the engine operates in. In other words, engine's remaining life is managed as a purely technical, of must-retain-integrity at any cost, problem. In contrast, this work proposes an approach in which lifing risks, as well as, electricity market conditions are equiponderant inputs to a model that offers a more balanced position between integrity risk and commercial incentives.

1.2 Aim and Objectives

Overall, scientific aim of this project is to optimize the remaining life of an engine that is exposed to increased lifing risk. The optimisation is realised when three critical conflicting requirements are balanced:

- I. commercial losses due to engine's derated operation are minimised,
- II. lifing risk is effectively managed and
- III. initial engine overhaul schedule is maintained, in other words no forced overhauls are introduced.

To achieve that, a modular process is developed that captures, assesses and quantifies the effect of engine's degraded cooling air system, in particular of the pre-swirl nozzles, on engine's life as this is defined and limited by turbine's 1st stage blade. Through this modular process, a number of individual objectives, which are illustrated in Figure 1-1, have to be

delivered in a sequential order. Starting from the top-left corner of the flow chart:

1. development of engine performance model that provides the boundary conditions in the hot gas path flow. It also quantifies the performance impact of engine's derated operation in a number of different firing temperature settings.
2. Development of analytical solution for engine's cooling air system in the area of pre-swirl nozzles cavity.
3. Development of computational fluid dynamics (CFD) model for the same area of turbine's pre-swirl nozzles cavity.
4. Development of analytical heat transfer model that calculates the metal temperature of engine's low-pressure-turbine 1st stage blade. The model is combined with Monte Carlo simulations.
5. Development of analytical model for calculation of the maximum stress that the same blade is subjected to. Similarly with above, the model is combined with Monte Carlo simulations.
6. Development of lifing model that assesses and quantifies the impact of cooling air effectiveness degradation on blade's creep life.
7. Finally, development of a techno-economic model; the model factors the incurred increase in blade's life consumption with electricity market conditions. It offers a tool that supports critical operation and maintenance decisions, such as degree of engine derate through firing temperatures and placement of maintenance events. In this way, engine's lifing risk remains at acceptable levels while its commercial position is optimised.

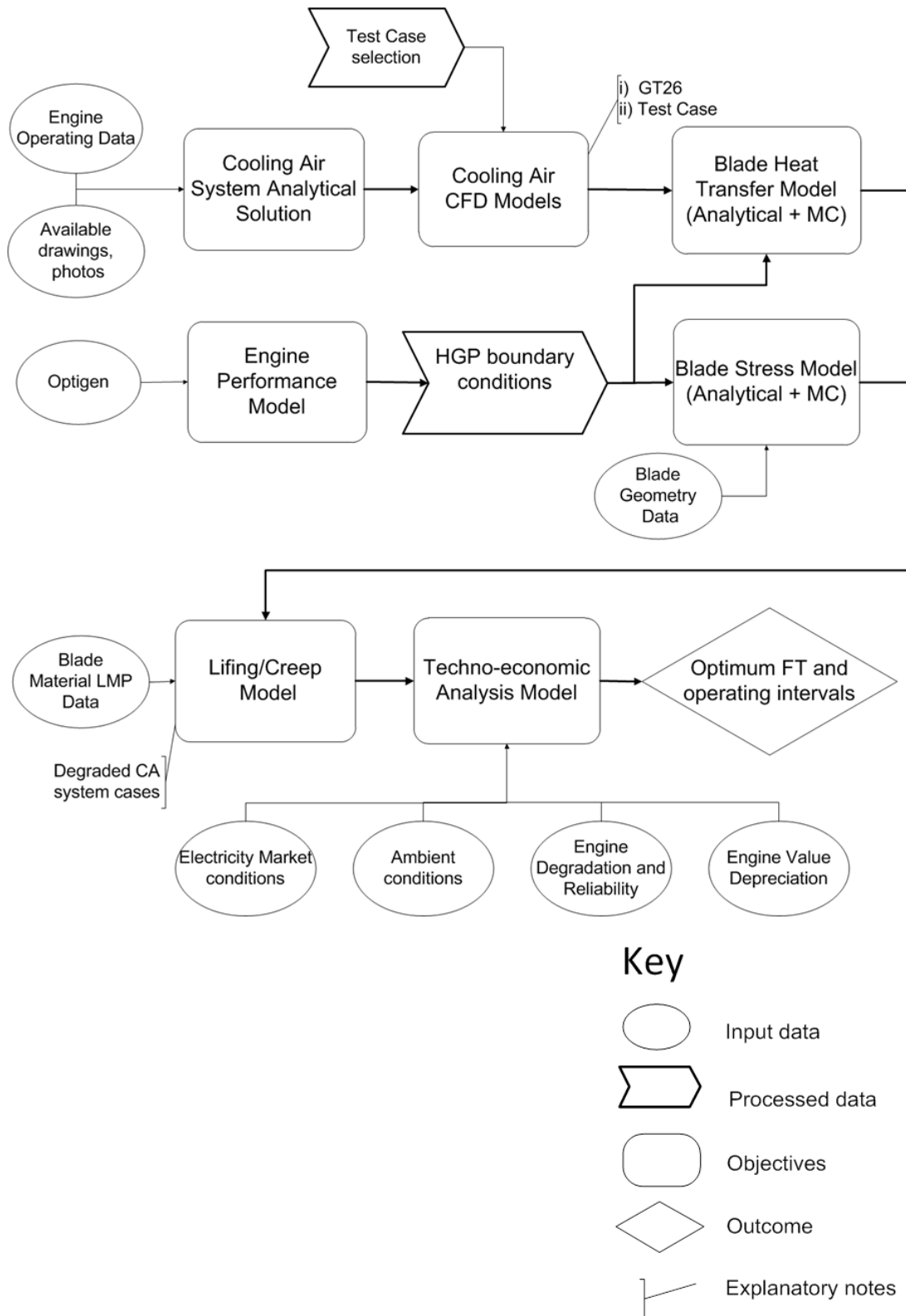


Figure 1-1: Research work flow chart

1.3 Thesis Structure

The structure of the thesis is outlined below:

- Starting from the second chapter, a review of relevant literature and theory is provided. The areas of gas turbine's pre-swirl nozzles, lifing and techno-economic analysis are discussed in detail and with a critic view. In the areas of blade heat transfer, blade stress and creep, only the applicable to this work parts of the theory are discussed. The chapter closes with a summary of the main gaps in literature, as they have been identified through the review.
- The third chapter provides a brief technical description of the key areas of this study. Gas turbine is described briefly, while more focus is given on the pre-swirl nozzles. The low-pressure-turbine 1st stage blade together with its material's creep properties are discussed. Gas turbine's maintenance model is explained briefly. Finally, Test Case's rig and Test Case's experiment results are discussed thoroughly, as they are critical for the validation of the CFD model, which follows in the methodology chapter.
- The fourth chapter is the longest one as it goes through the details of the methodology. It starts with the performance model that is developed for the gas turbine of this work. Details on the validation are discussed. Performance model's results in relation to hot gas path conditions are cited, while the impact of derated operation on engine's performance is captured. The results that correlate firing temperature with output and efficiency are listed in the techno-economic analysis chapter. In the same chapter, an analytical solution of the gas turbine cooling air flow in the area of pre-swirl nozzles cavity is developed. The solution provides a high-level insight to the flow conditions in this area of the engine. Case with increased cooling air flow through cavity's inner seal is explored and the impact on cooling air temperature inside blade cooling passages is quantified.

The CFD solution is the next section in the methodology chapter and it is described in full details. ANSYS software was used and all the details of the model (geometry, mesh, solver, results) are cited and discussed. Two models were developed, one for the Test Case, which was used for validation, and one for the gas turbine in discussion. Results from the latter model in relation to cooling air temperature increase feed to the blade metal temperature calculation model that follows.

A section dedicated to Monte Carlo simulations follows. Since extensive use of Monte Carlo simulations is made in this work, some basic principles and the purposes for using it are explained.

Following Monte Carlo, blade cooling calculation model is the next section that provides results on mean blade metal temperature. The model combines an analytical method with Monte Carlo simulations.

Similarly to blade metal temperature, the analytical solution for blade stress calculation, which takes place in the next section, is combined with Monte Carlo.

Finally, results from blade metal temperature and blade stress calculations feed to a blade lifing model. The impact of degraded cooling air conditions is mapped, with use of Monte Carlo, on a Larson-Miller graph. The increased lifing risk is quantified with the use of loading-strength diagram and methods to mitigate the risk are explored.

- In the fifth chapter, the concept of a techno-economic model is presented. The chapter starts with an introduction to electricity market while also discussion takes place around OEM-related costs and outage costs. The critical metric of Spark Spread value is explained and the factors which affect gas turbine's output, generated volume and revenue are described in detail. Finally, the predictive model for revenue generation is presented. The model runs the case with the derated, due to cooling air system degradation, engine. The results illustrate the dependence of predicted revenue value and remaining life hours on Spark Spread prices, which are selectable by the operator in the function

of cut-in prices. Conclusions and the risk-versus-value philosophy of the model are discussed thoroughly.

- Finally, in the last chapter a summary of the whole project is provided. Conclusions and impact of the research work are discussed, as well as thoughts and proposals on future work.

1.4 Assumptions and limitations

Assumptions are unavoidably made in every research project. It is however quite important to be succinctly listed, so they can offer ground for useful feedback and constructive criticism, but also to form part of future work. In this work, some of the key assumptions have been the below:

- *Low pressure turbine 1st stage blade as engine's life limiting factor;*
Throughout the course of this project, LPT1 blade is treated as the life-limiting component of the whole engine. It has to be stated that no Failure Mode and Effects Analysis (FMEA) has taken place for the purposes of this research work. However, and through other gas turbine users' experience, this assumption is considered to be quite close to reality.
- *Creep as the predominant failure mechanism;*
Steady state conditions only are considered in this work and creep is assumed to be the predominant failure mechanism. Discussion on other failure mechanisms takes place in paragraph 2.5.2. where the arguments for excluding them are presented. Transient analysis is at any way recommended as future work and in that case, the impact on low-cycle fatigue and thermomechanical fatigue has to be part of the assessment.

- *Miner's equation assumes that in equal fractions, equal damage takes place;*

This is a popular assumption that often met in the relevant literature and references, including those of the current work. The author of this work adopted unchallenged the same approach on Miner's lifing equation.

- *Isotropic and homogenous properties across LPT1 blade mass;*

Monte Carlo simulations were used in both blade metal temperature and blade stress calculations, so there is confidence that uncertainties were effectively captured. However, it remains a key assumption that the blade is expected to have isotropic and homogenous properties throughout its body and mass.

- *Impingement cooling effect on blade cooling calculation;*

Blade cooling calculations, especially when all types of blade cooling methods co-exist, are among the most complex tasks in gas turbine technology and in engineering in general. Current work uses an analytical approach combined with Monte Carlo so uncertainties are captured. However, only convective cooling film cooling and the effect of TBC have been explicitly captured in the analytical model.

- *Pre-swirl nozzles cavity inner seals relaxation;*

It is assumed that the pre-swirl nozzles cavity inner seals relaxation, as the root cause failure behind engine's cooling air system degradation, is diagnosed through borescope inspections. Appendix G offers some details on the borescope inspection path, which has been recently adopted by sponsor company of this project as a closed-engine inspection routine. Versions of the engine with additional installed instruments in the area as monitoring means are not considered in this work.

2 LITERATURE REVIEW AND THEORY

2.1 Introduction

This chapter provides a mixture of literature review on some topics and high-level discussion on theory on some others. More specifically, the topic of pre-swirl nozzles, as key part of gas turbine's cooling air system, is discussed in depth through a detailed literature review. This is because the specific component and its degradation mechanism, which is discussed later in this thesis, are critical parts of and the starting point for this project's scope. Similarly, the discussion on the lifing and techno-economic analysis parts takes place through relevant literature reviews. Both these themes, lifing and techno-economic analysis, are areas where the contribution and novelty of this work comes in. In contrast, topics like blade cooling, blade stress and creep are discussed through a relatively high-level theory review. Though the description of some of the theoretical principles is lengthy, the primary aim is to simply identify the applicable parts of the theory on project's purposes. On these topics no literature review takes place, apart from the reference of classic texts of these scientific fields which are used for guidance. Undoubtedly, blade cooling, blade stress and creep are wide topics, each one a science branch on its own, but only some of their basic principles are used in this work. Novelty and contribution by author's side do exist, but in the specific way that Monte Carlo is combined with the analytical models, rather than in reviewing and challenging the theory itself.

The literature review starts with the description of pre-swirl nozzles. Pre-swirl nozzles and their cavity are at the heart of the degradation mechanism that this work investigates. As such, a number of previous works in this area is cited, while also some of the theory behind the complex flow in this area of the engine is developed. Critical non-dimensional numbers are explained and the importance of pre-swirl nozzles location inside in the engine is discussed.

The chapter continues in section 2.3 with the theory around blade cooling. Basic principles of heat transfer are discussed and reference is made to most popular blade cooling methods, namely convective cooling, film cooling and impingement cooling. Also, important non-dimensional numbers are explained and their constitutive equations are cited. Classic texts in the field of blade cooling, like Lakshminarayana's and Wilson's, are referred as they have been used extensively through this project.

In the same tone, the theory behind blade stress calculations is developed in section 2.4. In this topic, the distinction between theory and methodology is vague and as a result the theory section has been kept quite brief, while in contrast, detailed methods on blade stress calculations are presented in the methodology respective section. Still, section 2.4 discusses the different types of stresses that a gas turbine blade is exposed to, together with some conclusions that are commonly found in other relevant research works.

Creep theory is discussed in section 2.5. Among the various approaches and methods for creep-lifing calculations, Larson-Miller's method is selected for this work and the arguments for this decision are discussed. Also in the same section, the exclusion of other failure mechanisms, for example high-cycle-fatigue, from being considered in the current study is explained in detail.

Lifing theory is discussed in section 2.6 with emphasis on Miner's equation. The different approaches that OEMs take on lifing are part of the discussion in this section. The discussion carries on with a literature review of works that approach the use of Miner's equation and Larson-Miller graph in a combined manner. One particular case that introduces Monte Carlo simulations in the Larson-Miller parameter graph is discussed in detail.

Finally, a review of the techno-economic analysis part takes place in section 2.7. The concept of Techno-economic and Environmental Risk Analysis (TERA) is introduced and a number of relevant works are cited and discussed. As the techno-economic analysis is key deliverable for this work, the review in this section goes on to identify some of the gaps in the existing works and approaches.

2.2 Pre-swirl Nozzles in Gas Turbine Cooling Air System

In the last decades, efficiency and performance of gas turbines – both aero and heavy-duty engines – have grown significantly. One of the main drivers behind this improvement is the effect of advanced cooling systems, which protect engine's hot gas path components from temperatures close to or above their material's limits. As a result, engines are allowed to operate at substantially elevated firing temperatures, improving by this way drastically their performance characteristics.

These systems are usually quite sophisticated; they effectively cool-down various hot gas path components by using air bled from different stages of the compressor. A critical challenge for cooling systems is to use as low as possible compressor air. By achieving that, losses for the main gas flow are kept reasonably low and the engine does not compromise improvement in efficiency from high firing temperatures. For the particular example of internally cooled blades, preservation of high pressure and low temperature levels inside the blades with the use of minimum compressor air flow are the main critical success design factors for cooling systems.

In most cases, gas turbine cooling air systems feature pre-swirl nozzles (PSNs) to deliver air to the rotating blades. Pre-swirling cooling air has three main advantages (1):

1. Difference of rotational velocity between cooling air and rotor is reduced and this means that also pressure losses are reduced.
2. Pre-swirl nozzles exit air velocity vector is not perpendicular to the rotor disc plane but angled, reducing substantially in this way impingement losses.
3. More importantly, pre-swirl nozzles induce a decrease in cooling air total relative temperature inside blade cooling channels, compared with the cooling air total temperature upstream of the nozzles.

Pre-swirl nozzles, being a critical point for the effectiveness of the whole turbine cooling system, have been studied by a number of researchers. Meierhofer and

Franklin (1) appear to be one of the earliest published works on pre-swirl systems. Meierhofer et al. (1) stressed the importance of pre-swirl nozzles, by stating that pre-swirl nozzles effectiveness is one of the most critical factors for the effectiveness of the cooling air system in a gas turbine. Typical values for temperature reductions in systems with pre-swirl nozzles are 20 – 100K lower (colder) when compared with systems, which do not incorporate pre-swirl nozzles (2).

Most researchers (1), (3), (4), (5), agree on the following non-dimensional numbers, as the main ones to describe pre-swirl nozzles operational characteristics:

- swirl ratio (β)
- turbulent flow parameter (λ_T)
- rotational Re number (Re_ϕ)
- non-dimensional mass flow rate (C_w)

Starting with *swirl ratio* β , it is defined as the ratio of tangential velocity of cooling air leaving pre-swirl nozzles to the speed of the rotating disk at the same radius.

The equation is:

$$\beta = V_{\phi,P} / \Omega r_{BCH} \quad (2-1)$$

Where,

$V_{\phi,P}$	tangential velocity of pre-swirl air
Ω	angular speed of rotor
r_{BCH}	blades cooling passages radius to the engine axis

Turbulent flow parameter λ_T , *non-dimensional flow rate* C_w and *rotational Reynolds number* Re_ϕ can all be defined with the below equation:

$$\lambda_T = C_w / Re_\phi^{0.8} \quad (2-2)$$

Where,

C_w	non-dimensional mass flow rate ($=\dot{m}/\mu r_{PSN}$)
Re_ϕ	rotational Reynolds number ($=\rho \Omega r_{PSN}^2 / \mu$)
$\dot{m}$	mass flow rate
ρ	density
Ω	angular speed of rotor
μ	dynamic viscosity
r_{PSN}	pre-swirl nozzle exit bore radius to the engine axis

Calculation of the above non-dimensional numbers is important for the selection of relevant Test Case, as it is explained in paragraph 4.3.2. Apart from the importance of non-dimensional numbers, researchers have been investigating the impact of nozzles' location inside the engine on their cooling air flow characteristics. In an interesting work, Granovski et al. (6) compared two cases of similar geometrically pre-swirl nozzles but located in very different areas inside the engine. Figure 2-1 illustrates the case where pre-swirl nozzles are located away from the blades cooling passages, axially in the stationary part. Figure 2-2 shows a more conventional design case, where pre-swirl nozzles are located in a radial level opposite to rotor's cooling holes. Granovski concluded that cooling air flow characteristics are heavily depending on pre-swirl nozzles themselves and less on their location. Therefore, he considered the design factor of pre-swirl nozzles location inside the engine as neutral.

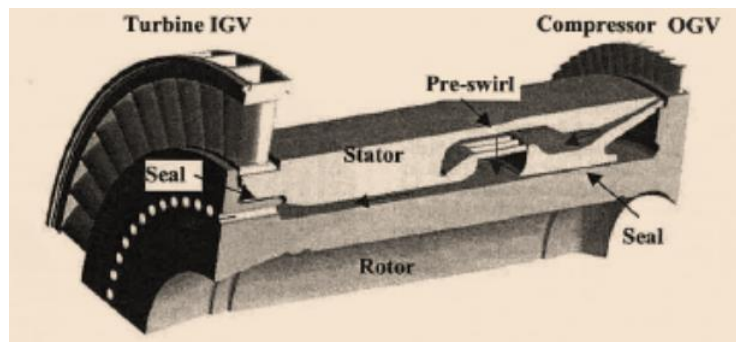


Figure 2-1: Schematic of pre-swirl nozzles located axially mid-channel **(6)**

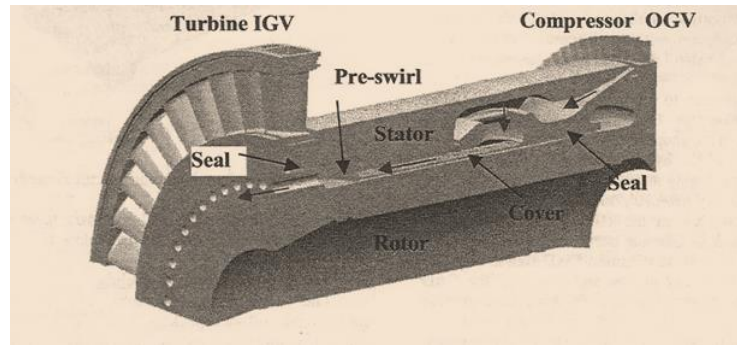


Figure 2-2: Schematic of pre-swirl nozzles located radially (6)

In another work, Kakade and Lock (7) investigated the effect of radial location of pre-swirl nozzles on heat transfer of the system. They took two cases, one where radius of pre-swirl nozzles to the engine axis r_{PSN} is lower than radius of cooling passages to the engine axis r_{BCH} and one where the two radii are equal. The first configuration was termed low-radius pre-swirl system ($r_{PSN} / r_{BCH} = 0.8$), while the second configuration ($r_{PSN} / r_{BCH} = 1$) was termed high radius pre-swirl system. With the use of experiments and CFD data, the authors concluded that high radius systems offer maximum decrease in cooling air temperature, but low radius systems appear to have less leakage flow through seals due to the different geometrical configuration between pre-swirl nozzles and cooling passages.

Apart from the above-mentioned factors, another critical design characteristic of pre-swirl systems is their differentiation in the so-called *cover-plate* system, shown in Figure 2-3, and *direct-transfer* system, shown in Figure 2-4. The difference between the two designs is that in direct-transfer pre-swirl system, cooling air is delivered from pre-swirl nozzles directly to the passages of the rotating blades. In cover-plate type, there is a rotating cover plate attached on the disc, which forms a rotating cavity. Through bores, the rotating cavity receives cooling air from pre-swirl nozzles and delivers it to the blades. Research has taken place for both types of pre-swirl systems. Geis et al. (5) concluded that the biggest difference between the two types is that direct-transfer system experience greater losses in total pressure due to strong mixing between pre-swirl flow and recirculating rotor-stator flow in the area between

pre-swirl nozzles exit and rotor disc. Pre-swirl nozzles both in this research work's gas turbine and in the selected Test Case are of direct-transfer type.

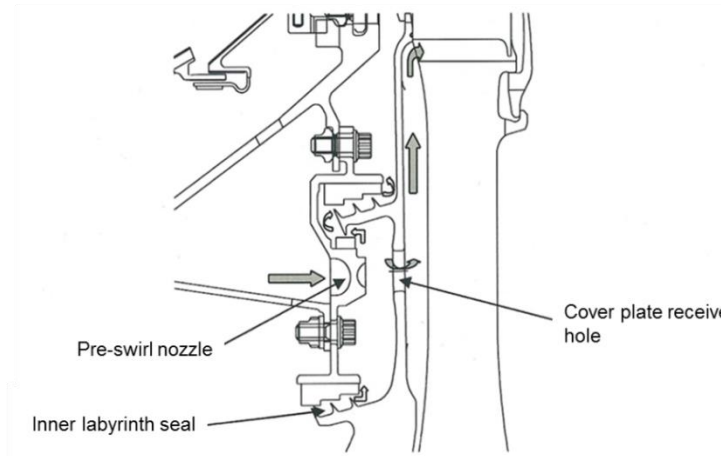


Figure 2-3 Typical pre-swirl nozzle with cover-plate arrangement **(8)**

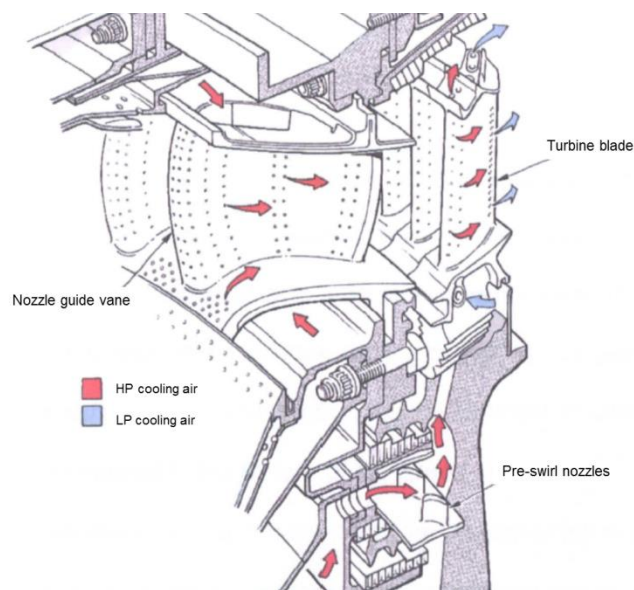


Figure 2-4: Typical pre-swirl nozzle with direct-transfer arrangement **(9)**

Finally, Ciampoli et al. (4) investigated how the geometrical characteristics of pre-swirl nozzles affect their operation and performance in an aero-engine application. The most influential characteristics were identified and illustrated in Figure 2-5, while in Table 2-1 the recommended baseline values, together with minimum and maximum ones are listed. According to Ciampoli, geometrical characteristics with values out of these minimum and maximum limits could not preserve adequate functionality of the specific pre-swirl system.

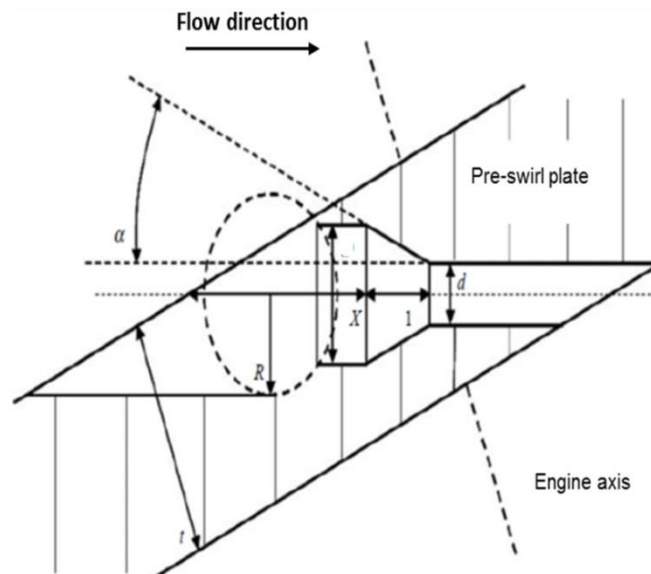


Figure 2-5: Parametric representation of pre-swirl nozzle (4)

Table 2-1: Typical geometrical characteristics of pre-swirl nozzles (4)

Parameter	Description	Baseline value	Min	Max
t	Pre-swirl plate thickness (mm)	13	10	20
d	Downstream nozzle parameter (mm)	7.2	6.7	12
l	Chamfer length (mm)	0.8	0.8	15
α	Chamfer angle (degrees)	45°	5	60
x	Chamfer distance from inlet (mm)	23	21	28

In a closing note, it can be said that the area of pre-swirl nozzles has justifiably attracted over the years significant amount of research works. This is explained by the fact that pre-swirl nozzles are critical for the effectiveness of cooling air systems. However, current literature review did not reveal any studies in which PSNs are approached holistically as part of the cooling air system and the engine. This in turn would have explored any effect of nozzles sub-optimum performance on downstream components. The impression is that these cases remain strictly within OEM's knowledge domain and probably cases publicly available are in scarce or do not exist at all.

2.3 Blade Cooling Theory

The drive for all-the-more higher turbine inlet temperatures derives from the explicit relationship between turbine inlet temperatures (TIT) and engine performance characteristics, such as thrust in aero-engines and energy output for land-based power generation gas turbines. However, hot gas path materials that are currently used by manufactures put specific limits in turbine inlet temperatures, typically in the range of 1800K (10). Blade cooling is used to maximise firing temperatures and, as a result, today's engines can operate at temperatures at least 300-350K higher than the temperature that they would have been operating in, if they were featured with un-cooled blades.

Cooling of gas turbine blades has been the subject of numerous research works over the last decades and three of the most influential works are those from Lakshminarayana (10), Han (11) and Wilson (12). Accurate prediction of cooling air flow that is required for keeping blade temperatures within metallurgical limits is of utmost importance for integrity of the blades and consequently of the engine itself. It is known that an increase in blade metal temperature by 10°C can reduce blade's estimated creep life by half. On the other side, but equally important, cooling air is sourced by upstream compressor bleeds and hence it is a very expensive medium for the engine's operational economy. Any inordinate amounts of cooling air flows incur significant penalties in gas turbine's performance characteristics, which are reflected in specific fuel consumption (SFC) or thrust for aero engines and heat rate (HR) or power output for power generation gas turbines. In Figure 2-6, Lakshminarayana (10) outlines from a historical perspective the most important blade cooling design concepts and the increase that they respectively allowed to turbine inlet temperatures. As the graph was taken from a late '80s paper, there is prediction for expected values of turbine inlet temperatures up to 2010. Engine developments of the last ten years proved this prediction to be right (13).

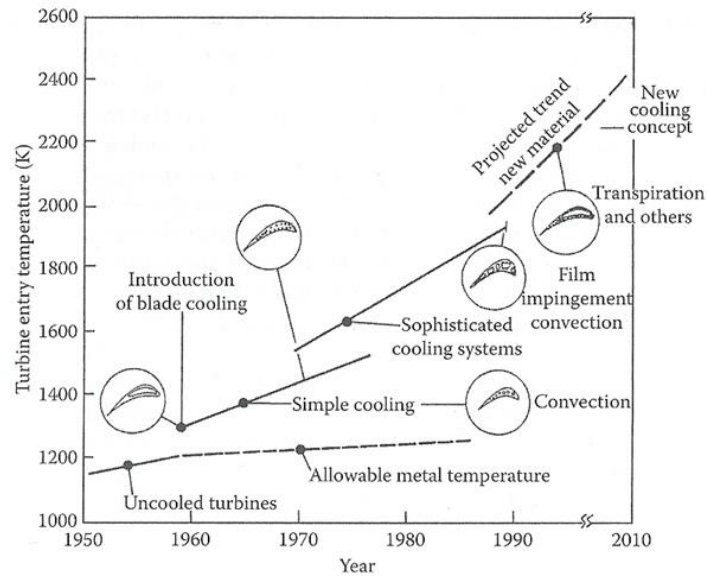


Figure 2-6: Increase of turbine inlet temperature following blade cooling technologies
(11)

2.3.1 Blade Cooling Methods

Introducing the basic theoretical principles for heat transfer on a gas turbine blade, all works (14), (11), (12), (10) agree on the coexistence of the three distinct heat exchange phenomena, namely heat transfer by conduction, by convection and by radiation. Starting from radiative heat transfer, it is profoundly important in the engine areas of combustor and first stage nozzles, where environment is also significantly more hostile in terms of gas path temperatures. However, for the purposes of current research project, the focus is exclusively on low-pressure-turbine 1st rotating blades, where radiative heat transfer is not considered. Hence only convective and conductive heat transfer phenomena are discussed in this section. In relation to them, there are three widely used design methods to provide cooling to gas turbine blades:

- I. convective cooling (CC),
- II. film cooling (FC) and
- III. impingement cooling (IC).

These three methods are considered to be industry standard for both aero-engines and heavy-duty gas turbines, since the concept of Energy Efficient Engine (E^3) in the early '80s (10). It has to be stated that only air cooling

methods are described in this paragraph, although other methods exist for different types of engines and different manufactures - for example steam cooling, water cooling etc. Besides, transpiration cooling method is not described in the below paragraph, as it is not used in GT26 blade cooling configuration¹. Moreover, it has to be clarified that thermal barrier coating (TBC) is approached as a passive cooling method in this research work, as it is shown in methodology section 4.6.

Overall aim of the combination of the three methods is to maintain a uniform as possible blade metal temperature, since the opposite conditions, namely strong temperature differentials across the blade, increases significantly thermal stresses and thermal fatigue. Figure 2-7 illustrates a typical gas turbine blade where all three methods apply and interact. Figure 2-8 illustrates the improvement in overall cooling effectiveness, when all cooling methods are combined (15). Overall effectiveness of blade cooling is defined by Lakshminarayana (10):

$$\varepsilon = (T_g - T_{bw}) / (T_g - T_{c, in}) \quad (2-3)$$

where,

- ε overall effectiveness
- T_g hot gas temperature
- T_{bw} maximum allowable blade wall temperature
- $T_{c, in}$ cooling air inlet total temperature

¹ Transpiration cooling in GT26 is used for cooling of EV and SEV combustors' walls

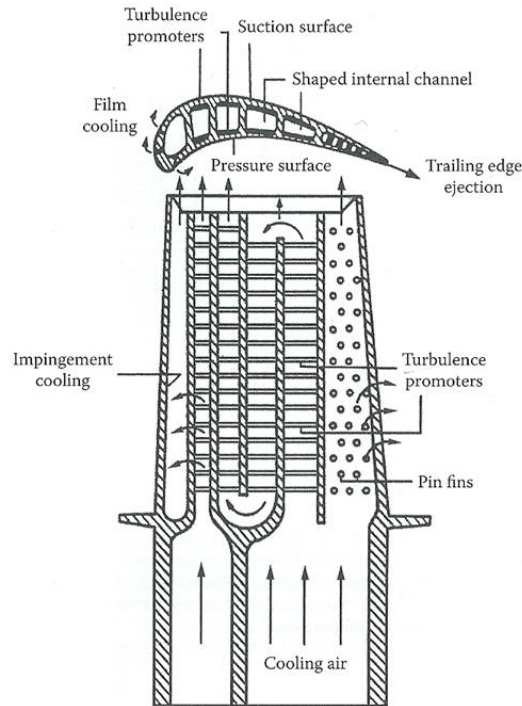


Figure 2-7: Typical blade with all three (convective, film and impingement) cooling methods (11)

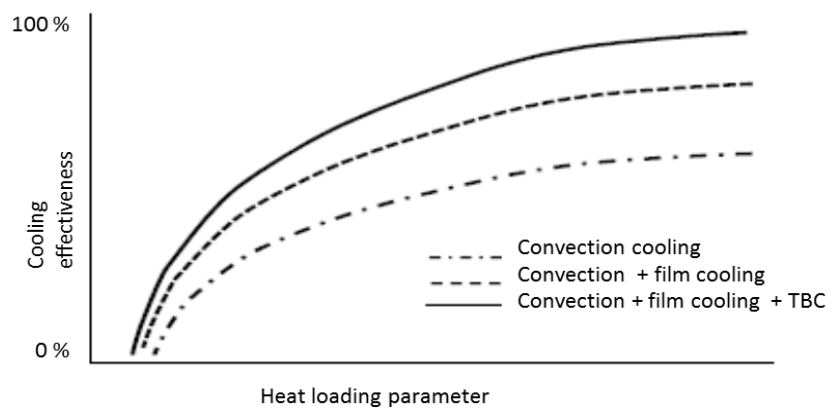


Figure 2-8: Cooling effectiveness with different cooling methods and combinations (15)

Convective cooling (CC) is the simplest method and the one that was conceived and applied first for blade internal cooling. Together with impingement cooling are classified as internal cooling methods, as opposed to film cooling which is classified as external. Convective cooling is achieved with the use of high velocity cooling air through blade's internal passages. Passages, almost always in modern engines, form a quite complicated internal

network where aim is to maximise internal surface of the blade without of course compromising blade's structural integrity. Spent air is ejected out at the trailing edge and/or the blade tip (Figure 2-9). Critical design details for any blade internal convective cooling system are:

- Effective and efficient interaction with the other two methods, namely film cooling and impingement cooling.
- Effect of Coriolis phenomenon, rotating buoyancy and centrifugal forces on the flow; these phenomena and forces alter the flow characteristics significantly. For example the effect of rotation, which has to be stated it is different on an inward to an outward channel, incurs a considerable heat transfer augmentation when compared with non-rotating passages (10). Lakshminarayana discusses these complex phenomena extensively in his work (10) and correlates the opposite sign (+/-) of Coriolis forces between inward and outward passages to heat transfer coefficient.
- Design features, such as pins and ribs inside cooling channels, that aim to promote flow turbulence inside passages; hence the term turbulence promoters (Figure 2-7) or turbulators.
- Pressure losses through coolant passages; as shown in Figure 2-7, a typical internal of a turbine blade is characterised by high complexity and narrowness of the passages. Curvature and roughness incur continuous changes in density and other flow characteristics. In addition, as it will be explained below, pressure ratio between cooling air and hot gas is of paramount importance not only for the effectiveness of film cooling but also for the integrity of the blade itself. A drop of cooling air pressure to lower than hot gas pressure levels would immediately mean hot gas ingestion inside cooling passages with detrimental effects for the blades.

It has to be stated that convective heat transfer particularly around turbine blades is quite complex. They can be identified at least six different ways of (subsonic) flow interaction between hot gas and blade body and this is because of the very different flow characteristics around blade surface. For example, the different heat transfer phenomena in a hot gas – blade system that have to be

studied would include the following separate cases: heat transfer at stagnation point, heat transfer in laminar boundary layer, heat transfer in transient boundary layer, heat transfer in turbulent boundary layer, heat transfer with separation of flow and re-attachment and finally with separation of flow and no re-attachment.

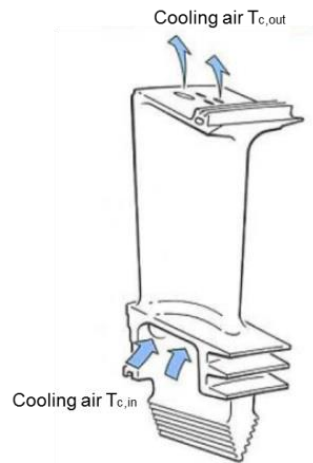


Figure 2-9: Typical convective cooling in turbine blade **(14)**

Film cooling (FC) involves development of a cooling air film around the blade that acts as a protective insulating layer against hot gases of main flow. Coolant is injected out of the blade through holes along blade body and protects from hot gases not only the immediate region but also the downstream blade area. Protection is achieved with a thin layer of cooling air, acting as a buffer between hot gas and blade material. Obviously, film cooling is used always together with convective cooling. Film cooling configuration can consist of a single slot or a single row of holes or multiple rows of holes on the blade surface, where the cooling air escapes from the inner passages to the gas path. Generally, film cooling is a very complex field. Apart from hot gas flow characteristics in the domain around the blade, designer has to consider numerous variables such as (10):

- cooling hole geometry characteristics – shape (circle/ellipsis/slot), diameter, angle of injection, spacing.
- Cooling hole rows, in terms of number and in-between spacing.
- Blade and cooling channel surface roughness.

- Configuration of *showerhead*; particular design (Figure 2-10) for cooling of blade's leading edge, where convective and impingement cooling methods are not sufficient. With showerhead method, cooling air penetrates hot gas flow to the inviscid zone, taking advantage of the stagnation point conditions that exist in the area.
- Choice of surface full coverage; design configuration where rows of cooling holes/slots cover full blade or nozzle surface both span and chord wise. Applicable where hot gas temperatures are very high, but it is also a design choice that incurs significant aerodynamic losses.
- Cooling air properties, such as velocity, density, pressure, temperature, turbulence characteristics and non-dimensional numbers, have to be understood in detail for every section of the tortuous cooling air path flow.

Finally, apart from heat transfer itself on the blade, another important aspect, which adds significantly to the complexity of the task, is interaction of the coolant, as it exits film cooling holes, with main hot gas. Mixing of coolant with hot gas is of paramount importance for flow characteristics of hot gas and calculation of cooling requirements of downstream stationary and rotating blades stages. Equally critical, mixing of coolant with hot gas has considerable effect engine's performance.

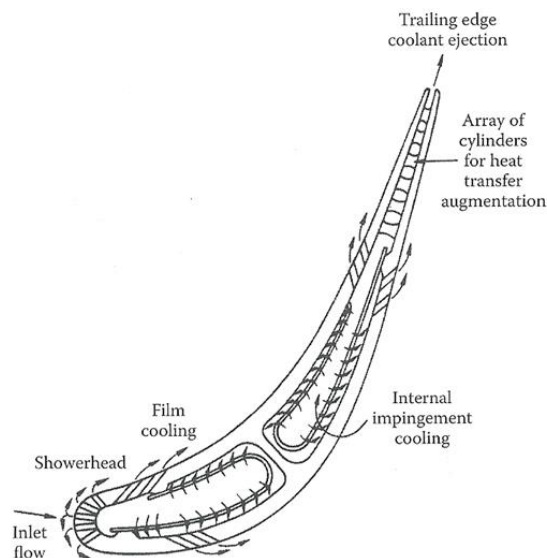


Figure 2-10: Showerhead configuration in film cooling **(11)**

Impingement cooling (IC) is the method, which both Lakshminarayana (10) and Han (11) agree that is the most effective method for increasing local heat transfer coefficients. With impingement cooling method, cooling air is jet-impinged through internal holes or slots directly on to internal blade faces in order to reduce metal temperatures. However, due to the particular construction that is required and to the added complexity that this construction brings to - an already complicated enough - blade internal configuration, impingement method's applicability is limited. Its biggest disadvantage is the fact that it results in considerable weakening of blade's structural strength. Therefore, it is used only for areas where thermal loads are excessive - typically blade's leading edge - or in nozzles (vanes) where due to absence of centrifugal stresses requirement for structural strength is substantially lower. Another construction limitation is due to additional space that impingement configuration requires and hence its use is limited to leading edge and to mid-chord regions of the blade only and not to trailing edge (Figure 2-12). Finally, another critical construction detail is the necessary offset between impingement holes and film cooling holes (Figure 2-11) in order to avoid escapement of the cooling air directly out of the blade. Impingement cooling is admittedly a very effective cooling method; however its performance is compromised by cross-flow effect between cooling jet and spent-air jets. This actually is considered the most important inhibiting factor to impingement effectiveness. Cross-flow from upstream jets interacts with downstream jets (Figure 2-12) and in cases where upstream cross-flow is strong enough serious deflection between jet flow and target plate takes place, which consequently reduces impingement effectiveness. Very similarly to film cooling, impingement cooling is a quite complex method especially in regards to heat transfer coefficients calculations. Numerous geometrical and flow characteristics need to be taken into consideration. The diameter of jet hole and the distance from target plate are two of the most critical dimensional ones.

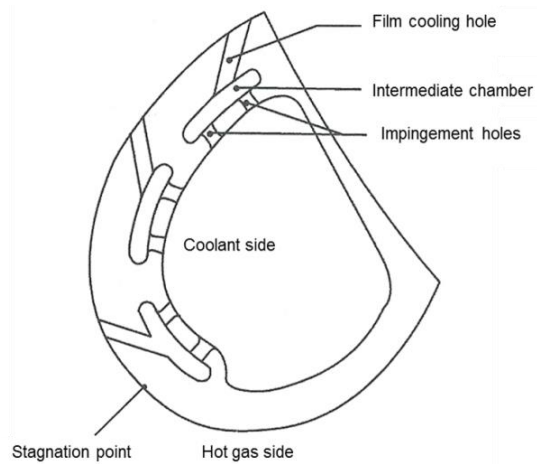


Figure 2-11: Detail of film and impingement cooling combination **(11)**

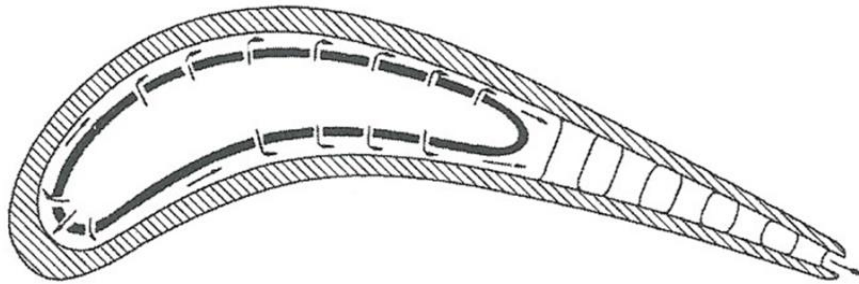


Figure 2-12: Typical impingement cooling configuration **(11)**

2.3.2 Critical Non-dimensional Numbers for Blade Cooling

As in almost all engineering calculations, similarly for those that involve heat transfer phenomena there are some important non-dimensional numbers that are essential for study and understanding of blade cooling (14). Nusselt number (Nu), Prandtl number (Pr), Stanton number (St) and Biot number (Bi) are described briefly in this paragraph.

Nusselt number (Nu) is the ratio between conductive and convective number. Nusselt number values are low (~ unity) when flow is laminar so heat transfer is predominantly conductive. They become high when flow is turbulent and consequently heat transfer becomes convective. The equation is:

$$Nu = (h L) / k \quad (2-4)$$

Where,

- h convective heat transfer coefficient
- L characteristic length
- k thermal conductivity

Prandtl number (Pr) expresses ratio between thermal and momentum diffusivity. Important detail is that Prandtl number is a characteristic of the fluid not the flow, as opposed to Reynolds (Re) and Nusselt (Nu) numbers. The equation is:

$$Pr = (\mu C_p) / k \quad (2-5)$$

Where,

- μ dynamic viscosity
- C_p specific heat capacity at constant pressure
- k thermal conductivity

Stanton number (St) gives the relation between heat capacity of the fluid and convective heat transfer. The equation is:

$$St = h / (\rho u C_p) \quad (2-6)$$

Where,

- h convective heat transfer coefficient
- ρ density of fluid
- u velocity of fluid
- C_p specific heat capacity at constant pressure

A useful equation that correlates the four important numbers is:

$$St = Nu / (Re Pr) \quad (2-7)$$

Where,

St	Stanton number
Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number

Finally, *Biot number (Bi)* is defined as the ratio of heat transfer resistance inside of and at the surface of a body. The equation is:

$$Bi = L h / k \quad (2-8)$$

Where,

h	convective heat transfer coefficient
L	characteristics length
k	thermal conductivity

The importance of the Biot number becomes apparent in the methodology chapter. In general, simple flows in terms of heat transfer have very small Biot number ($\ll 1$). The opposite case ($Bi \gg 1$) indicates that the temperature fields within and around the body are heavily non-uniform and the flow is complex (10).

2.4 Blade Stress Theory

As explained in chapter 1, a blade stress model has to be developed which, in combination with the blade cooling calculation, feed results in the blade lifing model. On the wide topic of blade stress calculations, relevant theory and methods are discussed extensively by Cookson (16), Laskaridis (17), Grkovic (18), Kaufman (19) and Singh (20). For the purposes of this section, only reference to some basic conclusions is made, while the details of the method and the relevant equations are presented in methodology's section 4.7. In summary, gas turbine blades are subjected to severe stresses during operation. Based on their origin, stresses can be classified as below:

- shear stresses
- momentum bending stresses
- gas bending stresses and
- centrifugal stresses
- stresses due to thermal gradients.

Shear stresses are produced from the untwisting effect of the blade itself. As they are considered relatively low comparing with the other four types of stresses, they are ignored for this study. Equally, stresses due to thermal gradients refer to transient operation of the engine, for example during start-up, and they are excluded from this work, which is focused on steady state conditions only. In all the works that have been reviewed, the remaining three types of stresses, namely momentum bending, gas pressure bending and centrifugal are calculated individually and they are vectorially summed in the end, providing the maximum stress levels that the blade is subjected to. Moreover, most workers agree on the fact that centrifugal stresses dwarfs the other two types of stresses (momentum bending and gas pressure bending) in this calculation. The explanation derives from the fact that centrifugal stress calculation involves the square of rotational speed, as opposed to linear products of the momentum and gas pressure bending calculations. Following the points above, a quite detailed methodology is developed in section 4.7.

2.5 Creep Theory

Creep is defined as the continuous deformation that occurs in a metal under temperature and stress and over a period of time. Creep failure mechanism is the most dominant failure mechanism and the most common cause of failure for heavy-duty gas turbines. Creep damage on a component typically starts when operating temperature is higher than 30-40% of its melting point. Important parameters that have to be considered for every creep calculation are stress, temperature, time and of course materials properties.

Figure 2-13 shows a typical creep time evolution graph under influence of strain (stress). There are three distinct phases that are termed as primary, secondary and tertiary creep. Primary phase is when first inelastic deformation appears in the material. This phase is quite steep but short, so soon it is followed by the secondary phase where creep rate is much lower, steady and for longer time. Secondary phase is followed by the tertiary phase, where creep rate increases rapidly and the material is lead to failure.

Figure 2-14 shows creep behaviour (resistance) that different engineering materials exhibit when they are under creep triggering conditions, namely stress and temperature over a period of time. Clearly, a creep limit for one material can be well within operationally acceptable conditions for another. Nickel alloys are base material for most gas turbine blades. Figure 2-14 shows that they can withstand a very desirable for gas turbines blades combination of very high temperatures and stresses.

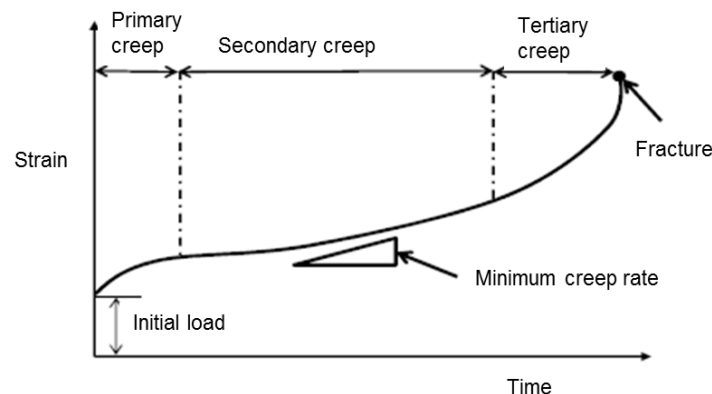


Figure 2-13: Typical time-strain creep graph

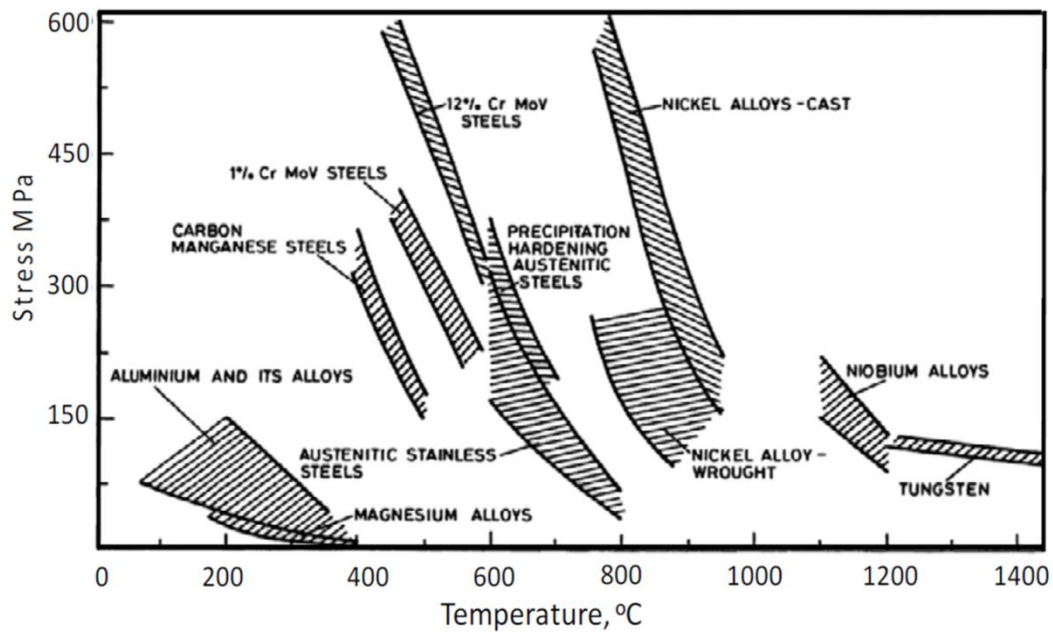


Figure 2-14: Stress-temperature-time (100h) results of different materials **(15)**

Creep is a subject with significant research and related literature on it. It is not the purpose of this work to analyse in detail all the different approaches and methods for studying creep life on materials. However, it remains important to understand the impact of degraded cooling air conditions on component's creep life, so it is critical the most appropriate method to be selected. Eshati (15) provides a brief summary of the most popular approaches and applicable methods for creep life prediction. In brief, these approaches are presented below:

- Design life approach;

under specific operating conditions (stress, temperature) a given life is prescribed for a component that is made of a specific material. At the end of the specified life, the component is retired.

- Creep strain approach;

actual strain itself, caused by creep, is measured dimensionally and evolution of creep is by this way monitored. When the component reaches unacceptable dimensional characteristics, it is retired from service. It is a very popular method in power plants, primarily applicable to heat recovery

steam generator (HRSG) headers, gas and steam turbine rotors etc. However it is not applicable on hot gas path parts (for example blades or vanes) where due to very tight by design operational clearances there is no allowance for creep-induced deformation of the material.

- Damage based approach;

this approach assumes that as the component, which assumedly is made by a material/alloy with known properties, operates in specific conditions, it accumulates damage in a linear way. There is a maximum limit of damage the component can accumulate, and upon reaching that point the component is retired.

- In service approach;

similar to creep strain approach, as non-destructive test (NDT) techniques are used to offer an evaluation of components status. The difference is that no dimensional checks take place, as the critical components remain within a closed engine and can only be accessed visually with the use of borescope/endoscope tool. Critical for this approach is the fact that it has to be combined with relevant crack prediction and propagation models.

- Statistical approach;

probably the most simplified approach; through pool of statistical data, explicit relationships are developed that describe expected creep life. Weibull distribution is commonly used in this approach. Simplistically explained, when a component reaches a point in its life that falls outside of confidence intervals of previously surviving components, it has to be retired.

It has also to be stated that in a number of cases OEMs and/or operators use combinations of the approaches above depending on the application and its criticality. Apart from the overarching approach, which can vary depending on application, there are also specific techniques and methods that are used to provide more tangible answers to creep-life questions. An interesting cluster of methods is the so-called parametric methods, where stress, temperature and time are co-related in the same graph. There are numerous different parametric

methods. They are all based on stress – temperature – time curves, which are derived from experiments. As creep experiments are expensive and time consuming, their results refer to a specific and usually limited envelope of temperature-stress-time conditions but through extrapolation and projection, wider range graphs can be produced. The user needs to extrapolate accordingly the conditions of interest and extract conclusions from the graphs. The most known creep parametric methods are: Orr-Sherby-Dorn parameter (OSDP), Manson-Haferd parameter (MHP), Goldhoff-Sherby parameter (GSP) and Larson-Miller parameter (LMP) (21). All these methods share common nomenclature:

- T_m creep initiating temperature (40% of total melting point) and
- t_f time to creep failure due to unacceptable plastic deformation or rupture.

2.5.1 Larson-Miller Parameter Method

Larson-Miller parameter (LMP) is probably the most popular and applicable method for creep calculations. It is used for a wide range of materials, including the blade material of the current project. As it is shown in Figure 2-15, LMP defines a constant point C where all iso-stress curves are converged. Constant C takes value of 20 for a wide range of applications and industrial materials (15), (21). Larson Miller parameter combines temperature, stress and time factors with the below formula:

$$LMP = (T_m/1000)(\log t_f + C) \quad (2-9)$$

Where,

- LMP Larson-Miller Parameter
- T_m metal temperature
- t_f time to failure
- C material constant

As already explained, selection of appropriate creep parametric method is important for prediction of blade's creep life and consequently a key step for the success of this research work. Always a critical decision factor for selecting creep parametric method is the material of the component under study. In the cases of common engineering materials and alloys, such as carbon steel, cast-iron, stainless steel and others, there is a plethora of publicly available creep related research and creep tests results, using almost all of the available creep parametric methods. However, for turbine blade materials, also called “exotic” materials, available references that discuss lifing characteristics are in scarce, primarily due to proprietary policies, and the use of creep parametric methods is more specific and selective. Blade material of the current project is CMSX4, which is a second-generation Nickel superalloy (22). Available references for CMSX4 use almost exclusively Larson-Miller method (22), (23). Conclusively for this project, Larson-Miller Parameter (LMP) is the chosen method for the next steps of lifing analysis and calculations. Paragraph 3.2.2 in technical description goes through the details of the available creep life data for the CMSX4-made blade of the current work.

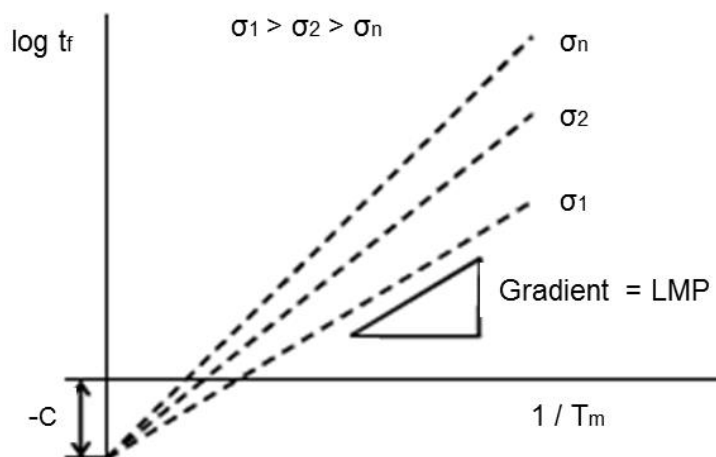


Figure 2-15: Typical Larson-Miller Parameter graph

2.5.2 Other Failure Mechanisms

Apart from creep, a number of other failure mechanisms can also pose threats on heavy-duty gas turbines integrity and life expectancy. Erosion, corrosion, High Cycle Fatigue (HCF), Low Cycle Fatigue (LCF), oxidation and Thermo-

mechanical fatigue are the most important mechanisms for further consideration (24) and the ones that are discussed below. A general comment, which has to precede the discussion, is the fact that hot gas path flow conditions, namely pressure, temperature and velocity of hot gas are unaffected by the changes in cooling air system conditions. In other words, as the blade has its inherited by design failure-resistant characteristics - for example protective coatings, high Ni-Cr content etc - any consideration of different behaviour against failure mechanisms and potentially reduced life expectancy, should be made in the context of cooling air temperature increase only.

External protection from erosion and corrosion on the blade is offered by its combination of MCrAlY and Thermal Barrier Coatings (TBC). These coatings do frequently suffer from spallation and these failures are considered by OEMs as life-limiting mechanisms (25). However, they are not classified as linear failure mechanisms and in most cases (26), especially in aero-engines, failures are caused by foreign object damage (FOD)² or domestic object damage (DOD). As land-based, heavy-duty gas turbines operate, almost always, behind a stringent arrangement of air intake filters, their environment of operation and primarily their quality of ingested air is significantly more controlled. An increase in cooling air temperature is extremely unlikely to affect external coatings integrity. Equally, the blade's internal passages surfaces, though in the case of the current blade are not featured with any protective coatings, are not expected to suffer any accelerated erosion/corrosion induced failure, following the above described increase in cooling air temperature.

High Cycle Fatigue (HCF) is a well-documented failure mechanism. The resistance of hot gas path components to HCF has been extensively studied and documented over the years in both industry and academy. A number of references investigate temperature's impact on material's HCF properties. In two cases, Laskaridis (17), (27) and Sampath (28) discuss the temperature incremental steps that the component has to experience before any meaningful

² Sand and even dust particles of particular size are also classified as FOD.

change in HCF properties can be recorded. The steps were in excess of 350°C, more specifically they were from 30°C to 400°C and from 400°C to 870°C. Conclusively, It can be said with confidence that the order of blade metal temperature change that is explored in this project, and it is lower than 15K, is not going to affect HCF life to a point that a quantifying approach is required.

Oxidation is an important mechanism to be considered if the resulting increase in blade metal temperature was originated by changes in hot gas path conditions. However and as already described, the changes under study in this project are in relation to the cooling air system of the blade, where a change in air temperature has not been found to be linked with a deterioration of oxidising behaviour. The main anti-oxidation protection feature for the blades is the MCrAlY coating, and if its integrity remains intact as it has been assumed above, then any changes in blade's interior of this magnitude are unlikely to affect and accelerate oxidation rates.

The conclusions that were reached above on the HCF case apply partially on Low Cycle Fatigue (LCF) and Thermo-mechanical Fatigue (TMF) considerations. The changes in conditions are relatively small and there is no reference describing the impact on LCF or TMF triggered by this variation. However and as already stated in the assumptions section 1.4, only steady state conditions are explored in this work. In other words, engine's transient operation during start-ups, shut-downs and rapid load changes is excluded from the scope of this work, as they can form a separate research topic on their own. Moreover, the relative expansion between rotor and casing during engine's start-ups and shut-downs alters significantly the dimensional characteristics of the PSNs seals, so any case of relaxed seals has to be approached from a totally fresh point of view. And as for the engine of this project the total interval of transient operation is quite low compared with the base-load one, this analysis can be omitted from the current work. Transient analysis is recommended as future work in section 6.3 and in this case, impact of cooling air changes on LCF and TMF should be integral parts of the scope. The same conclusion applies on the presence of corrosion-fatigue.

2.6 Lifting approach and Miner's Equation

Almost all gas turbine original equipment manufacturers (OEMs) have developed their own lifting methodology and models. As described above, there are different philosophies, approaches and methods for creep life calculations. They span from basic safe-life approach to more sophisticated methods that employ neural networks and fuzzy logic. These methods are out of this work's scope and have not been discussed at all. Most OEM- or user-developed models include a number of different characteristics and features, trying to combine the most important, and applicable to their engine or component, advantages from every creep analysis approach and method above.

In heavy-duty gas turbines industry, OEMs are more aligned in their philosophy and approach on lifting. Their approach starts with the aim to capture accurately the effects of engine's operational profile such as number of operating hours, number of cyclic events, type of cyclic events etc. Then with the use of an appropriate model, the impact of ageing of the engine - namely the accumulated operating hours - is combined with the severity of operational events - namely operation in specific temperature ranges, number of engine start-ups, number of trips etc. This combination is reflected in an explicit relation, usually in the form of a mathematical formula. Knowing the safe life limits of their engines, lifting formula's outcomes drive decisions as to when the engine is due for inspection or overhaul. OEMs' used terminology is quite similar as GE use the term Factored Hours (FH), Siemens uses the term Firing Hours (FH) and Alstom the term Equivalent Operating Hours (EOH). The latter of the above models is the one that is used for the gas turbine of this project (GT26) and it is presented in some more details in paragraph 3.2.3. As a general comment, it can be said that all OEMs and researchers agree that one of the most challenging problem in lifting calculations is to sum various operating intervals of an engine all in different settings of firing temperature and/or stress. It can also be said that most lifting models are based on the fundamental principal of Miner's equation (17), (16), (29):

$$\sum \frac{t_k}{L_k} = 1 \quad (2-10)$$

Where,

t_k time that the engine/component spend at particular temperature-stress conditions

L_k total safe life that the engine/component has at the above temperature-stress conditions.

Miner's equation applies the principle of cumulative creep damage. The limitation is that it assumes linear damage in all cases, in other words similar fractions of time, under similar conditions, cause similar damage. Miner's equation is widely used in both academy and industry and this assumption has not affected its popularity. In this work, the deltas in blade metal temperatures, which are investigated, are relatively low, in the order of 5°C to 7°C. The author considered this fact also as supportive of the view that the incurred creep damage is expected to have linear behaviour. Conclusively, the assumption was used unchallenged, being however recorded in sections 1.4 of Assumptions and Limitations and 6.3 of Future Work.

Following on the theory above, a brief literature review reveals that the combination of creep parametric methods and Miner's equation is a popular research topic. Cranfield's researchers Eshati (15) and Obonyegba (21) both used combination of LMP and Miner's equation analytically in their works to derive life predictions. Similarly, Christensen in his work (30) uses Miner's equation, though his primal focus is cumulative fatigue calculations; however he expands his conclusions on creep also. It is interesting that Christensen recognises the limitation and the inherent uncertainty of the analytical models and hence the importance that empirical safety factors have. Very similar conclusion is reached by Tinga et al. (31). In his work, operational data³ is recorded and fed to analytical lifing models and subsequently lifing prediction

³ The paper refers to aero-engines; operational data in terms of flying recorded data.

can be made. However, the authors recognise that there is high inaccuracy in analytical lifing models and large scatter in material properties and that future work should focus on improving them. Shepherd and Williams (32) in a famous paper of the previous decade, went further and associated high fidelity stress calculation models with statistical models. Again, the focus of the authors was on fatigue, though most of their conclusions are transferable and applicable also to creep calculations. Recently, Choudhary et al. (33) presented a work on creep assessment⁴, in which lifing models are combined with Monte Carlo simulations. In this way, thousands of points are produced on a Larson-Miller graph, which then are statistically processed. The author concludes that increased confidence can be assumed by this way, deriving from the fact that uncertainty is captured through use of thousands of Monte Carlo Points. Choudhury's is the only found study where analytical models are combined with Monte Carlo and probability density functions. He goes on to quantify the risk in terms of expected drop of reliability as temperature and operating hours progress.

In summary, most studies use the combination of LMP and Miners equation analytically. Hence they are exposed to uncertainty, which unavoidable lies with a number of factors like material properties, experimental data etc. In addition, the above-described works do not quantify the risk of increased life consumption. Choudhury does it, but in a quite different way comparing with the approach in this work which is presented in section 4.8.

⁴ The paper refers to P91 material, which is a popular for its creep-resistant properties material in power industry.

2.7 Techno-Economic Analysis

The techno-economic analysis is the part where largely novelty and contribution of this work come in. Starting with the already available tools and techniques, Cranfield University has developed over the years a holistic tool for techno-economic analysis, named as Techno-economic and Environmental Risk Analysis, TERA (34). The TERA framework was initially conceived to be used on aero-engines, but soon its applicability expanded so as today to be considered a valuable and popular engineering asset management tool.

A number of works involve, to different extent, TERA. Khan et al. (35) links effectively engine lifing risks with the economics of the maintenance events (Figure 2-16). Maintenance events are approached statistically in terms of Mean Time Between Failures (MTBF) and “bath-tube” curves are derived. The aim is the development of a tool that calculates total lifecycle costs. Isaiah et al. (36) in another work integrates lifing calculations to derive maintenance costs, shown in the flow chart of Figure 2-17. An interesting characteristic of his work is that the modularised approach allows for varying levels of sophistication to be applied on every module. Creep life is calculated by considering seasonality effect, for example higher ambient temperatures in the summer. This then is factored to the maintenance planning process. In the same context, El-Suleiman (37) uses TERA in his work. Gas turbine performance characteristics are used to derive revenue, while maintenance and outage costs are also calculated and are factored in the equation. The final economic model uses the concept of Net Present Value (NPV) to conclude on optimum gas turbine selection. Finally, Lorenzo (38) makes exemplar use of the TERA framework, as shown in Figure 2-18. Engine’s performance characteristics, though output and efficiency, feed to the economic model. In the same model, risk is factored and its variability is captured through Monte Carlo simulations. This results in a financial risk model whose outcome is in the form of Net Present Value (NPV), Internal Rate of Return (IRR), Pay-Back Time (PBT) etc.

At this point, it has to be clarified that the environmental part of the TERA analysis, which is used extensively in the above works, is only partially

applicable in the current work. Through methodology's results that follow later in the thesis, it becomes apparent that the perceived impact of cooling air system degradation is only on engine's lifing. It is assumed that NO_x, SO_x or CO emissions remain within normal expected levels and no excursion takes place that would have been sizeable enough to be factored in any techno-economic and environmental assessment. CO₂ emissions are increased, as part of the proposed in chapter 5 engine derate, however this increase is factored effectively in the Spark Spread calculation, since lower thermal efficiency means higher CO₂ emissions. Moreover, CO₂ prices are considered constant through the timeline horizon that this work explores.

Apart from Cranfield's approach on techno-economic analysis, there are numerous other works, which combine the performance and technology elements of gas turbines with risk analysis and economic aspects. Marini in her work (39) identifies the accentuated, due to renewables, volatility as one of the key characteristics of the electricity market. Volatility plays a catalytic role in engine's Reliability, Availability and Maintainability (RAM), but it also affects heavily predictions made by economic models, especially when those have analytical approach. In another interesting study, Majid et al. (40) calculates all the costs associated to a gas turbine failure, including the ones due to lost-generation. Failure probability curves are then used to output total costs through engine's lifecycle. Finally, Breeze (41) in his report "The Cost of Power Generation" provides an inclusive guidance of costs calculation in relation to electricity generation. Extensive historical data is used and the split among the various cost centres is detailed.

In summary, both Cranfield and non-Cranfield studies approach the techno-economic analysis with an open-mind. A wide range of technology, environmental and risk factors are considered and their contribution to costs and to decision-making processes is analysed in depth. However, the gap in the above works is their approach to market's volatility. Though it is recognised as a factor that induces high levels of uncertainty into any calculations, no study really attempts to tackle it. The research is rightly focused on engines

economics but in a deterministic and macroeconomic only, it can be said, manner. The difference between current work and the studies above becomes more explicable in chapter 5.

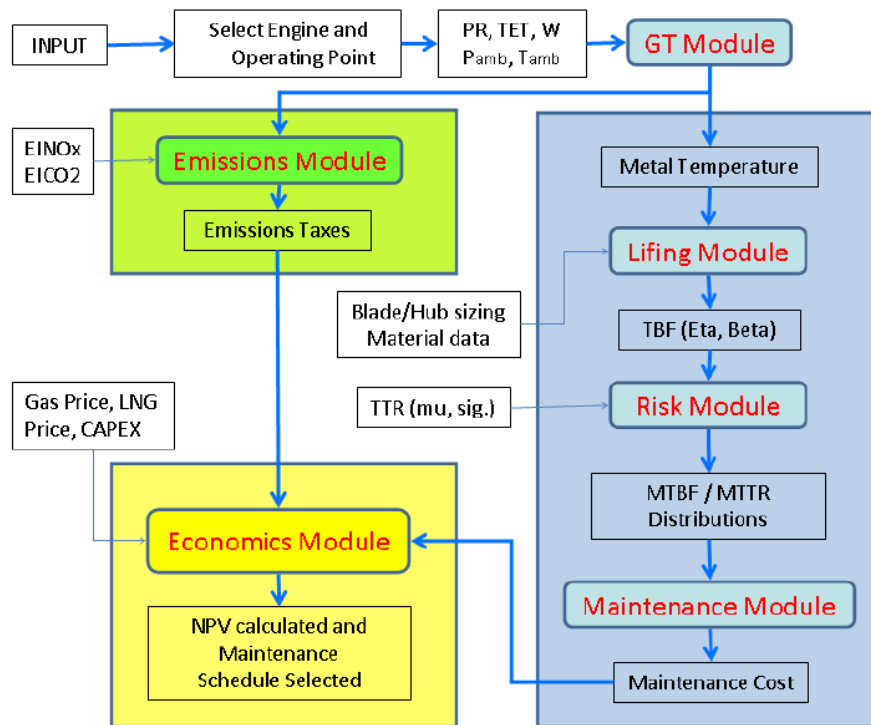


Figure 2-16: Khan's (35) approach on TERA framework

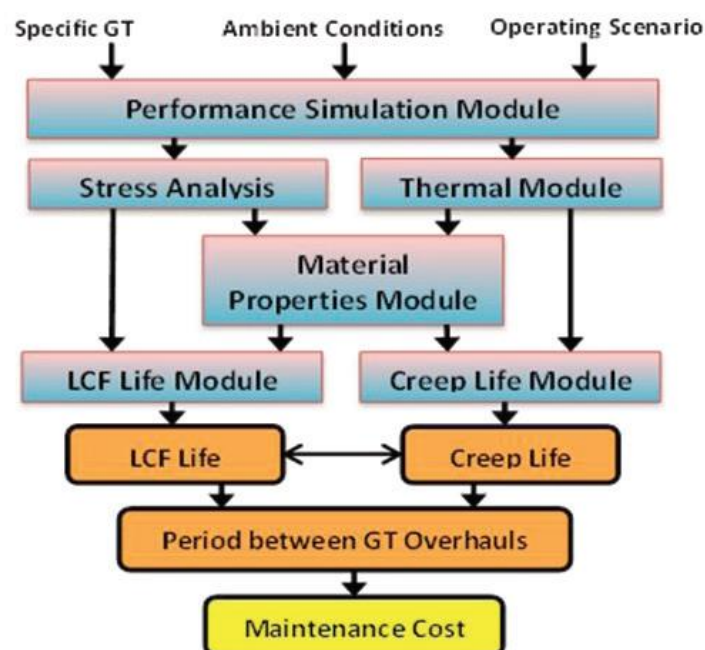


Figure 2-17: Isaiah's (36) approach on TERA framework

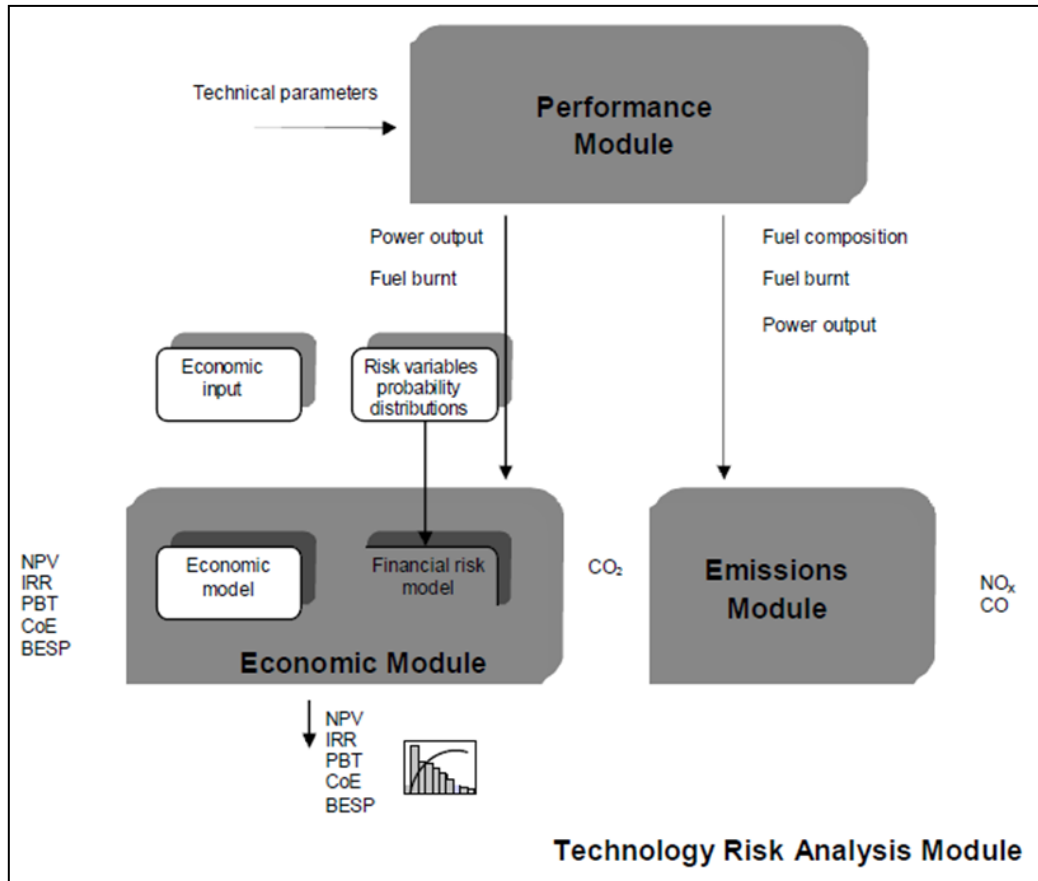


Figure 2-18: Lorenzo's (38) approach on TERA framework

2.8 Summary

In a brief summary, the literature review chapter started with reference in the pre-swirl nozzles. The purpose of the nozzles was explained in detail. Critical non-dimensional numbers, importance of nozzles location inside the engine and key geometrical characteristics of the nozzles were also discussed.

Section 2.3 was about blade heat transfer and in primarily blade cooling. The principles of heat transfer together with the three distinctive methods of convective, film and impingement were discussed. Cooling effectiveness was also defined.

In section 2.4, theory of the stresses on turbine blades has been very briefly discussed. Some commonly found conclusions in relation to the origins of stresses are cited. Detailed methodology on stress calculation is presented in section 4.7.

A summary of creep theory followed in section 2.5. As it is a quite wide topic, only some of the approaches were discussed. Various parametric methods were mentioned, while emphasis was given on Larson-Miller Parameter method as the method of choice for this work. The section closed with a brief discussion on other failure mechanisms that can have, in theory, an impact on low-pressure-turbine blade. The arguments behind the decision not to consider any other failure mechanisms apart from creep were presented.

Section 2.6 offered a very brief summary of OEM's lifing approach, while GT26's lifing calculation method is described in more details in the Technical Description chapter. Miner's equation, as one of the most useful and applicable equations in lifing theory, was cited. The section included also a literature review of similar works.

A literature review on techno-economic analysis took place in section 2.7. The concept of TERA was described in detail as a holistic tool with many applications. A number of interesting works were discussed, some of them using TERA and some using different techno-economic models.

In a closing note, it can be said that the three areas of blade cooling, blade stress and creep were only discussed at a high level. From each area, the principles that apply in this work were identified and thereupon are used in the methodology. The approach was different for pre-swirl nozzles, lifing and techno-economic analysis, where a more critic view was offered for reasons explained above. The most important gaps that were found in the relevant literature are summarised as below:

- Pre-swirl nozzles have been studied extensively but in isolation of the cooling air system and the engine in general. No case was found in the public domain where PSN's performance is linked with further downstream cooling air conditions or with downstream components temperature.
- Lifing calculations combine LMP and Miner equation, however the vast majority of the studies remain analytical and unavoidably exposed to the

inherently high uncertainty of the involved process, for example scattered experimental and material data.

- Combining the above two points, no case was found that studies the relation between cooling air system and lifing in an explicit way. This is well in contrast with the abundance of cases describing the relation between hot gas path conditions and lifing.
- Finally, the techno-economic analysis in all studies involves a quite holistic model. Gas turbine performance characteristics, in particular, are examined in a wider techno-economic and environmental context. Also in some cases, purely technical issues, for example lifing risks, are approached holistically, so as not only the integrity threat but also financial and/or environmental impact to be effectively assessed. However, all reviewed cases approach the economic part macroscopically and mainly with the use of financial indicators like NPV, IRR and PBP. Market's volatility is not factored effectively and remains a heavily random and difficult-to-employ parameter for most of the researchers. This particular gap is going to be more evident in the last chapter of the thesis, when it is contrasted with this work's approach.

3 TECHNICAL DESCRIPTION

3.1 Introduction

This chapter provides the necessary technical description of the key components, which are involved in this project, namely the low-pressure-turbine pre-swirl nozzles and 1st stage blade. This deemed to be an essential part of the thesis as, by this way, critical input information to the project's methodology is explained and can be traced back. The chapter also describes the existing maintenance schedule of the engine, as well as OEMS's lifing philosophy. Similarly, these details are important for the technoeconomic analysis in chapter 5. Figure 3-1 illustrates in the form of a flow-chart the information-transfer links between Technical Description and Methodology chapters. More specifically, this chapter provides:

- a brief description of project's gas turbine, which is Alstom's GT26, and its main parts. The description is also supported by Appendix A, which offers a synoptic description of its parts.
- A detailed description of GT26 low-pressure-turbine pre-swirl nozzles; the cooling air flow conditions in the area upstream of the pre-swirl nozzles are also quoted. Details of the nozzles are critical for the analytical solution and the CFD models. As pre-swirl nozzles are part of a particularly sophisticated and complex system, a more detailed description of the cooling air system and an explanatory schematic are cited in appendix B.
- A brief description of low-pressure-turbine 1st stage blade; Larson-Miller graph for blade's material is also cited and discussed. These details provide the input to blade metal temperature, blade stress and lifing calculations in the methodology chapter.
- A brief summary of engine's planned maintenance schedule, in terms of events and their durations. In addition, a brief summary of OEM's lifing

approach and the principal lifing equation are described and the expected operational intervals between maintenance events are listed. In the same topic, Appendix H offers some more details on the most important maintenance events and their typical scope of work.

- Finally, section 3.3 offers a detailed description of the Test Case rig that is used for the validation of the CFD model. Test Case experiment's results are also cited and discussed. They are important as for the first time in the project the quantified impact of degraded cooling air system on cooling air temperature is presented.

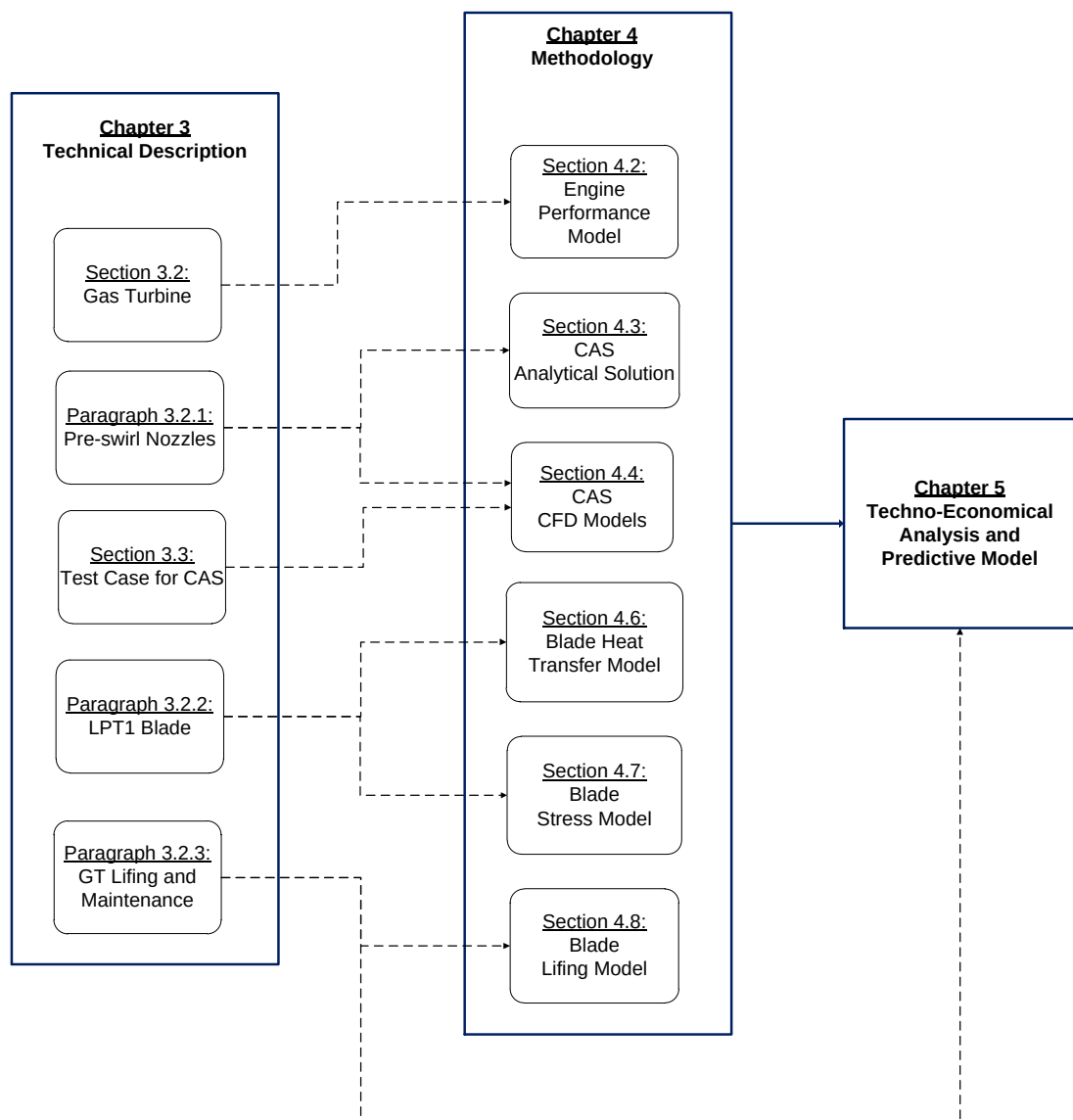


Figure 3-1: Interaction of Technical Description with Methodology

3.2 Gas Turbine

Alstom's GT26 is one of the four large frame⁵ heavy-duty gas turbines in the global power industry. GT26 was initially launched by Alstom (then ABB) in the early '90s and today the fleet has more than 100 engines in all over the world (42). Applications vary from electrical power generation in base load, intermediate or peak duty and industrial applications such as paper mills, off-shore oil rigs, desalination plants etc. Main operating data of GT26 is shown in Table 3-1.

Quite distinct feature of GT26, which actually differentiates it from the other three OEMs, is the sequential combustion. As it is shown in Figure 3-3, combustion in GT26 takes place in two different and separate stages, which are called EV and SEV combustors. This is a unique configuration, since no other heavy-duty gas turbine or aero-engine has been developed with similar architecture.

Gas turbine is part of a combined cycle gas turbine power plant CCGT(PP). Steam is produced in the heat-recovery-steam-generator (HRSG) and is directed to a steam turbine (ST). A detailed cross section of the gas turbine is illustrated in appendix A, while the main parts of the engine are described below and shown in Figure 3-3.

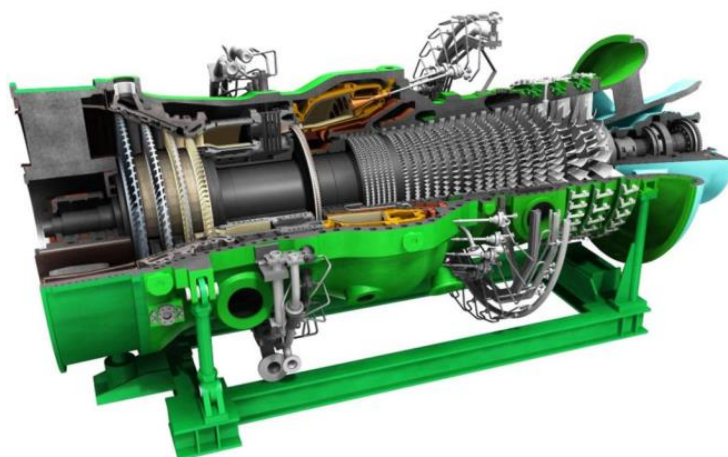


Figure 3-2: GT26 typical (42)

⁵ The other three are GE 9FA, Siemens SGT and Mitsubishi-Hitachi M

Table 3-1: GT26 typical operating data (42)

Fuel		Natural Gas
Frequency	Hz	50
Turbine speed	rpm	3,000
Gross electrical output	MW	326
Gross electrical efficiency	%	40.3
Gross heat rate	kJ/kWh	8,933
Compressor pressure ratio		35.00 : 1
Exhaust gas flow	Kg/s	692
Exhaust gas temperature	°C	603
	K	876
Weight	tones	406
Dimensions (L x W x H)	m	12.00 x 4.9 x 5.5

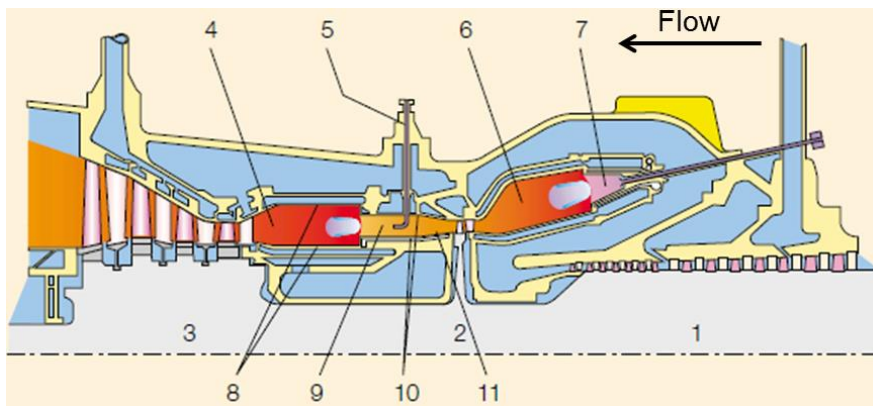


Figure 3-3: Main parts of GT26 (42)

Where,

1. Compressor
2. High pressure turbine (HPT)
3. Low pressure turbine (LPT)
4. SEV combustor
5. SEV burner fuel supply
6. EV combustor
7. EV burner
8. SEV combustor walls
9. SEV combustor mixing zone
10. SEV burner walls
11. SEV vortex generator

3.2.1 Pre-swirl Nozzles

Pre-swirl nozzles are in the stationary part that is called *rotor cover*. They are located in the same radius with the first stage LPT blades' cooling passages. The design configuration is of direct transfer type. Table 3-2 lists the most important geometrical characteristics, while Figure 3-4 shows a photo of the pre-swirl nozzles from an open-engine maintenance event. The cooling passages (holes) of the first stage LPT blades are also shown. For the avoidance of any confusion, it has to be clarified that the upper half of the part rotor cover is not shown in this photo.

The LPT pre-swirl nozzles provide cooling air to 1st stage of LPT blades. Cooling air is bled from compressor's 16th stage and is delivered to pre-swirl nozzles after being cooled in an external heat exchanger⁶. Cooling air upstream of pre-swirl nozzles has approximately temperature of 643K and gauge pressure of 2150kPa, when engine operates at design point (base load, ambient temperature 284.15K). Total mass flow, which is bled from compressor's 16th stage, is around 145kg/sec at design point, but not all this amount of flow is delivered to pre-swirl nozzles. It is assumed that approximately 24% of the flow finally arrives to low pressure pre-swirl nozzles, while remaining 76% cools 1st stage nozzles, SEV casing and SEV burners. These operating conditions are known through engine's monitoring software Optigen, which is described in more details in paragraph 4.2.1. Also the above details feed to the performance model that is developed with the use of GasTurb and it is also discussed in paragraph 4.2.

⁶ This bleed is called MBH30 according to KKS nomenclature system. Appendix B offers some more details on GT26 cooling air system as well as on the use of KKS system. The external heat exchanger is called Once Through Cooler (OTC) and is also described in some more details in Appendix B.

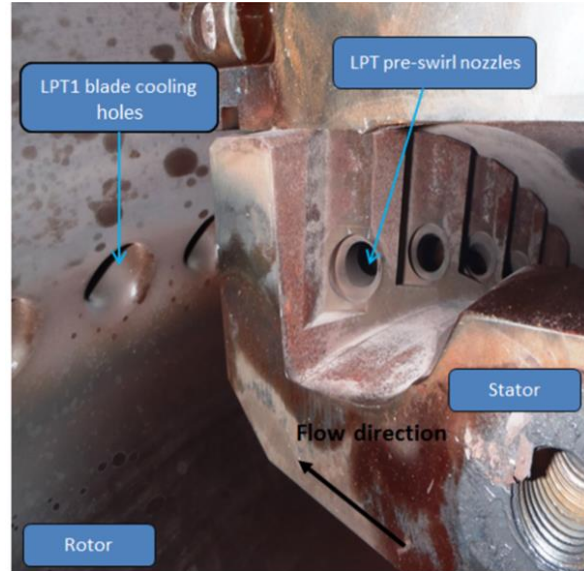


Figure 3-4: Open-engine view of LPT pre-swirl nozzles

Table 3-2: GT26 pre-swirl nozzles main geometrical characteristics

Nomeclature	Description
NPSN	Number of nozzles (pcs)
DPSN	Pre-swirl nozzle hole diameter (mm)
LPSN	Length of nozzle (mm)
Ra	Roughness (mm)
β	PSN angle (x-z)
α	PSN angle (x-y)
rPSN	Outlet hole location radius refer to turbine rotor axis (mm)

3.2.2 Gas Turbine Low-Pressure-Turbine Blade

Important geometrical characteristics of LPT 1st stage blade are listed in Table 3-3, while the blade itself is shown in Figure 3-5. These details are used in methodology's sections 4.6 and 4.7 for blade metal temperature and blade stress calculations. Moreover, Harris et al. (22), Rowe (23), Schneider et al. (43) and Pollock et al. (44) provide significant details on blade's material, which is CMSX4. In particular, CMSX4 behaviour under creep-promoting conditions (stress, temperature, time) is discussed extensively in these two works. Figure 3-6 shows 108 points of CMSX4 creep test results, collating more than 20 years of research on the particular alloy (23). It is important to be clarified that these

points refer to specimen failures, in the form of detectable⁷ crack formation or in the form of unacceptable permanent dimensional deformations. Moreover, these points on the LMP graph delimit an area where actual operating points should not approach, as clearly presence in the area poses unacceptably high lifing risk for the component(s) and consequently for the engine.

The stress values of the 108 points in Figure 3-6 cover a significant range from as low as 60MPa up till values higher than 900MPa, which also explains the need to be captured in a logarithmic graph. However, and as it is discussed in paragraph 4.7, LPT blade stress levels are in a region lower than 200MPa. Conclusively, not all the population of historical data points for CMSX4 is needed and a maximum stress value filter was applied. This simplified significantly the graph, as illustrated in Figure 3-7. By applying a filter of <200MPa, a linear Larson-Miller graph is formed showing a limited number of historical data points. This in turns allows for useful statistical analysis of the points such as the development of linear regression points and their respective linear regression line. Further statistical analysis as well as generation of Monte Carlo points based on the population of limited historical data points is discussed in paragraph 4.8.1.



Figure 3-5: Low-pressure-turbine 1st stage blade

⁷ Through Non-Destructive Tests (NDT)

Table 3-3: Blade geometrical characteristics

Nomenclature	Description	Origin
Nbl	Number of blades	Counted
Lspan	Blade span	Measured
Lchord	Blade chord	Measured
Sg	Blade external perimeter	Calculated
Lequiv	Blade equivalent length	Calculated
Ncc	Number of cooling channel passes	Measured
dcc	Diameter of cooling pass	Measured
RDS	Root Datum Section	Measured

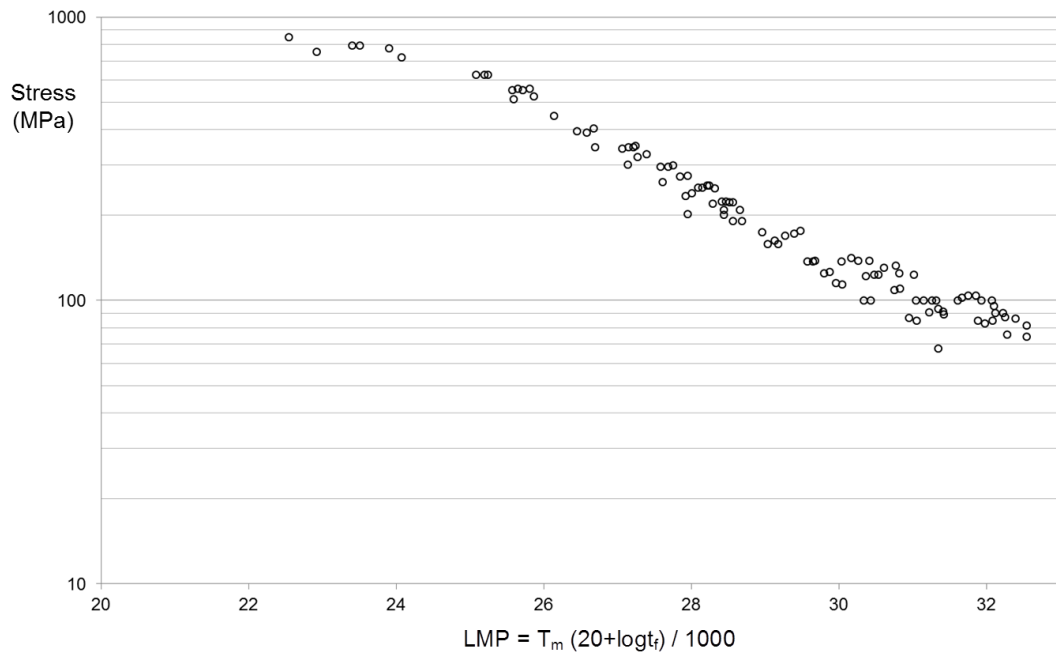


Figure 3-6: Larson-Miller graph for CMSX4 (23)

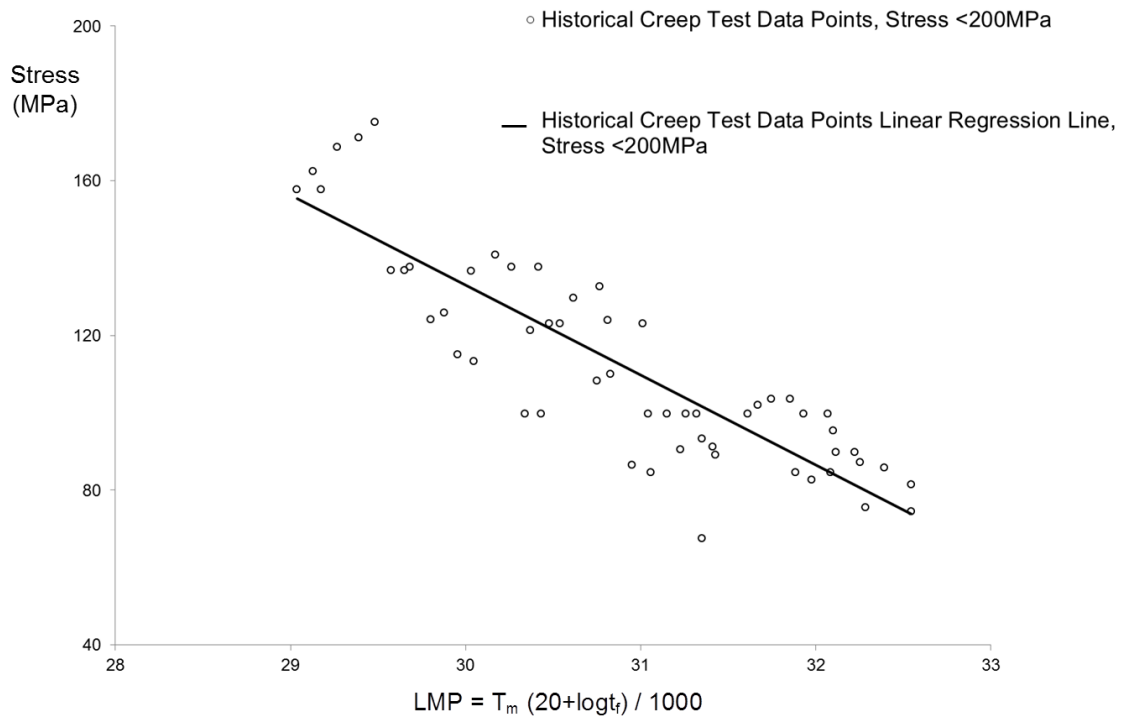


Figure 3-7: Larson-Miller plot for CMSX4, Stress <200MPa

3.2.3 OEM's Maintenance approach and Lifting Philosophy

Key deliverables of this work are the development of a lifting model that is able to capture the impact of cooling air system degradation, and the development of a techno-economic analysis that offers as a result the optimum commercial operation for the engine in the light of the above-mentioned lifting risk. As such, it is important the OEM's approach in terms of maintenance and lifting philosophy to be discussed. Not only because they interact closely with the above deliverables, but also because the reasons the existing lifting model fails to capture and reflect lifting risks when these are imposed by engine's cooling air system need to be understood.

Starting with the maintenance approach, as part of the contractual operational and maintenance (O&M) support, OEM prescribes three types of scheduled maintenance events. They are called A, B and C Inspections and are described in some more details in Appendix H. Through the cycle and between the events,

borescope inspections with closed-engine are employed to provide visual (remote) access to hot gas path (HGP) parts.

Table 3-4 below lists typical durations for the three types of inspection events. It has to be stated that listed durations are indicative, as the exact duration is defined by the details of the scope, which may vary⁸. However, the duration of these events, and primarily C Inspection's, is critical input to the techno-economic analysis discussion, due to the lost-generation cost that is unavoidably associated with these events. Among the three, C Inspection is the major overhaul, and hence the biggest, event of the engine. It includes removal of all upper compressor and turbine casings, removal of rotor, detailed inspection of all compressor parts with use of non-destructive tests (NDT) and replacement of all hot gas path parts. At the end of their operational cycle components are retired or rejuvenated and return for a second cycle of operation⁹. Physical access to the hot gas path (HGP) parts is possible only every end of the cycle¹⁰, which is usually between 4 and 5 years. C Inspections are key events in the engine's lifecycle and the largest cost-centres in the operators financial sheet. The origin of costs associated with C Inspections is discussed extensively in paragraphs 5.2.1 and 5.2.2.

In terms of lifing philosophy, OEM's lifing model for GT26 is relatively simple. Operational profile and history are monitored and captured and the details of events are used to calculate accumulated damage life. OEM's principal lifing formula is described by the following equation (45):

$$EOH = WOH + 10 WCE \quad (3-1)$$

Where,

EOH equivalent operating hours

⁸ Durations of events refer to GT scope only. Steam turbine and generator works (eg removal of ST rotors and casings) can extend critical path of the outages to over 45-50 days.

⁹ GT26 HGP components are classified into 2-cycles and 1-cycle life components. The first ones are rejuvenated at the end of 1st cycle and are re-used. The second ones are retired and scraped at the end of 1st cycle and are replaced by new ones.

¹⁰ It is called C Cycle.

WOH	weighted operating hours
WCE	weighting cycling events

The equation combines engine's ageing, namely operating hours, with severity of operational events such as start-up and shut-down cycles, trips from load etc. The weighting factors are provided by the OEM through relevant tables and classify the severity of operating hours and the cyclic events, according to the firing temperature. The accumulated number of EOHs and OHs drives the frequency and type of maintenance events, for example C Inspection every 32,000 OHs. Table 3-5 lists equivalent and operating hours limits for the maintenance events.

Critical outcome of the above description is the fact that OEM's lifing methodology uses only firing temperature to classify engine's operation and assign accordingly lifing penalty factors. In all the above cases, cooling air system is considered to be in as-design and probably as-new condition. Potential degradation of cooling air system can expose blades and other hot gas path components to significantly higher "felt" temperatures, while actual firing temperature remains constant. In this scenario, OEM's lifing model fails to capture a change in engine's operational characteristics, while this change has considerable effect on HGP components life.

An important clarification for the next steps of the project is the fact that only operating hours are used in the lifing analysis that takes place in the methodology chapter. Equivalent operating hours are derived, as explained, by the inclusion of transient or cyclic events, for example number of start-ups. Since the focus in this work is on steady state conditions and creep only induced lifing risks, operating hours and not equivalent operating hours are considered.

Table 3-4: Typical duration for maintenance inspections

Inspection type	A	B	C
Duration (days)	5 - 7	5 - 7	30 - 35

Table 3-5: Operating Hours and Equivalent Operating Hours limits for gas turbine maintenance inspections

Inspection type	A	B	A	C
Operating hours (OH) limit	8,000	16,000	24,000	32,000
Equivalent Operating Hours (EOH) limit	9,000	18,000	27,000	36,000

3.3 Test Case for Cooling Air CFD Model

As explained in the first chapter, the development of a CFD model for the flow in the area of pre-swirl nozzles is a key objective of the project. The complexity of the flow in this area of the engine makes unsound any other than CFD approach, for example an analytical solution combined with Monte Carlo. Use of high fidelity model is key pre-requisite in the effort to quantify the impact of cooling air system degradation on cooling air temperature. As a consequence, the task of selecting an appropriate Test Case for the validation of the GT26 CFD model becomes also a critical one.

Within a plethora of available research works on pre-swirl nozzles, two important criteria had to be met. Firstly, the geometrical and configuration similarities between the Test Case and GT26. Secondly, the Test Case should ideally involve a study on the impact of increased seal flow inside pre-swirl nozzles cavity, as this is the failure mechanism under investigation for GT26.

As part of the literature review a number of relevant cases were identified. The most relevant were Geis et al. (5), Kakade et al. (7), Meierhofer et al. (1), New (9), Chew et al. (8) and Peng et al. (46). Amongst these cases, Peng et al. (46) and New (9) were the ones that found to be closer to GT26's configuration, mainly due to existence of seals' flows (inner and outer) which they, as it is explained in section 4.3, play significant role in the mass and heat balance of the cavity. Another important geometrical similarity is the fact that Peng's Test rig appears to have the same radius between pre-swirl nozzles outlet and blade cooling passages inlet, this being the case also in GT26. More importantly,

Peng's work (46) is the only one that investigates impact of increasing inlet/inner seal flow to cooling air's effectiveness.

3.3.1 Description

Figure 3-8 outlines the layout of Peng's and New's test rig, while Figure 3-9 focus on details of the cavity itself. Apart from the existence of seals' flows, another attractive similarity with GT26's cavity configuration is the fact that pre-swirl nozzles and blades' cooling holes have the same radius to the engine axis. The same test rig is also being described in Peng's work (46), so a number of geometry, configuration and operational details were collectively available. For the purposes of the CFD model, detailed geometry models have been developed; these models are discussed in paragraph 4.4.2.1. Two important details have to be stated in regards with Test Case rig's instrumentation. The first one is the fact that static pressure P_2 instruments inside the cavity (plane 2, Figure 3-10) have an uncertainty of $\pm 2.5\%$ (9). The second is the fact that total relative temperature $T_{03,rel}$ thermocouples inside blade cooling passages (plane 3, Figure 3-10) had a recovery factor of 0.85 (9). New provides definition for instrument's recovery factor as the characteristic of the thermocouple to measure lower total relative temperature than the real total relative temperature $T_{03,rel}$ inside rotor's receiver holes (plane 3, Figure 3-10). Both these instruments' features are critical for interpretation of any results, and hence are discussed in more details in CFD results paragraph 4.4.2.4.

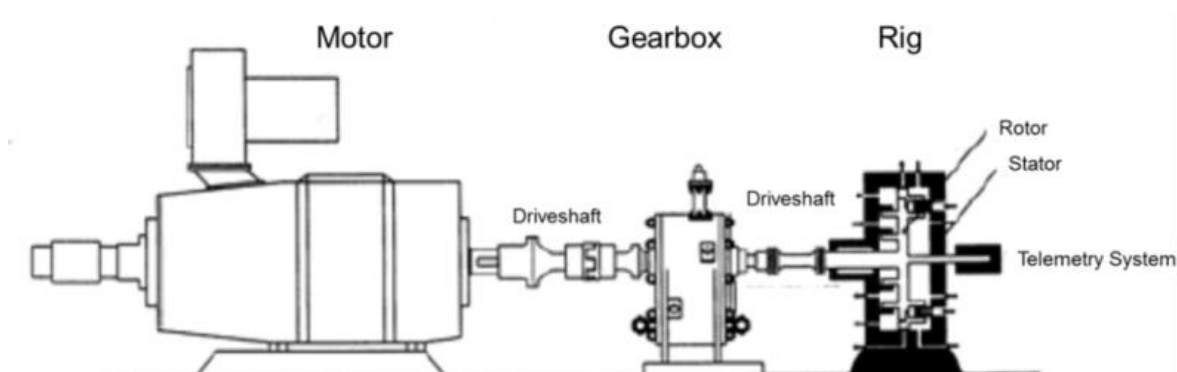


Figure 3-8: Layout of Test Case rig (9)

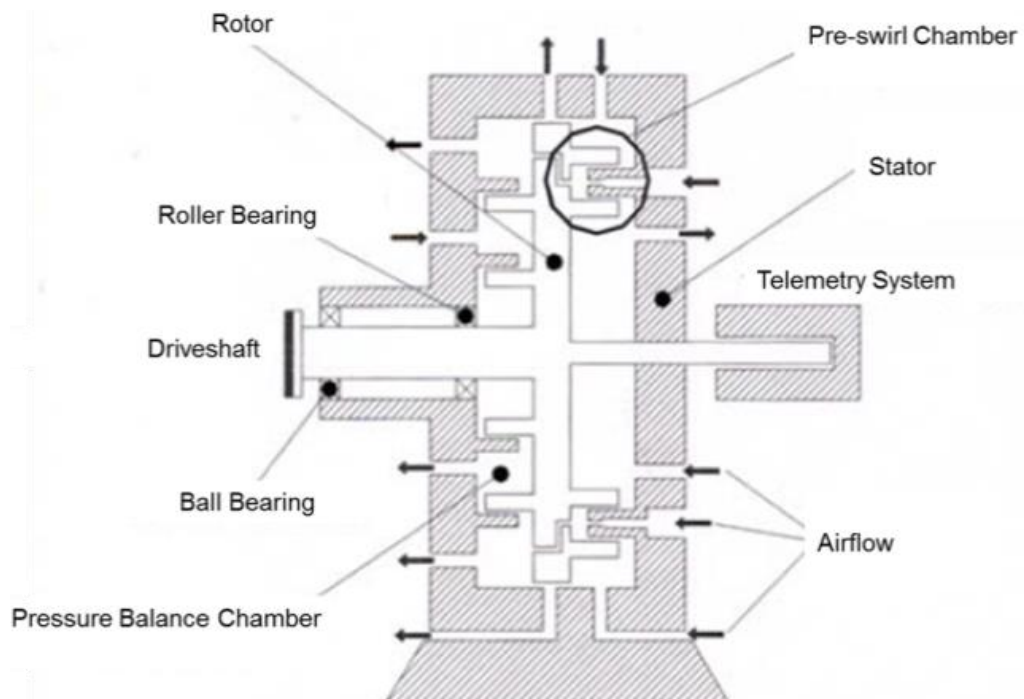


Figure 3-9: Detail of Test Case rig (9)

Table 3-6: Test Case rig specifications (9)

Max Re_ϕ	0.31×10^7
Disc outer rim diameter	450 mm
Max rotational rated speed	13,325 rpm
Pre-swirl nozzle	total 18 in number, 6mm in diameter
Receiver holes	total 72 in number, 5.8mm in diameter
Angle of injection	20° to the plane of rotation
Rotor-stator cavity distance	6.5 mm
Air supply max mass flow	1.132 kgr/sec
Air suppl max temperature	200°C
Air supply max. pressure	4 bara
Air supply min temperature	ambient
Drive motor rating	90 kW
Drive motor max speed	3000 rpm
Gearbox ratio (output:input)	5:1
Gearbox rating	225 kW
Thermocouples	K type

3.3.2 Theory of flow in Test Case

Figure 3-10, Figure 3-11 and Table 3-7 provide the necessary orientation and details for the theory equations that follow. Pre-swirl nozzles are not parallel to the engine axis but angled (20°) to the plane of rotation (Figure 3-11). Starting with the swirl ratio, this is defined as:

$$\beta = V_{PA} \cos \alpha / U \quad (3-2)$$

Pre-swirl effectiveness is ratio of actual velocity V_{PA} as it exits pre-swirl nozzle over ideal velocity V_{Pi} . The equation is:

$$E = V_{PA} / V_{Pi} \quad (3-3)$$

Actual V_{PA} and ideal V_{Pi} velocities are calculated from following equations:

$$V_{PA} = [2c_p(T_{01} - T_{03,rel}) + U^2] / 2 U \cos \alpha \quad (3-4)$$

$$V_{Pi} = M (\gamma R T_{2i})^{0.5} \quad (3-5)$$

Where T_{2i} , M and U from the below ones:

$$T_{2i} = T_{01} (P_2 / P_{01})^{(\gamma-1)/\gamma} \quad (3-6)$$

$$M^2 = [(T_{02} / T_{2i}) - 1] 2 / (\gamma - 1) \quad (3-7)$$

$$U = r \omega \quad (3-8)$$

Table 3-7: Symbols and definitions for Test Case domain

Symbol	Description
T_{01}	Total temperature in the inlet reservoir
T_{02}	Total temperature at the pre-swirl nozzle exit
T_{2i}	Ideal pre-swirl nozzle exit static temperature
$T_{03,rel}$	Total relative temperature at the exit of rotor receiver hole
P_{01}	Total pressure in the inlet reservoir
P_2	Static pressure in the rotor-stator cavity
V_{PA}	Actual exit velocity of pre-swirl nozzle
V_{Pi}	Ideal exit velocity of pre-swirl nozzle
U	Tangential speed of the rotor
M	Mach number
ω	Rotor rotational speed
α	Pre-swirl nozzle angle to the plane of rotation
r	Radius
R	Universal gas constant
c_p	Specific heat capacity
γ	Ratio of specific heats (C_p / C_v)
ν	Kinematic viscosity

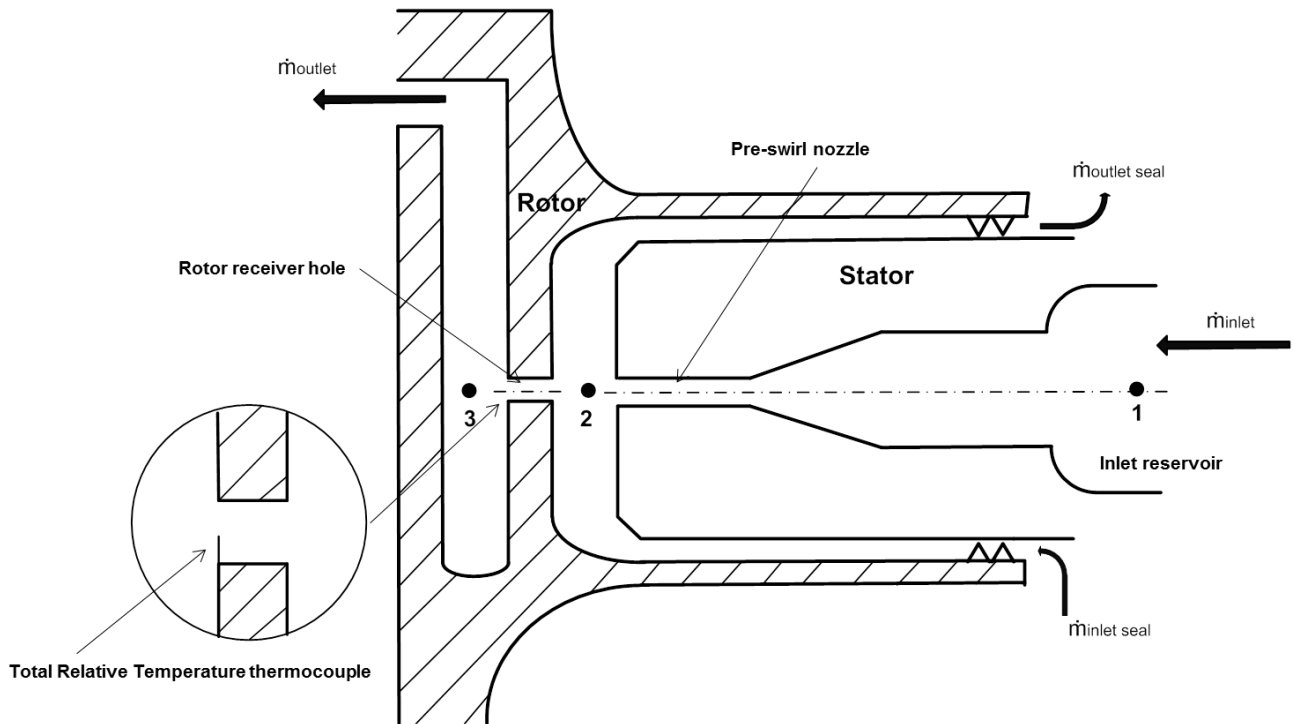


Figure 3-10: Test Case planes of measurement points

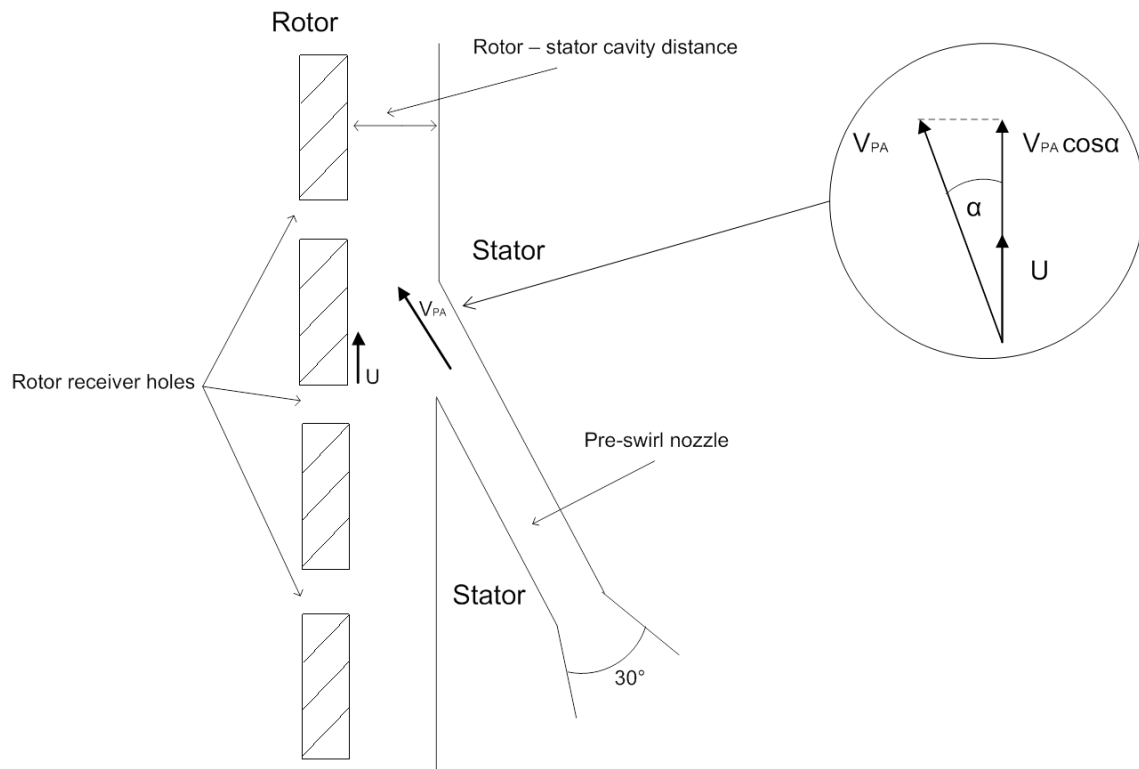


Figure 3-11: Coolant exit velocity from pre-swirl nozzle (46)

3.3.3 Measurements and Experiment Results

The three most important measurements are total temperature T_{01} in point 1, static pressure P_2 in point 2 and total relative temperature $T_{03,rel}$ in point 3 (Figure 3-10). In all cases, mass flows were measured with help of orifice plates designed to BS 1042. Also in all cases air temperature of the flows was the same (46). In some more details:

- Measurement of T_{01} ;

Point 1 is the inlet reservoir where total pressure and temperature measurements are taken. Coolant velocity inside reservoir is quite low ($\sim 1.5\text{m/sec}$) so there is negligible dynamic contribution to temperature and pressure (46).

- Measurement of P_2 ;

Point 2 is inside rotor-stator cavity. In this area static pressure measurements are taken with help of six pressure instruments fitted three higher than pre-swirl nozzle exit and three lower than it. The average of these measurements is used.

- Measurement of $T_{03,rel}$;

Finally, point 3 is in the exit of rotor receiver's holes where total relative temperature measurements are taken with thermocouples fitted inside the rotating holes (Figure 3-10).

Both Peng (46) and New (9) run a series of experiments with the above rig, produced a wealth of results and derived useful conclusions. For the current work, the focus was on the impact of increasing inlet seal flow on pre-swirl effectiveness (equation 3.4). Also, while the rig was capable of running in different rotational speeds (namely: 5000rpm, 7500rpm, 9500rpm, 11000rpm), only results from 5000rpm are cited below. The reason is that GT26 model refers to steady state condition only, with no transient (ramp-up/down) or unsteady simulations. In both below cases, outlet seal flow is constant. The following results from Peng are used for validating CFD model:

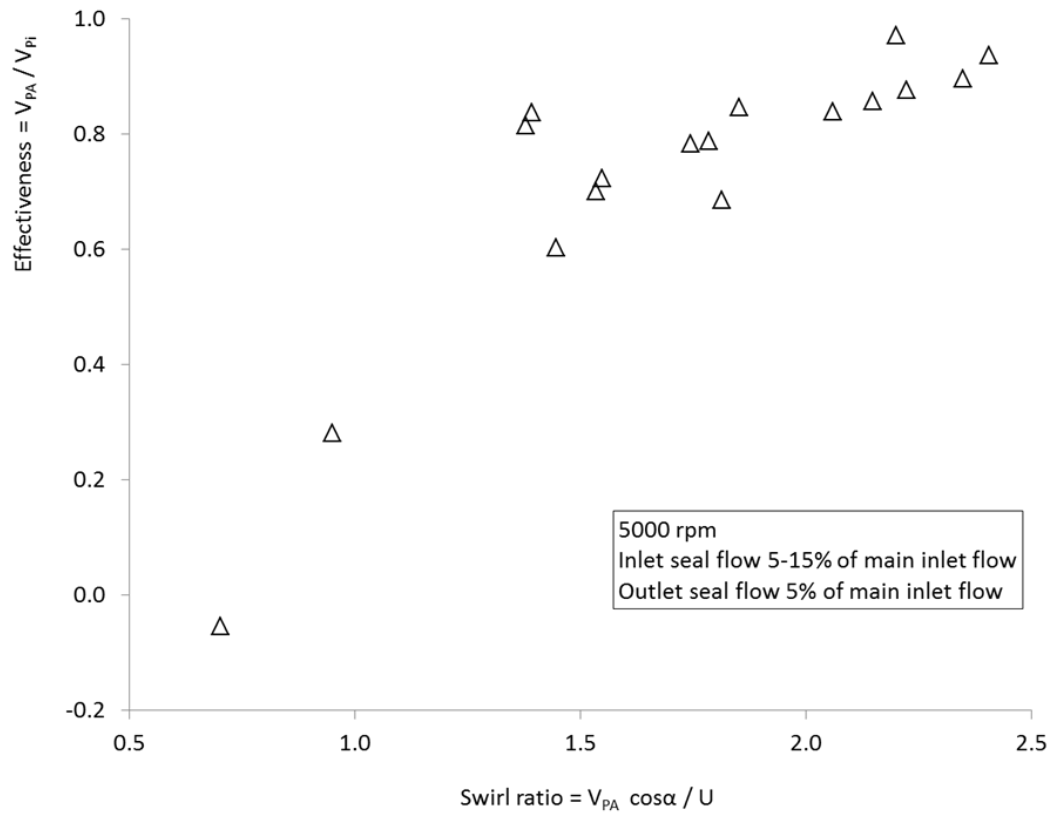


Figure 3-12: Effect of increased swirl ratio on pre-swirl effectiveness (46)

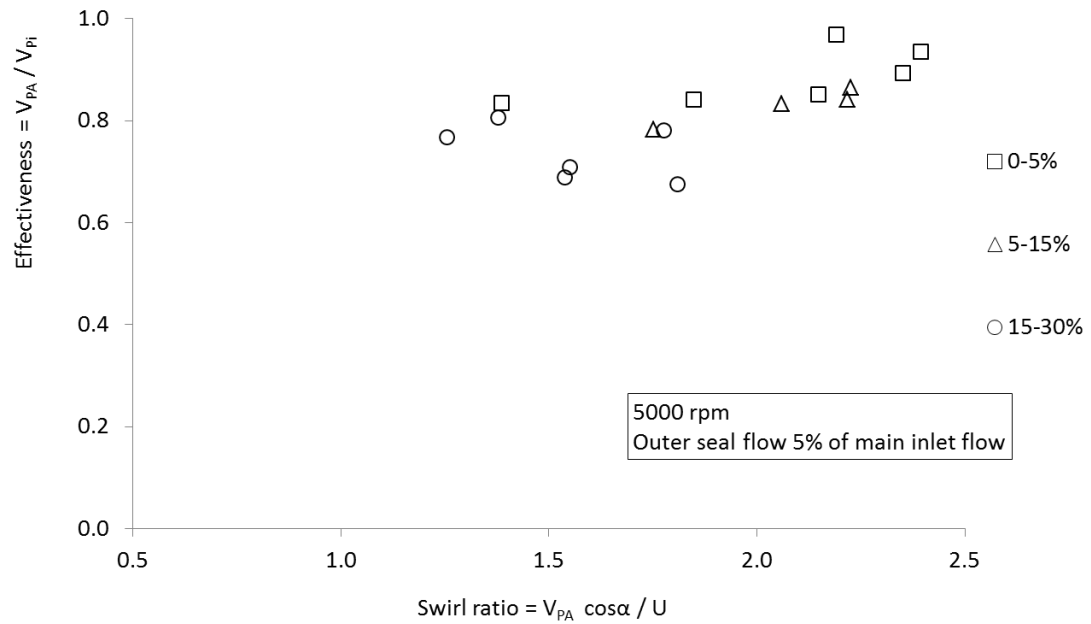


Figure 3-13: Effect of increased inlet seal flow on pre-swirl effectiveness (46)

Two important conclusions are extracted from the results above:

1. Figure 3-12 shows that pre-swirl effectiveness is higher at high swirl ratios. Though GT26 model refer to steady state runs (base load), this is a useful conclusion which implies that for increased load and consequently increased cooling air flow through pre-swirl nozzles, increased cooling air effectiveness can be expected.
2. Figure 3-13 shows that increased inlet seal flow has a negative impact on cooling air pre-swirl effectiveness. This is quite important as this is the basic failure mechanism scenario that has to be explored and confirmed with GT26 model.

3.4 Summary

Technical Description chapter offered the necessary description of the project's key components, the low-pressure-turbine pre-swirl nozzles and 1st stage blade. As explained, this chapter deemed to be an essential part of the thesis, discussing critical input information to the project's methodology that follows.

Starting with the engine of the project, Alstom's GT26 was described in some details. The interested reader can find more details for the engine and its sophisticated cooling air system in Appendices A and B accordingly. Emphasis was given on the description of engine's LPT pre-swirl nozzles, as they are at the heart of the cooling air system analysis that follows. Following cooling air system and pre-swirl nozzles, a brief description of the low-pressure-turbine 1st stage blade was provided, as it is the particular component where the lifing considerations of the following chapters apply. Blade's material was made known and creep life data in the form of a Larson-Miller parameter graph was cited and discussed.

Following the descriptions of components, the maintenance schedule was discussed and the duration of maintenance events was cited, as they contribute in the techno-economic analysis discussion of the last chapter. The description of GT26 was accomplished with engine's lifing model, where the principal

equation and OEM's approach was discussed. An important conclusion is that OEM's lifing equation uses engine's firing temperature as a key parameter for lifing calculations. Cases with degraded cooling air system and increased, as a result, cooling air temperature, are not affecting by any explicit or implicit way the EOHs counting. And this in spite of the fact they have substantial impact on creep life consumption, as it is discussed in the next chapter.

Following GT26, the Test Case's rig was described in detail in section 3.3. The selection of Test Case is important for the validation of GT26's CFD model, so a significant amount of details were offered. Configuration and geometry details as well as operating data were described, together with relevant theory and applying equations. The reasons behind the selection of the specific Test Case were explained. Finally, the experiment's results were discussed. The main conclusions are that pre-swirl effectiveness is sensitive to swirl ratio and to cavity's inlet seal flow. An increase to swirl ratio improves effectiveness. The opposite happens with inlet seal flow, as an increase of this flow brings a decrease in pre-swirl effectiveness. This exact phenomenon needs to be qualitatively and quantitatively assessed in GT26's cooling system with the use of CFD, as it is the starting point for this project's investigation and research.

4 METHODOLOGY

4.1 Introduction

This chapter goes through the details of the project's methodology. It is the longest chapter in this thesis as it covers all the objectives, as those were discussed in section 1.2, and are illustrated again in Figure 4-1. The Techno-economic Analysis due to its importance and weight is capture separately in chapter 5.

Starting point is the development of engine's performance model. The model offers the boundary conditions in hot gas path, but also the impact of derate operation in power output and thermal efficiency. The latter results are used and cited in the Techno-economic Analysis chapter.

An analytical solution of the gas turbine cooling air flow in the area of pre-swirl nozzles cavity is described in section 4.3. The results are used in conjunction with the CFD ones.

The CFD approach is described in section 4.4. ANSYS CFX software was used and all the details of the model (geometry, mesh, solver, results) are discussed in detail. Two models were developed, one for the Test Case and one for the gas turbine.

Since Monte Carlo simulations are used extensively in this research work, section 4.5 goes briefly though it. Principles of monte Carlo are discussed, while the particular algorithms which are used are explained.

Blade cooling is discussed in section 4.6. A detailed analytical model is combined with Monte Carlo and is used in the calculation of blade metal temperature. Blade stress is calculated in section 4.7. Results were again produced through a combination of analytical solution combined with Monte Carlo.

Finally, blade lifing is discussed in section 4.8. The available creep life data for blade's material is used as the starting point. On this graph, the actual operating points are added from the above blade stress and temperature calculations results and the available OEM life. Impact of degraded cooling air system case is explored and the increase of risk is quantified. Methods to manage risk are also discussed.

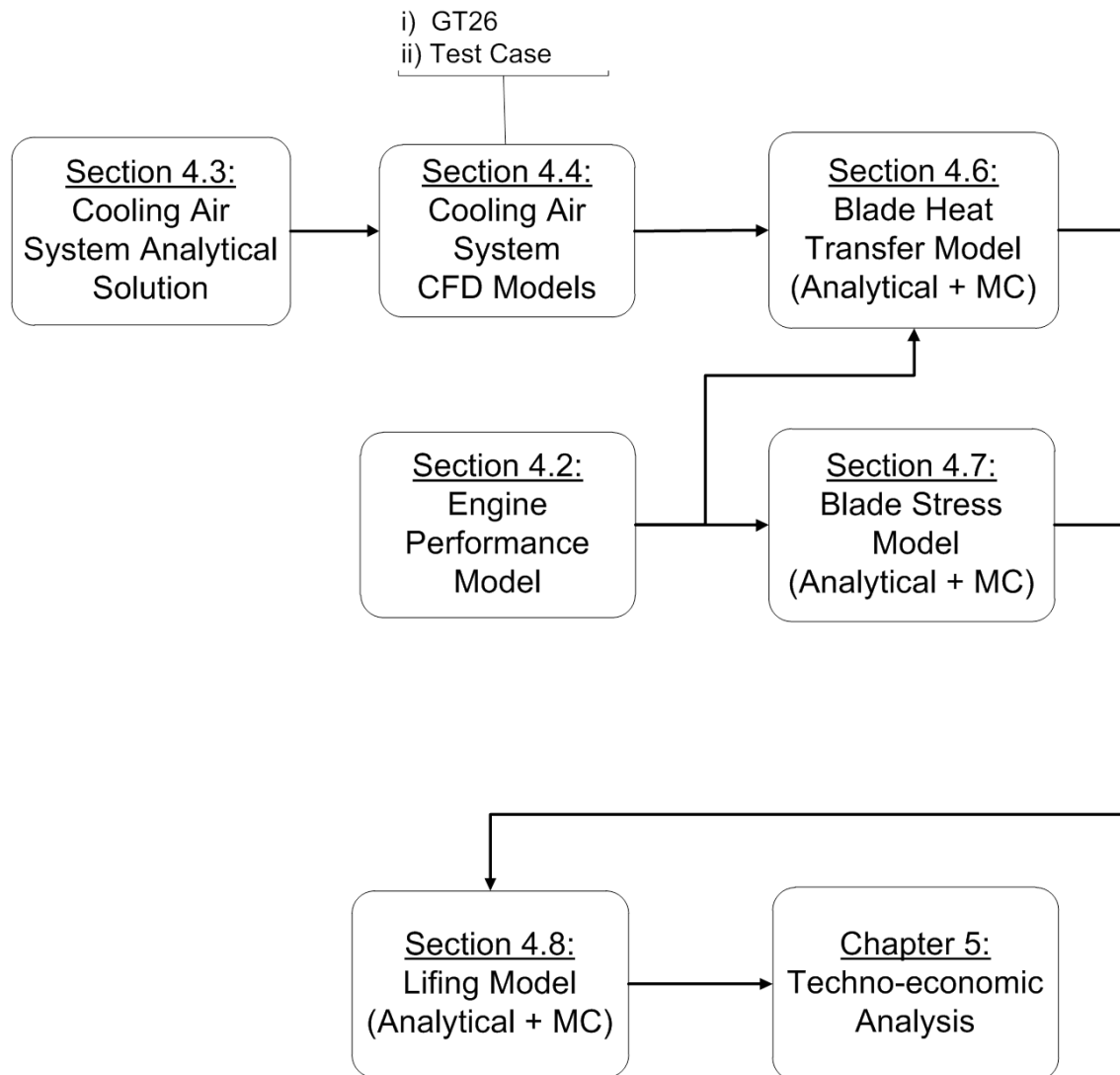


Figure 4-1: Methodology flow chart

4.2 Engine Performance Model

As already mentioned, one of the key objectives for the successful accomplishment of the current research work is the development of a realistic and accurate GT26 performance model. The model is used to offer insight to boundary conditions in hot gas path flow stream. Moreover, the engine derate cases due to cooling air degradation are also explored in regards with their impact on engine's performance characteristics. These results are discussed and cited in the Techno-economic Analysis chapter.

For the development of GT26 performance model, two gas turbine performance modelling tools has been initially used, Cranfield's Turbomatch (47) and the commercially available software GasTurb (48). GasTurb results were found in good agreement with Optigen, which is a performance monitoring tool used for validation. Turbomatch results were significantly diverged from Optigen. Consequently, GasTurb model is the selected model and its results and comments are discussed in the below paragraphs.

4.2.1 Optigen

Optigen is a performance monitoring program. It has been developed and used by the sponsoring company of the current research project. It is an in-house developed software which monitors/captures live operating data from gas turbines' operation, runs calculations and provides performance related output. Figure 4-2 shows a typical Optigen screen.

Optigen's code is written in VBA programming language (49). Live signals from engine's operation, such as fuel flow, compressor discharge temperature and pressure, power output and others, feed as input to Optigen's code. Built-in thermodynamic calculations are executed by the code and performance related data, for example thermal efficiency, is delivered as output. Critical calculations for Optigen's fidelity and validity are firing temperature calculation, which is based on BS ISO2314:2009 (50), and compressor mass flow calculation, which is based on a heat balance method (49). Optigen's validation is achieved

through comparison with gas turbine heat rate (HR) tests results. These tests are repeated on a planned periodicity, for example twice a year.

It is important to be stated that Optigen is a read-only program without ability to allow active engine performance simulations, which can be done by altering input variables for example ambient conditions, fuel flows, mass flows etc. In other words, Optigen cannot be used for parametric runs as opposed to GasTurb or Turbomatch, which they can both run “what if” scenarios. Hence Optigen’s existence and use does not remove the need for a GT26 performance model, which has to be able of running parametric cases.

Moreover, Optigen’s code is not based on a modularised architecture like Turbomatch or based on pre-defined engines’ templates like GasTurb. This means that Optigen offers “black box” values in terms of HPT and LPT output and efficiency and not thermodynamic station results or stage values which GasTurb and Turbomatch offer. Having results of thermodynamic stations is important for understanding conditions in hot gas path area and defining boundary conditions for blade heat transfer and stress models, that are required in the next phases of the project.

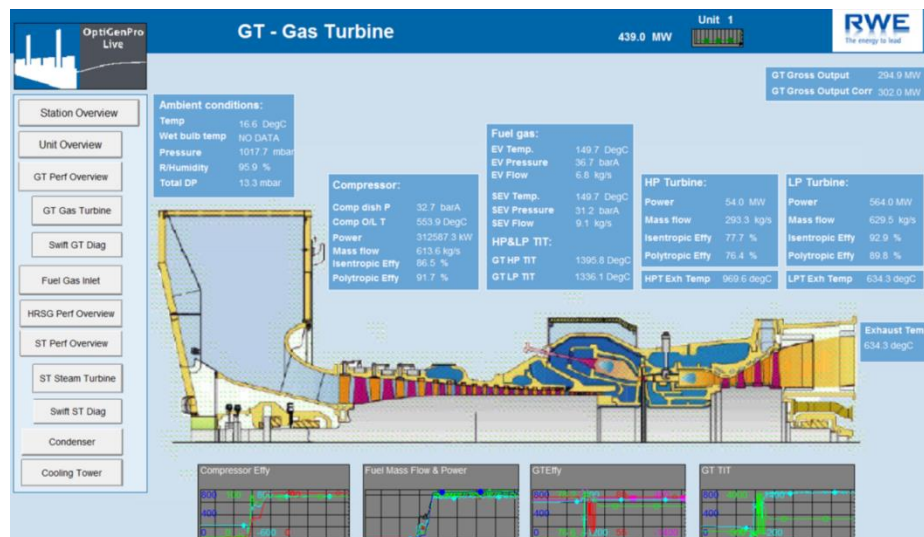


Figure 4-2: Typical Optigen screen¹¹

¹¹ Screen readings purposely skewed.

4.2.2 GasTurb Model

GasTurb is a commercially available software (48) which has been employed to simulate performance of GT26. GasTurb was selected as it is the only commercially available software that offers template for sequential combustion engines. Scarcity of relevant to GT26 performance models in the public domain is not unexpected considering the unique combustion configuration of GT26, which has been already discussed in section 3.2.

Apart from offering single-spool, sequential combustion template, GasTurb provides also a number of useful functions to the user. Visualised engine schematics, T-S, H-S and P-V diagrams, thermodynamic stations results tables and options for parametric analysis are some of the useful GasTurb features.

Figure 4-3 illustrates a GT26 schematic in GasTurb, where sequential combustion configuration is shown. Number of compressor's and turbines' stages which are shown in the schematic are for illustration only, as the actual number of stages were input to model. Also, functions such as recirculating, handling bleed and overboard bleed which are shown in Figure 4-3 are not applicable to current case and were set as 0%. Table 4-1 lists thermodynamic stations results for GT26 model.

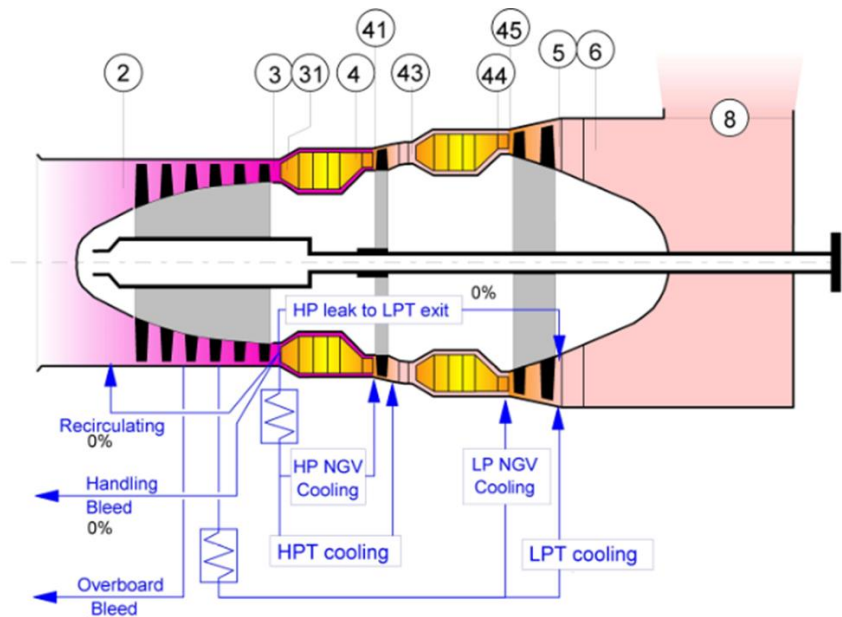


Figure 4-3: GasTurb schematic of sequential combustion template

4.2.3 Results

Table 4-1 below shows a summary comparison between Optigen values and GasTurb model results. The function of compressor bleeds MBH30 and MBH40 is explained in the Technical Description section 3.2 and also with some more details in appendix B. The input values to GasTurb model are marked with green font, while excessive differences (higher than 2%) between Optigen and GasTurb values are marked with red font.

In the majority of the table's rows, the differences between the two models are well within acceptable ranges. As an exception HPT output stands out, being higher in GasTurb by almost 2.8MW. This delta builds-up also the difference in the total GT output, which is also higher in GasTurb by almost 4.2MW. In contrast, GasTurb LPT output is much closer to Optigen with difference being only ~1.2MW. As LPT is the focus area of this work, GasTurb results provide confidence that hot gas path conditions inside LPT have been captured sufficiently. Overall GT efficiency is very close between the two models with a delta of 0.09% only.

An important point that has to be clarified is the fact that only the gas turbine was modelled with GasTurb and not the whole GT-ST unit. Hence the results below refer to gas turbine output and efficiency only, with typical figures lying in the range of ~290MW for power output and of ~39% for thermal efficiency. However, the Technoeconomic Analysis of chapter 5 uses unit's performance characteristics, namely the combined gas turbine and steam turbine configuration, which has been described in section 3.2. In this case, the expected operational figures are in the ranges of ~450MW for power output and of ~60% for thermal efficiency, since steam turbine's contribution is also accounted. Following the parametric runs results, in which output and thermal efficiency are mapped against gas turbine's firing temperature, the respective deltas in GT performance were applied to unit's performance. This is an assumption which has to be made for simplification purposes. It is a safe assumption, as steam turbines typically have indifferent behaviour in gas turbine firing temperature changes up to 40K.

As a summary, GasTurb model is the closest to GT26 actual performance from any other available relevant tools and software. Overall performance characteristics of the engine are reflected sufficiently and realistically. There are small differences in the HPT output and consequently in the overall GT output, but they do not impact on the purposes of this model within the context of the current research work. The unique configuration of sequential combustion and the complex heat and mass balance between the two combustors render the task of modelling GT26 a quite challenging one.

Table 4-1: GasTurb and Optigen results comparison

No	Description	Units	Optigen	GasTurb	Optigen - GasTurb	
			Results	Input / results	Value	%
1	Ambient Temp	°C	15.1	15.0	0.1	0.37%
2		K	288.2	288.2	0.1	0.02%
3	Relative Humidity	%	67.2	67.2	0.0	0.07%
4	GT Output	MW	280.321	284.516	-4.2	-1.50%
5	GT Efficiency	%	39.29	39.38	-0.1	-0.23%
6	Flow	kg/s	609.078	609.632	-0.554	-0.09%
Compressor	Compressor work	MW	327.368	324.729	2.6	0.81%
	Isentropic Effi	%	85.99	86.00	0.0	-0.01%
9	Flow	kg/s	145.753	146.160	-0.407	-0.28%
10	Temp before OTC	°C	458.0	460.0	-2.0	-0.43%
Bleed MBH30 (16th stage)		K	731.2	733.2	-2.0	-0.27%
	Temp after OTC	°C	369.4	370.0	-0.6	-0.16%
13		K	642.6	643.2	-0.6	-0.09%
14	Pressure	barg	21.96	22.22	-0.26	-1.16%
15	Flow	kg/s	50.091	50.547	-0.456	-0.91%
16	Temp before OTC	°C	547.5	553.2	-5.7	-1.04%
Bleed MBH40 (22nd stage)		K	820.7	826.4	-5.7	-0.70%
	Temp after OTC	°C	340.0	342.9	-2.8	-0.84%
19		K	613.2	616.0	-2.8	-0.46%
20	Pressure	barg	31.93	32.42	-0.5	-1.54%
EV Combustor	Efficiency	%	99.82	99.82	0.0	0.00%
	Fuel Flow	kg/s	6.656	6.690	-0.034	-0.51%
SEV Combustor	Efficiency	%	99.82	99.82	0.0	0.00%
	Fuel Flow	kg/s	8.228	8.173	0.055	0.67%
Total Fuel Flow	EV+SEV Fuel Flow	kg/s	14.884	14.863	0.021	0.14%
HP Turbine	Power	MW	77.640	80.456	-2.816	-3.63%
	Isentropic Effi	%	76.39	76.00	0.4	0.51%
28	Power	MW	530.049	528.789	1.260	0.24%
29	Isentropic Effi	%	91.15	91.21	-0.06	-0.07%
LP Turbine	St45 Total Temp	°C	N/A	1509.9	#VALUE!	#VALUE!
		K	N/A	1783.0	#VALUE!	#VALUE!
32	St45 Velocity	m/sec	N/A	147.3	#VALUE!	#VALUE!
33	St Density	kg/m3	N/A	3.1	#VALUE!	#VALUE!
34	Mass Flow	kg/s	623.965	624.401	-0.436	-0.07%

4.3 Analytical Solution for Gas Turbine Cooling Air System

An analytical solution of flow conditions (pressure, temperature, mass flows and velocities) upstream and downstream of GT26 low-pressure-turbine (LPT) pre-swirl nozzles follows in below paragraphs. Equations from energy and mass conservation and compressible flow theory are used to develop the solution. The dimensional characteristics of pre-swirl nozzles and the operating conditions of the area upstream to the nozzles were discussed in paragraph 3.2.1 and they are used as input to the analysis which follows below.

4.3.1 Calculation of Pre-swirl Nozzles Velocities

The upstream operating conditions ($\dot{m}_1$, $T_{t,1}$, $P_{t,1}$) are known from paragraph 3.2.1, so the objective in this chapter is the analysis of the flow characteristics through the nozzles and inside the cavity area downstream of them. Downstream cavity is formed by rotating and stationary walls. It has incoming flows through pre-swirl nozzles and inlet seals and outgoing ones through blade cooling passages and outlet seals (Figure 4-4). For the purposes of this analysis, the flow is considered steady and no heat transfer phenomena are taken into consideration between fluid and walls of cavity and nozzles. Total temperature $T_{t,1}$, total pressure $P_{t,1}$ and total mass flow $\dot{m}_1$ in the area upstream of pre-swirl nozzles (Figure 4-4) are known from Optigen, which provides operation data. Also, engine's technical description in paragraph 3.2.1 provides total number N_{PSNs} and diameter D_{PSN} of LPT pre-swirl nozzles.

Mass flow per nozzle is calculated from equation:

$$\dot{m}_{PSN} = \dot{m}_1 / N_{PSNs} \quad (4-1)$$

Mass flow per nozzle $\dot{m}_{PSN}$, total temperature $T_{t,1}$, total pressure $P_{t,1}$ and nozzle's diameter D_{PSN} are known. Flow function Q is calculated from equation, ref. (51):

$$Q = \dot{m}_{PSN} T_{t,1}^{0.5} / (A_{PSN} P_{t,1}) \quad (4-2)$$

$$A_{PSN} = \pi D_{PSN}^2 / 4 \quad (4-3)$$

Q tables give Mach number Ma inside pre-swirl nozzles. Static pressure $P_{s,2}$ and static temperature $T_{s,2}$ at the exit of pre-swirl nozzles are calculated from the below formulae, ref. (52), (53):

$$P_{t,1} = P_{s,2} \left(1 + \frac{\gamma - 1}{2} Ma^2\right)^{\gamma / \gamma - 1} \quad (4-4)$$

$$T_{t,1} = T_{s,2} \left(1 + \frac{\gamma - 1}{2} Ma^2\right) \quad (4-5)$$

Static pressure $P_{s,2}$ and static temperature $T_{s,2}$ have been calculated above. Cooling air's density $\rho_{s,2}$ and velocity $V_{PSN,out}$ at outlet of pre-swirl nozzles are calculated from equations, ref. (52):

$$\rho_{s,2} = P_{s,2} / (R T_{s,2}) \quad (4-6)$$

$$V_{PSN,out} = \dot{m}_{PSN} / (\rho_{s,2} A_{PSN}) \quad (4-7)$$

It is also know from technical description, paragraph 3.2.1, that GT26 LPT pre-swirl nozzles are angled in two directions, α_1 to y-z plane and α_2 to x-z plane. Figure 4-5 illustrates the vector of $V_{PSN,out}$ velocity and its analysis to axial $V_{P,ax}$, tangential $V_{P,tn}$ and radial $V_{P,r}$ vectors. As the inlet velocity to pre-swirl nozzle is of no interest any more, outlet pre-swirl nozzle velocity will be symbolised as V_P in the interest of brevity. Velocity V_P is the equivalent of experiment's velocity V_{PA} in paragraph 3.3.2 (Figure 3-11).

Axial $V_{P,ax}$, tangential $V_{P,tn}$ and radial $V_{P,r}$ velocities are calculated with use of trigonometry:

$$V_{P'} = V_P \cos\alpha_2 \quad (4-8)$$

$$V_{P,r} = V_P \sin\alpha_2 \quad (4-9)$$

$$V_{P,ax} = V_{P'} \cos(90-\alpha_1) \quad (4-10)$$

$$V_{P,tn} = V_{P'} \cos\alpha_1 \quad (4-11)$$

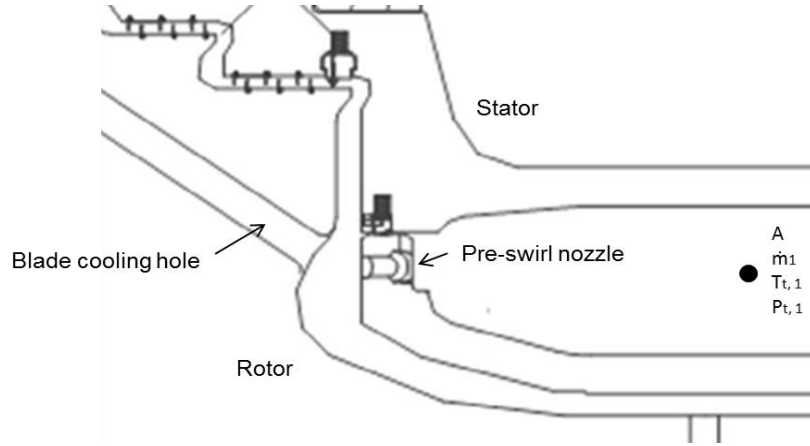


Figure 4-4: Cooling air flow conditions upstream of pre-swirl nozzles

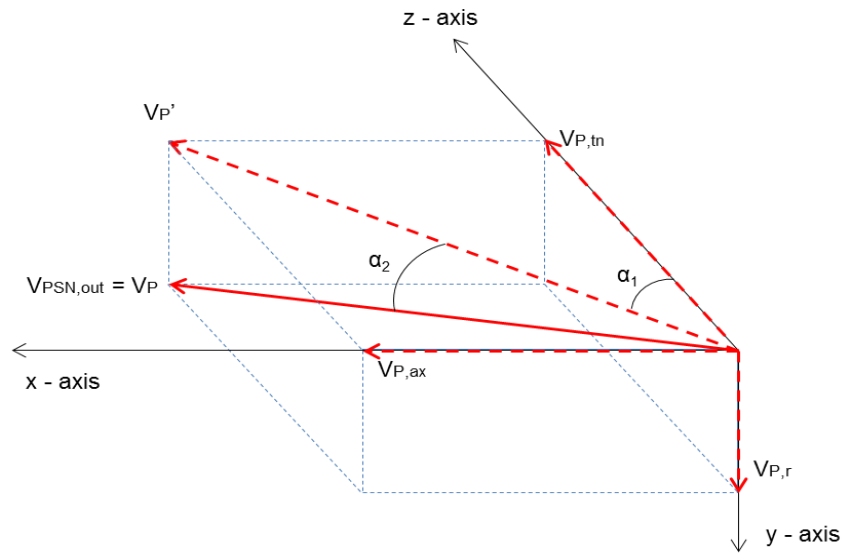


Figure 4-5: Velocity V_P at the exit of pre-swirl nozzle

4.3.2 Calculation of Non-dimensional Numbers

The velocities, mass flow, pressure and density of cooling air upstream and downstream of LPT pre-swirl nozzles have been calculated in the above paragraph. Using these results, critical non-dimensional numbers, as those were identified in section 2.2, are calculated in this paragraph.

More specifically:

- **swirl ratio β** with the use of equation:

$$\beta = V_{P,tn} / (\omega r_{PSN}) \quad (4-12)$$

Where,

$V_{P,tn}$ cooling air tangential velocity at pre-swirl nozzle exit
(equation 4-11)

ω angular velocity of rotor

r_{PSN} radius of pre-swirl nozzle exit bore-center to turbine
axis

▪ **rotational Reynolds number Re_ϕ :**

$$Re_\phi = \rho_{s,2} \Omega r_{PSN}^2 / \mu \quad (4-13)$$

Where,

$\rho_{s,2}$ static density at outlet of pre-swirl nozzles

ω angular velocity of rotor

r_{PSN} radius of pre-swirl nozzle exit bore-center to turbine
axis

μ dynamic viscosity (for $T_{s,2}$)

▪ **non-dimensional mass flow rate c_w :**

$$c_w = \dot{m}_{PSN} / (\mu r_{PSN}) \quad (4-14)$$

Where,

μ dynamic viscosity (for $T_{s,2}$)

r_{PSN} radius of pre-swirl nozzle exit bore-center to turbine
axis

▪ **turbulent flow parameter λ_T with use of equation:**

$$\lambda_T = c_w / Re_\phi^{0.8} \quad (4-15)$$

Where,

c_w non-dimensional mass flow rate

Re_{ϕ} rotational Reynolds number

Results for the above non-dimensional numbers were produced also for other relevant cases, which had been identified as similar to GT26 and cited in section 3.3. Table 4-2 below lists three of the most relevant cases, together with GT26 and selected Test Case results. In the critical non-dimensional number of swirl ratio β , the comparison confirms that GT26 and the relevant cases operate at the same range.

Table 4-2: Comparison table of non-dimensional factors for Alexiou et al. (54), Wilson et al. (55), Chew et al. (56), Peng et al. (46) and GT26 at base-load operation, ISO conditions

Cases	C_w	Re_{ϕ}	λ_{τ}	β
Alexiou	Not calculated	Not calculated	Not calculated	$0.5 < \beta < 3$
Wilson	$5 \cdot 10^3 < C_w < 2 \cdot 10^4$	$6 \cdot 10^5 < Re_{\phi} < 2.4 \cdot 10^6$	$0.03 < \lambda_{\tau} < 0.06$	$0.5 < \beta < 3.7$
Chew	$1.2 \cdot 10^4 < C_w < 9.5 \cdot 10^4$	$1.22 \cdot 10^6 < Re_{\phi} < 5.8 \cdot 10^6$	Not calculated	$0.6 < \beta < 2.5$
Peng	Not calculated	$0.31 \cdot 10^7$	Not calculated	$0.5 < \beta < 2.5$
GT26 (@base load)	$3.94 \cdot 10^4$	$4.92 \cdot 10^7$	0.0276	1.215

4.3.3 Cavity Mixing Flows

Calculation of flows mixing inside the cavity downstream of pre-swirl nozzles was performed with a methodology developed by Chew et al. (8) . In a brief summary, the cavity from Figure 4-4 and Figure 4-6 was converted to a simple model, which is illustrated in Figure 4-7.

Chew's method assumes that fully mixed conditions apply inside the cavity and tangential inner seal velocity $v_{innerseal}$ equals half the rotor speed at inner seal radius. In addition, GT26 mass flow through seals $\dot{m}_{innerseal}$ should be less than 2% of the total flow through pre-swirl nozzles at design conditions, according to OEM.

Following Chew's method, below equations apply:

$$V_{mix,tn} = \frac{\dot{m}_{PSN} N_{PSN} V_{P,tn} + \dot{m}_{innerseal} V_{innerseal}}{\dot{m}_{PSN} N_{PSN} + \dot{m}_{innerseal}} \quad (4-16)$$

And

$$T_{t,inlet} = \frac{\dot{m}_{PSNs} T_{t,PSN} + \dot{m}_{innerseal} T_{t,innerseal}}{\dot{m}_{PSNs} + \dot{m}_{innerseal}} \quad (4-17)$$

Where,

$V_{mix,tn}$	cooling air tangential velocity after mixing (Figure 4-7)
$V_{P,tn}$	cooling air tangential velocity at exit of pre-swirl nozzle (Figure 4-5, Figure 4-7, equation 4-11)
$\dot{m}_{PSN}$	mass flow through pre-swirl nozzle (equation 4-1)
N_{PSN}	number of pre-swirl nozzles
$\dot{m}_{PSNs}$	total mass flow through pre-swirl nozzles
$\dot{m}_{innerseal}$	mass flow through inner seals (Figure 4-7)
$V_{innerseal}$	cooling air exit tangential velocity from inlet seal (Figure 4-7)
$T_{t,inlet}$	averaged cooling air total temperature in the cavity after mixing (Figure 4-7)
$T_{t,PSN}$	cooling air total temperature at exit of pre-swirl nozzles (Figure 4-7)
$T_{t,innerseal}$	cooling air total temperature at exit of inner seal (Figure 4-7)

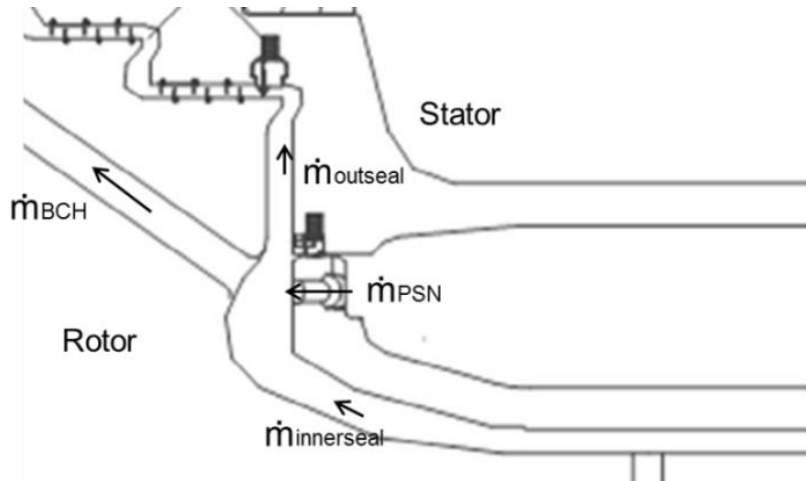


Figure 4-6: Incoming and outcoming flows in pre-swirl nozzles cavity

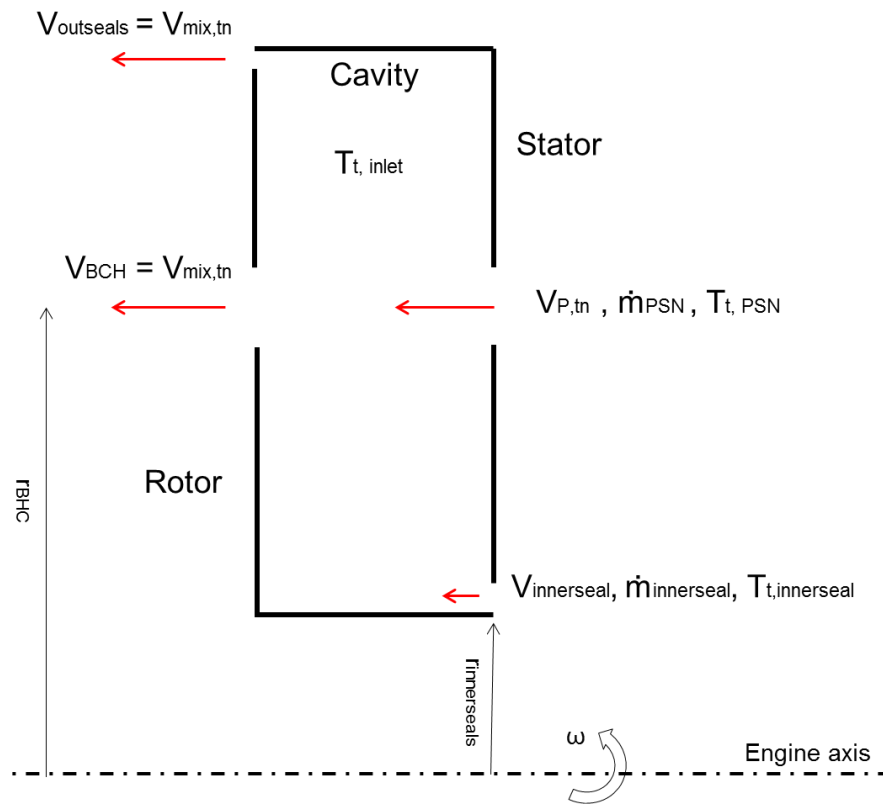


Figure 4-7: Simplified model of pre-swirl nozzles cavity

4.3.4 Decrease of Cooling Air Temperature Inside Blade Cooling Passage

Appendix C goes through the derivation of stationary and rotational reference frame theory equations (57), (58). Figure C-1 from Appendix C can be applied to GT26 pre-swirl nozzles cavity and produce Figure 4-8. Also, combining equations 4.16, 4.17 and A-4, the below equation is derived:

$$T_{t,inlet} - T_{t, rel, BCH} = (2 U V_{mix,tn} - U^2) / 2C_P \quad (4-18)$$

Where,

$T_{t,inlet}$ averaged total temperature in the stationary frame after mixing (equation 4-17)

$T_{t, rel, BCH}$ total relative cooling air temperature inside blade cooling passage (Figure 4-8)

$V_{mix, tn}$ averaged tangential velocity in the stationary frame after mixing (equation 4-16)

U passage wall velocity (blade speed), $U = \omega r_{BCH}$

where,

ω engine rotational speed

r_{BCH} radius of blade cooling passage center to engine axis

Some critical assumptions for the above equations are:

- total temperature remains constant upstream and downstream of pre-swirl nozzles. This assumption was made also for the flow through Test Case's pre-swirl nozzle.
- Total temperature remains constant upstream and downstream of inlet seals.
- In GT26 configuration, pre-swirl nozzles and inlet seals are supplied from common cooling air source (MBH30), so it can be said that

$$T_{t, PSN} = T_{t, innerseal}$$

As it is always preferable to work with dimensionless properties, equation 4-18 is re-arranged and dimensionless ΔT is defined as:

$$\text{Dimensionless } \Delta T = \frac{2 C_p (T_{t,\text{inlet}} - T_{t,\text{rel,BCH}})}{U^2} = \frac{2 V_{\text{mix,tn}}}{U} - 1 \quad (4-19)$$

Swirl ratio for blade cooling passage can be calculated from equations 4-12 and 4-16:

$$\text{Swirl ratio} = V_{\text{mix,tn}} / (\omega r_{\text{BCH}}) \quad (4-20)$$

As only steady state conditions are considered and since GT26 is a single spool, constant rotational speed engine, no cases with variable rotational speed or variable flow through pre-swirl nozzles are explored. Only cases of increased inlet seal flow are explored. From equations 4-16, 4-19 and 4-20, the impact of increased inlet seal flow on dimensionless ΔT and swirl ratio can be plotted. Both dimensionless ΔT and swirl ratio are expressions of the averaged tangential velocity in the stationary frame after mixing $V_{\text{mix, tn}}$ and of the rotational speed. However, and in line with the used references, the unit of interest is the dimensionless ΔT as this is used to calculate the increase of cooling air temperature inside blade cooling passage.

As mentioned above, OEM's guidance for design conditions inlet seal flow is to be lower than 2% of total pre-swirl nozzles flow. Figure 4-9 and Figure 4-10 below illustrate the impact on both dimensionless ΔT and swirl ratio of inlet seal flow experiencing values higher than 2% of total mass flow through pre-swirl nozzles.

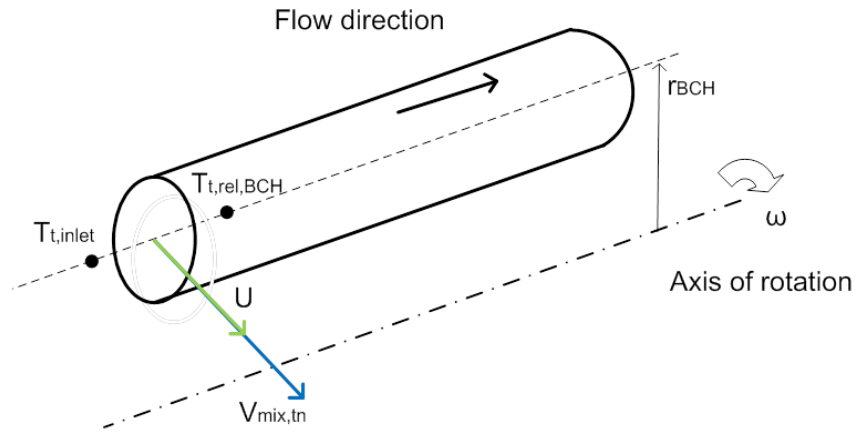


Figure 4-8: Typical rotating passage model

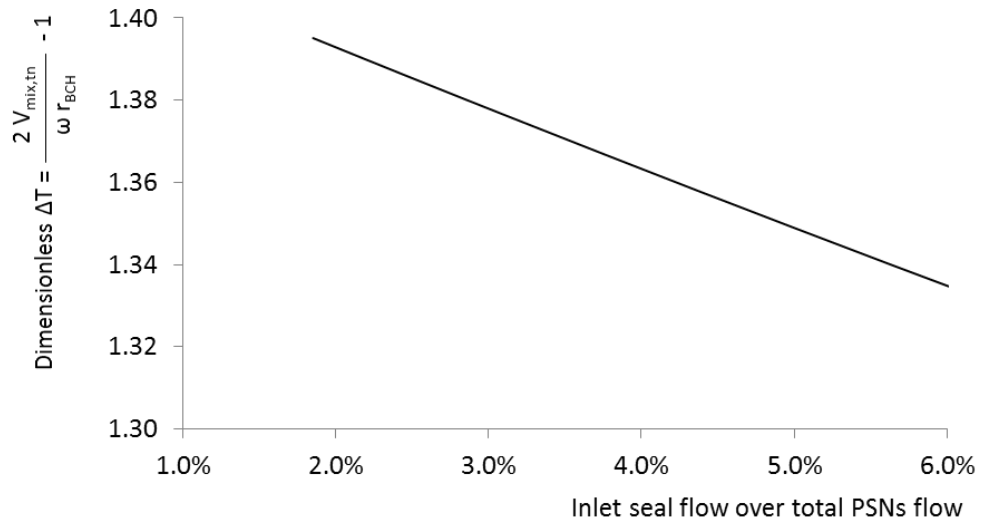


Figure 4-9: Effect of inlet seal flow increase on dimensionless ΔT

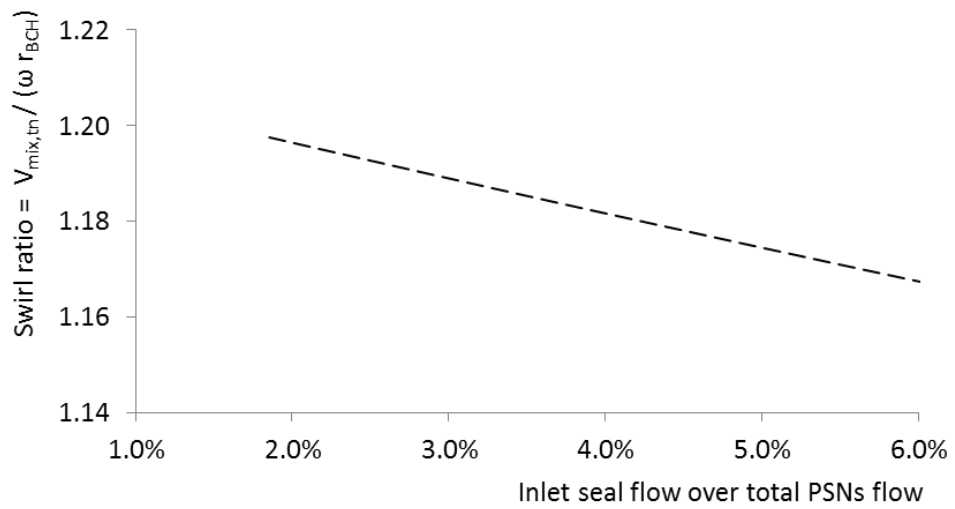


Figure 4-10: Effect of inlet seal flow increase on swirl ratio

4.3.5 Summary of Gas Turbine Cooling Air Analytical Solution

An analytical solution for GT26 low-pressure-turbine pre-swirl nozzles area was developed with use of fundamental equations of compressible flow and energy and mass conservation.

As a starting point, the critical non-dimensional numbers, as those have been identified in literature review, were calculated. Results were compared with Test Case's ones, showing encouraging similarities between conditions in GT26 pre-swirl nozzles cavity and Test Case.

To model the pre-swirl nozzles cavity, Chew's (8) approach was used which assumes isentropic and full-mixture conditions. The drop of total relative temperature $T_{t, \text{rel, BCH}}$ inside blades' cooling passages was calculated in the form of dimensionless ΔT , assuming Chew's basic cavity model.

More importantly, cases with increased inlet seal flow were explored and the effect on dimensional ΔT and swirl ratio has been captured. As observed, increased inlet seal flow has a negative effect on both dimensional ΔT and swirl ratio. In particular, increase of inlet seal flow from design value of 2% up to 6%, incurs a drop in dimensionless ΔT of almost 0.06. The meaning of the drop in ΔT is that the cooling air total relative temperature, as it is delivered inside blade cooling passages, has increased as a result of the increase of inlet seal flow. In other words, the flow through the inner seals inside the cavity has a clearly unwanted effect on cooling air temperature, as it increases it. The value of 0.06 in dimensionless ΔT equates to approximate 3K.

As a closing note, the above-described analytical solution offered two useful conclusions. Firstly, it confirmed the expected polarity of this phenomenon, namely that cooling air temperature increases as inlet seal flow increases. Secondly, it offered an understanding of the order of magnitude of this change; a decrease of almost 0.06 in ΔT is expected when inlet seal flow increases by approximately 4%. These results will be discussed extensively in collation with the CFD ones in section 4.4.

4.4 Computational Solution for Gas Turbine Cooling Air System

As it has been described in project's aim and objectives paragraph, a computational fluid dynamic (CFD) model will be developed to simulate the flow phenomena in the area of GT26 low-pressure-turbine pre-swirl nozzles and their downstream cavity. As the pre-swirl nozzle cavity has a relatively complex geometry, the CFD solution is expected to offer higher fidelity results and a more detailed insight to the flow conditions in the area, as opposed to the simplistic approach of the analytical solution, which was discussed in the previous section. CFD's contribution to the project is crucial as accurate prediction of cooling air temperature, as it is delivered inside LPT blade, is necessary for critical downstream processes of the project, such as blade metal temperature calculation, blade lifing estimation etc.

Leading to that, and as necessary pre-requisite, a CFD model has also to be developed for the selected Test Case from section 3.3. This is normal research process as by this way the CFD approach and methods can be validated and then applied in areas where results from experiments or other sources (for example operational signals) are not available. In this research work, CFD model was firstly developed for the selected Test Case and then for GT26 domain. Computational results from Test Case are compared to experiment's results. Results from GT26 CFD model are compared to results from analytical solution in section 4.3.

ANSYS CFX is the commercial CFD software for this part of research work. Selection of parameters, various settings, results and conclusions are discussed below. As explained, CFD models have been developed for both Test Case and GT26. As the Test Case is the leading task and involves a CFD model that has to be validated, more details on the methodology are included in Test Case's respective paragraphs below. For GT26 model a number of these details are omitted in the interest of brevity, as they are very similar or even identical to Test Case' ones.

CFD simulations were performed with ANSYS CFX, versions R16 and R17. Workbench platform was used, instead of the stand-alone package, as for both models (Test Case and GT26) there were requirements for parametric run studies, which is an area where Workbench offers substantially increased flexibility and straight-forward function.

ANSYS CFX is a finite volume, pressure based, implicit, coupled solver¹². This means that the domain is discretized into a set of control volumes and all general conservation equations (momentum, energy, mass and species) are solved on the set of discrete volumes simultaneously. The impact of this characteristic is reflected in convergence history graphs (Figure 4-22, Figure 4-34), where clearly the number of iterations to reach convergence is relatively low, fewer than 100 iterations, but equally the whole process time was considerably lengthy, between 2 and 3 hours per operational point.

4.4.1 Brief Historical Review of Relevant CFD Works

One of the key objectives of the current research work is to model the flow phenomena in the area of the GT26 LPT pre-swirl nozzles with use of computational flow dynamics. In almost all similar cases of research work in this area, CFD packages have been used in combination with experiments or theoretical study.

In general terms it can be said that there is mostly agreement in the assumptions and various parameters selection among researchers. In all relevant cases, flow in pre-swirl cavity is assumed to be compressible and steady. Reynolds-averaged Navier-Stokes (RANS) partial differential equations are used with a model of turbulence in all cases. K- ϵ appears to be the most common turbulence model. It has been used in the works of Chew et al. (56), Wilson et al. (55), Yan et al. (59), Lewis et al. (60), Karabay et al. (61) and (62), Granovskiy et al. (6), Ciampoli et al. (4) and Farzaneh (63). In an exception, Jarzombek et al. (64) used κ - ω . There is reference in a number of researchers

¹² As opposed to ANSYS Fluent, which is an explicit, velocity based solver.

for using Large Eddy Simulation (LES) models in the near future, but there is no present case (5).

Interesting research works are those from Yan et al. (59) and Karabay et al. (61). Yan used a steady, compressible 3D flow model with the SIMPLEC¹³ pressure-correction scheme solver. Turbulence model was k- ϵ and the domain was one pre-swirl nozzle. Grid's size was approximately one million cells and achieved non-dimensional distance from wall y^+ lower than unity (<1). Results were compared with experimental ones and the agreement was good. Yan investigated the effects of rotational speed, pre-swirl ratio and flow rate on flow through-the-nozzle conditions. In another case, Karabay (61) used steady-state, axisymmetric elliptic flow SIMPLEC solver, again with k- ϵ turbulence model. His grid was smaller in size but due to smaller domain he achieved y^+ values lower than 0.5. In Karabay's case, CFD results were compared with theoretical calculation results and the agreement was satisfactory. His investigation was in relation to pre-swirl ratio and its effect on cooling air temperature and Nusselt number.

Apart from the above briefly described cases, there are some other cases where researchers explore specifically pre-swirl nozzles effect on total relative temperature inside blade cooling passages or simply rotating passages. Conclusions from these cases are discussed separately in some more details in Test Case's results discussion paragraph 4.4.2.5.

Finally, it has to be stated that CFX solver has the capability of running Direct Numerical Simulations (DNS) or Large Eddy Simulations (LES). In both cases, turbulence in flow can be resolved with instantaneous values, as opposed to averaged ones that are resulted from RANS combined with turbulence model(s). Using DNS or LES is always tempting for every flow simulation. However for the current work, a sub-grid detail of the flow that both these methods can offer is not required. As discussed in the results paragraph, the

¹³ Semi-Implicit Method for Pressure Linked Equations-Consistent; CFD algorithm developed by Doormal and Raithby in 1984.

good agreement between RANS and experiment results removes the need for further resolution of the model by inputting much finer grid or using DNS or LES methods. A DNS or LES approach in Test Case's or GT26's domains would probably require 7-10 days running time per operational point with the currently available computational power. A parametric run - for example involving incremental inlet seal flow) with 8 to 10 design points would last for over 3 months. However, when the scope of research is the design optimisation of pre-swirl nozzles or hosting cavity with its seals, as opposed to the current project where aim is to understand flow conditions on a given design, then DNS and LES models are recommended.

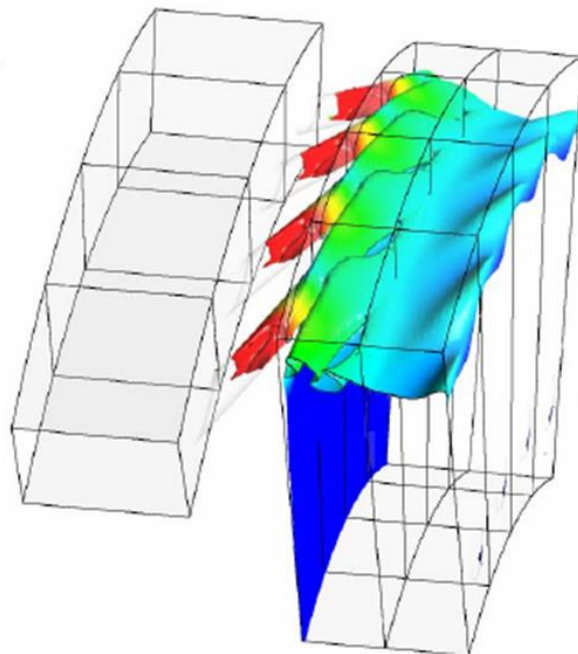


Figure 4-11: Typical CFD model for pre-swirl nozzles (64)

4.4.2 Computational Solution for Test Case

Test Case's experiment (46) which is described in detail in section 3.3, was first to be modelled with the use of CFD for this work. Geometry, meshing, settings and results are discussed in the below paragraphs. Some important definitions and explanatory notes are also included, and some provided in the appendices, as the author considers them useful for explaining the thought and decision-making processes on various steps of the model development.

4.4.2.1 Geometry and Domain

Test Case's geometry was derived from the works of New (9) and Peng (46). It has to be stated that there was no available detailed drawing of Test Case's rig. With use of sketches and technical details from the above two works and through extensive communications with the primary (were possible) authors and co-workers, an accurate re-production of Test Case rig in CAD was achieved. Figure 4-12 and Figure 4-13 show the pre-swirl nozzle sketch that was reproduced, while Table 4-3 lists the main dimensional nozzle characteristics. With the below 2D sketches, a 3D CAD model was developed with the use of Siemens NX CAD package. Next step was to import NX model in ANSYS DesignModeler and convert it into a fluid domain. As a full 360° model of the rig would have been heavy and time-consuming for the solver, minimum possible number of pre-swirl nozzles and blade cooling holes, one and four respectively, were selected to form the fluid domain, maintaining of course the necessary rotational periodicity of the model. Figure 4-14 depicts final version of fluid domain in ANSYS DesignModeler.

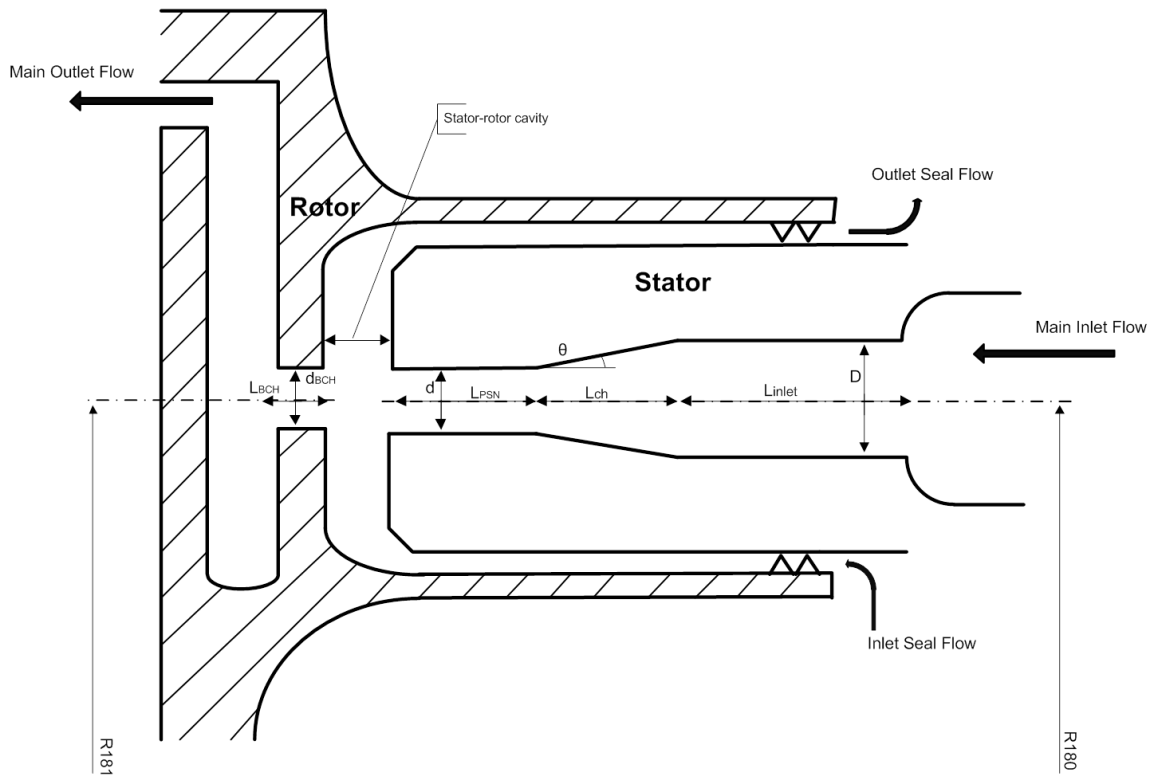


Figure 4-12: Test Case pre-swirl nozzle and cavity sketch

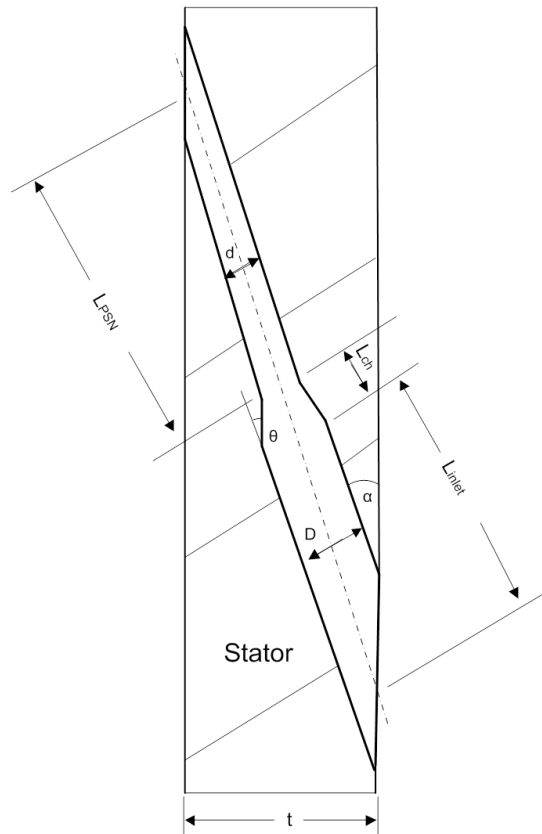


Figure 4-13: Test Case pre-swirl nozzle geometrical characteristics

Table 4-3: Test Case pre-swirl nozzle description of nomenclature

Nomeclature	Description	Origin
t	Pre-swirl plate thickness	Scaled from sketches in references
D	Inlet nozzle diameter	Scaled from sketches in references
d	Pre-swirl nozzle diameter	Provided by references
dBCH	Blade cooling hole diameter	Provided by references
L_{inlet} / L_{PSN}	Inlet nozzle over PSN nozzle lenghts ratio	Scaled from sketches in references
L_{inlet}	Inlet nozzle length	Calculated
L_{PSN}	Pre-swirl nozzle length	Calculated
L_{ch}	Chamfer length	Calculated
L_{BCH}	Blade cooling passage length	Provided by references
θ	Chamfer angle	Provided by references
α	Pre-swirl nozzle angle to the plane of rotation	Provided by references

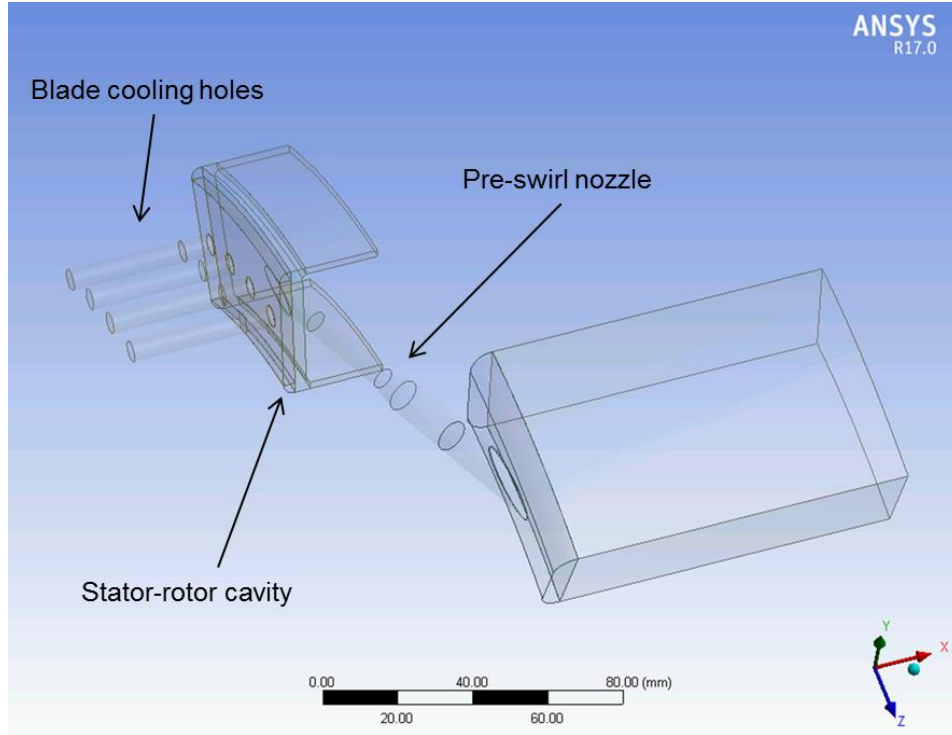


Figure 4-14: Test Case fluid domain in CFD

4.4.2.2 Mesh

As generation of an appropriate mesh (grid) is the most important part in any CFD model (65), (66), this paragraph goes into a number of details, while also Appendix D makes special reference to critical mesh characteristics that define the quality of the mesh.

ANSYS mesher was the tool that was used for meshing the Test Case and GT26 domains. A strategy of hybrid mesh was selected. A fully structured mesh was not practical, due to complexities in geometry and primarily the existence of acute angles between the pre-swirl nozzle and upstream and downstream domains. Consequently, structured mesh (hexagons) was applied with method of *sweep* in the areas of inlet domain (Figure 4-16), inlet seal and outlet seal. Unstructured mesh (prisms and tetrahedrals) was applied in the blade cooling passages, in the cavity and in the areas where pre-swirl nozzle meets inlet and cavity domains. Unstructured mesh was developed with *automatic method*, which is based on the Delaunay triangulation algorithm.

Use of Shear-stress-transport (SST) model, which is discussed extensively in paragraph 4.4.2.3.2, relaxed requirement for non-dimensional distance from wall y^+ values close to unity (65). Instead, values of 10-100 were used with sufficient capturing of boundary layers flow particularities as it is discussed below. To achieve these values, 10 inflation layers were used in all walls with first layer having a height of 0.05mm, while remaining inflation layers had a growth rate of 1.2. Figure 4-17 shows details of inflation layers in a blade cooling passage, while Figure 4-24 in the results paragraph illustrates the achieved y^+ values in the domain. Also *proximity and curvature function* was used which allowed mesher to automatically increase number of cells (thus reducing their size) in domain areas of increased curvature, namely close to angles, areas of changes of diameters etc. A typical representation of such a domain area where mesh becomes significantly finer is the area close to and around the angle between blade cooling passages and rotor wall is shown in Figure 4-15.

In terms of mesh quality characteristics, ANSYS mesher uses a number of different mesh quality metrics. The two most important ones are the *skewness* and the *orthogonal quality*. Appendix D provides detailed description of the two metrics and their equations. The average skewness value for the Test Case mesh was 0.20661, in a scale from 0 (perfect) to 10 (poor). This value is considered “excellent” in ANSYS scale (65). Figure 4-18 shows mesh elements distribution according to their skewness values. The average orthogonal quality value for the test case was 0.89901, in a scale from 0 (poor) to 1 (perfect). This value is considered “very good” in ANSYS relevant scale (65). Figure 4-19 shows mesh elements distribution as per their OQ values.

Apart from mesher’s quality metrics, which are *skewness* and *orthogonal quality*, ANSYS solver’s uses also its own quality criteria for the imported mesh. This is explained by the fact that CFX can import mesh files from different mesh generation algorithms (for example ICEM) so maintaining a separate independent mesh quality assessment process is important for the solver. CFX’s mesh quality metrics are the *orthogonal angle*, the *expansion factor* and

the *aspect ratio*. These metrics are also described in detail together with their relevant thresholds in Appendix D. Table 4-4 lists the results of Test Case mesh quality characteristics. It has to be clarified that the terms rotor and stator were used for simplifying the set-up process and their referring subdomains are shown in Figure 4-21. There are areas in rotor's mesh with poor values of *expansion factor*. As they are less than 1% of the total number of rotor elements, no further action is required (65).

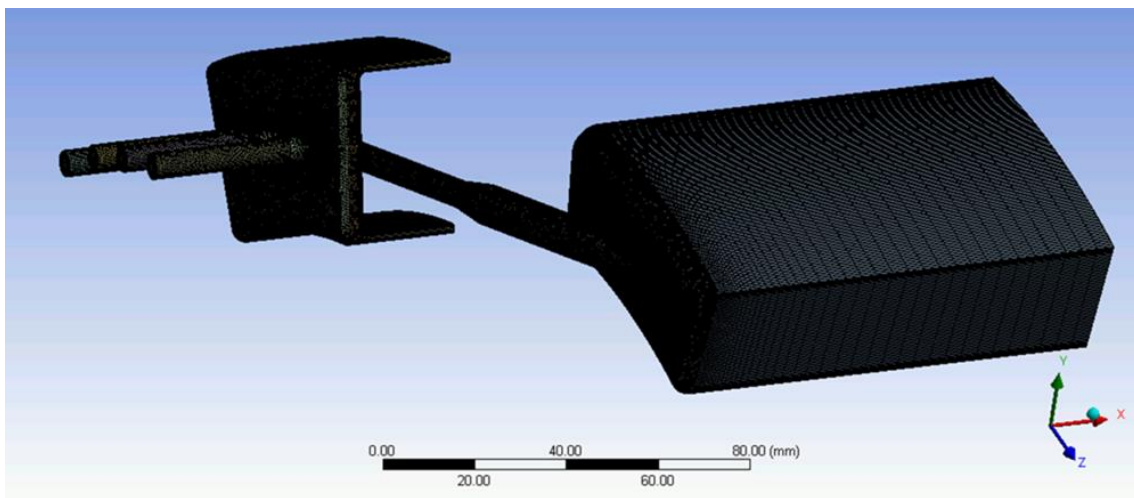


Figure 4-15: Test Case mesh

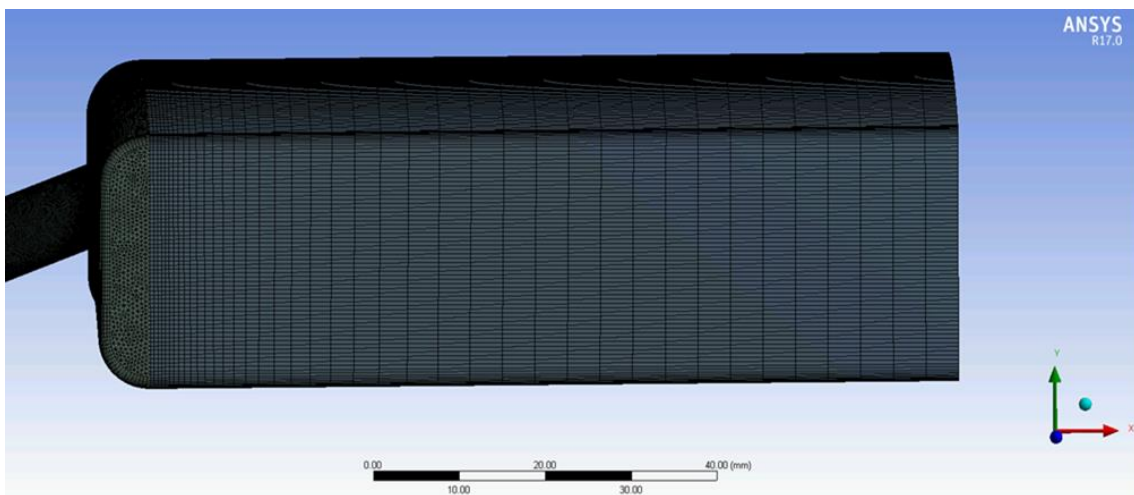


Figure 4-16: Detail of structured mesh at Test Case inlet domain

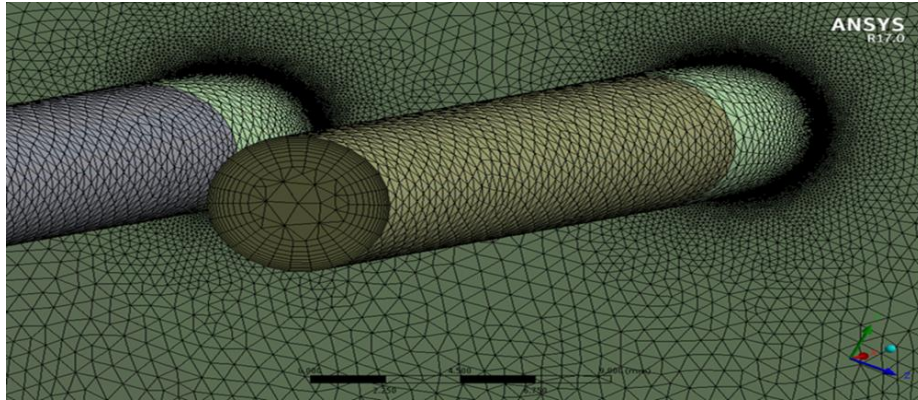


Figure 4-17: Detail of mesh at Test Case blade cooling passage outlet

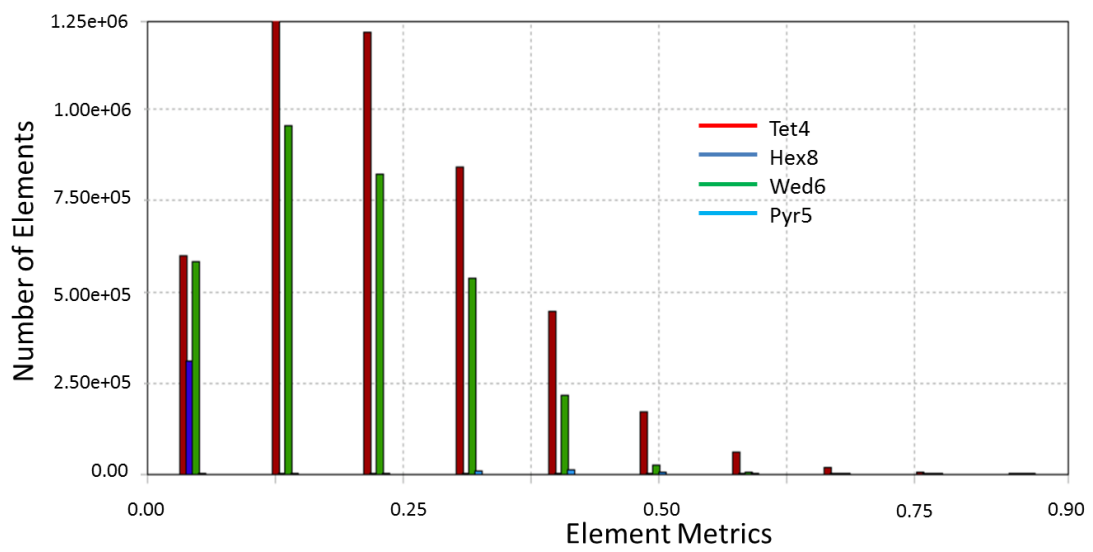


Figure 4-18: Skewness values for Test Case mesh elements

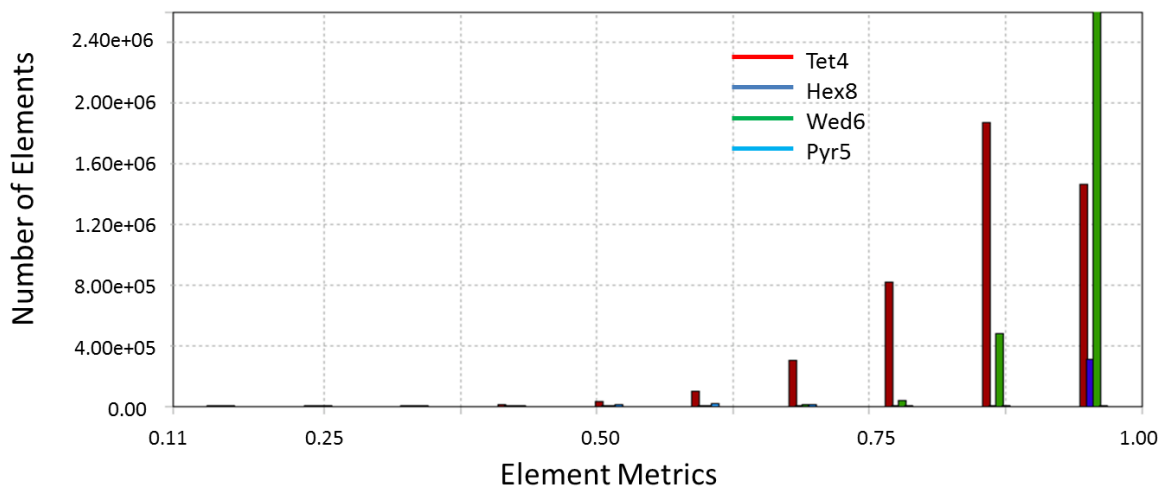


Figure 4-19: Orthogonal quality values for Test Case mesh elements

Table 4-4: Solver mesh quality characteristics for Test Case

Domain Name	Orthog. Angle	Exp. Factor	Aspect Ratio
	min (deg)	max	max
Rotor	39.8 ok	30 !	23 OK
Stator	38.9 ok	18 ok	76 OK
Global	38.9 ok	30 !	76 OK
	%! %ok %OK	%! %ok %OK	%! %ok %OK
Rotor	0 <1 100	<1 2 98	0 0 100
Stator	0 <1 100	0 <1 100	0 0 100
Global	0 <1 100	<1 1 99	0 0 100

4.4.2.3 Solver Settings

The main areas of interest for setting-up CFX solver are the boundary conditions, the stator-rotor interface, the turbulence model and the convergence criteria. All these, together with some other settings and parameters, are discussed in the below paragraphs.

4.4.2.3.1 Boundary Conditions and Stator-Rotor Interface

Specifying appropriate boundary conditions is of great importance for successful CFD models. As explained above, both Test Case and GT26 are 4-arm models, which mean that both domains have 4 ports, namely main inlet, main outlet, inlet seal and outlet seal. All ports are connected to a common cavity. For specifying boundary conditions in any port or opening, CFX offers four options i) mass flow, ii) velocity, iii) total pressure or iv) static pressure. For simpler models, for example of one inlet and one outlet, ANSYS provides detailed guidance and examples (65). For more complicated cases, like Test Case and GT26 ones, there is no established guidance and the advice in most cases is for trial-and-error approach.

Through numerous trial runs, it was concluded that inlet, outlet and both seals ports have all to be specified with mass flows. For a 4-arms cavity it proved quite difficult to specify pressure boundary conditions, as solver kept crashing with overflow or non-convergence errors. For side boundaries, *rotational*

*periodicity*¹⁴ was used. The generally monotonic and smooth behaviour of convergence history curves (Figure 4-22) during solution confirmed that this configuration worked.

The interface plane between rotor and stator is shown in Figure 4-21. The location of interface was selected to be in half distance between stator and rotor cavity walls. This is a common and proven technique for interfaces in similar cavities, such as blades-vanes interaction in compressors and turbines (65). To manage the interface between the two domains, CFX offers four types of frame change models: *frozen rotor*, *stage*, *transient rotor-stage* and *none*. Starting from the last one, *none* is selected when there is no pitch change or frame change between domains, so clearly it was not an option for the current model. *Transient rotor-stage* is a method that aims to capture effects on unsteady flows of varying rotational speed of one or both domains. This is not an option for Test Case or GT26, as both models refer to steady state conditions, including also steady rotational speed.

Remaining options *frozen rotor* and *stage* refer to steady state models. *Frozen rotor* assumes fixed relative positions between stator and rotor, while *stage* method calculates a circumferential average of all fluxes. As there is no pitch change between stator and rotor, which can be a differentiator between the two methods with *stage* being the recommended one in that case, the two methods were found to give very close results. *Frozen rotor* was finally selected as the option that requires less computational effort and less CPU memory.

Important note is the fact that mesh connection between two interfaces was done automatically by the solver. Through *automatic method* selection, solver tries to connect the two meshes in a 1:1 connection, in other words node to node. Most of the times this is not feasible for the full common surface between two sub-domains. In this cases, solver switches automatically to an interpolation method between faces of two meshes (stator side and rotor side). Critical detail

¹⁴ Rotational periodicity is used when a sector out of a 360° domain is modelled and it defines the two sides of the sector's periodic interface; further details are available in (69).

for successful connection between the two meshes is that both sides have similar sizes of cell edges. As it is shown in Figure 4-15, this key requirement was met with appropriate elements size selection for both stator and rotor subdomains.

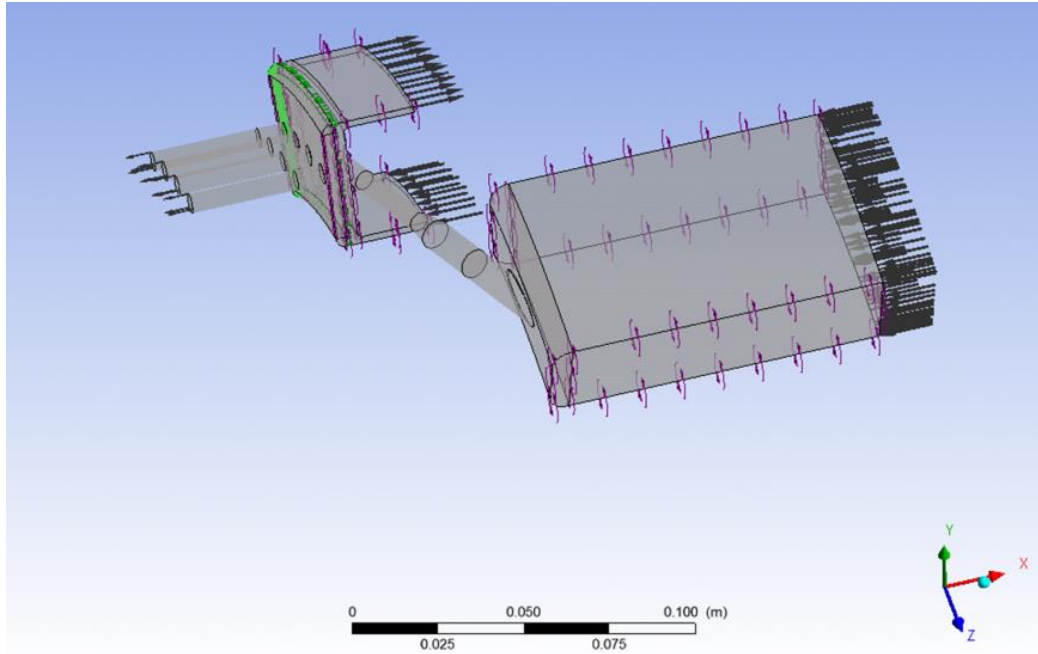


Figure 4-20: Test Case model set-up in solver

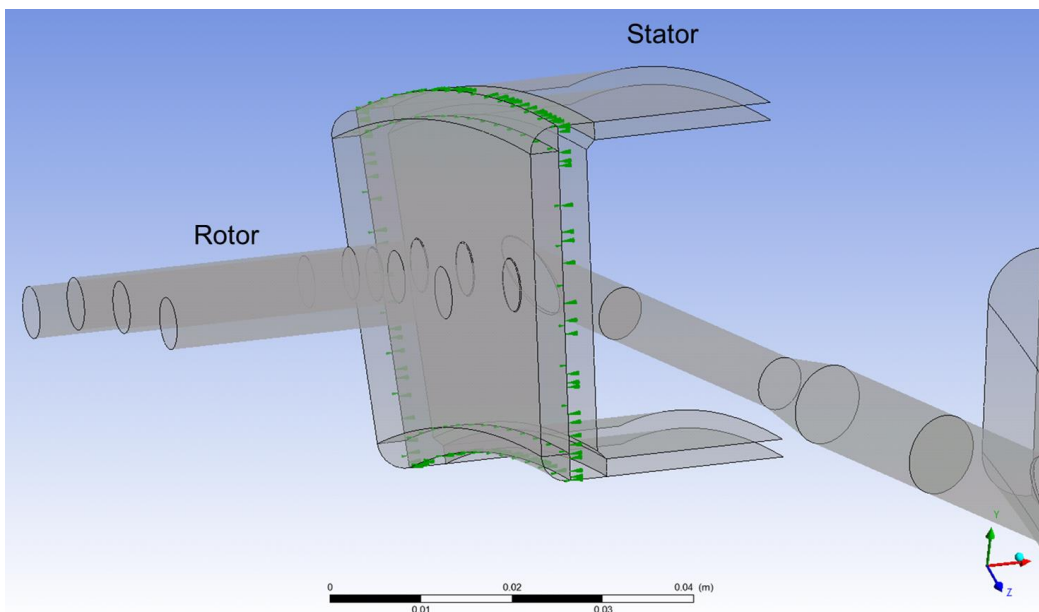


Figure 4-21: Interface plane between rotor-stator in Test Case

4.4.2.3.2 Turbulence Model and Near-wall Treatment

One of the most important settings in CFD solvers that solve Reynolds-averaged-Navier-Stokes (RANS) equations is the turbulence model selection. This is because Navier-Stokes (NS) equations are resolved with a Reynolds averaging procedure, so it can be said that the full information from NS equations is lost without use of appropriate turbulence model. And in any turbulent flow, turbulence has a significant effect on flow characteristics. In absence of a LES or DNS approach, selection of the most appropriate to the specific flow turbulence model is key for understanding the flow details.

CFX offers a number of different models that treat turbulence. CFX models are classified into two categories, Reynolds-Stress Models (RSM) and Eddy Viscosity Models (EVM). To an extent very similarly to LES versus RANS discussion in paragraph 4.4.1, RSM models are significantly more complicated and computationally intensive than EVM ones (10), (67), (65), as they employ up to seven (note: sometimes up to 12) partial differential equations (PDEs) to cover all components of turbulent flow (10), (68). They are primarily used in very complex flows, where full 3D understanding of flow domain is required. Expectably, they also involve much longer computational times. From the other side, EVM models offer a fair compromise between acceptable fidelity and computational effort. They are models capable of capturing flows in boundary layers, cross flows, mixing flows and jets (65). For current research work, the three candidate EVM models were $k - \omega$ ($k - \omega$), $k - \epsilon$ ($k - \epsilon$) and Shear-Stress-Transport (SST) model, all being two – equations models. Remaining EVM models, that CFX offers, were not recommended by ANSYS for the specific application, but also no references using them were found.

In some more details:

- $k - \epsilon$ model;

model's two equations resolve kinetic energy (k) and turbulence eddy dissipation (ϵ). It is a popular model, used in a number of references, with a wide range of engineering applications. However, it is not recommended for

modelling flow close to boundary layers, flow separations and also flow in rotating fluids.

- $k - \omega$ model;

it employs kinetic energy, similarly to $k - \epsilon$, but second equation resolves turbulence eddy frequency (ω). It has all the advantages of $k - \epsilon$ but it can also capture separation of flow, boundary layers and flow impingement on walls. Its main disadvantage is that is more sensitive than $k - \epsilon$ in free-stream flows.

- Shear-Stress-Transport (SST) model;

SST is an eddy frequency (ω) based model that is considered to be a combination of $k - \epsilon$ and $k - \omega$ models, as it blends the strong points of the two. This is achieved through SST's feature to switch between the two models ($k - \epsilon$ and $k - \omega$) according to the distance from wall.

ANSYS recommend use of *automatic wall function* when SST is selected, so as full advantage of model to be materialised (69). The *automatic wall function* method allows a gradual switch between the two globally accepted methods for resolving near-wall flows, which are the *wall function method* and the *low-Re method*. By this way the rapid variation in flow variables near the wall is captured, while the requirement for y^+ values close to unity is relaxed¹⁵. Combination of SST with *automatic wall function* brings requirement for y^+ values to be between 20 and 100 with sufficient capture of the flow in the boundary layers. This is quite important for the mesh economy of model, as potential requirement for y^+ values close to unity (~ 1) would unavoidably necessitate use of first cell height with values lower than 0.001mm. In turn, this would have a dramatic impact on mesh size and computational time, as number of mesh elements would probably increase to more than 25 million for the specific domain and running time to more than 8-9 hours per operational point.

¹⁵ y^+ values close to unity is still a requirement when heat transfer phenomena between the wall and the flow are of interest.

4.4.2.3.3 Convergence Criteria and Other Settings

Default values for convergence criteria were ignored and stricter criteria were selected for convergence residual target and, more importantly, for mass and energy conservation target. Aim was through lower residuals and stricter conservation target, a more accurate solution to be calculated. This led to longer running times (2-3 hours per design point) but it re-assured that especially critical total relative temperature calculations inside blade cooling holes were as accurate as possible. More specifically, the applied criteria were asking for residual target of 0.001 and conservation target for mass and energy <0.1%, with convergence residual type RMS.

In terms of discretised advection term calculation, CFX offers three options: *1st Order Upwind*, *Specified Blend Factor* and *High Resolution*. Appendix E provides some more details on all three options. *High Resolution* scheme was selected in order to improve accuracy of the solution and as the solver did not encounter any robustness issues, it proved to be a correct selection.

Finally, some other important settings for Test Case's model were:

- fluid selection;

air ideal gas was selected as fluid medium. Despite the relatively low temperature and pressure in boundary conditions, use of constant viscosity values was avoided. Instead, Sutherland's formula was used with the below settings (70): $T_0 = 273\text{K}$, $\mu_0 = 1.71 \times 10^{-5} \text{ kg / (m s)}$ and $S_{\text{air}} = 110.4\text{K}$

- timescale;

automatic timescale was selected. The fact that solution was smooth and not bouncy removed any need for further optimisation of timescale settings.

- side interface model;

rotational periodicity was selected for all side interfaces of both stator and rotor sub-domains (Figure 4-20). It has to be noted that rotational periodicity selection can cause solver to crash, when geometry model is not carefully

sectored out of the 360° model. Both Test Case and GT26 models proved that CAD models were accurately sectoried out of their initial 360° models.

4.4.2.4 Results

The model's results are discussed below in terms of convergence history, mesh sensitivity study, results in CFX Post and, critically, comparison with the experiment ones.

4.4.2.4.1 Convergence History

Figure 4-22 illustrates convergence history of model. Both convergence criteria in residuals and imbalance were met in fewer than 100 iterations. It is also important that curves in both graphs are smooth and monotonic and this is a confirmation, amongst other factors, for correct setting of realistic boundary conditions.

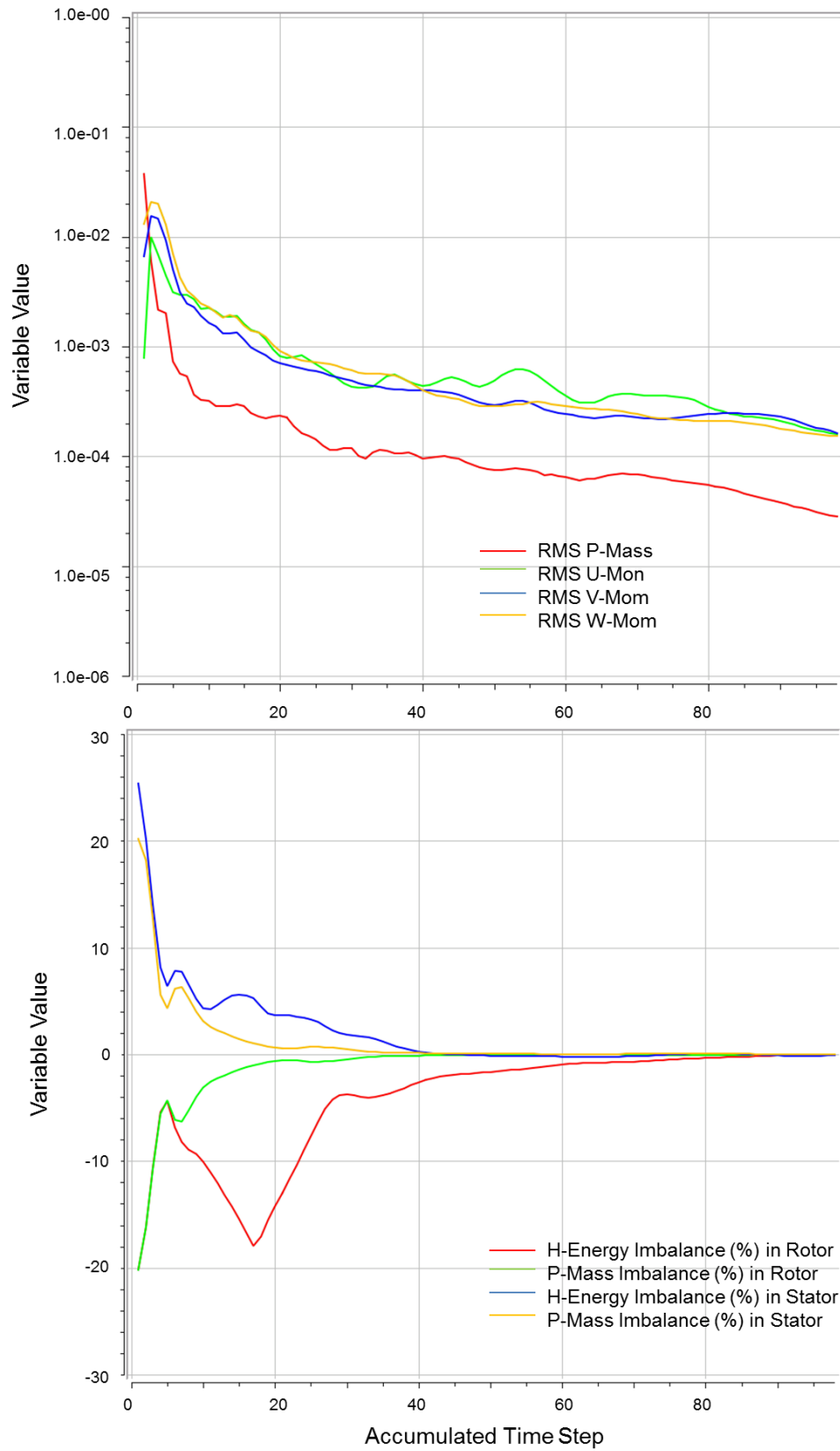


Figure 4-22: Test Case model, convergence history, mass and momentum residuals (top) and mass and energy imbalances (bottom)

4.4.2.4.2 Mesh Sensitivity Study

For the necessary mesh sensitivity study, two methods were used. Firstly, the property of interest, namely dimensionless ΔT was plotted over different mesh populations. The graph in Figure 4-23 illustrates that mesh populations above 6.5 million elements did not alter total relative temperature and consequently dimensionless ΔT results.

In addition, Richardson's Extrapolation method was also used. Chaplin's work (71) was used as reference for the use of this method and Appendix F describes the whole process. Coarse, medium and fine mesh populations were used from Table 4-5, resulting in an asymptotic range (AR) value was 1.066, which is considered excellent.

Table 4-5: Test Case model, mesh elements populations, values of $T_{\text{tot,rel,BCHs}}$ and dimensionless ΔT

	Coarse		Medium		Fine
Number of elements	4,833,557	5,966,780	6,410,339	12,448,774	14,168,721
$T_{\text{tot,rel,BCHs}}$ (K)	308.333	308.037	308.014	308.004	307.994
Dimensionless ΔT	3.341123	3.408552	3.413791	3.416069	3.418347

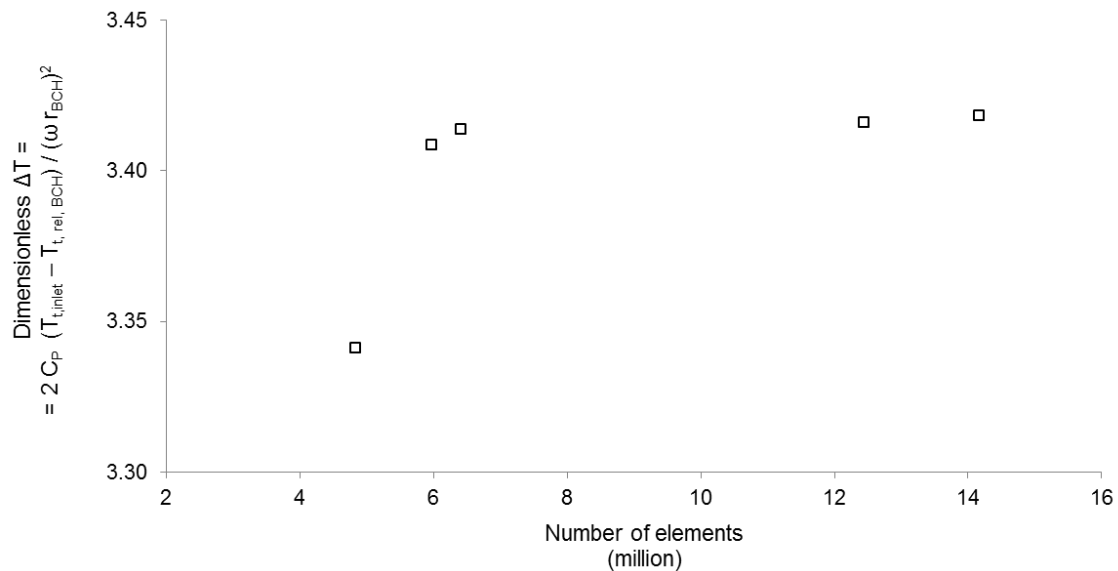


Figure 4-23: Evolution of dimensionless ΔT with increase of mesh elements population

4.4.2.4.3 Results in Post

The below results were produced with ANSYS Post, which is CFX's post-solver processor. Y^+ values, cooling air velocity and Mach number are three parameters that were plotted in fluid domain.

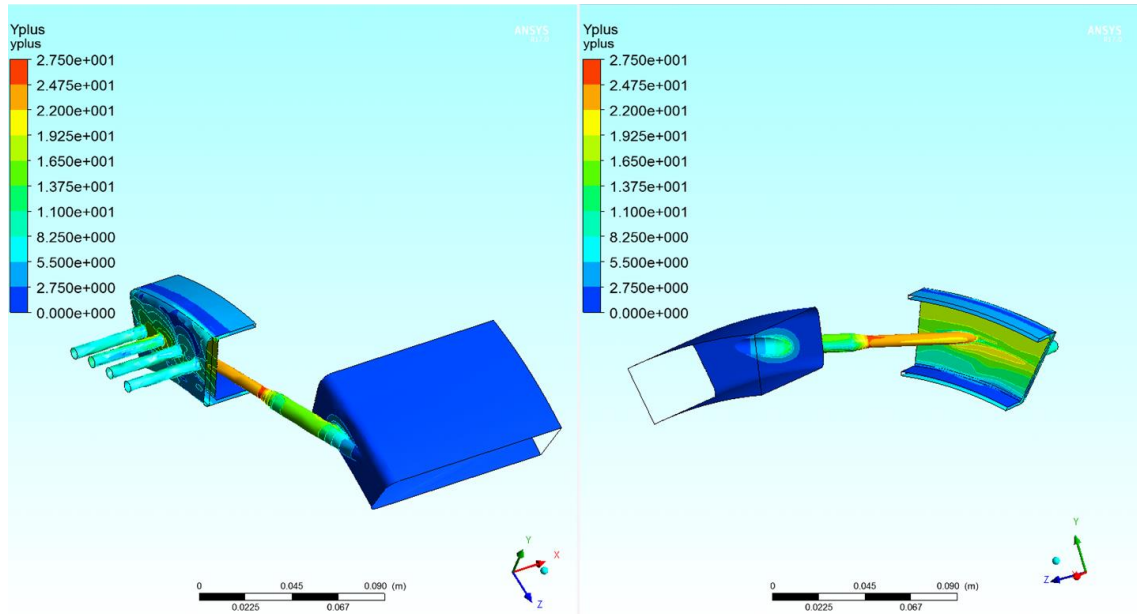


Figure 4-24: Y^+ values for Test Case

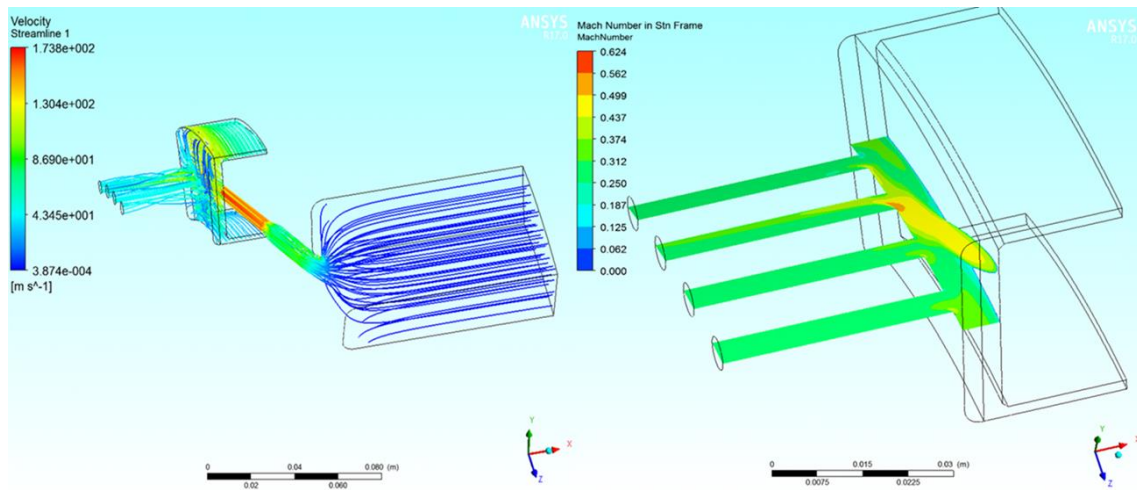


Figure 4-25: Test Case model, flow velocity (left) and Mach number (right)

4.4.2.4.4 Comparison with the Results form Experiment

As expected, total relative temperatures inside blade cooling passages were also calculated. Initially, calculation of total relative temperatures where made with mass flow average function at the respective plane. However, in order to simulate more realistically the function of Test Case rig's thermocouples as these are shown in Figure 3-10, probe points were used. Figure 4-26 shows the location of probes inside blade cooling passages.

Results from total relative temperatures inside blade cooling passages and static pressure inside cavity were converted to effectiveness and swirl ratio as per paragraph 3.3.2 discussion. Effectiveness and swirl ratio graphs are shown in Figure 4-27 and Figure 4-28, together with Test Case's results. These graphs are discussed in more details in the next paragraph

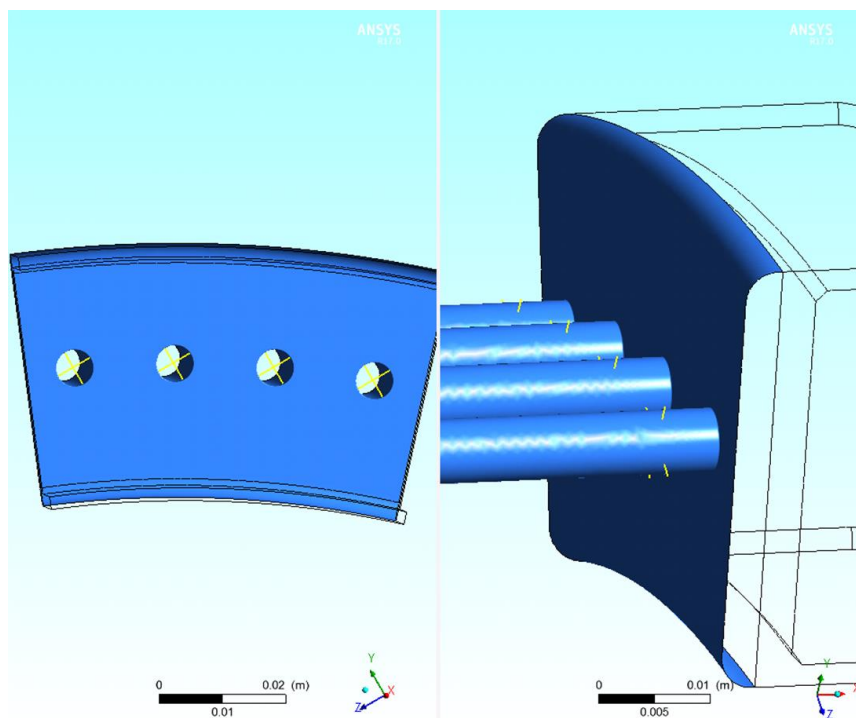


Figure 4-26: Location of probe points for total relative temperature measurements inside Test Case blade cooling passages

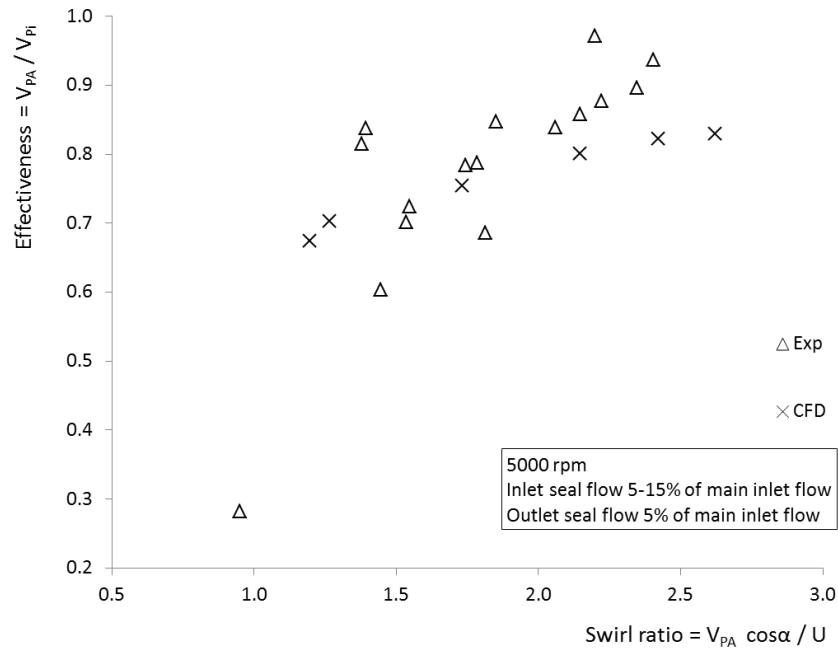


Figure 4-27: Test Case model, comparison of experiment with CFD results

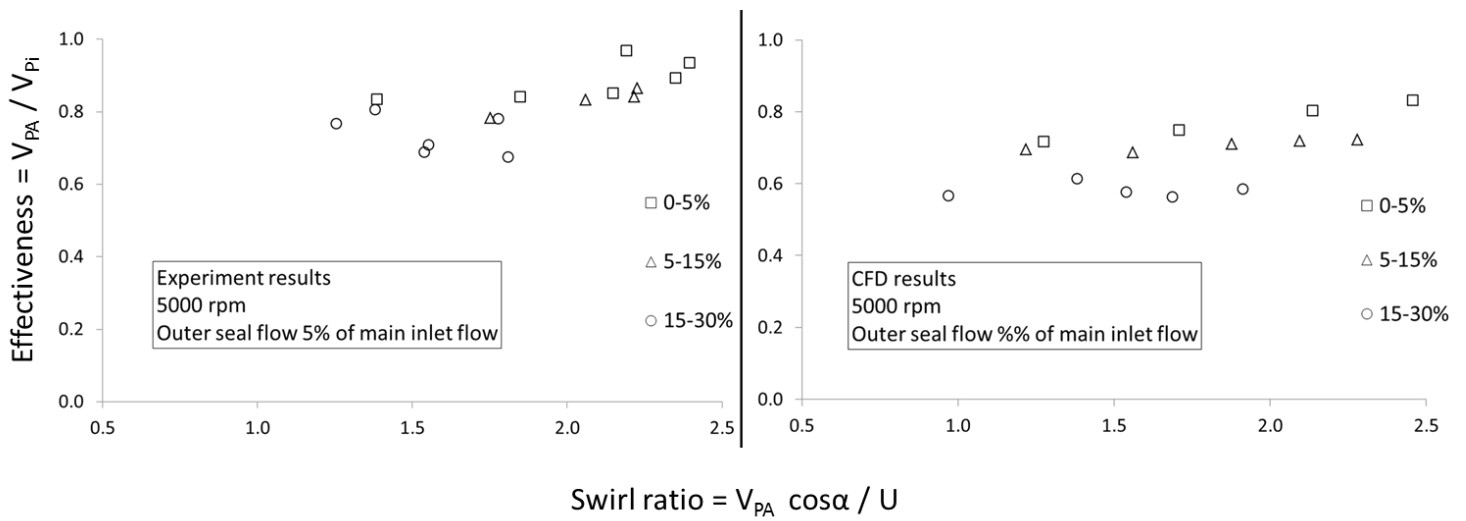


Figure 4-28: Test Case model, effect of inlet seal flow on pre-swirl effectiveness, experiment (left) and CFD (right) results

4.4.2.5 Discussion on Test Case CFD Solution Results

Summarising, the main conclusions and comments from results in Figure 4-27 and Figure 4-28 are:

- i. In Figure 4-27, CFD results agree with Test Case's experiment results as effectiveness increases in higher swirl ratios.
- ii. In figure 4-28, CFD results agree with Test Case as increase in inlet seal flow, decreases effectiveness.
- iii. In Figure 4-27, CFD results overestimate effectiveness for swirl ratios <1.3 and underestimate effectiveness for swirl ratios >2.1 .
- iv. In the case of CFD effectiveness-underestimate (right side of graph in Figure 4-27), the maximum difference in effectiveness between CFD and experiment is approximately 0.1; this equates to a value of almost 2.5K in total relative temperature inside blade cooling passages (BCHs).
- v. In the case of CFD effectiveness-overestimate (left side of the graph in Figure 4-27) the difference in effectiveness is similar and close to 0.1.
- vi. Experiment's value at bottom left of the graph in Figure 4-27 with effectiveness 0.28 and swirl ratio 0.95 has probably to be ignored as an outlier; in CFD model, such low effectiveness was achieved with inlet seal flow well over 60% of main inlet flow.
- vii. Experimental values are quite scattered in both graphs. This can be explained by the uncertainty of the static pressure instrument, which is in the order of $\pm 2.5\%$ (9), as it has already been discussed in paragraph 3.3.1.
- viii. Critical detail remains the fact that total relative temperature thermocouples, which are located inside blade cooling passages, as shown in Figure 3-10, have a recovery factor of 0.85 (9). In other words, this means that the thermocouples "feel" lower temperature – which results in higher effectiveness - than the actual temperature inside blade cooling passages, as it was discussed also in detail in paragraph 3.3. New (9) expanded a bit more and argued that thermocouples feel even lower temperature, in higher swirl ratios due to increased turbulence inside blade cooling passages.

- ix. A number of similar works report that total relative temperature predictions and measurements inside blade cooling passages have some limitations (72). Chew et al. (8) reports that even with a recovery factor of 0.97, total relative temperature thermocouples accuracy was estimated as 2.5K. Cagan et al. (73) reports differences between predicted and measured total relative temperature from 2K up to 5K, with the trend between the two being diverging as swirl ratio increases. A very interesting work by Rolls Royce (2) concluded that the difference between predicted and measured total relative temperature was between 2 and 3K. Finally, Wilson (55) reports difference between experiment and CFD up to 8.7°C, which is considered to be well above the experimental uncertainty. In a rather disappointing tone, Wilson concluded that there are too many factors contributing to this, but it is encouraging that the CFD values show the correct trend¹⁶.

Following the above, current CFD results are considered to be accurate and acceptable. Further alignment with experiment results can be probably achieved with use of DNS or LES solvers. Instruments' uncertainty of the rig has also to be considered in all cases.

As a summary, Test Case's CFD model was validated with the use of the experiment's results. The dimensionless quantities of pre-swirl effectiveness and swirl ratio, as those were defined in paragraph 3.3.2, were used and CFD results show good agreement with experiment's ones. A decrease in effectiveness was confirmed, when inlet seal flow increases. Three different groups of inlet seal flows were modelled, 0-5%, 5-15% and 15-30%, with the percentage referring to inlet seal mass flow over total mass flow of cooling air through pre-swirl nozzles. In agreement with the experiment, CFD results showed that effectiveness is higher when inlet seal flow is 0-5%, lower when it is 5-15%, and lowest when it is 15-30%, across a wide range of swirl ratios. Quantitatively, the maximum difference in effectiveness between CFD and

¹⁶ It has to be acknowledged that Wilson's work is of the earliest CFD ones (1997) in the area, so limitations in technology (solver, turbulence models etc) have to be accounted for.

experiment values was approximately 0.08, with experiment values being higher than CFD. This difference was observed in high swirl ratios (>2.2). The derived total relative temperature inside blade cooling ducts from this delta in Effectiveness is approximately 2.1K, with the domain temperature in both experiment and CFD cases being between 300K and 323K. This is considered acceptable given the uncertainty of the experiment's instruments ($\pm 2.5\%$) and especially the recovery factor of the total relative temperature thermocouples which can experience values as low as 0.85, when they operate inside highly swirled turbulent flows.

Test Case's CFD model was verified with the use of a Grid Sensitivity Study and Richardson's Extrapolation methods. Grid's Sensitivity Study showed that mesh population higher than 6.5 million elements did not alter the quantity of interest, namely the total relative temperature inside blade cooling duct. Richardson's Extrapolation method resulted in an Asymptotic Range value of 1.066, which is considered excellent.

4.4.3 Computational Solution for Gas Turbine

Following conclusions from Test Case model above, similar approach was applied to develop CFD model for GT26 LPT pre-swirl nozzles domain. Some of the details on geometry, mesh, solver settings and results are discussed in below paragraphs. However, as GT26 model is in essence a replication of Test Case's one, a number of details on definitions and settings are not repeated in the interest of brevity.

4.4.3.1 Geometry, Domain and Mesh

Geometry is shown in Figure 4-29. The domain consists of two pre-swirl nozzles and three blade cooling passages. Reproduction of geometry model was achieved through extended reverse engineering activities, namely by capturing geometry from open-engine maintenance opportunities, and available photos and drawings.

As expected, the same meshing strategy to Test Case was followed for GT26 domain. A hybrid mesh was applied with structured and unstructured parts. A fully structured mesh was again not practical, due to similar complexities in geometry and primarily the existence of acute angles between pre-swirl nozzles and upstream and downstream domains. Consequently, structured mesh (hexagons) was applied with method of *sweep* in areas of inlet domain, inlet seal and outlet seal. Unstructured mesh (prisms and tetrahedrals) was applied inside blade cooling passages, inside cavity and in the areas where pre-swirl nozzle joins inlet and cavity domains. The method for development of unstructured mesh was the *automatic* one, based on Delaunay triangulation algorithm. Total number of elements was approximately 17.6 million.

Similarly to Test Case, use of Shear-stress-transport (SST) model relaxed requirement for non-dimensional distance from wall (y^+) values close to unity (65). Instead, values of 20-200 were used for the area of interest which in this case is the blade cooling passages outlet. To achieve current y^+ values, 10 inflation layers were used in all domain walls with first layer having a height of 0.1mm, while remaining inflation layers had a growth rate of 1.2. Figure 4-31

shows details of inflation layers in a blade cooling passage, while Figure 4-36 in the results paragraph 4.4.3.3.3 shows the achieved y^+ values in the domain.

Proximity and curvature function was used which allowed mesher to automatically increase number of cells, by reducing their size, in domain areas of increased curvature, namely close to angles, areas of changes of diameters etc.

Mesh quality characteristics were discussed in detail in paragraph 4.4.2.2. For GT26 domain, *skewness* distribution is shown in Figure 4-32. Average value was 0.21862, which is considered excellent. *Orthogonal quality* is also shown in Figure 4-32. Average value was 0.89666, which is considered very good. In solver's mesh quality criteria, less than 1% of the total elements were poor (Table 4-6), so no further action was required.

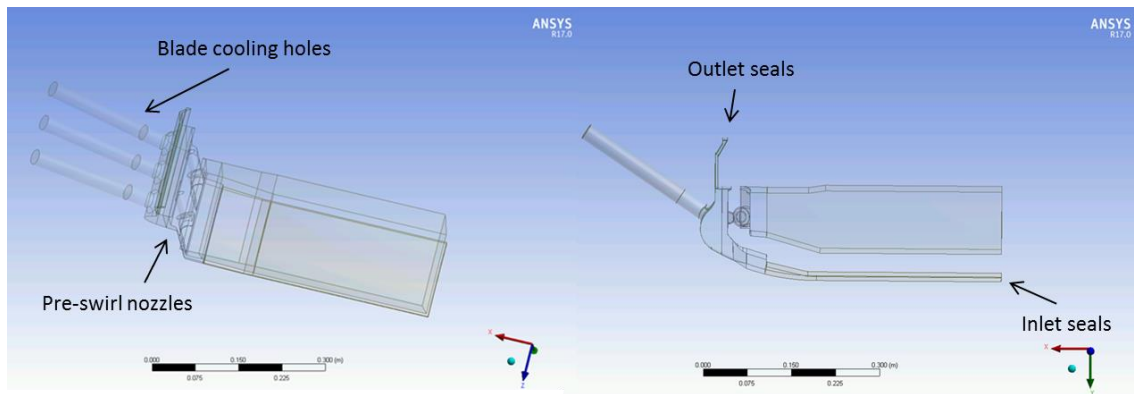


Figure 4-29: Gas turbine domain geometry

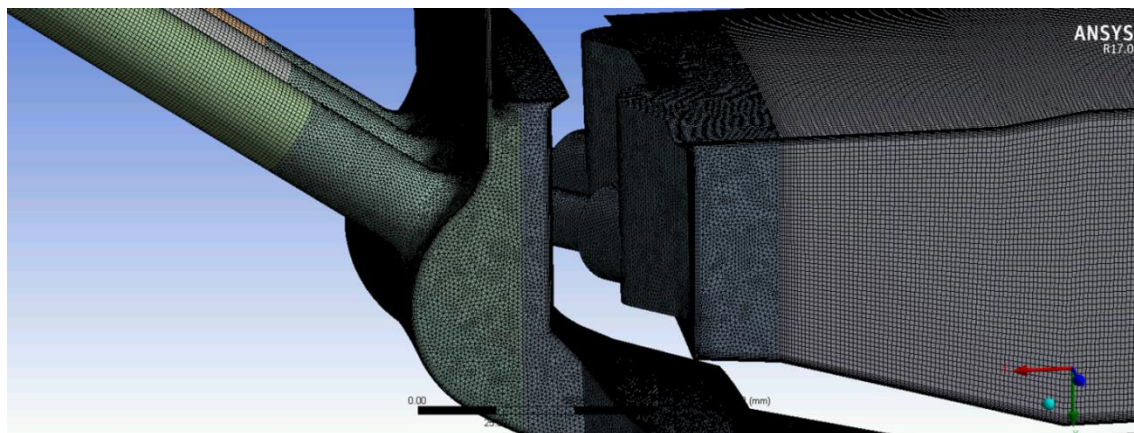


Figure 4-30: Detail of gas turbine model mesh

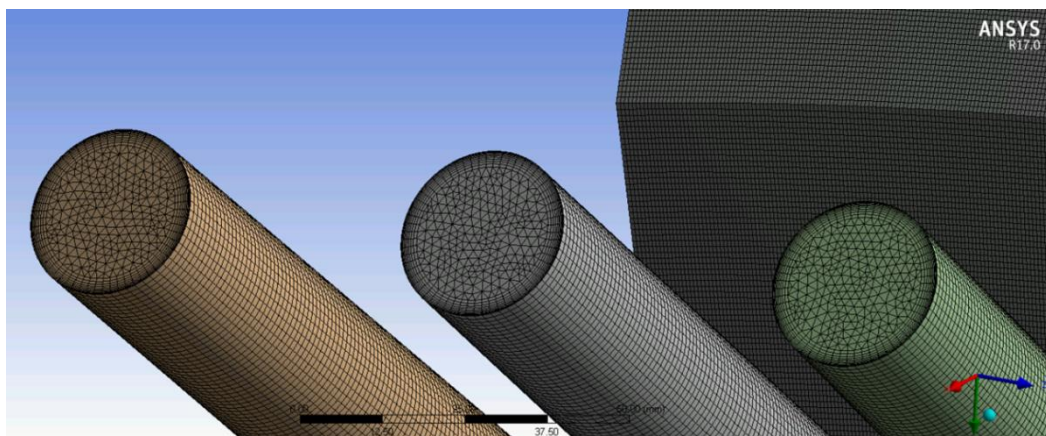


Figure 4-31: Detail of inflation settings in blade cooling passages

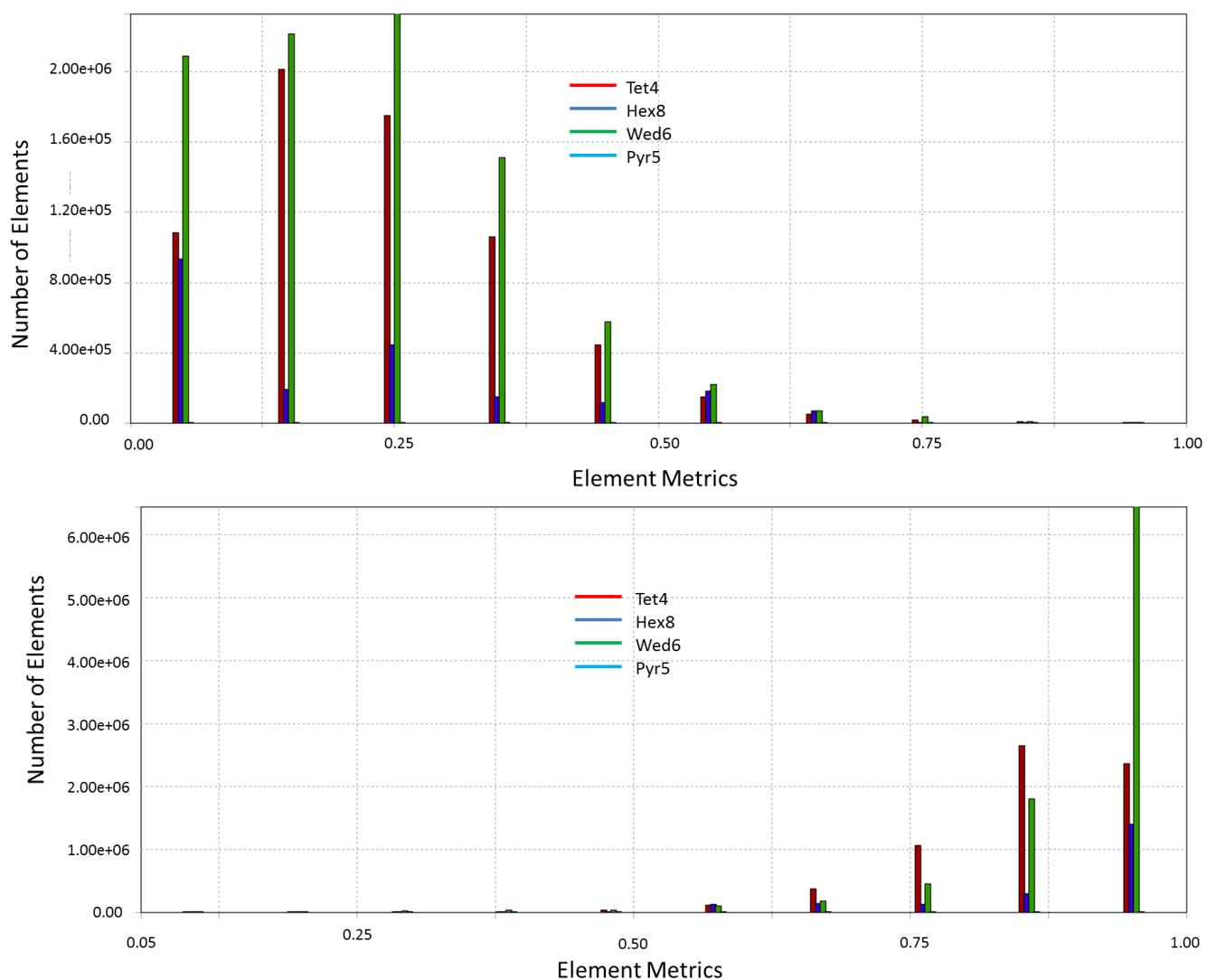


Figure 4-32: Skewness (top) and Orthogonal Quality (bottom) for gas turbine mesh elements

Table 4-6: Solver mesh quality characteristics for gas turbine model

Domain Name	Orthog. Angle	Exp. Factor	Aspect Ratio
	Min (deg)	Max	Max
Rotor	36.8 ok	75 !	25 OK
Stator	12.6 !	53 !	93 OK
Global	12.6 !	75 !	93 OK
	%! %ok %OK	%! %ok %OK	%! %ok %OK
Rotor	0 7 93	<1 1 99	0 0 100
Stator	<1 1 99	<1 1 99	0 0 100
Global	<1 3 97	<1 1 99	0 0 100

4.4.3.2 Solver Settings

Detailed discussion over solver settings took place in paragraph 4.4.2.3, where the arguments behind selection of various settings can be found. Due to the similarities with the Test Case, most settings were kept the same for GT26, so there are only referred in this paragraph.

In summary:

- for stator – rotor interface, *frozen rotor* was selected.
- For side interface, rotational *periodicity* was selected.
- For turbulence model, SST model was selected.
- As fluid selection; *air ideal gas* was used with viscosity values as per Sutherland formula (70).
- For advection scheme, *high resolution* was selected; solver encountered no robustness issues or delays during convergence.
- For boundary conditions, total pressure in main inlet, instead of mass flow that was used in Test Case, was applied. Pressure values came from Optigen and GasTurb values. Remaining three ports (inlet seals, outlet seals and main outlet) were all defined with mass flows (Figure 4-33).

In terms of time step selection, the approach was different to Test Case. In order to optimise further running time in both aspects of total duration and number of iterations, a revised time scale was used. Time step setting was not

automatic as in Test Case but defined by $1/\omega$ (ω =rotor rotational speed). The impact on convergence history of the above setting was obvious, as it is shown in Figure 4-34. Convergence was achieved in fewer than 30 iterations.

Similar to Test Case, default values for convergence criteria were ignored and stricter criteria were adopted for convergence residual target and for mass and energy conservation target. Aim was to re-assure that especially critical total relative temperature calculations inside blade cooling holes were as accurate. For GT26 model, the convergence criteria were 0.001 for residual target and <0.1% for mass and energy target.

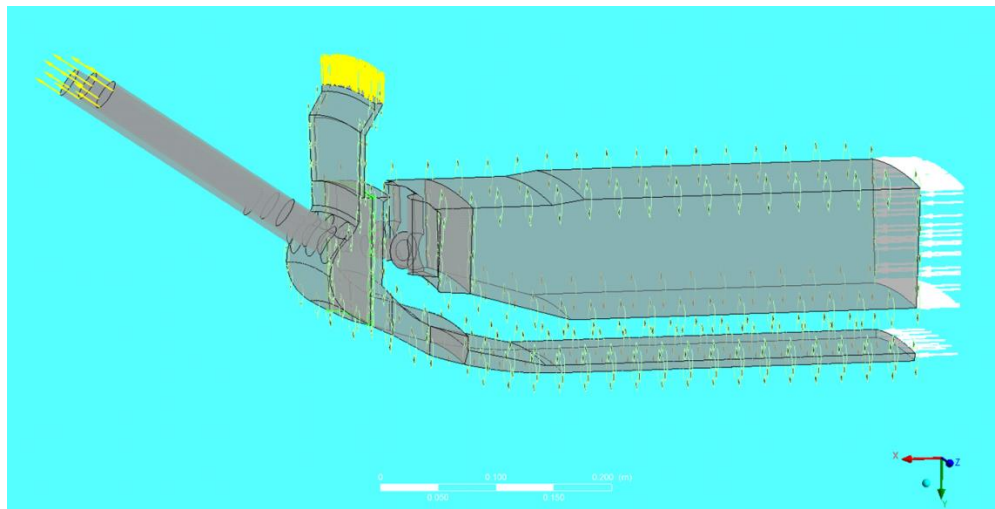


Figure 4-33: Gas turbine model set-up in solver

4.4.3.3 Results

Similarly to Test Case, this paragraph discusses the model's results in terms of convergence history, mesh sensitivity study, results in Post and comparison with analytical solution ones.

4.4.3.3.1 Convergence History

Convergence history is shown in Figure 4-34. Optimised time scale, as discussed in paragraph 4.4.3.2, improved further the number of required iterations and consequently the total running time. As shown, with convergence criteria of RMS residuals at 0.0001 (left graph) and mass and energy conservation at 0.1% (right graph), convergence was achieved in fewer than 30

iterations. In addition, the smooth and monotonic convergence is a re-assuring factor for correctly-posed boundary conditions.

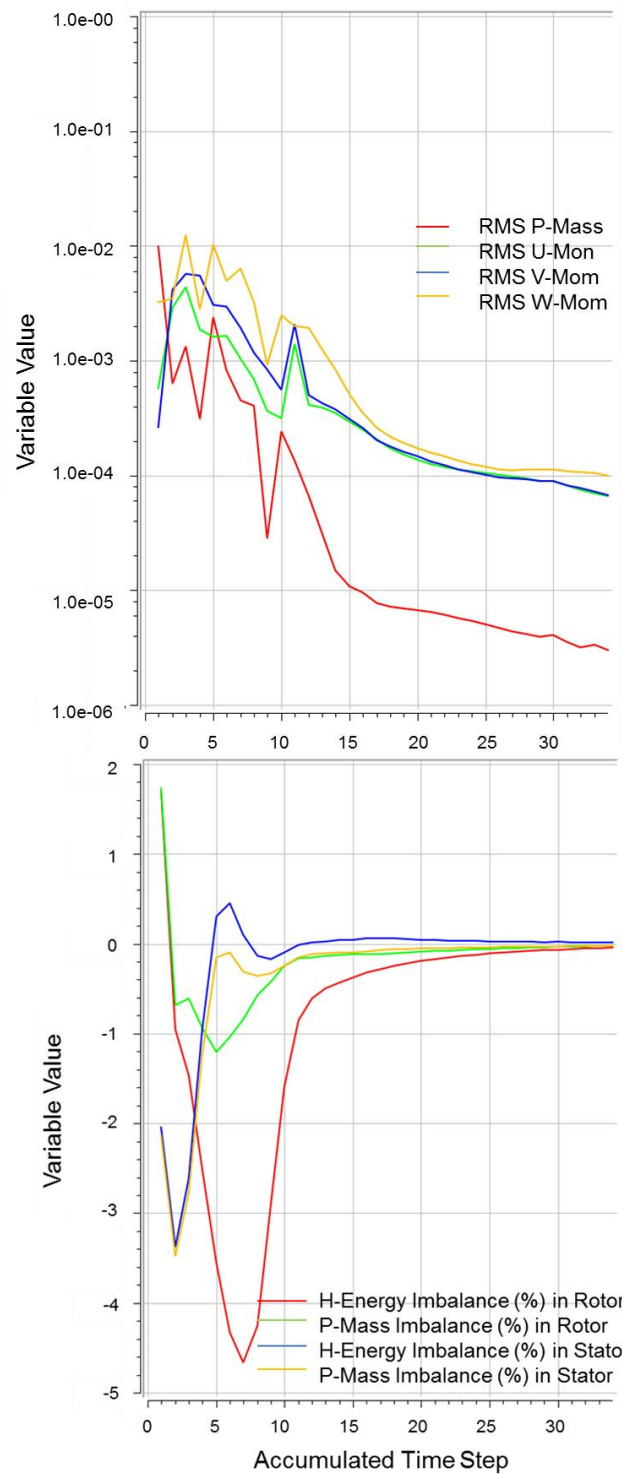


Figure 4-34: Gas turbine model, convergence history for mass and momentum residuals (top) and mass and energy imbalance (bottom)

4.4.3.3.2 Mesh Sensitivity Study

Table 4-7 shows the five different populations of mesh elements that were used for mesh (grid) sensitivity analysis. Change in total number of elements was achieved by altering global elements size. Figure 4-35 shows that mesh populations higher than 17.5 million elements did not have any substantial change in the total relative temperature inside blade cooling holes and consequently in the dimensionless temperature ΔT .

Richardson's Extrapolation method (71) was also used again, with the coarse, medium and fine populations from Table 4-7. As a result, asymptotic range value for this study was 5.037, which is considered very good.

Table 4-7: Gas turbine model, mesh elements populations, values of $T_{\text{tot,rel,BCHs}}$ and dimensionless ΔT

	Coarse		Medium		Fine
Global element size (mm)	2	1.7	1.4	1.3	1.2
Number of elements	11,735,920	13,621,142	17,599,211	19,578,951	22,578,020
$T_{\text{tot,rel,BCHs}}$ (K)	607.311	607.208	606.474	606.749	606.556
Dimensionless ΔT	1.191	1.195	1.219	1.210	1.217

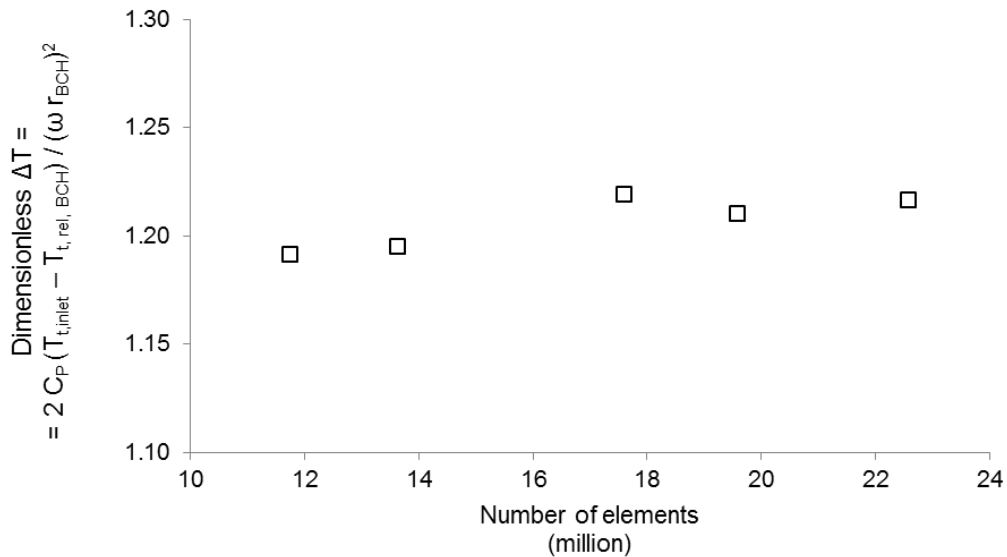


Figure 4-35: Evolution of dimensionless ΔT with increase of mesh elements population

4.4.3.3 Results in Post

With CFX Post, results were produced for Y^+ values, Mach number inside pre-swirl nozzles and cavity and Mach number inside blade cooling passages.

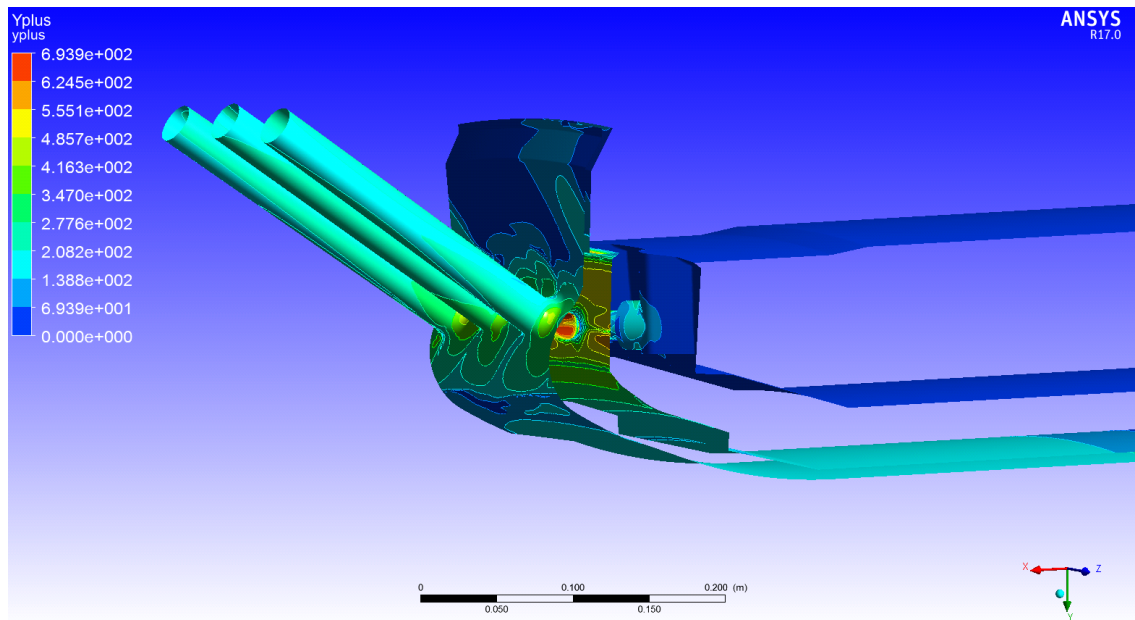


Figure 4-36: Y^+ values for gas turbine

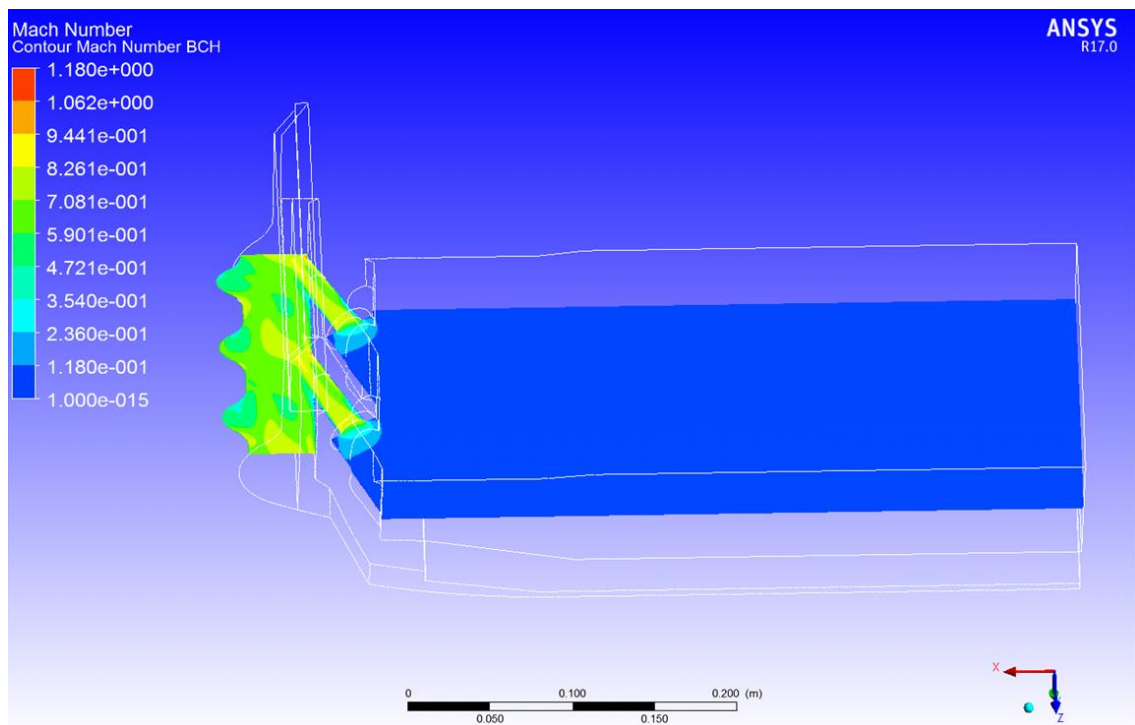


Figure 4-37: Gas Turbine, Mach number inside pre-swirl nozzles and cavity

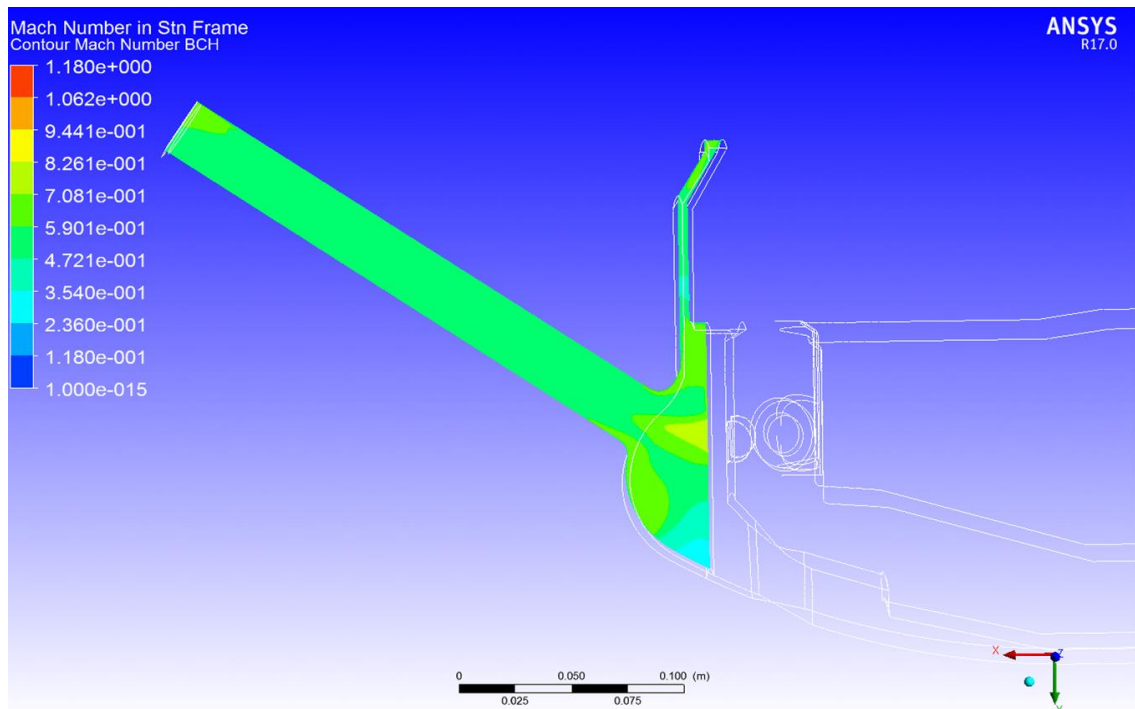


Figure 4-38: Gas Turbine, Mach number inside blade cooling passage

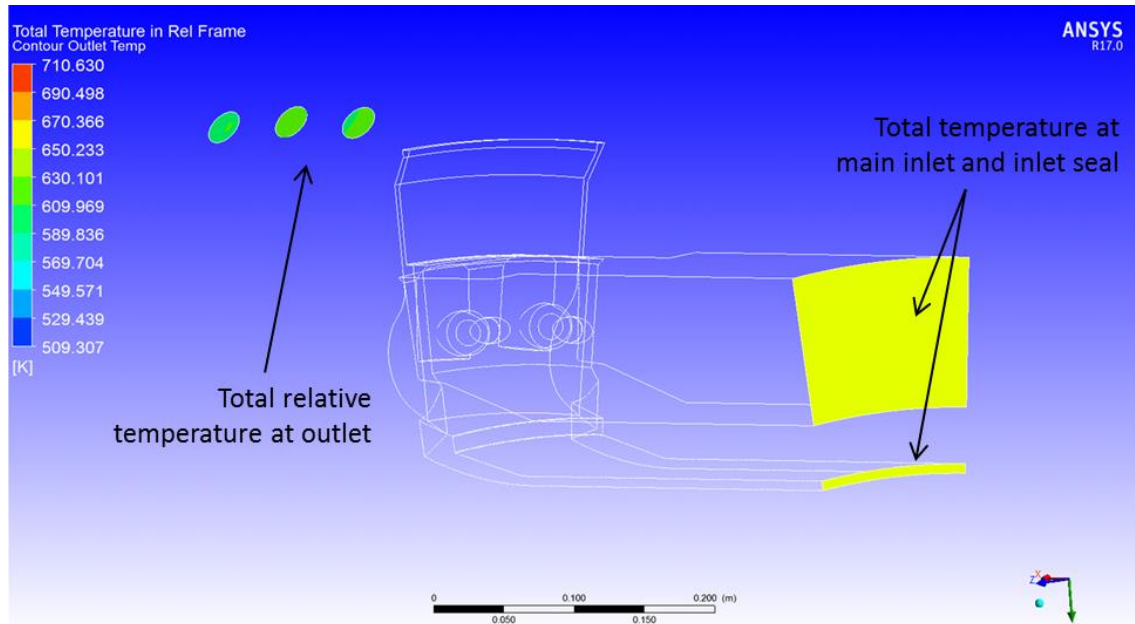


Figure 4-39: Gas Turbine, difference between inlet total and outlet total relative temperatures

4.4.3.3.4 Comparison with Analytical Solution

Finally, a graphical comparison between analytical solution results from paragraph 4.3.5 and CFD results is shown in Figure 4-39, while this comparison is discussed in some more details in paragraph 4.4.3.4. Table 4-8 summarises the impact on inlet seal flow increase over cooling air's dimensionless ΔT and total relative temperature inside blade cooling passages.

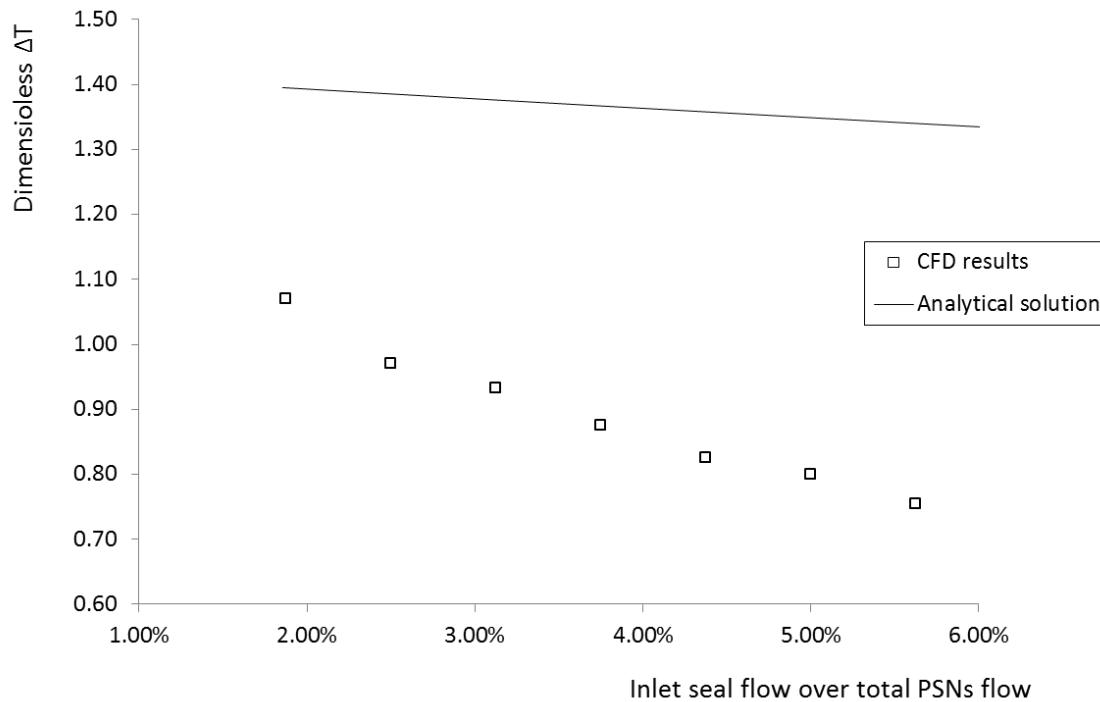


Figure 4-40: Gas turbine model, comparison of CFD and analytical solution results

Table 4-8: Gas Turbine CFD results - impact of inlet seal flow increase over cooling air dimensionless ΔT and total relative temperature inside blade cooling passages

Inlet Seal Flow over Pre-swirl Nozzles Flow	1.88%	2.50%	3.13%	3.75%	4.38%	5.00%	5.63%	6.25%	6.88%	7.50%
Dimensionless ΔT	1.0704	0.9708	0.9324	0.8748	0.8253	0.7998	0.7546	0.7229	0.6798	0.6617
Cooling Air Total Relative Temperature inside Blade Cooling Passage (K)	610.938	613.921	615.070	616.797	618.280	619.044	620.398	621.348	622.637	623.180

4.4.3.4 Discussion on Gas Turbine CFD Solution

For GT26's model, no experiment results were available. Instead, comparison of the CFD results took place with the analytical solution ones. Qualitatively, both sets of results agree that cooling air total relative temperature inside blades' cooling ducts increases, when inlet seal flow in the cavity increases. Quantitatively, the difference in dimensionless temperature difference (ΔT) between analytical solution and CFD was considerable, almost 0.3 in the design conditions (2% inlet seal flow), with analytical solution results being higher than CFD ones. This difference equates in approximately 9.7K in actual total relative temperature inside blade cooling ducts, with the domain temperatures being between 600K and 643K. The difference becomes bigger as inlet seal flow increases, resulting in almost 0.6 in ΔT and 18.2K respectively, when inlet seal flow is around 6%. Predicting higher ΔT values means that the analytical solution calculates lower total relative temperature values inside blade cooling ducts. This is expected, considering the over-simplistic approach of the analytical solution, which does not consider the complexities of geometry, the effect of cross jet flows between pre-swirl nozzles exits and the impact that discharge coefficients for pre-swirl nozzles and blade cooling duct inlets have on the flow. In addition, it assumes fully-mixture and isentropic conditions inside the cavity. Considerable difference between analytical and CFD solutions has also been reported in other works.

Similar to Test Case, Grid Sensitivity Study and Richardson's Extrapolation methods were used for GT26's model verification. Grid's Sensitivity Study showed that mesh population higher than 17.6 million elements did not alter the quantity of interest, namely the total relative temperature inside blade cooling duct. Richardson's Extrapolation method resulted in an Asymptotic Range value of 5.037, which is considered very good.

4.4.4 Summary of CFD Solution

In both Test Case and GT26 models, CFD results confirmed that an increase of inlet seal flow in the pre-swirl nozzles cavity incurs an, undesirable from blade's lifting perspective, increase of cooling air total relative temperature inside blade cooling ducts. This happens primarily because the inner seal flow velocity hampers and effectively decreases pre-swirl nozzle exit velocity, which is the driving factor behind total relative temperature drop inside blade cooling ducts.

In the Test Case, the agreement between experiment results and CFD is very good and the differences in effectiveness that were observed in high swirl ratio flows are explained above.

In GT26, the difference between analytical solution and CFD in Dimensionless Temperature Difference is considerable with the analytical solution overestimating the drop of total relative temperature of cooling air inside blade cooling ducts. This is expected and discussed in the above paragraph. Table 4-8 lists the important results in terms of cooling air temperature that have to be fed to the next objective of the project, which is the calculation of blade metal temperature.

4.5 Use of Monte Carlo Simulations

As opposed to the cooling air system solution, where the use of CFD provided high fidelity results and insight to the flow conditions of the area, the next tasks of the project are progressed with the use of analytical models. Unavoidably, in these cases the uncertainty is considerable and cannot be ignored. Hence Monte Carlo simulations are used in combination with these deterministic models.

The use of Monte Carlo (MC) in engineering problems is described in detail in the works of Singh (20), Di Lorenzo (38) and Parsons (74). Figure 4-41 shows schematically the principle in Monte Carlo models. A number of independent input variables, all described by different distributions, are combined in an algorithm or any mathematical relation and through numerous simulations they result in an output which is described by a new distribution. The details of Excel's Random Number Generator (RNG) algorithm, which is used to enable MC simulations in this work, can be found in (75).

In all the cases of blade cooling, blade stress and blade lifing calculations, the deterministic analytical models have been derived from the theory in chapter 2. The models are described in detail in every respective section (4.6, 4.7 and 4.8). The models are followed by tables which list the uncertainty boundaries and the type of distribution which is employed. In most cases, approximately 8000 simulations are performed as beyond this number no meaningful change in the result(s) is observed.

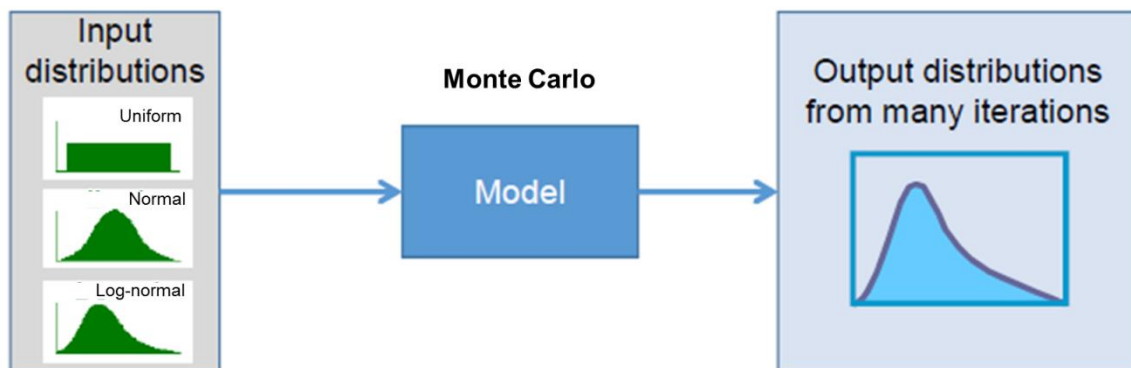


Figure 4-41: Typical schematic of Monte Carlo (74)

4.6 Blade Cooling Calculation

Following the theory of section 2.3, mean blade metal temperature is calculated. Two cases are explored; the first one assumes nominal conditions and no degradation in the cooling system. The second one refers to cooling air system degradation conditions and uses the increased cooling air temperature, as it was calculated in section 4.4, results in Table 4-8. Rubini's (14) and Apostolidis' (76) works are used for this step. Rubini in particular developed an algorithm derived from the theory and principles of paragraph 2.3, which is illustrated in the form of a flow chart in Figure 4-42. In addition, Table 3-3 in Technical Description's paragraph 3.2.2 offers the necessary geometrical details of the blade. Table 4-9 below lists the quantities that are subjected to Monte Carlo simulations. Mean values for cooling air flow (MBH30), hot gas velocity, hot gas temperature and hot gas density are results from Optigen and GasTurb models. The typical uncertainties (%) and the fact that the distributions are normal is suggested as common industrial practise for flow, temperature and density measurements. However, Biot number's mean value and selection of distribution and uncertainty is suggested by Rubini (14). Similarly, mean value, type of distribution, uncertainty and uncertainty values for film effectiveness ϵ_f are suggested by Han (11), as applicable to typical blades which combine presence of TBC with convective, film and impingement cooling. Total number of Monte Carlo points was 8000 and the stability of the average blade temperature was satisfactory. Blade metal temperature results are cited in Table 5-3 of paragraph 5.4.3.

Table 4-9: List of Monte Carlo uncertainties for blade cooling calculation

Property	Distribution	Mean Value	Standard Deviation
MBH30 CA Flow (kg/s)	Normal	35	0.05
Hot gas velocity (m/s)	Normal	147	0.02
Hot gas temperature (K)	Normal	1783	0.01
Hot gas density (kg/m ³)	Normal	3.3	0.01
Bi number	Normal	1	0.05
Film cooling effectiveness ϵ_f	Gamma	0.294	alpha = 8.9466 beta = 0.0328

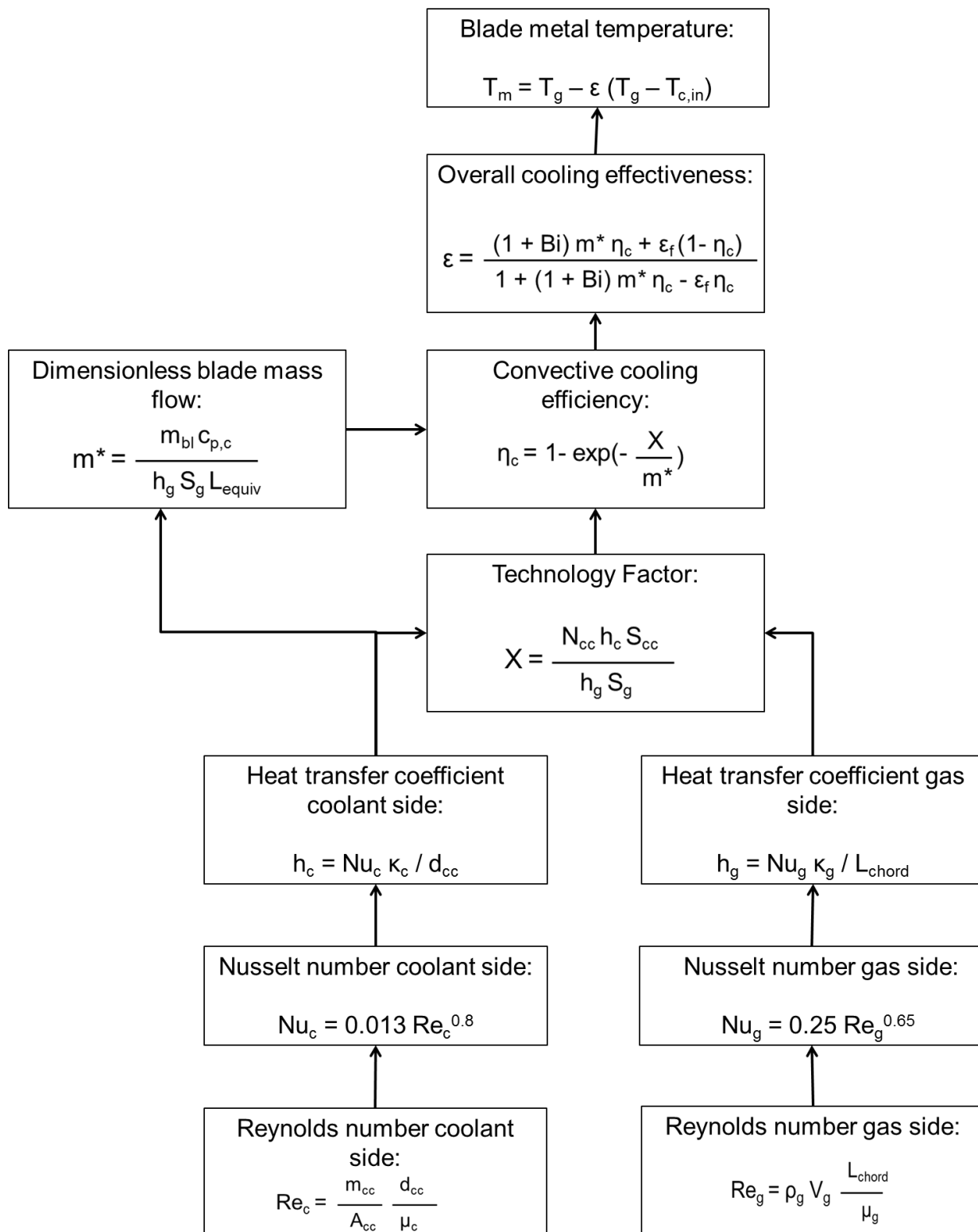


Figure 4-42: Blade metal temperature calculation flow chart **(14)**

Where,

A_{cc}	blade cooling channel cross sectional area
d_{cc}	blade cooling channel diameter
L_{chord}	blade chord length
m_{bl}	cooling air mass flow per blade
m_{cc}	cooling air mass flow per cooling channel
μ_c	hot gas viscosity
μ_g	cooling air viscosity
ρ_g	hot gas density
V_g	hot gas velocity
k_c	thermal conductivity cooling air side
k_g	thermal conductivity hot gas side
N_{cc}	number of blade cooling channels
S_{cc}	blade cooling channel perimeter
S_g	blade external perimeter
L_{equiv}	equivalent length of blade cooling channels
T_g	hot gas temperature
$T_{c,in}$	cooling air inlet temperature
Bi	Biot number
ϵ_f	film cooling effectiveness
$C_{p,c}$	cooling air specific heat

4.7 Blade Stress Calculation

Theory from paragraph 2.4 is used for the calculation of LPT blade stress. The technical description of the blade (paragraph 3.2.2) and in particular the geometric characteristics that are listed in Table 3-3 provide the necessary input to stress calculations. In contrast to the blade cooling calculations of the previous paragraph, stress calculations are largely based on blade's geometrical data, so it can be assumed that uncertainty is limited, since the geometrical characteristics of the blade can be accurately measured. It has to be stated though, that momentum and gas bending stress calculations are derived by quantities that are subjected to uncertainty, for example hot gas velocity and mass flow. But even if these properties take extreme values their contribution to the total stress was found to be less than 1%, so they can be omitted without a compromise to method's validity.

However, it was decided that Monte Carlo is still employed in the stress calculation in order to capture uncertainties in relation to materials isotropic properties and density, given also the fact that the stress calculation refers to a total number of 88 1st stage blades. Hence, the approach in this paragraph is different and Monte Carlo is applied on the analytical calculation result, which is shown in Table 4-10, as opposed to blade cooling calculation where Monte Carlo was applied on the constituent quantities of the algorithm.

Conclusively, following calculations of gas bending, momentum and centrifugal stress, it was found that centrifugal stress accounts for more than 99% of the total stress that the blade is subjected to, fact which is in agreement with available references.

Table 4-10: Monte Carlo uncertainty over blade stress calculation

Property	Distribution	Mean Value	Standard Deviation
Blade maximum stress (MPa)	Normal	115	0.05

The methodology that was followed is based on the works of Cookson (16), Laskaridis (17) and Eshati (15) and it is described in detail in the paragraphs below.

Momentum bending stress is generated as hot gas traverses blades and it incurs a difference in gas velocity between inlet and outlet of the blade flow domain (Figure 4-44). This difference in velocities results in momentum forces acting on blade sections in both axial and tangential directions. The describing equations of these forces are (17):

$$FV_{ax, sc(i)} = (m_{sc(i)} A_{sc(i)} \Delta V_{ax}) / N_{bl} \quad (4-21)$$

$$FV_{tn, sc(i)} = (m_{sc(i)} A_{sc(i)} \Delta V_{tn}) / N_{bl} \quad (4-22)$$

Where,

$FV_{ax, sc(i)}$	force on blade section i due to axial velocity difference
$FV_{tn, sc(i)}$	force on blade section due to tangential velocity difference
$m_{sc(i)}$	specific mass flow per section area i (calculated from the total mass flow in the area divided by blade annulus sections)
$A_{sc(i)}$	annulus blade section area
ΔV_{ax}	axial velocities difference across blade ($=V_{in, ax} - V_{out, ax}$, Figure 4-44)
ΔV_{tn}	tangential velocities difference across blade ($=V_{in, tn} - V_{out, tn}$, Figure 4-44)
N_{bl}	number of blades
$sc(i)$	blade section i (i = 1,2,3,4, Figure 4-43)

From axial and tangential forces, momentum bending moments can be calculated with the following equations:

$$MV_{ax} = \sum (FV_{ax, sc(i)} d_{cg(i)}) \quad (4-23)$$

$$MV_{tn} = \sum (FV_{tn, sc(i)} d_{cg(i)}) \quad (4-24)$$

Where,

MV_{ax}	bending moments due to axial velocity difference
MV_{tn}	bending moments due to tangential velocity difference
$d_{cg(i)}$	distance of blade section's centre of gravity to blade root datum (Figure 4-45)
$cg(i)$	blade section i center of gravity ($i = 1,2,3,4$, Figure 4-45)

Gas bending stress is generated by moment, which is produced by change in static pressure, as hot gases pass across blade(s). Notional division of blade into sections helps also with the calculation of relevant forces and moments. Hence, equation for applied force is (16):

$$F_{P, sc(i)} = (A_{sc(i)} \Delta P_{s, sc(i)}) / N_{bl} \quad (4-25)$$

And for bending moment:

$$MP_s = \sum (F_{P, sc(i)} d_{cg(i)}) \quad (4-26)$$

Where,

$F_{P, sc(i)}$	force on blade section i due to static pressure difference across blade
$A_{sc(i)}$	annulus blade section area
$\Delta P_{s, sc(i)}$	static pressure difference across blade section i
N_{bl}	number of blades

MP_s	bending moments due to static pressure difference across blade sections
$sc(i)$	blade section i ($i = 1,2,3,4$,
$d_{cg(i)}$	distance of blade section's centre of gravity to blade root datum section (Figure 4-45)

Static pressure values upstream and downstream of the blade are results from engine's performance model in section 4.2.

Centrifugal stress is stress applied to the blade due to its mass and to the fact that it is rotated with engine's rotational speed. Centrifugal force that each blade's section is subjected to is given by the below formula (17):

$$F_{C, sc(i)} = \rho_{bl} A_{c, sc(i)} H_{sc(i)} \omega^2 R_{cg(i)} \quad (4-27)$$

Where,

$F_{C, sc(i)}$	centrifugal force applied on section i center of gravity
ρ_{bl}	blade material density
$A_{c, sc(i)}$	blade average cross sectional area between top and bottom areas of section i
$H_{sc(i)}$	blade section height (Figure 4-46)
ω	angular speed
$R_{cg(i)}$	radius between engine's rotating axis and section's center of gravity (Figure 4-46)
$sc(i)$	blade section i ($i = 1,2,3,4$)

Knowing centrifugal forces, total direct centrifugal stress can be calculated from the sum of all centrifugal forces over the cross section area of root datum section (RDS). The equation is:

$$\sigma_{F_{c, sc}} = \sum F_{C, sc(i)} / A_{c, sc(RDS)} \quad (4-28)$$

Where,

$\sum F_{C, sc(i)}$ sum of centrifugal forces at each blade section

$A_{C, sc(RDS)}$ cross blade section area at root datum

Following guidance from Cookson (16), it is assumed that axial velocity remains constant along blade's span, while forces on blade act on the centre of gravities of each blade section.

$$M_{xx} = (MP_S + MV_{ax}) \sin\theta + MV_{tn} \cos\theta \quad (4-29)$$

$$M_{yy} = (MP_S + MV_{ax}) \cos\theta + MV_{tn} \sin\theta \quad (4-30)$$

$$\sigma_M = (M_{xx} I_{min}) Y + (M_{yy} I_{max}) X \quad (4-31)$$

Where,

M_{xx} bending moment

M_{yy} bending moment

θ stager angle (Figure 4-47)

I_{min} moment of inertia minimum value

I_{max} moment of inertia maximum value

X distance from X axis (Figure 4-47)

Y distance from Y axis (Figure 4-47)

Finally, **total stress** is calculated from formula:

$$\sigma_{tot} = \sigma_{Fc, sc} + \sigma_M \quad (4-32)$$

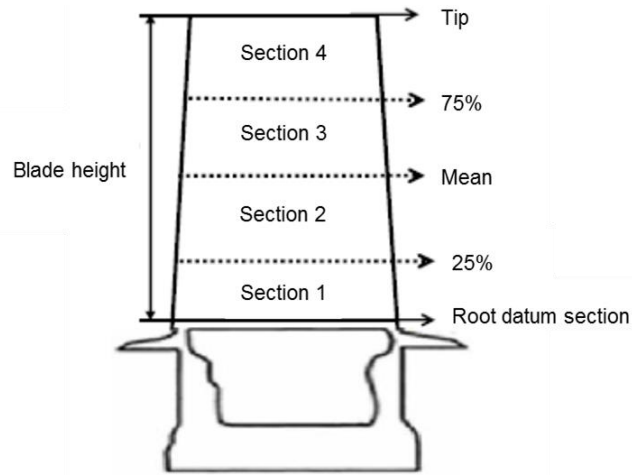


Figure 4-43: Typical blade sections (15)

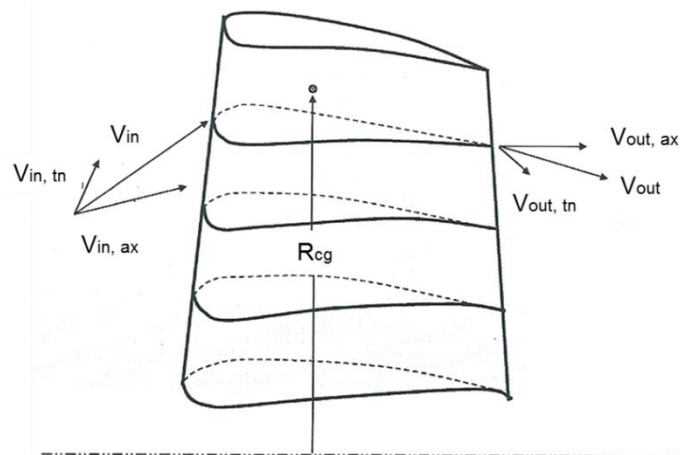


Figure 4-44: Hot gas inlet and outlet velocities (17)

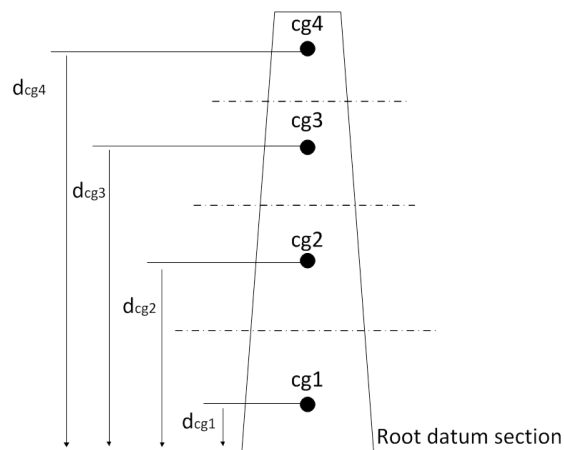


Figure 4-45: Blade sections center of gravity distances to Root Datum Section (16)

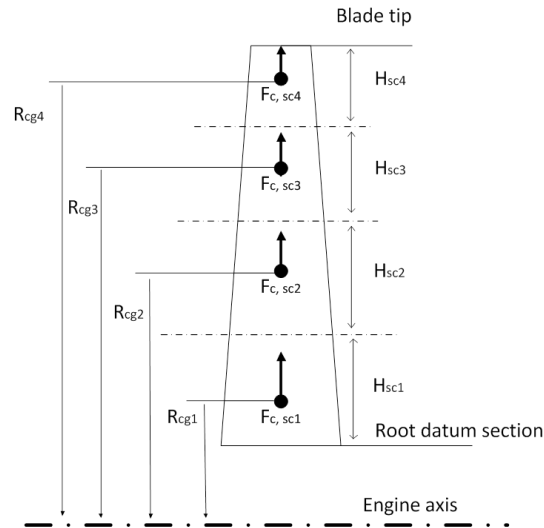


Figure 4-46: Calculation of blade centrifugal stresses (16)

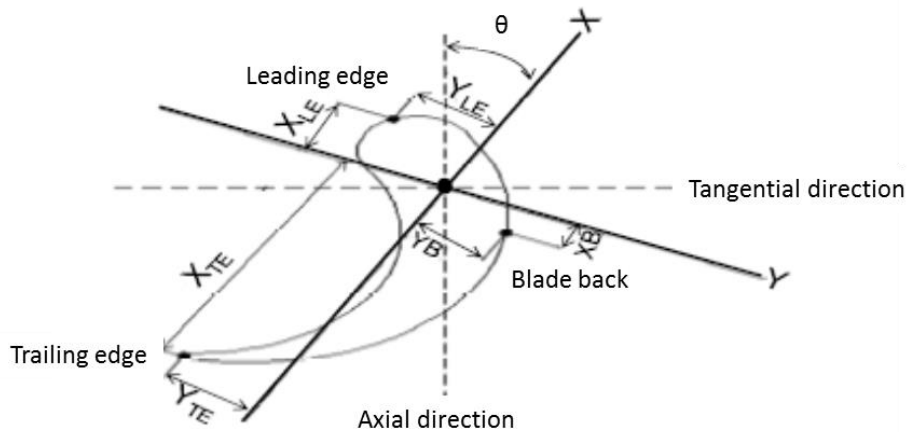


Figure 4-47: Schematic of maximum stress calculation (16)

4.8 Blade Lifting Calculation

The starting point for lifing calculations is Figure 3-6 in paragraph 3.2.2, where historical data from tests and experiments are illustrated, showing the combination of creep conditions (stress, time, metal temperature) in which CMSX4 specimen fail (23). As it was discussed in the same paragraph, the initial population of 108 test results was possible to be narrowed down to a significantly smaller number of test data points, following the results from blade stress calculation in section 4.7 (Figure 3-7).

In addition, sections 4.6 and 4.7 have provided results of operational blade metal temperature and maximum blade stress value. As for both calculations of blade metal temperature and stress Monte Carlo was used, 8000 points were produced capturing all the possible variations of temperature and stress that the blade is subjected to. These points with the use of Larson-Miller Parameter equation and the time interval for safe life (32000 hours) provided by the OEM, formed the population of *actual operational points* and were plotted on the same graph with the historical points.

As a comparison between two populations of 8000 actual points and a few only test data points is statistically unpaired, a similar Monte Carlo simulation was applied also to the historical test data points. Statistical processing of the limited number of historical test data points provided with mean and typical variation values (Table 4-11), while also linear regression points and their respective line was drawn (Figure 4-48). Monte Carlo was applied on the mean and typical variation point of the initially limited test data population and a much more representative population of 8000 points was produced (Figure 4-48).

This was an important step, as the use of a limited number of historical test data points could lead to inaccurate and constraintly determinist approach when compared with large populations of actual operational points. This is even more critical as limited knowledge over the experiment conditions are known and available, so a relatively high uncertainty should be assumed.

As the large population of Monte Carlo points provided with confidence that limitations and uncertainties over the use of analytical methods for blade metal temperature and blade stress were overcome effectively, similarly the adoption of the same high number for test data points provided with confidence that uncertainty over experiment's conditions were captured. By this way, two equal in number populations were compared and was made possible to further process them statistically.

Table 4-11: Mean value and typical variation for limited number of Historic Test Data Points

Initial number of Historic Test Data points (figure: 3.7)	Limited number of Historic Test Data points (within stress criteria)	Mean Value of Limited Historic Test Data points		Typical Variation of Limited Number of Test Data points	
		Stress (MPa)	LMP	Stress (MPa)	LMP
108	35	114.321	30.7825	16.600	0.7238

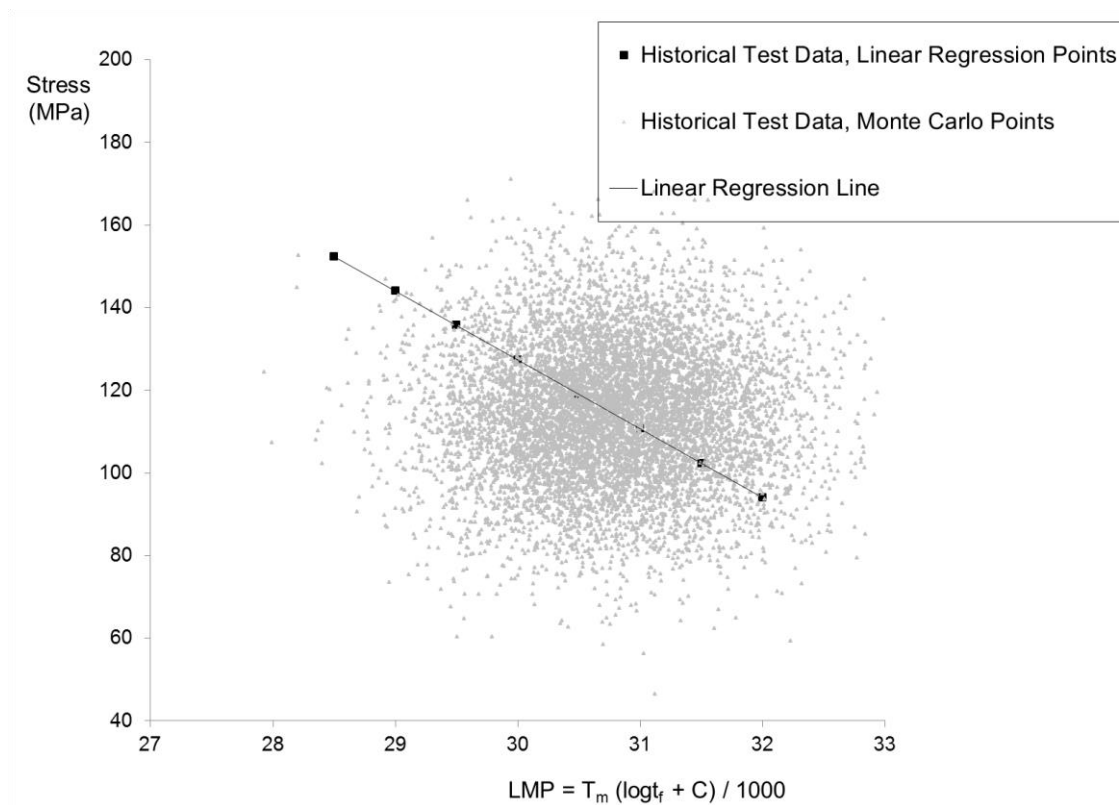


Figure 4-48: Generation of Monte Carlo points for CMSX4 Larson-Miller parameter Historic Test Data Points

4.8.1 Operational Actual Points and Degraded Cooling Air System Case

In Figure 4-48, test data points produced by Monte Carlo were mapped on the Larson-Miller Parameter graph. Next step was the mapping of the actual operational points. Those were firstly produced for a clear case, namely with no cooling air system degradation to be considered. The development of a

degraded cooling air system case followed, where an increase of cooling air temperature of +15K was taken into account in the blade metal temperature calculation. Table 4-12 lists the increase of blade metal temperature following a stepped increase in cooling air temperature.

Figure 4-49 shows the interaction between the two populations of actual operating points and test data points. Both populations count 8000 points each. The linear regression points from historical data and their respective line were shown as useful statistical reference. The mapping of all the points, which were generated by MC, on the LMP graph was used as a method. By this way, the visual effect of the cooling air system degradation and the incurred closeness of the two populations was more obvious. In Figure 4-50, the actual operating points are subjected to increased cooling temperature of +15K. Since all other creep parameters (stress, time) remain the same, the population of actual points shifts to the right of the LMP graph, coming closer to the test data points and increasing the overlapping with this population. Visually evident, it can be said that the blade, and consequently, the engine now runs with increased risk of creep failure, despite controllable and observable operating parameters such as load, rotational speed, firing temperature and exhaust temperature remain unaltered and phenomenically the engine operates in a safe mode.

Table 4-12: Impact on blade metal temperature of cooling air temperature increase

Cooling Air Temperature (K)	Hot Gas Temperature (K)	Blade Metal Temperature (K)
610.938	1783.150	1048.232
615.938	1783.150	1051.872
620.938	1783.150	1055.492
625.938	1783.150	1059.092

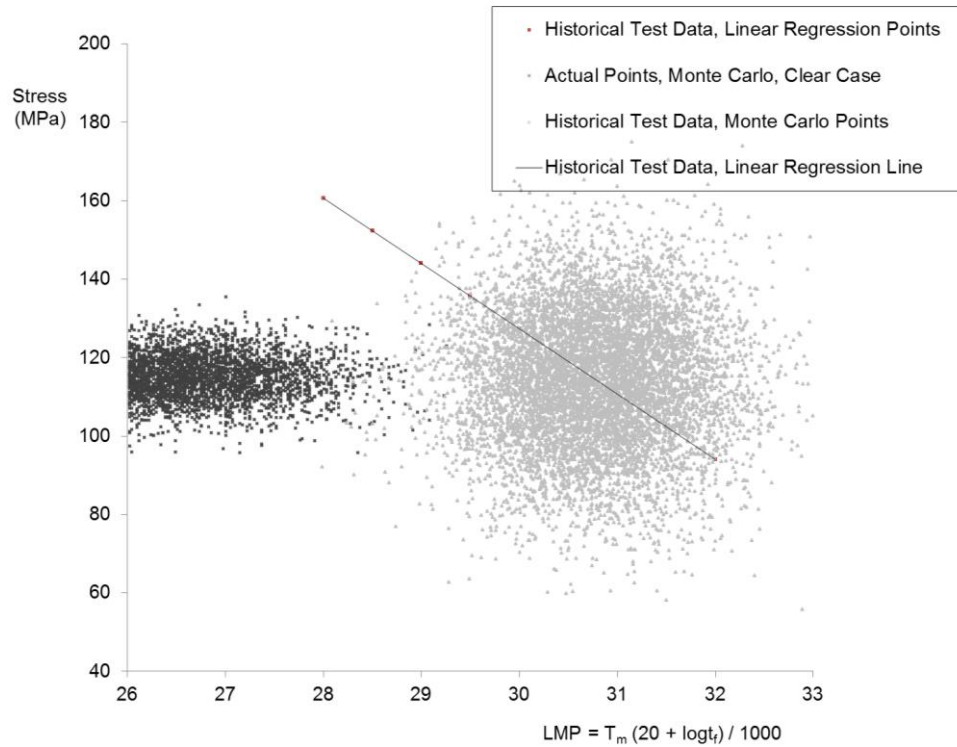


Figure 4-49: Larson-Miller Parameter graph for Actual and Historic Data Points, Clear Case

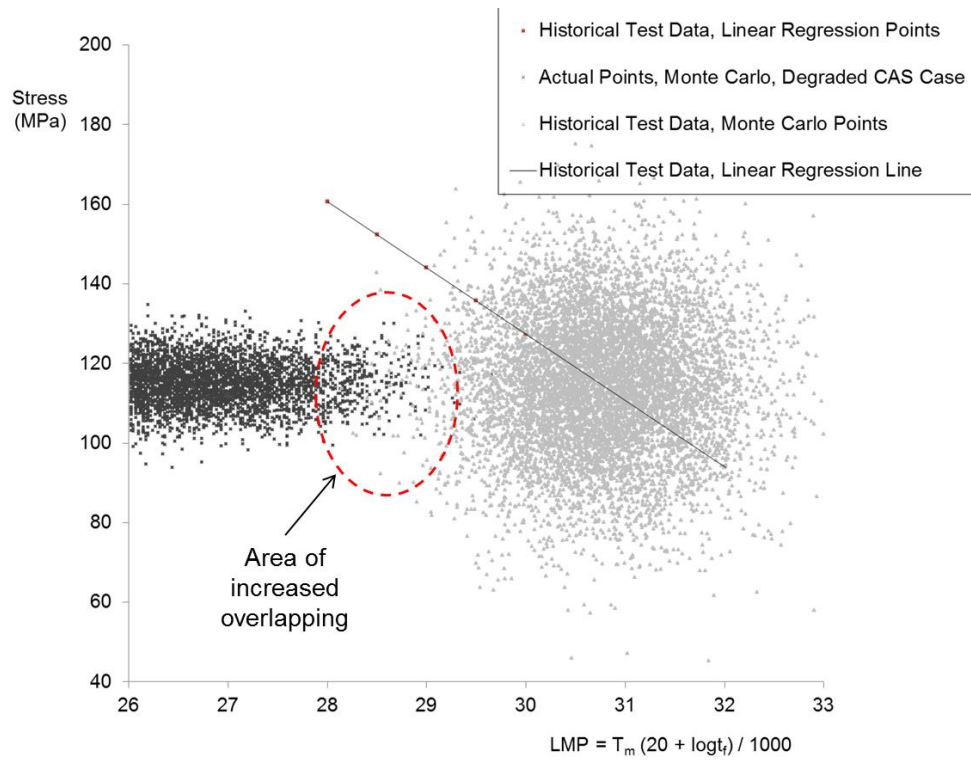


Figure 4-50: Larson-Miller Parameter graph for Actual and Historic Data Points, Degraded Cooling Air System Case

4.8.2 Use of Loading-Strength Diagram

Figure 4-50 illustrates the shifting of actual points to the right side of the Larson-Miller graph and the consequent increase of the overlapping area between the two populations. To quantify the visual impression of the overlapping area increase, the loading-strength diagram methodology was used.

Loading-strength (L-S) diagrams are frequently used in reliability engineering statistics as they can apply on large populations and provide quantified outcomes. Reference (77) provides details on the use of L-S diagrams and in particular on the calculation of overlapping areas. Overlapping areas results can be converted to a percentage of reliability with the use of relevant tables of statistics (77). Figure 4-51 illustrates a typical loading-strength diagram, where loading and strength curves are normal distributions.

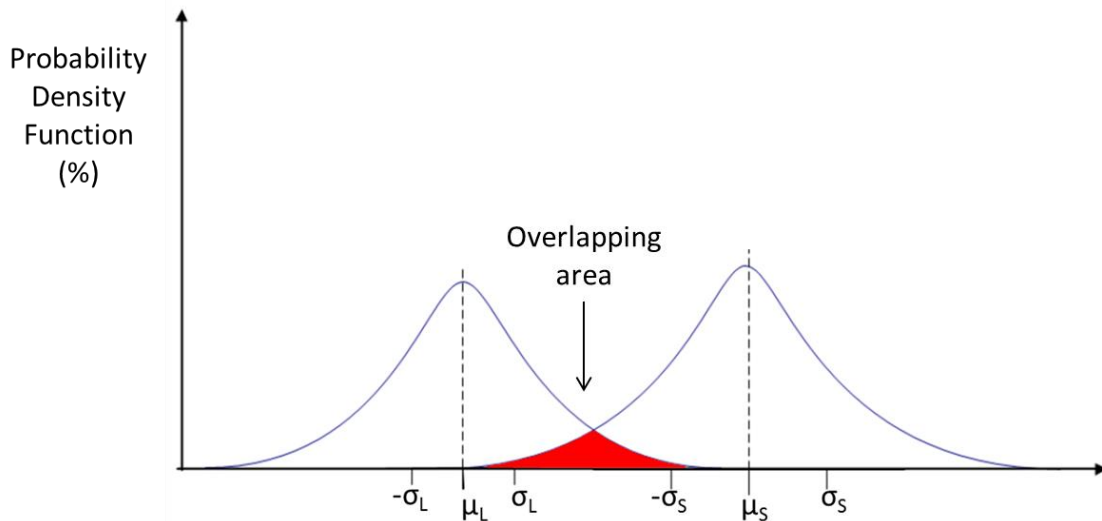


Figure 4-51: Typical loading-strength diagram

The distance between the two distributions is also called Safety Margin (SM) and it can be calculated from the equation (77):

$$SM = \frac{\mu_S - \mu_L}{\sqrt{(\sigma_S^2 + \sigma_L^2)}} \quad (4-33)$$

where,

SM	safety margin
μ_S	mean strength value
μ_L	mean loading value
σ_S	typical strength variation
σ_L	typical loading variation

Clearly the larger the distance between mean strength μ_S value and mean loading μ_L value, the more reliable a component is. Similar observation applies on the profile of the curves, as distributions with low typical variation values σ_S and σ_L do not spread-out considerably and their in-between overlapping area is small. In the hypothetical case where there is no overlapping area between two distributions at all, then a component can be considered as 100% reliable and never expected to fail. In practice, a very reliable component (>99.99%) appears to have an overlapping area of >3.69 (dimensionless number), ref. (77).

On applying loading-strength diagrams on Larson-Miller graphs, there is an additional layer of complexity that needs to be considered. Larson-Miller are 2-entry graphs with stress values in the vertical axis and LMP values in the horizontal axis. Both stress and LMP are characterised by their individual mean and typical distribution values, but it is the combination of stress and LMP value that defines a component's safety margin or in other words the distance between loading and strength. For example a blade can fail while exposed to low stress values if temperature or time are high enough to move the operational point to the right side of the graph. Separate loading-strength diagrams for stress and LMP do not provide a full answer.

This challenge was overcome by introducing the *product* of stress values multiplied by LMP values, as expressed by the following equation:

$$\text{Product} = (\text{stress}) \times (\text{LMP}) \quad (4-34)$$

Two new populations were formed by this way, one with the actual operating products and one with the historic data results products. The two new populations of points were statistically processed, so mean value μ_L and typical variation σ_L were derived for the actual operating products, and μ_S and σ_S for the historic data products.

Safety Margin was calculated and it firstly referred to clear case as no cooling air system degradation was considered. The degradation conditions were applied as the next step, and the increase in cooling air temperature incurred an expected increase in overlapping area. This is reflected in a decrease of Safety Margin and consequently in decrease of component's reliability percentage (Table 4-13). As a conclusion of the use of loading strength diagram method, it can be said that the LPT blade is 1.09% less reliable since being exposed to increased cooling air temperature.

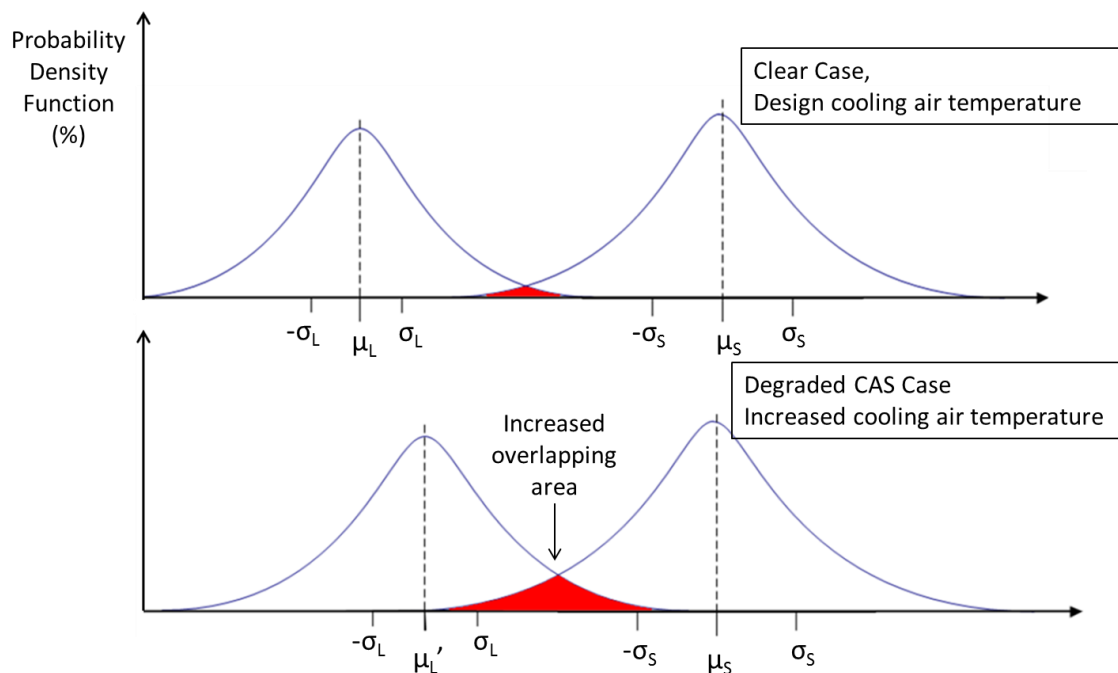


Figure 4-52: Increase of overlapping area due to blade metal temperature increase

Where,

- μ_L mean value actual operational products, clear case
- μ_L' mean value actual operational products, degraded cooling air system case
- μ_s mean value, historical test data products
- σ_L typical variation, actual operational products
- σ_s typical variation, historical test data products

Table 4-13: Safety Margin and reliability for clear case and degraded cooling air system case

	Safety Margin	Reliability
Clear Case	2.9095	99.82%
Degraded Cooling Air System Case	2.2397	98.73%

4.8.3 Methods to Manage Increased Lifting Risk

The discussion in the above paragraph concluded that the degradation of engine's cooling system incurs a higher risk of creep failure to the engine, which was quantified with the use of a loading-strength diagram and the calculation of its overlapping area.

It has to be stated that the use of loading-strength diagrams on Larson-Miller graphs has not been referred in any other relevant research works. Hence, and despite the fact that the above described approach lies on sound statistical methodology, the proposed in this work reaction to and management of the increased creep failure risk follows methods where the increase in blade metal temperature (Table 4-12) is the predominant alarming indicator, rather than the decrease in reliability percentage as shown in Table 4-13.

Historically in similar creep related issues, three methods are used to manage any risk. The methods derive from the key constituent factors of any creep correlation, namely stress-time-temperature, and are discussed in some details in the following paragraphs.

4.8.3.1 Reducing Stress Levels

Section 4.7 went through the principles and steps of blade stress calculation. As it was concluded, that centrifugal stress dwarfs any other type of stress, as it accounts for more than 99% of the total stress levels that a turbine blade is subjected to. The calculation proved that the remaining applicable types of stresses, which are gas bending stress and momentum stress, account for less than 1% of the total stress, so they can safely omitted from any further calculations. It has to be stated that these conclusions are in agreement with relevant references (16), (17), (12).

This is an important point, as gas bending and momentum stresses derive from mass flow and velocity calculations on the blade. Following the above argument, the option to reduce stress by reducing, for example, engine's mass flow and aim by this way to remove the actual operational points in Figure 4-50 lower on the graph, is not applicable.

In contrast, centrifugal stress is not derived by mass flow or any other flow parameter in the engine. As explained in section 4.7, rotational speed is the only operational parameter in this calculation. So in theory, the only way to control dynamically the levels of centrifugal stress is by controlling, lowering in this case, the rotational speed of the engine. However, it has to be clarified that land-based gas turbines operate continuously and constantly synchronised to the electricity grid, and hence they are subjected to grid's frequency (50Hz). In directly coupled engines (no gear box between gas turbine and generator), this equates to rotational speed of 3000 rpm. As grid's frequency is often unstable the turbine-generator pair is called to operate usually within a band from 49.5 to 50.5Hz. However, it has to be repeated that the engine can never rotate at different to grid's frequency, but only to follow. Conclusively, and as opposed to

aero engines where active control of rotational speed is possible, land-based gas turbines do not have any means to control their rotational speed.

In a closing note, it is interesting that the impact of momentum and gas bending stress do not really relieve the maximum stress, so it is a safe assumption that an increase in mass flow through the engine¹⁷, will not equally increase stress levels alarmingly. Though it has not been considered in this work, it is proposed in Future Work's section 6.3, that increase in mass flow and the resulting increase in engine's output, can be considered as a mitigation measure when engine's derate to manage lifing risks, in the form of firing temperature decrease, is applied.

4.8.3.2 Reducing Operational Time Intervals

In the above paragraph's discussion, stress was ruled-out as a controllable factor to manage creep-induced lifing risk. The other two parameters that were explored were time and temperature. Following Larson-Miller equation,

$$LMP = (T_m/1000)(\log t_f + C) \quad (2-9)$$

when blade metal temperature T_m increases, and given the fact that LMP and C constants retain the same value, the time to failure t_f has to be reduced, in order the operational point to remain in an operationally safe and remoted from the test data points area (Figure 4-50). Figure 4-53 illustrates the necessary reduction in blade's operational life that the owner/operator has to adopt, when the mean blade metal temperature increases, while C and LMP constants remain unaltered. It is noticeable, that small increments in metal temperature require substantially large reductions in operational life and this is critical for any risk management approach. Time to failure t_f (or useful operational time) is clearly not an engine's operational characteristic such as blade's metal temperature or stress, but rather an engine's technological feature with major impact on maintenance planning. Moreover, as it is discussed below in paragraph 5.2.2, outage planning is a key activity for every operator which

¹⁷ Through further opening of V(I)GV, when applicable.

impacts on engine's affordability and general economics around it. Comment on outage costs below paragraph 5.2.2 provides some quantified data in relation to outage moves and why these events are so expensive and difficult to move around. It has to be stated that this is in contrast to aero-engines world, where their size and weight allow for much more flexible approach when it comes to overhauls and major repairs, with options such on-wing repair or whole engine replacements etc. However, the time element in lifing risks remain a quite important and controllable feature, as it is in detail discussed in paragraph 5.4.2. The explicit relation between blade metal temperature and life is extensively used in Miner's equation to optimise commercially engine's operation while at the same time lifing risk is effectively managed.



Figure 4-53: Remaining life of CMSX4 over increasing metal temperature

4.8.3.3 Decreasing Blade Metal Temperature

Among the three so far proposed methods to manage lifing risk in a gas turbine, decreasing blade metal temperature is the most frequently applied method by operators and OEMs. It is predominantly achieved through control of firing temperature, which is a key operational parameter well-within, in most of the cases, the operator's sphere of influence, as opposed to, for example, engine's rotational speed. Firing temperature is also a relatively straight-forward

adjustment that can be performed manually or automatically through engine's control system¹⁸.

Reducing the heat burden on the blade through firing temperature can bring mean blade metal temperature down to nominal levels, as those have been calculated in section 4.6. This then is reflected by moving the μ_L' point to the left side of the loading-strength diagram and the cluster of actual operational points to the left side of Larson-Miller graph accordingly (Figure 4-50 and Figure 4-52).

In theory, blade metal temperature can be also decreased if cooling air temperature is decreased. In most gas turbines, aero-engines or heavy-duty ones, there is no mean to directly control cooling air temperature, as cooling air is sourced through compressor bleeds with no feature such as control valve or heat exchanger between the bleed point and the cooled component (blade, vane etc). In GT26, and as described in Appendix B, cooling air is externally cooled with the use of heat exchangers. However, cooling air temperature settings are not consider adjustable for reasons whose analysis is beyond the scope of this work, so this option remains inapplicable.

Reinstating cooling air system's degraded condition is another theoretical option. To repair/replace the pre-swirl nozzles cavity inner seals and restore, by this way, design cooling air temperature involves an open-engine maintenance event (outage). The scale and magnitude of these events in heavy duty gas turbines, and in particular GT26, is discussed in detail in paragraph 5.2.2, where it becomes apparent why these events take place strictly on a 4-5 years cycle.

Another important observation comes from Figure 4-54, where the different impact on blade's metal temperature between hot gas temperature and cooling air temperature can be inferred. To retain constant blade metal temperature, when cooling air temperature increases for example 10K, hot gas temperature and consequently firing temperature has to be decreased by almost 25K. This can be explained with blade heat transfer configuration in mind, as it was

¹⁸ In the case of GT26, the term which is used to describe engine's control system is Distributed Control System (DCS).

explained in section 2.3. Internally, blade's parent material is in direct contact with the flow of cooling air¹⁹. Externally, the blade is protected through layers of TBC and MCrAlY coatings, while also film cooling applies a thermal insulation effect. This is a critical observation, as decreasing firing temperature incurs lower output and, more importantly, lower efficiency. Table 5-4 in the Technoeconomic Analysis chapter cites the performance penalties in output and efficiency when firing temperature is decreased.

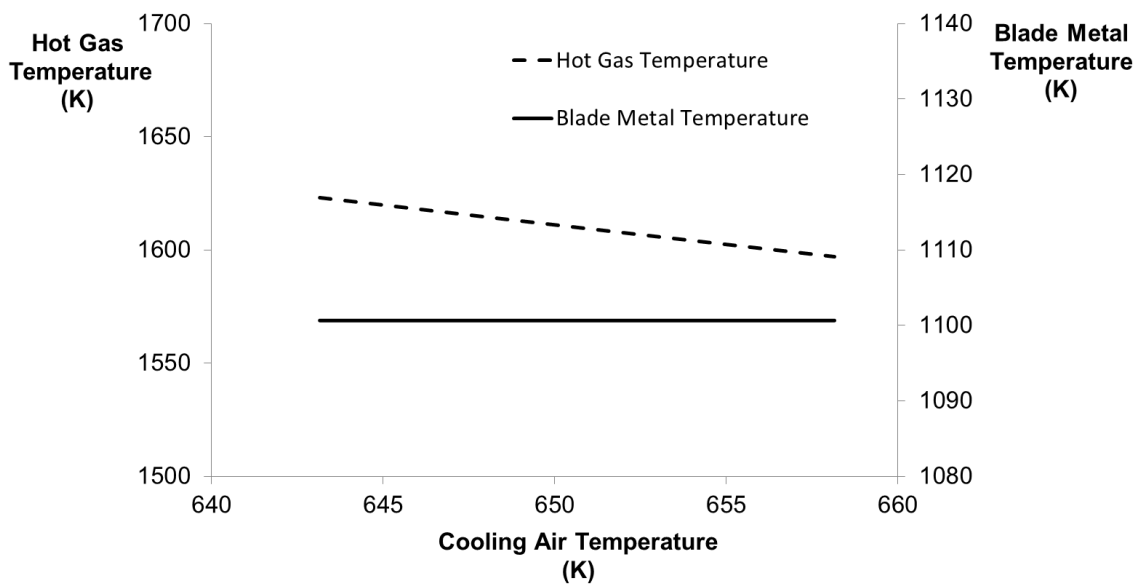


Figure 4-54: Decrease of hot gas temperature to maintain constant blade metal temperature, over increasing cooling air temperature

4.9 Summary

The chapter of methodology was expectably the longest of this thesis. Starting from the engine's performance model in section 4.2, GasTurb software was used for the development of GT26 model, while validation took place with Optigen, an in-house developed performance monitoring software. The model had twofold value, as it offered the necessary boundary conditions for blade stress and blade cooling calculations, but also, engine's output and efficiency

¹⁹ In GT26 LPT blades, no internal coatings are used.

were mapped against firing temperature and this is critically important for the technoeconomic analysis that follows in the next chapter.

In section 4.3, engine's cooling air system was resolved analytically. Critical non-dimensional numbers, as identified in theory's section 2.2, were calculated and showed similarities with relevant cases from the available literature. The end results showed a drop in cooling air's total relative temperature inside blade cooling passages, as it was expected, when cooling air's flow through pre-swirl nozzles cavity inlet seals increase.

In section 4.4, the CFD solution was developed. Firstly for the Test Case, as the purpose was to validate the CFD methodology with an available in the public domain experiment case. Following successful validation, the same principles and methodology was followed for the GT26 model. Results in GT26 model showed a drop in cooling air's total relative temperature inside blade cooling passages of more than 13K, when flow through pre-swirl nozzles cavity inlet seals increase from the design value of 2% up to 7%, percentage refers to inlet seals flow over pre-swirl nozzles flow. Comparison between analytical solution and CFD results proved that the analytical solution, though over-simplistic in approach, predicted sufficiently the order of magnitude as well as the correct polarity of the change (decrease) in cooling air total relative temperature.

In section 4.5, the principle of Monte Carlo simulation technique was discussed, as Monte Carlo was used for blade metal temperature and blade stress calculations. In addition, Monte Carlo is also used in the Technoeconomic Analysis chapter for electricity market and ambient conditions predictions.

In section 4.6, blade metal temperature was calculated with the use of Rubini's method, based on the theory from section 2.3. The analytical deterministic model was combined with Monte Carlo, so all the uncertainties were captured, A total of 8000 points were produced and mean blade metal temperature was calculated to be very close to 1048K, at design, base load and no degradation conditions. According to the degraded cooling air system scenario, increased cooling air temperatures were applied in steps, and the resulting increase in blade metal temperature was also calculated.

In section 4.7, blade stress was calculated with the use of theory from section 2.4. The results showed that gas bending and momentum induced stresses can be ignored, as they accounted for less than 1% of the maximum stress that the blade is subjected to. In other words, centrifugal stress dwarfed the other two types of stresses. Monte Carlo was used again to capture any uncertainty primarily around blade's density and material's isotropic properties. 8000 points were produced.

Finally, with input from blade cooling and blade stress results, lifing calculations took place in section 4.8. Material's historical test data points in Larson-Miller graph were used as the starting point. As they were finite in number, Monte Carlo was used to produce a population of 8000 points which statistically satisfied the conditions of test data points. On this graph, the actual points as those are mapped by known stress levels, blade metal temperature and OEM's creep life for the component. A clear case was mapped first, assuming no degradation in cooling air system. It was followed by a degraded cooling air system case, where the actual operational points shifted to the right of the graph since blade metal temperature increased, following the increase in cooling air temperature. The increase of overlapping area was quantified with the use of loading-strength diagram. Closing the section, methods to reduce lifing risk were discussed, concluding that controlling engine's firing temperature is the most effective mean to manage creep-induced lifing risk. However, the explicit relation between temperature and time in Larson-Miller's equation remains critical for lifing calculations, especially when combined with Miner's equation.

In the next chapter a techno-economic analysis follows. The objective is to provide the turbine's operator with a model that enables effective management of the increased lifing risk and at the same time commercially optimised operation of the engine.

5 TECHNO-ECONOMIC ANALYSIS AND REVENUE GENERATION PREDICTIVE MODEL

5.1 Introduction

A technoeconomic analysis is developed and discussed in this chapter. It follows the outcomes of the lifing model development. The chapter starts with a description of the electricity market in the UK, aiming to offer the necessary background information of the regulatory and commercial framework that today's gas turbines operate in. Most comments and observations apply to European but also USA markets. The overview of the electricity market includes a section on Original Equipment Manufacturers (OEMs), as well as an insight to outage costs. OEMs related costs in terms of hot gas path parts and services and outage costs are the two most important cost centres for gas turbine operators. Continuing, the concept of Spark Spread value is explained in detail as it is a key notion in the electricity market and also an expression of how utilities factor their costs in their generated-electricity dispatch strategy.

The generated revenue calculation algorithm is an important contribution of this project. The key parameters that define gas turbines output, volume and revenue are described in detail. More specifically, impact of ambient temperature, engine's degradation, reliability and life usage, as well as market conditions in the expression of Spark Spread prices are described. Most of the parameters are factored in the calculation algorithm through polynomials, though Spark Spread values and ambient temperature are captured with the use of Monte Carlo due to their stochastic and random nature.

Following the description of its critical parameters, the model is used to run different cases. Firstly a case with no degradation – Clear Case - is run so as expected revenue to be calculated and used as benchmark. Then cases with degraded cooling air system are explored. Author's terms of Constant Derate

(CD) and Dynamic Derate (DD) and then Deep Dynamic Derate (DDD) and Shallow Dynamic Derate (SDD) are introduced. A particular real-life scenario is adopted, where an engine with approximately 20,000 OHs is faced with a degraded cooling air system issue and the operator is called to decide on the strategy of the next 16 - 18 months of committed operation until the next major overhaul that offers opportunity for repair. The model offers the optimum position between lifing risk and commercial losses.

5.2 Electricity Market Overview

Facing a lifing risk on the engine, even if this involves high probability of catastrophic failure, is not the only critical consideration for operators. Another significant input in their business is the timing of the repair or failure. This input derives from the electricity market, as always having an engine out of service for inspection or overhaul, incurs loss of electricity generation and consequently loss of revenue.

Table 3-4 in paragraph 3.2.3 cites typical durations for GT26's maintenance events, and it can be assumed for other of the same frame size heavy-duty gas turbines. As table shows, major overhaul durations for heavy-duty gas turbines are, in most of the cases, in excess of 1 month. This is a considerable length of time for a commodity product, which cannot be stored and can only be sold when generated. The contrast is vivid with the aero-engines sector, where their scale and size allow for on-wing repairs or whole engine replacements within hours or few days.

Electricity market has changed and evolved a lot in the last two decades. 30 years ago, nuclear and coal power stations used to produce more than 80% of electricity supply. Since then and as part of a general decarbonisation policy in Europe and UK, coal power stations have started to give their place in combined cycle power plants, as the latter operate with significantly higher thermal efficiency. Moreover, in the last 10-15 years the development of renewables, primarily due to very favourable incentives and subsidising, has been exponential. In the last five years, percentage of electricity generation produced by renewables has increased by more than 14%, as Figure 5-1

illustrates. It is also noticeable that all the more European countries manage to achieve full days of generation powered only by wind, solar and hydro-generated electricity (78). The targets for most European countries for the next years are even more ambitious as energy from renewables must be at least 30% of total generated energy by 2030 in UK and Europe.

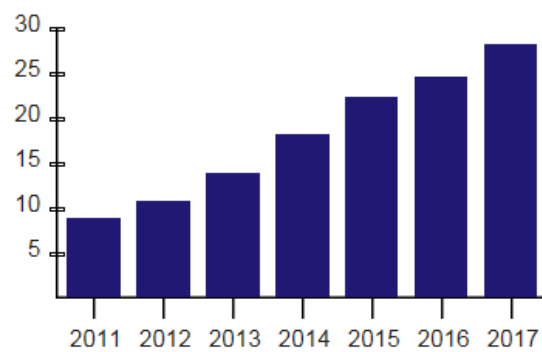


Figure 5-1: Percentage of renewables installed capacity in UK (79)

In terms of impact on generation profile, renewables intermittency to the grid brings new challenges to conventional producers. Modern combined cycle gas turbine (CCGTs) plants have now to play a more balancing than base-load role. Fast start-ups and rapid responses to grid's frequency instabilities, that are imposed by renewables and primarily wind turbines, are the main characteristics of CCGTs' operation. Between nuclear plant's base-load capacity and renewables' unpredictability and intermittency, CCGTs have to operate as the necessary buffer of the system.

Finally, electricity market is also characterised by seasonality, which is an expression of the different demand in electricity between summer and winter months. This is also reflected in pricing with winter electricity prices being significantly higher than summer's ones²⁰. Renewables increase had an even more accentuating effect on seasonality. Wind generation is difficult to be held responsible for this, as generally wind profile does not fit in predicted seasonality patterns. However, solar generation is on an increasing tendency

²⁰ Remark applies to UK and north-European countries. South Europe and Mediterranean countries exhibit the opposite seasonality with high demand and prices in summer and lower in winter.

for a number of years now in UK (Figure 5-2), while it has to be stated that this increase is much more impressive in north Europe and especially in Germany. Solar factor has accentuated further the difference between winter and summer Spark Spread prices. Figure 5-3 illustrates the typical seasonal variation of electricity pricing in UK.

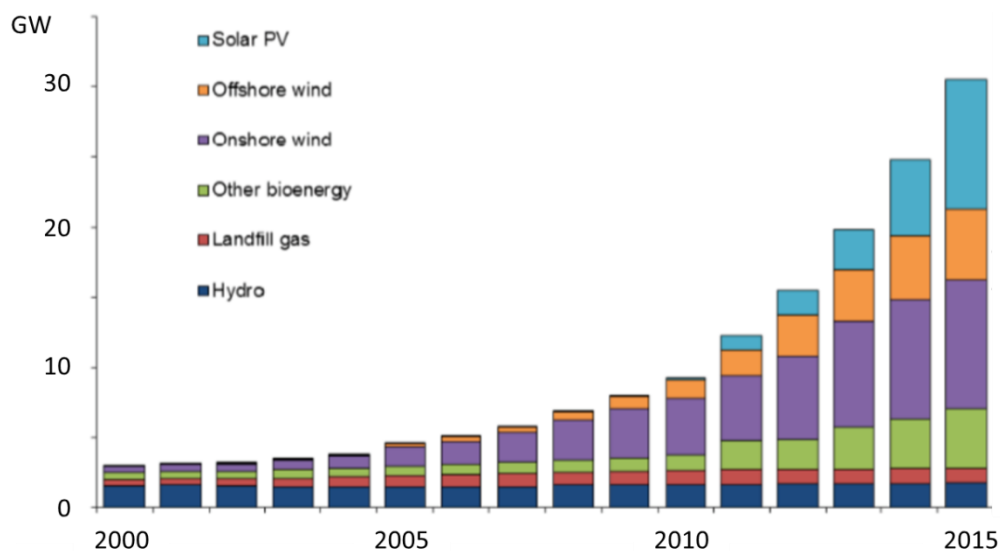


Figure 5-2: Renewables installed capacity in UK (79)

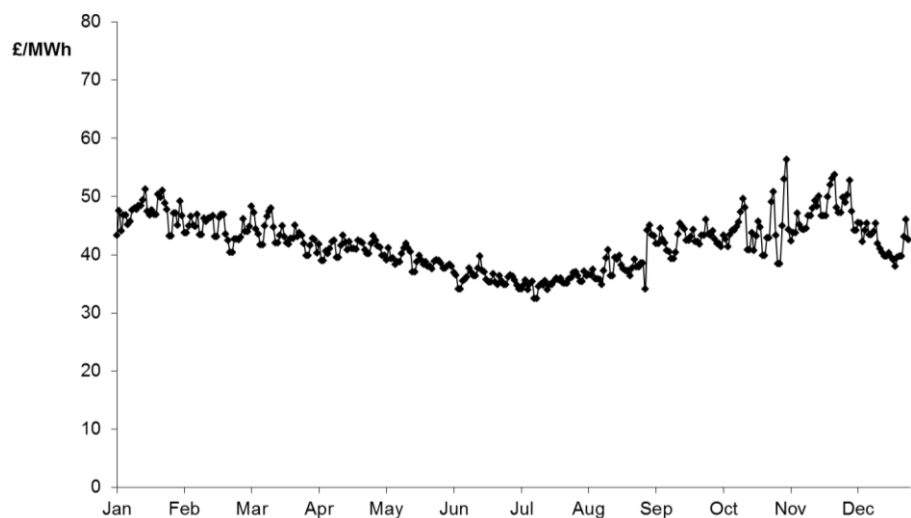


Figure 5-3: Typical seasonality on electricity prices, UK market (80)

5.2.1 Original Equipment Manufacture costs

Apart from the above-described changes in electricity market, another noticeable evolution is the all-the-more increasing dependency of operators - namely utility companies - on Original Equipment Manufacturers (OEMs). Historically, coal-fired power stations used to be operated and maintained largely through an in-house developed competency and capability approach. The relatively approachable technology of steam turbines (STs) and conventional boilers has allowed operators to manage their assets in a more independent and contractually relaxed way and to explore options for reverse engineering and improvements, with the reward being the suppression of OEM costs by this way. Things are different for combined cycle gas turbine plants. Gas turbine technology is substantially elevated in terms of complexity and sophistication, when compared with steam turbines and conventional boilers. It is also a technology that keeps evolving over the years, being primarily driven by advances in aero-engines. Typical example is the race among OEMs for higher firing temperatures, improved materials, improved performance etc. This is in contrast with steam turbines where very few things have changed in the last 30 to almost 40 years. Gas turbines technology has become an even more restricted-access knowledge domain and the technological barrier between OEMs and utilities has kept increasing.

This has unavoidably brought a different, more conservative, perception of risk. Gas turbines by their design are fragile pieces of equipment, where a seemingly humble cause - for example a foreign object damage (FOD) caused by a small stone - can incur disproportionately large damage. It is a remarkable fact that gas turbines are assets that attract very high premiums in the assets insurance market, as they considered to be of very high risk, since small failures can incur large losses.

Table 3-4 in paragraph 3.2.3 shows that a typical outage event for a gas turbine can be up to 35 days (or more) but it has to be clarified that this intervals refer to planned events. Planned events are defined as events where all spare parts have been delivered to site, working teams and experts have been mobilised

and arrived in time and most importantly the scope of work - for example extent of hot gas path parts replacement - has been pre-defined. In an unplanned event or worse in a catastrophic failure of a gas turbine - for example liberation of a turbine blade due to insufficient quality control during blade's casting process²¹ - the unplanned outage that is required will most probably last for some months depending on spare parts availability, as a large number of them are not "on-the-shelf" parts. Apart from repair costs, lost generation will also incur significant revenue losses to operator. As a result, the majority of utilities choose to bind themselves with OEMs in contractual service agreements. By this way, they share risk, mainly around hot gas path parts liability and spares availability, but from the other side they have to abide to quite conservative, OEM driven, lifing models.

In OEMs world things are not less exciting. The shift of electricity market described in paragraph 5.2, has brought a significant shift in engines' design approach. Gas turbines in CCGTs are not designed as base-load plants any more and more importantly, they will never probably be base-load plants again as very few energy analysts perceive these market changes as reversible. The intermittency described in paragraph 5.2 requires engines that can respond rapidly to grid fluctuations, as those are now generated by ambient conditions, primarily wind, and not so much by demand for electricity which it was and it is always a much more predictable factor. Modern heavy duty gas turbines, despite their size and mass of metal, must be able to start-up from cold standstill conditions to full load (thus in GT26 is approximately 290MW) in less than 30 minutes and respond to grid's frequency imbalances by increasing or decreasing load with rates of 50MW/min. These prerequisites that materials for HGP parts must cope with overwhelmingly aggressive thermal transients.

Adding to the mixture, environmental legislation becomes, justifiably it has to be said, stricter every 3 to 4 years. For CO₂ emissions things are more or less

²¹ Author's real life example from Pembroke Unit 41, December 2012

straight forward. Through a pan-european agreed CO₂ trading system²², policy makers provide incentives to operators and manufactures to run and design accordingly more efficient gas turbines and power plants. It can be said that in terms of improving efficiency, CCGTs are getting more efficient by a 1% every 2 to 3 years and a modern CCGT can run today with > ~ 61% thermal efficiency. With the way that the CO₂ trading system works, less efficient plants of older generation and technologies - for example 55% thermal efficiency or lower – are still allowed to run. They just have to bear and absorb the cost of lower efficiency (£/CO₂tn) in their operational (OPEX) costs. Operators are incentivised by this way to invest in more modern technology for their plants. However, things are more complicated and demanding with NOx emissions. NOx formation follows linearly firing temperatures with favourable temperatures being in the range of > 800°C. Current EU legislation asks for NOx emissions from CCGTs to be lower than 50ppm, while there are also cases (for example state of California in USA) where local environmental legislation asks for NOx emissions to be lower than 10ppm. What is more important is that NOx legislation has a very different approach, due to the fact that the threat to environment is very different, compared with CO₂ legislation. NOx legislation does not reward or punish in a financial context, but acts on a purely punitive basis. Breaches in NOx emission limits can result in CCGT's operational permit being revoked. The technological aspect of this piece of legislation is that gas turbines' combustor and burners are now significantly more complicated in order to meet these compliance requirements. This adds one more layer of complexity and fragility on modern gas turbines, which in turn it is reflected in higher costs for the end user. It can also be said that for low NOx technologies - for example DLN – most of heavy-duty gas turbine manufactures²³ are on their own to drive research in this difficult field. NOx legislative limitations are substantially more relaxed in aviation for obvious safety reasons, so there is no strong incentive for aero-engine manufactures to invest heavily on related R&D programmes.

²² For Europe reflected through European Union Emissions Trading System (EUETS)

²³ Comment applies on Siemens, Alstom and Mitsubishi-Hitachi

Finally, fuel variability is something that heavy-duty gas turbine OEMs have to accommodate within their engines' combustors and burners, as modern engines are called nowadays to burn efficiently and reliably, fuels that span from liquid oil to natural gas from different sources all over the world. Today's gas turbines have to operate with Wobbe Index²⁴ of more than 10MJ/Nm³, while only 10 years ago, specified WI by manufacturers was 8 MJ/Nm³. Fracking method for gas exploitation and use of H₂ as fuel are another two challenges for combustion technology that are coming in the very near future.

All the above, as it is easily understood, are reflected into capital costs (CapEx) of gas turbines, their noble parts but also OEMs after-sales services. Moreover, in heavy-duty GT market for engines larger than 200MW (Frame 9), there are only three major manufacturers, GE²⁵, Siemens and Mitsubishi-Hitachi. Having only three manufacturers for a globally spread market is a clear indicator of how difficult and R&D intensive is to design, manufacture, sell and maintain heavy-duty gas turbines. Absence of alternative options, in terms of manufacturers, suppliers and (reliable) reverse engineering firms, keeps unavoidably the costs quite high in all aspects of gas turbines lifecycle - purchase, operate, maintain, overhaul. A serious impact of the above is the cost of services but mainly cost of hot gas path parts that OEMs put on the end-user. In a typical modern combined cycle power plant, cost of HGP parts and cost of gas turbine overhauls are by far the biggest operational expenses for the business. Paragraph 5.3.4 provides indicative costs of a large frame heavy-duty gas turbine set of hot gas path parts and how these costs are factored in the revenue generation algorithm.

5.2.2 Outage costs

Electricity market was explained in some details in section 5.2, while paragraph 5.2.1 went through the costs incurred by OEMs due to the nature of their

²⁴ Wobbe Index (WI) is a metric used to express fuel variability; definition can be found in (83).

²⁵ Inclusive of Alstom's heavy duty turbines.

technology and business. This paragraph discusses some other, equally important, drivers behind the cost of a typical outage for an operator.

As a starting point, Figure 5-4 illustrates the significant difference in cost when an outage event moves from winter to summer months. It has to be repeated that the case refers to a C Inspection event with typical duration of 30-35 days. According to Figure 5-4, if an outage event has to commence earlier, because of, for example, lifting risk concerns, then a shift of the starting date from June to February will incur additional costs to the operator of almost £1.2m. The main driver behind this cost is the seasonality pattern of electricity prices. For almost all gas turbine operators the seasonal window from November to March is an extremely profitable period, which can equally turn to be an extremely costly period if an unplanned event takes place within it. Summer months window is the exclusive period that power stations chose to run their outage events²⁶.

In terms of scale and magnitude, gas turbine outage events, which involve full engine overhauls, epitomize the definition of complex, lengthy and expensive projects. For all large frame gas turbines, these events are planned to take place in a cycle of every 4 to 5 years. Their duration can vary from 1 to 2 months, while the number of personnel involved can be up to some hundreds of engineers, technicians, specialists etc. The planning of a major overhaul in a large gas turbine is an activity which starts 2 to 3 years before the event itself. It involves ordering of spare parts, which in the case of hot gas path parts or engine's structural parts, for example turbine's rotor or casing, they have delivery times in excess of 18 months. Booking of the specialised personnel is another task that has to take place well in advance. The outage commencing date is usually agreed among all stakeholders more than 12 months in advance of the event. After that point, any deviation from the agreed date is in most of the cases linked with expensive contractual clauses and penalties.

The arguments discussed above paint the picture of one of the most important events in the lifecycle of a heavy-duty gas turbine. As opposed to aero-engines,

²⁶ Comment applies to UK and north Europe countries.

major overhauls are events that are planned to take place only 4 or 5 times in the life of the engine. Any unplanned open-engine events are almost in all cases disastrously expensive events, hence most operators try to avoid these eventualities at all costs.

Following all the above points, it becomes evident that any operational issues which pose mid- to long-term risk have to be managed with respect to existing, already-in-place outage planning. Degraded cooling air system and the risks that poses is a typical example of an issue that does not necessitate a forced opening of the engine in an interim basis and out of the sequence of planned events. However it has to be managed carefully and effectively as there is lying down risk of a catastrophic failure and unplanned outage.

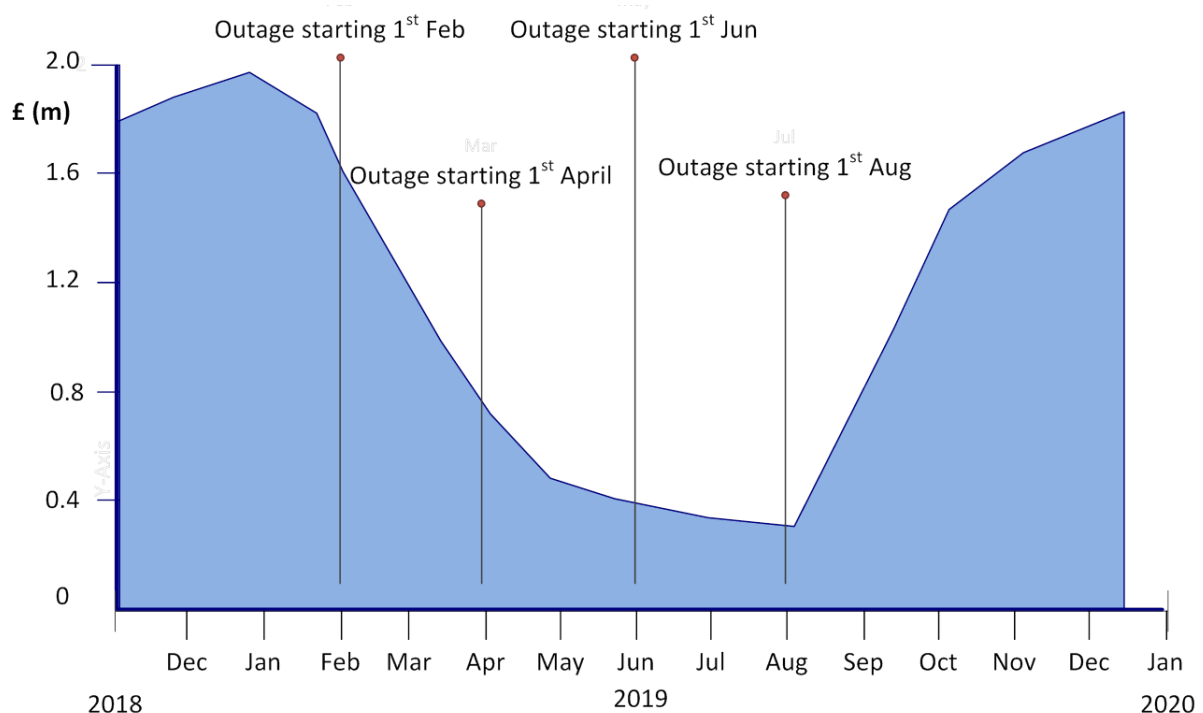


Figure 5-4: Cost of outage due to lost generation through a typical year **(80)**

5.2.3 Spark Spread Value

While Figure 5-3 provides indicative electricity prices (£/MWh) over a typical year, for electricity revenue calculations a different metric is used, which is

termed Spark Spread^{27,28}. Electricity price is a purely market metric, which follows demand and supply mechanisms in the electricity market and it does not refer to fuel costs. In contrast, Spark Spread incorporates two additional critical factors, which are fuel gas price and engine's thermal efficiency. In this way, Spark Spread prices can be used directly for electricity income and revenue generation calculations.

In Figure 5-5, the Spark Spread value calculation equation is illustrated:

$$\text{Spark Spread Value (}\text{£/MWh}\text{)} = \left[\text{Electricity Price (}\text{£/MWh}\text{)} \right] - \frac{\left[\text{Natural Gas Price (}\text{£/MWh}\text{)} \right]}{\left[\text{GT Efficiency (\%)} \right]}$$

Figure 5-5: Spark Spread value calculation

Spark Spread links the two exogenous, non-controllable factors, which are market's parameters of electricity price and natural gas price, with the controllable and endogenous to the business factor of gas turbine efficiency²⁹. Following the equation, higher gas turbine efficiency has a positive increasing impact on Spark Spread value.

Spark Spread's importance lies with the fact that is used to define gas turbines, and consequently CCGTs, dispatchability, which is the ability of the engine/power station to operate profitably in the market. In other words, every operator defines a minimum profitable Spark Spread value/price for their engine. When Spark Spread prices in the market are higher than this minimum, the engine operates and makes profit. The higher the Spark Spread prices are in the market above the set minimum, the higher is the profit for the engine and operator. When Spark Spread prices in the market are lower than the set

²⁷ Clean Spark Spread (CSS) includes also cost of CO₂ (£/tn CO₂); for simplification purposes, CSS is not used in this work as CO₂ prices can be considered constant.

²⁸ Dark Spread (DS) is the equivalent metric referring to coal –fired power stations.

²⁹ Gas turbine efficiency is used for simplification purposes, as it typically is CCGT's efficiency. However, steam turbine's (ST) and heat recovery steam generator (HRSG) efficiencies can be considered constant and not affected by GT's cooling air issues.

minimum, the engine has to stay idle, not synchronised with the grid, as otherwise it will operate at a loss. Engines with better thermal efficiency are rewarded, since they can make profit with lower Spark Spread values. In contrast, engines with low efficiency are commercially forced to stay idle, not dispatched, when Spark Spread prices are low, since otherwise they will run at a loss.

In setting the minimum profitable price for Spark Spread, the operator can factor or include overheads of all costs, for example cost of personnel and salaries etc. By this way, Spark Spread value can be a multiplier factor for net revenue calculations since all the other cost factors are included in the equation of Figure 5-5.

As electricity market operates on a supply-and-demand basis, Spark Spread values vary continuously on an hourly basis, following demand – for example high demand for electricity on Monday morning peak time - but also supply – for example high supply in a windy day where wind turbines generation suppress the need for CCGTs to operate. There are also cases when Spark Spread prices become negative. This can happen when low demand is combined with high by renewables supply, and by this way the market tries to keep most or all fossil-fuel electricity producers idle.

Figure 5-6 shows daily average of Spark Spread values through a typical year. Figure 5-7 shows natural gas prices through a typical year, where again seasonality of the same profile is observed with higher prices in winter and lower in summer.

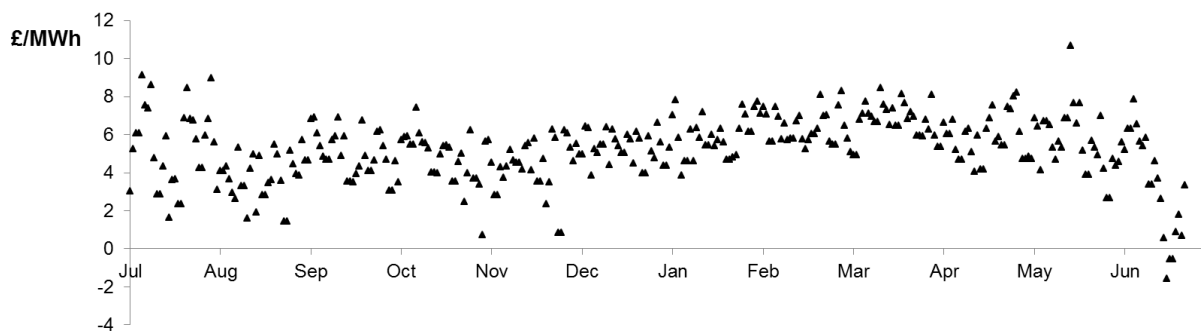


Figure 5-6: Typical daily average of Spark Spread values through a year **(80)**



Figure 5-7: Typical daily average natural gas prices³⁰ through a year **(80)**

5.3 Factors affecting Engine's Generated Revenue

A number of factors define and influence the expected by a gas turbine generated revenue. They are quite different in principle and type, as they span from market conditions, in the form of Spark Spread prices, and ambient temperature to reliability and general degradation characteristics of the engine, as well as engine's overhaul and hot gas path parts costs. In the below paragraphs, all the critical factors are described in detail, both in qualitative and quantitative aspects, especially as to how they are expressed mathematically and how they contribute to the calculation algorithm of paragraph 5.4.

5.3.1 Ambient Conditions Correction

For every "air-breathing" engine ambient conditions are of great importance for output and thermal efficiency calculations. In most cases, and similarly in regards to the current project, OEM provides data for the impact of ambient

³⁰ Note and caution over the distinctive unit that European and UK gas market use, p/therm

conditions on engine's performance in the form of correction curves. These curves are usually validated by the operators through engine performance tests, also termed Heat Rate (HR) tests, at different periods through the year.

It has to be stated that only ambient temperature as critically influencing factor is considered in this work. Ambient humidity can affect, in theory, engine's performance but to a significantly lower extent, as land-based gas turbines are in most cases equipped with extensive configuration of air filters, which among other duties perform also the task of keeping humidity out of the compressor. Ambient pressure is also another affecting factor, however ambient pressure varies randomly by nature and it does not follow any seasonal patterns with expected average values etc, as opposed to temperature. As a result, most gas turbine operators choose to omit ambient pressure from any predictive models and tools.

Figure 5-8 illustrates the typical seasonal variation of ambient temperature in Pembroke³¹, on a monthly average analysis. Figure 5-9, more interestingly, illustrates gas turbine's variation of output and thermal efficiency following the seasonal variation of ambient temperature. Expectably, ambient temperature plays a quite significant role in engine's performance. The variation from summer to winter in output can be in the order of +10MW, while in thermal efficiency the variation can be in the order of +0.5%.

It has to be clarified that, in Figure 5-9, output refers to a CCGT unit, in other words to the combined cycle configuration of a gas turbine + heat recovery steam generator + steam turbine (GT-HRSG-ST). However, the discussion on ambient impact is related only to GT performance as both the HRSG and the ST are closed systems (not "air-breathing") and relatively indifferent to ambient conditions^{32,33} changes.

³¹ Pembroke in Wales, UK; location of the gas turbine in discussion of this work.

³² Steam Turbine performance can be heavily affected by cooling water temperature, especially in direct-cooled (DC) power plants, as in most cases cooling water is sourced from the sea or adjacent lakes, rivers etc. However, for simplification purposes the impact of seasonal cooling water temperatures on steam turbines is not considered in this work.

³³ Pembroke CCGT is a direct-cooled power plant.

As already mentioned, the ambient temperature versus output and thermal efficiency curves of Figure 5-9 are used as correction curves. In other words, the output or thermal efficiency of a gas turbine in a particular day of the year is corrected with the use of these curves, so as the ambient temperature effect to be deducted or added accordingly. Both output and thermal efficiency curves are expressed in polynomial forms for the purposes of the revenue calculation algorithm of section 5.4. In terms of predicting ambient temperature for modelling or other purposes, Monte Carlo is used. On the historical monthly or daily average ambient temperature records that exist for every location, an uncertainty margin applies, for example $\pm 5^{\circ}\text{C}$, and then prediction through Monte Carlo algorithms can take place. The predicted values of future ambient temperatures are used to correct the future output and thermal efficiency that the engine exhibits.

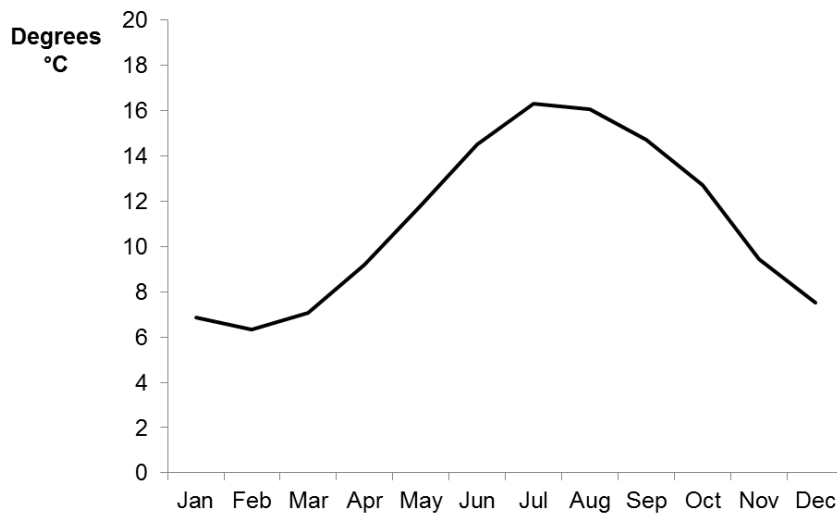


Figure 5-8: Typical monthly average ambient temperature for Pembroke, Wales, UK

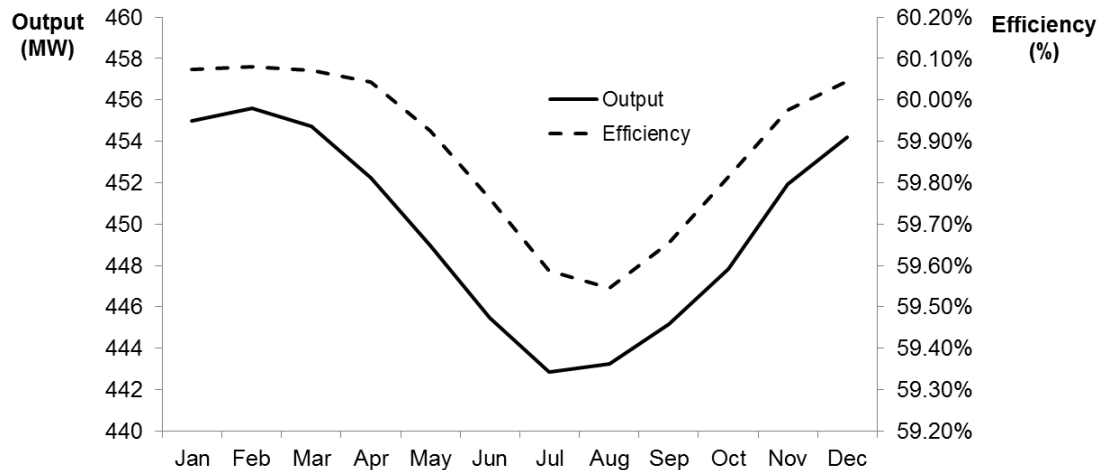


Figure 5-9: Typical monthly variation of gas turbine output and efficiency following ambient temperature

5.3.2 General Engine Degradation

Another critical factor, that has to be considered when prediction of future engine performance characteristics is in pursuit, is engine's general degradation. With this term, OEMs and operators describe the expected drop in output and thermal efficiency that every engine experiences as it ages. It refers to linear wear-and-tear mechanisms, primarily in the area of compressor and turbine seals between rotating parts and casings.

Similarly to the ambient correction curves, general degradation curves are in most cases provided by OEMs. They are usually expressed in the form of a polynomial using operating hours as the independent variable (Figure 5-10). The drop in output and efficiency is reflected through percentage values, which refer to new engine condition when no degradation applies.

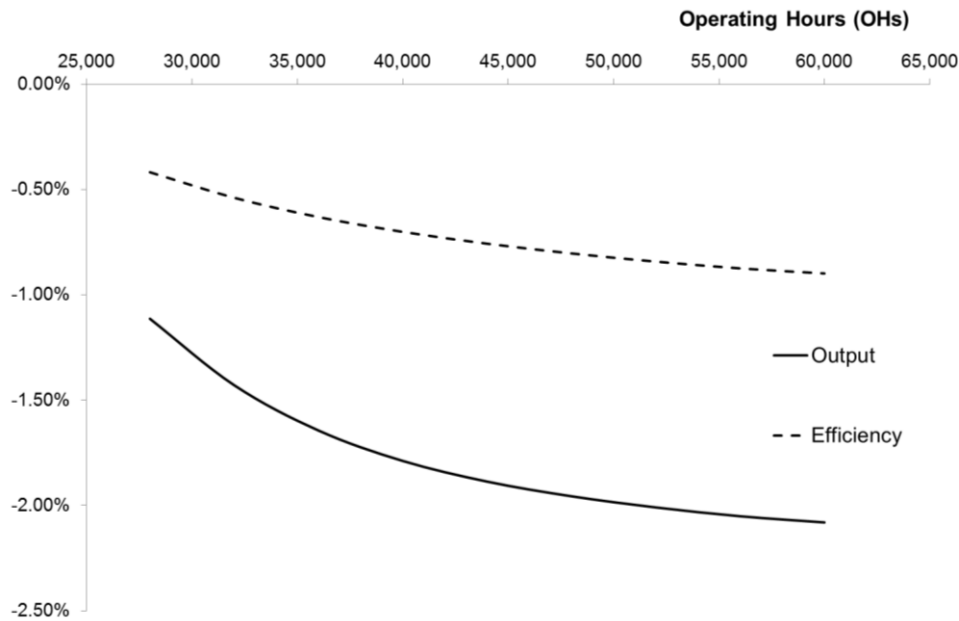


Figure 5-10: Gas turbine output and efficiency degradation with operating hours

5.3.3 General Engine Reliability

By general reliability, OEMs and operators term the reliability that all the peripheral, but necessary for the safe and efficient operation of a gas turbine, systems, components and instruments exhibit.

More specifically, a gas turbine unit can be described as the package of the main shaftline, typically comprised by a compressor-combustor-turbine configuration, and the peripheral equipment which indicatively can be the lubrication oil pumps, the hydraulic oil pumps, engine's cooling water system, vibrations monitoring system, various actuators and valves, monitoring instruments and sensors etc. For the shaftline itself, reliability is expected to be constantly ~100% through the whole cycle of operation (this can be between 4 and 5 years), hence the criticality and importance of reliable lifing models which then guide the intervals for hot gas path parts or structural components replacements. However for the peripheral equipment, and despite the fact that there is always a form of redundancy in a number of systems and components, for example in the duty/stand-by configuration of lubrication oil pumps, reliability

is not expected to be constantly through the cycle at levels of 100%, as this would have been disproportionately expensive for engine's lifecycle costs. Single points of failures unavoidably exist in the gas turbine package and they can result in a gas turbine unit being out of service for repair or replacement of that particular piece of equipment.

Usually general reliability curves are derived from experience, as they have to combine behaviors from multiple different pieces of equipment. Figure 5-11 shows the reliability curve which is used for the gas turbine of the current project. As it is illustrated, there is an increasing steepness towards the end of the cycle. This is expected and it is reflection of a typical "bathtub curve" behavior that the majority of the components, such as pumps and valves, appear to have through their lifecycle. It has to be clarified though that the initial steepness of a typical bathtub curve is not applicable in the current case, as it is widespread strategy for gas turbine operators to have extended testing and commissioning periods for their engines, in order to alleviate significantly the profile of the general reliability curve in the early phase of engine's life and by this way to enter the commercially operational phase of engine's life with high percentages of general reliability (>96%).

Another clarifying comment on Figure 5-11 is that it illustrates a full cycle between major overhauls of the engine. The engine came out of a major overhaul, which took place at around 27,000 OHs, with high levels of general reliability ~ 96%. As it accumulates operating hours through the cycle and as it approaches the next major overhaul event at around 60,000 OHs the predicted general reliability drops to <90% values. Though it is not shown in the graph, it can be assumed that after the second major overhaul, general reliability percentage will be reinstated to high levels, ~ 96%, similar to post first major overhaul operation.

In terms of factoring general reliability in revenue calculations, the approach is slightly different when compared to ambient correction curves. The expected percentage of general reliability at a specific point of operating hours, for

example 93% at 45,000 OHs, is multiplied to any volume (MWhs) or revenue (£) figures that correspond to the same point of operating hours.

As an example, if a gas turbine unit has 100% general reliability and prediction of generated electricity volume of 10,800MWh in a day at 45,000 OHs, then the corrected volume when general reliability applies will be,

$$10,800 \times 0.93 = 10,044\text{MWh.}$$

In this way, the expected losses in volume or revenue due to general reliability are effectively captured and a more realistic revenue calculation is achieved.

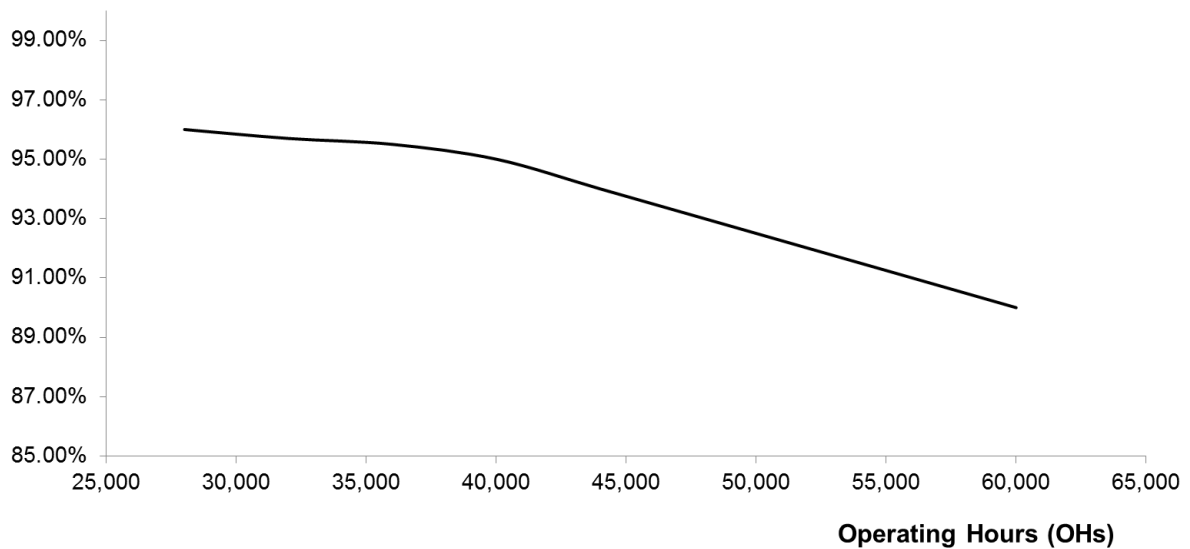


Figure 5-11: Gas turbine general reliability over operating hours

5.3.4 Engine Life Usage

In paragraph 5.2.1, the cost drivers behind OEM related costs, primarily around hot gas path parts and services, were discussed in some details. The scale of these costs in the gas turbine business is quite high, so it is critical for any revenue value calculation model to factor them in.

Whenever an engine goes through its major overhaul event, a new set of hot gas path parts is installed and all relevant works and adjustments take place. In most cases, the OEM performs these tasks. The completion of the overhaul is

the exact point the new operational cycle commences, with an expected duration of 4 to 5 years. This is also the point in engine's lifecycle that the biggest cost is incurred, reflecting primarily the cost of the new hot gas path parts and OEM services. Within the forecoming cycle this initial value depreciates, reaching a minimum value at the end of the cycle. The term that is used in this work for this particular type of depreciation is *engine life usage* (ELU).

Always key objective for every utility is to maximise the value out of their asset. In the context of gas turbines and in particular of hot gas path parts and engine life usage, this objective is reflected through utilities operational strategy to extract maximum value out of the engine, in other words the engine to complete its nominal cycle with no value left to depreciate, always in relation to hot gas path parts. This is expected, as most value-generating assets are managed in this way. However, this objective becomes particularly important when lifing risk issues, similar to what this project explores, pose questions on earlier withdrawal of hot gas path parts through forced outage events. In these cases significant remaining value can be left undepreciated which means that the engine does not make the most of its hot gas path parts value and the remaining value is in reality lost for the operator. Depreciation is a considerable factor for engine's economics and profitability and together with the cost of lost generation and the lifing risk are the three most important factors to drive decisions on if and when an engine should have a forced outage.

It has to be stated that engine's life usage is based on operating hours, as opposed to conventional actual time based depreciation curves that apply, for example, to civil assets. Also, no inflation correction has been applied³⁴.

Table 5-1 lists the key points that form the depreciation curve. Indicative costs have been used in this case, though it has to be said that the figures are quite reflective of actual costs of the heavy-duty gas turbines industry, in particular of

³⁴ The current project explores a scenario that is developed in a time-period of 1 to 1.5 years; for longer timelines, impact of inflation on future prices and values should be considered.

Frame 9 size of engines. The value of hot gas path parts at the end of the cycle was put as 10% of the initial value. This percentage is indicative, however it reflects realistically the value of the parts at the end of their useful life and it is a percentage that is commonly used in similar calculations in industry. Figure 5-12 shows the depreciation curve for one C cycle, namely from zero to 32,000 OHs. Similarly to general reliability curves, a new cycle resets the clock with high initial value, which starts to depreciate again through the course of the operational cycle.

Table 5-1: Engine life usage parameters

Description	Value	Unit
Estimated value of new set of HGP parts	22,000,000	£
C Cycle duration	32,000	OHs
Value of HGP parts at the end of the C cycle	10%	%
Value of HGP parts at the end of the C cycle	2,200,000	£

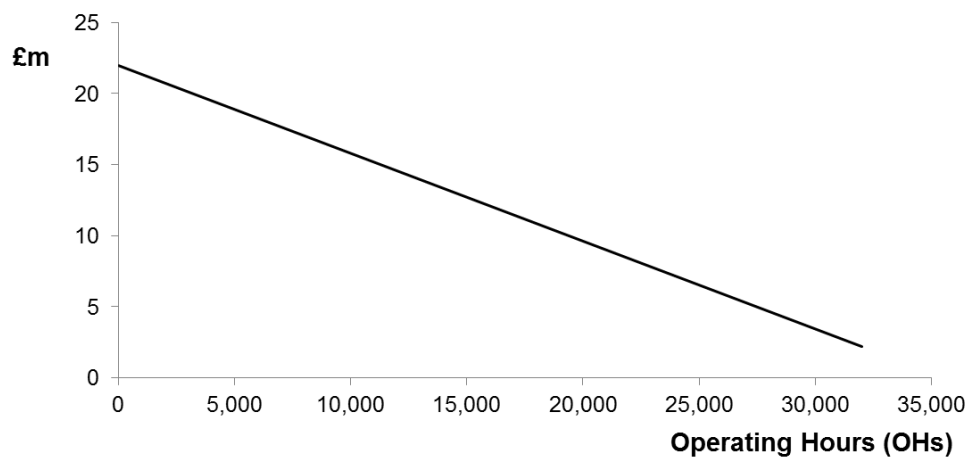


Figure 5-12: Engine life usage with operating hours

5.4 Predictive Model for Revenue and Lifting

Following the description of the critical influencing factors, a model is developed which produces results with the expected revenue generation by predicting Spark Spread values and ambient temperatures and integrating all the above-described factors. The model is applied on an engine with degraded cooling air system and the algorithm is combined with lifting calculations, by using Miner's equation. An optimum operating profile of the engine is proposed which minimises losses and manages effectively the lifting risk.

5.4.1 Revenue Generation Algorithm

Figure 5-13 illustrates the revenue generation algorithm in the form of a flow chart. Starting from the bottom side, Monte Carlo is used to predict system's³⁵ future Spark Spread prices. Historical Spark Spread values have been used to formulate the monthly average ones as well as the uncertainty, which is cited in Table 5-2 (80).

The predicted Spark Spread prices define engine's running profile, as explained in paragraph 5.2.3. A realistic minimum cut-in price has been set, in the current example is £3/MWh. Monte Carlo produces two different predictions, one for day-time (07:00-23:00) prices and one for night-time (23:00-07:00) prices. So depending on the value of the predicted price in comparison with engine's cut-in price, results are produced in terms of engine' running hours. One of the below four operating scenarios always apply:

- If system's Spark Spread prices are lower than engine's cut-in price during both day- and night-time, then the engine is not expected to run at all, hence it does not accumulates any OHs.
- If system's Spark Spread prices are lower than engine's cut-in price only during night-time, then the engine is expected to run only through the day, hence it accumulates 16 OHs.

³⁵ As explained in paragraph 5.2.3, Spark Spread prices come up from the combination of electricity and natural gas prices. So by the term "system" is meant the electricity and natural gas grid regulators.

- If system's Spark Spread prices are lower than engine's cut-in price only during day-time, then the engine is expected to run only through the night, hence it accumulates 8 OHs.
- If system's Spark Spread prices are higher than engine's cut-in price during both day- and night-time, then the engine is expected to run through for 24h, hence it accumulates 24 OHs.

The next step in the calculation algorithm is the multiplication of the predicted operating hours by engine's nominal output. Engine's nominal output, which in current example is 450MW, is unit's specified power output in design and as-new conditions. The product of this multiplication is the nominal electricity volume in MWhs.

Next block in the algorithm's sequence is the multiplication of nominal electricity volume by the corrections factors of ambient temperature, normal/general degradation and general reliability. The product is again electricity volume in MWhs, however the volume figure is now adjusted (corrected), based on the predicted ambient temperature at the time of generation and on engine's cumulative operating hours, which define values for reliability and degradation factors. As already mentioned, ambient temperature is predicted with the use of Monte Carlo, based on previous historical records. General reliability and degradation are defined through polynomic functions, which are described by their respective curves in paragraphs 5.3.2 and 5.3.3.

Following the calculation of electricity volume, the next step is the multiplication of the volume by system's Spark Spread prices, which were applied at the time of generation, and the production of gross revenue in (£). The same Spark Spread prices, which defined engine's operating hours, are used for the calculation of gross revenue.

Finally, the total net revenue is calculated with the deduction from gross revenue of engine's life usage value (£/OH). Engine life usage has a decreasing value as the operating hours accumulate, also shown in Figure 5-12. So clearly, towards the end of the cycle, its value is low and it is unlikely to influence decisions on maintenance events. However, the economics around a scenario

of a force outage early in the cycle will be heavily affected by the remaining undepreciated value of engine's life usage at that point in the cycle.

In a summary, the algorithm in Figure 5-13 provides an inclusive model for gas turbine revenue calculations. As also explained in paragraph 5.2.3, operational costs, other than natural gas cost, can be factored in the engine's Spark Spread value calculation as an overhead factor. This means that an engine with higher operational costs sets unavoidably a higher cut-in Spark Spread value and needs to receive higher system Spark Spread prices in order to operate profitably. The extent of capturing these costs in amount and type in the Spark Spread calculation defines the degree of how close the revenue value is to the definitions of *gross* or *net* revenue. If for example all the associated costs of running a CCGT are factored in the Spark Spread value (O&M costs, chemicals, etc) then the product of the algorithm is clearly net revenue. As it has become even more evident now, the concept of Spark Spread is pivotal, as it offers the operator the input to factor all or some of the operational costs.

Figure 5-14 repeats the steps of the above calculation but for a derated engine. Since the gas turbine has been founded with degraded cooling air system, a derate has been applied, in the form of decreased firing temperature, that allows the engine to operate safely until its planned major inspection. The degree of derate depends on the extent of cooling air temperature increase. Table 5-3 and Table 5-4 list the necessary decrease in firing temperature that has to be applied, showing also the preceded increase in cooling air temperature and the resulted blade metal temperature. Following the flow chart, the derated engine performance block affects the volume calculation, since the output is lower than nominal, and the Spark Spread value, since thermal efficiency is dropped.

Table 5-2 lists the details that were used for the Monte Carlo predictions, in regards with system's Spark Spread prices and ambient temperatures. In both cases, historical data was used from Figure 5-6 and Figure 5-8 accordingly to formulate average and uncertainty values.

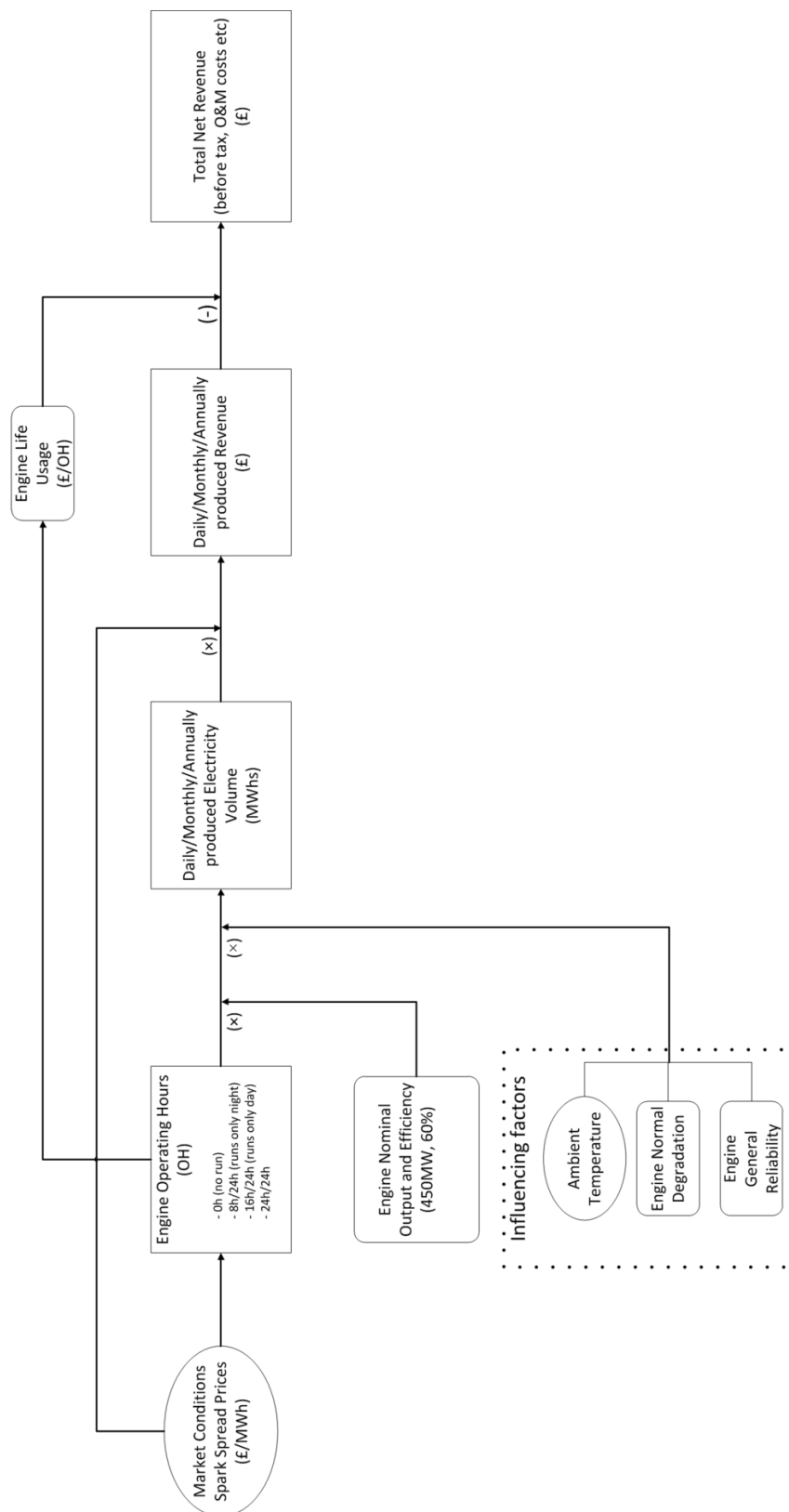


Figure 5-13: Revenue calculation algorithm for clear case – no engine derate

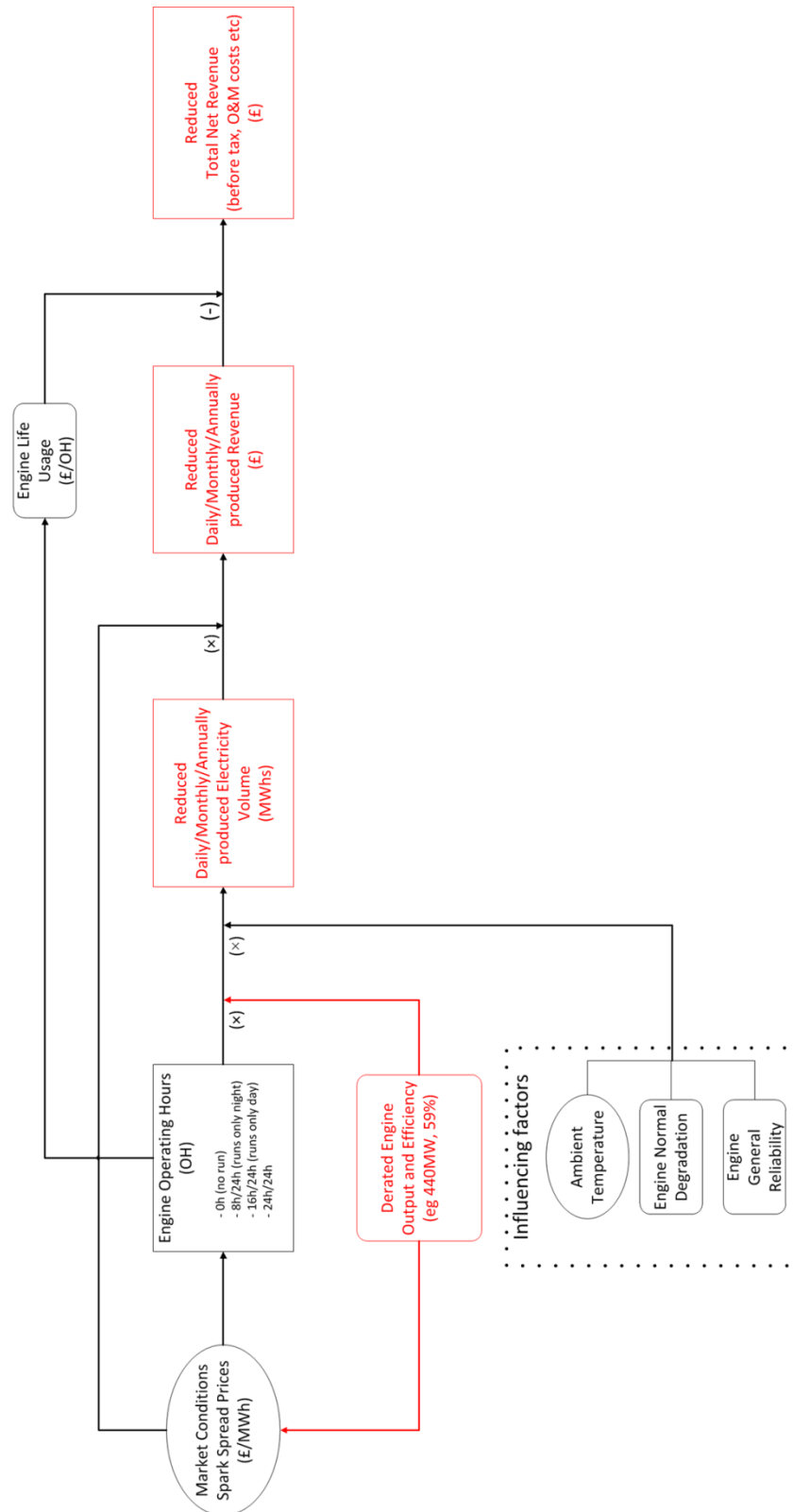


Figure 5-14: Revenue calculation algorithm for degraded case – derated engine

Table 5-2: List of Monte Carlo uncertainties for revenue calculation algorithm

Property	Distribution	Mean Value	Standard Deviation
Spark Spread Price (£/MWh)	Normal	Monthly Average	+/- 2 £/MWh
Ambient Temperature (C°)	Normal	Monthly Average	+/- 3 C°

5.4.2 Revenue and Lifting

The algorithm of the above paragraph provided the steps of calculating revenue in two different cases. In the first case, there is no applied derate on the engine, in other words there is no degradation in engine's cooling air system or any other similar issue and the case is termed Clear Case. In the second case, the engine is derated and operates with decreased firing temperature. This case is termed Constant Derate case. Both cases can be illustrated with use of time versus revenue (value) curves.

Delving deeper in the above-described concept, Figure 5-15 illustrates three different curves:

- Clear Case curve is, as explained above, the case where there is not any scenario of engine derate, in other words there is no cooling air system degradation or similar. Algorithm from Figure 5-13 is used to calculate the expected revenue until engine's next planned overhaul; for the purposes of the current scenario the overhaul is assumed to take place in August 2020. The revenue value curve provides momentary results, for example value at any specific time-shoot of a day, week or month, but also total cumulative results for any interval with a given duration of time, for example from January 2019 to August 2020.
- Constant Derate (CD) curve shows the case in which the engine has been found with degraded cooling air system, so a constant derate on firing temperature has been applied, following the process described in

paragraphs 4.6 and 4.8. In this case, algorithm from Figure 5-14 is used. Decreased firing temperature has an impact on engine's output, which affects generated volume. It has also an impact on thermal efficiency, which affects engine's Spark Spread values and consequently generated revenue. As expected, the Constant Derate curve is lower than the Clear Case on the graph, following a constant pattern through the time with no adjustment on market conditions. The interpretation of the Constant Derate curve is that the engine generates lower revenue at any given time and therefore the total cumulative revenue is lower when compared with Clear Case.

- Dynamic Derate (DD) case curve is the third curve on the graph. It remains always lower than the Clear Case one on the graph, but not in a constant offset pattern. It crosses Constant Derate curve, being in higher position through winter months and in lower position through summer months. The explanation behind curve's behaviour is as follows: the engine operates between two different firing temperature (FT) settings, following market's Spark Spread prices. When Spark Spread prices are higher than a specific pre-defined value, the engine operates with a mild derate (termed as Shallow Dynamic Derate, SDD) as by this way it takes advantage of the favourable market conditions. When Spark Spread prices are low, in other words there is no market reward, then the turbine operates at deeper derate (termed as Deep Dynamic Derate, DDD) as by this way it saves lifing without significant costs. In this way, the firing temperatures and therefore engine's performance are driven by Spark Spread prices, in other words by market conditions. The two different firing temperatures have different life consumption allocations, which are shown in Table 5-3. Dynamic Derate is a more intelligent way to manage engine's operation and to trim the otherwise unavoidable losses.

It is important to be clarified that Figure 5-15 simplifies graphically the concept of Dynamic Derate, as it illustrates it following a seasonal pattern only. In reality, the change between Deep and Shallow Derate can happen multiple times

through the day, depending always on market's Spark Spread values. Though it can be said, there is an expectation that the engine will have more hours in Shallow Derate through winter and accordingly more hours in Deep Derate in summer months.

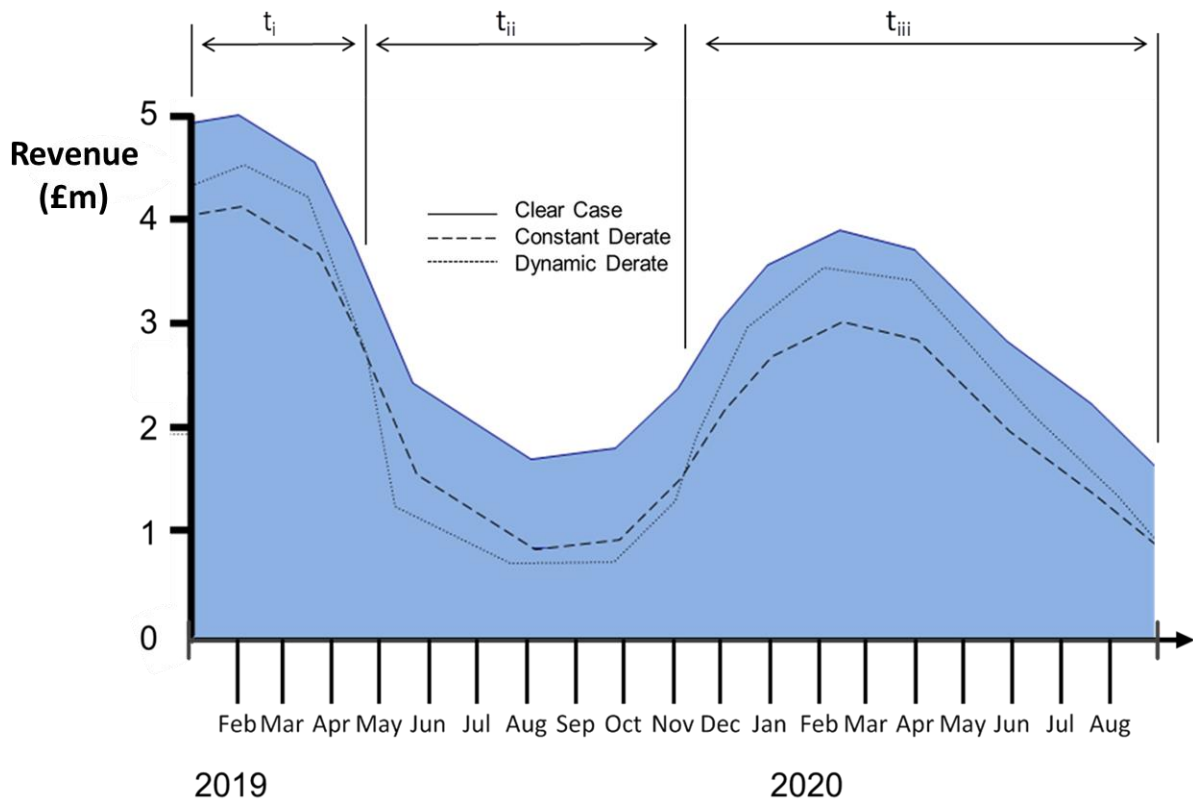


Figure 5-15: Concept of Dynamic vs Constant engine Derate

In a simplified mathematical expression, the enhanced revenue and lifing model aims to satisfy two conditions:

1. To maximise revenue, with the equation:

$$\text{Max Value} = \sum \int f(t_n, \text{Output}, \text{Spark Spread}) \quad (5-1)$$

Where,

f function to calculate revenue, following principles of the algorithm in Figure 5-13 and Figure 5-14

t_n	the time that the engine has spent at any firing temperature setting
Output	engine's output, according to firing temperature and corrected for ambient temperature, general reliability and degradation
Spark Spread	market prices (£/MWh) that gas turbine operates with

And critically,

2. To maintain the lifing constraints, as those expressed by the equation:

$$1 - \frac{t_0}{L_0} > \frac{t_i + t_{iii}}{L_1} + \frac{t_{ii}}{L_2} \quad (5-2)$$

Where,

t_0	operating hours that have already consumed at the point when derated engine settings apply
L_0	nominal operating hours, 32000 OHs
t_i, t_{iii}	operating hours at Shallow Dynamic Derate
L_1	total life at Shallow Dynamic Derate
t_{ii}	operating hours at Deep Dynamic Derate
L_2	total life at Deep Dynamic Derate

in the next paragraph, results that have been produced following graph of Figure 5-15 and the concepts of Constant and Dynamic Derate are discussed in detail. Moreover, the adopted scenario together with its assumptions is described and discussed in detail.

5.4.3 Scenario and Results

The above model is applied on a hypothetical, but very close to real-life, scenario:

- an engine with nominal output 450MW and thermal efficiency 60%³⁶ respectively is found³⁷ with degraded cooling air system, after 20,000 hours of operation. The flow through the pre-swirl nozzles cavity inner seals is calculated to be higher almost 5% of the nominal values.
- The process, which has been described thoroughly in chapter 4, is initiated. The increase in cooling air temperature incurs an increase in blade metal temperature, shown in second row of Table 5-3. The engine is forced to operate in derated state, in order to manage the increased lifing risk, until its next planned major overhaul in June 2020. Three different firing temperatures are used, all lower than the nominal one, to manage lifing risk by decreasing hot gas temperature and consequently blade metal temperature. Using the terminology that was introduced in the above paragraph, Shallow Dynamic Derate refers to -20K comparing to nominal firing temperature, Constant Derate (CD) to -30K and Deep Dynamic Derate to -40K. The decreased firing temperatures entail performance penalties in output and thermal efficiency, which are listed in Table 5-4. But they also have different remaining life allocations which are listed in both Table 5-3 and Table 5-4.
- Table 5-3 summarises the stepped increase in cooling air temperature as it has been fed from the relevant results in paragraph 4.4.3. In the same table, the second row shows the impact on blade metal temperature as well as the resulted creep life, if no mitigation measure is taken. The next three rows show the positive effect on blade metal temperature and creep life total hours of decreasing firing temperature in steps of 10K.

³⁶ Number refers to whole unit (GT-ST) output

³⁷ Through use of borescope inspection; Appendix G offers some more details on this.

The three cases of decreased firing temperatures have been termed above as Shallow Dynamic Derate, Constant Derate and Deep Dynamic Derate.

- In Table 5-4, the steps in firing temperature decrease are shown in relation with their respective drops in output and thermal efficiency as they were produced by engine's performance model. The drop in thermal efficiency incurs also a Spark Spread penalty, which is also listed in reference to firing temperature.
- Figure 5-16 illustrates the difference in revenue (value) between the three curves. *No Derate* curve shows the Clear Case revenue. In other words, the theoretical revenue that the engine would have made if there was no cooling air degradation issue and hence no reason to operate in derated state. The cut-in Spark Spread price for the current scenario is £3/MWh, so it can be assumed that the engine operates through this period whenever the market offer prices above £3/MWh. Following this market prediction, by the end of the period illustrated in Figure 5-15 the engine is expected to have clocked in total approximately 32,000 operating hours. This is within OEM's recommendations for the operational intervals between major overhauls, so the next overhaul will take place as planned in August 2020. More importantly, with the current predictions on Spark Spread prices and ambient temperatures, the engine is expected to generate more than £20.5m in revenue. This value and its associated curve are used as benchmark for the comparison of the two methods, Constant Derate and Dynamic Derate.

Constant Derate curve is the bottom line on the graph. As, similarly to *No Derate* curve, there is no dynamic adjustment to market conditions, the two lines are parallel to each other. Constant derate reflects the above scenario, where as a mitigation to increased blade metal temperature an engine derate is applied. Following the blade cooling calculations from section 4.6 and lifing calculations from section 4.8, a new firing

temperature has been derived which can satisfy the requirement of approximately 32,000 operating hours. The Constant Derate temperature and the resulted blade metal temperature and creep life are shown in Table 5-3, while the associated penalties in output, thermal efficiency and Spark Spread value in Table 5-4. As the new creep life is very close to the nominal one, there is no need for the planned overhaul to start earlier, for example in spring 2020. By this way, the operator firstly is not exposed to increased market prices and generation-lost revenue, as Figure 5-15 explains. Secondly, there is not exposure to lost engine life usage value as discussed in Figure 5-15. And thirdly, there is not exposure to any of the outage-date-shift issues, as those were discussed in paragraph 5.2.2. All the above, with an effective lifing risk management in place, as the expected total operating hours by August 2020 are not going to be significantly higher than the nominal ones (Table 5-3). However, with the current scenario of predicted Spark Spread prices and ambient temperatures, the engine is predicted to generate total revenue slightly lower than £18.5m, which when compared with the Clear Case it equates to a loss of approximate ~£2.7m. It has to be said, that Constant Derate is the preferred mitigation measure by OEMs in all cases of increased lifing risks.

Finally, the *Dynamic Derate* curve is the middle line on the graph. It is not parallel to the other two lines and it shifts from higher value to lower, as Spark Spread prices increase. The explanation is that the Dynamic Derate value is the result of a continuous adjustment of engine's firing temperature to market Spark Spread prices. Following the concept of Figure 5-16, the operator defines a second Spark Spread price, over and above the primary one which is £3/MWh in this scenario. As a result of the primary cut-in price, the engine operates for prices higher than £3/MWh and it stays idle (de-synchronised) for prices lower than £3/MWh. As a result of the secondary cut-in price, for example £3.5/MWh, the engine operates in Shallow Dynamic Derate when prices are higher than £3.5/MWh and operates in Deep Dynamic Derate, when

prices are lower than £3.5/MWh, but still higher than £3/MWh. Table 5-4 lists the outcomes in output, thermal efficiency and Spark Spread penalty for Shallow and Deep Dynamic Derate firing temperature settings respectively. By this way, the concept of equation 5.1 is materialised, in other words maximum value, given the firing temperature limitations, is achieved. As the graph in Figure 5-16 shows, the lower the secondary Spark Spread cut-in price is, the higher is the revenue. As the secondary cut-in price increases, this means that the engine operates in Shallow Dynamic Derate less time, so the revenue decreases. If high prices are selected, for example higher than £5.5/MWh, then the operation in the profitable Shallow Dynamic Derate becomes limited. In these range of Spark Spread prices, the difference between the two derate methods, Constant and Dynamic, vanishes. But since commercial optimisation is not the only and unique purpose of the above model, equation 5.2 comes now into apply. The two firing settings have different creep life consumption, as shown in both Table 5-3 and Table 5-4. Every time that a secondary cut-in price is selected, the algorithm calculates the remaining safe operating hours of the blade, a function which is illustrated in the below bullet point.

It is imported to be clarified, that in all three curves above the same assumptions for Spark Spread prices and ambient temperature predictions were applied.

- Figure 5-17 epitomises the concept and contribution of this project. The solid line shows the difference in revenue value between the two derate methods, Dynamic and Constant. The difference is reduced as Spark Spread prices increase, as explained in above bullet point. If a low secondary cut-in price is selected, for example £3.6/MWh, the difference between the two derate methods is substantial, almost £0.9m. However, this means that the engine operates more frequent in Shallow Derate, since the operator aims to reap benefit from the market even with low prices. This in turn is reflected with very high lifing risk, as the remaining

operating hours, rectangular and triangular points, will be close to zero or less, which in essence means that the engine is predicted to run beyond its safe creep life. If higher Spark Spread prices are selected, a more balanced operation is achieved. For Spark Spread prices between £4.6/MWh and £5/MWh, the difference in the two methods is considerable, ~£0.5m, while also the remaining hours are now considered as adequate, between 300 and 400 OHs. As the operator chooses to operate the engine even more conservatively, namely in higher Spark Spread prices, the lifing risk is reduced drastically but also the financial benefit from the Dynamic Derate method diminishes entirely. A good use of the model can be in combination with frequent borescope inspections of the HGP parts. By this way, the operator can choose how conservative or not, in terms of Spark Spread prices selection, can be depending on the results of the visual inspections.

Table 5-3: Cooling air temperature, hot gas temperature, blade metal temperature, decrease in firing temperature and cree life in operating hours

Cooling Air Temperature (K)	Hot Gas Temperature (K)	Decrease in Firing Temperature (K)	Blade Metal Temperature (K)	Creep LMP Life (OH)	Comments
610.938	1783.150	0	1048.232	32,000	Clear Case, no CAS degradation
625.938	1783.150	0	1054.967	22,279	CAS degradation, +15K in CA temperature
625.938	1763.150	-20.000	1051.000	27,576	FT in Shallow Dynamic Derate
625.938	1753.150	-30.000	1048.500	31,536	FT in Constant Derate
625.938	1743.150	-40.000	1047.000	34,190	FT in Deep Dynamic Derate

Table 5-4: Constant and Dynamic Derate firing temperature settings and impact on output, efficiency, Spark Spread value and creep life

	Shallow Dynamic Derate	Constant Derate	Deep Dynamic Derate
Decrease in Firing Temperature (K)	20	30	40
Decrease in Load (MW)	15	22.5	30
Decrease in Efficiency (%)	0.30%	0.60%	0.90%
Spark Spread Value Penalty (£/MWh)	0.202	0.405	0.607
New Creep Life (OHs)	27,576	31,536	34,190

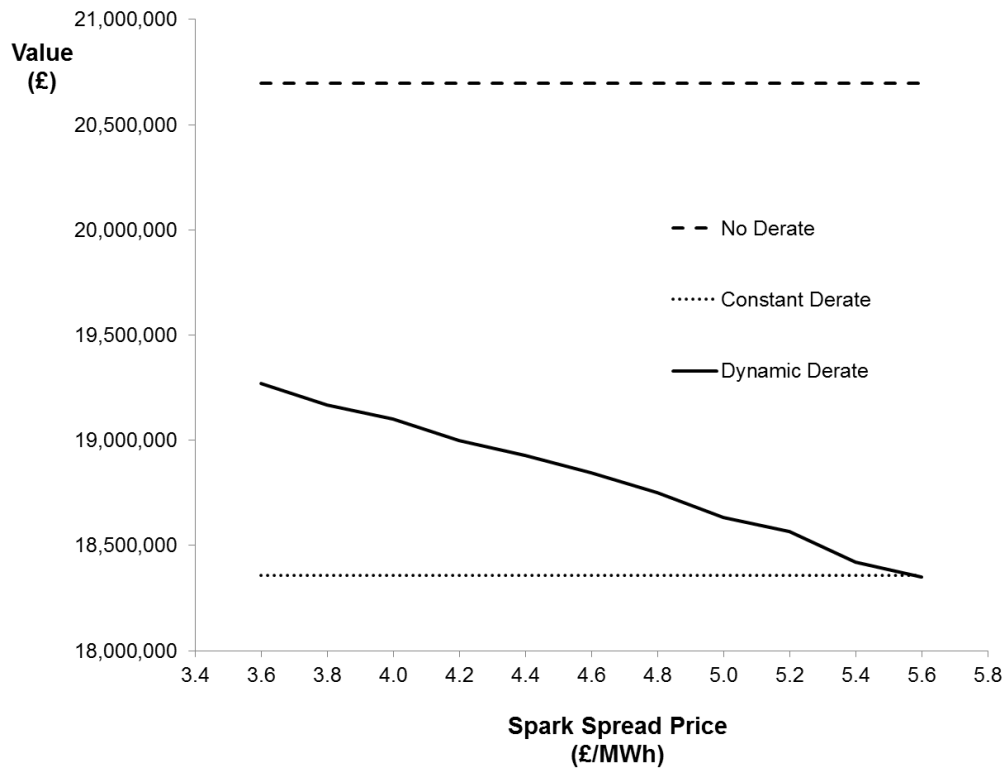


Figure 5-16: Revenue value with No Derate, Constant Derate and Dynamic Derate

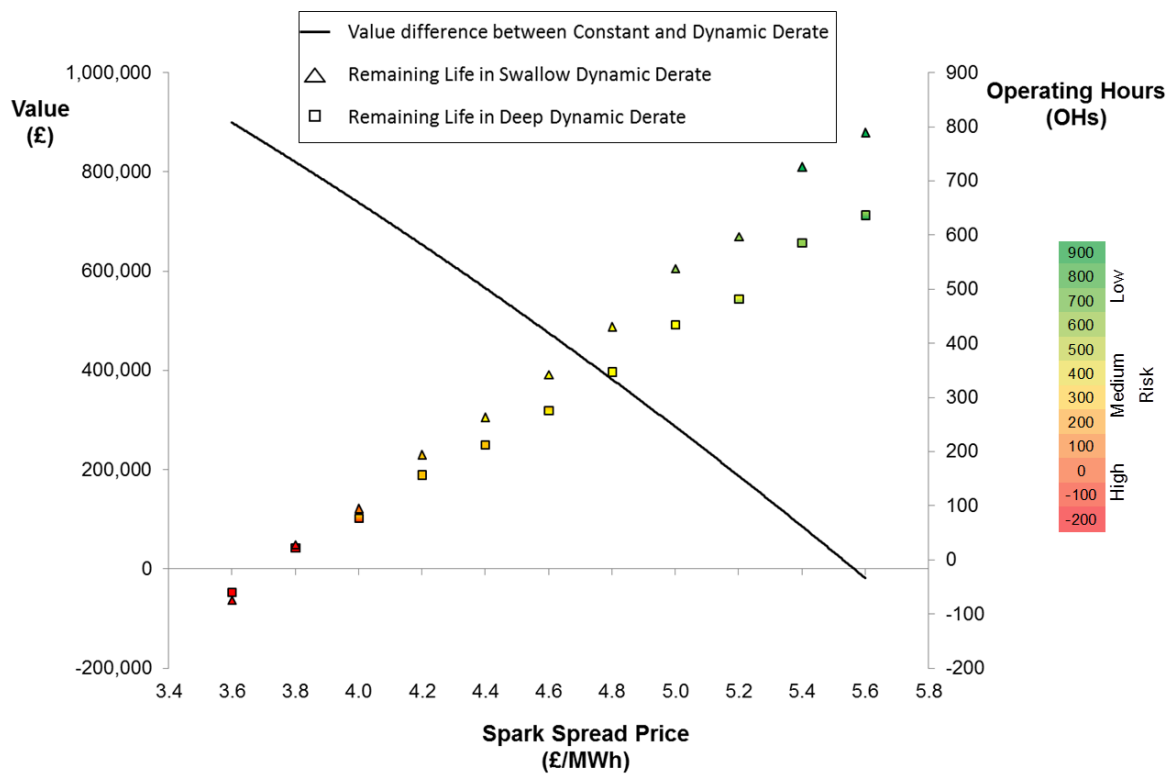


Figure 5-17: Diference in revenue value and remaining creep life between Constant and Dynamic Derate

5.5 Summary of Techno-economic Analysis and Predictive Model

Following outcomes from the lifing model of section 4.8, a techno-economic analysis was undertaken and a revenue generation predictive model was developed in this chapter. The chapter started with some important details of the electricity market in UK, though most of the comments apply also to a number of European countries and USA states. Special reference to OEM costs was made as those costs burden significantly utilities and operators. Outage costs were described in some details as together with OEM costs, they are the most important cost centres for utilities. Spark Spread value was explained thoroughly, as it is a key concept for understanding the electricity market and how operators factor their costs in their dispatch strategy.

In section 5.3, the factors that affect gas turbine's output, volume and revenue were described. Ambient conditions, engine's reliability, degradation and life usage were discussed. Ambient temperature is predicted based on historical monthly average values and the use of Monte Carlo. For engine's reliability, degradation and life usage polynomial are used, which are based on operational experience or provided by the OEM.

In section 5.4, the revenue calculation algorithm was presented. It is an important contribution of this project as it combines all the key factors that drive and influence an engine's generated revenue. The developed algorithm was described in detail, initially referring to a Clear Case with no engine derate to consider. Then a case with degraded cooling air system was explored, as this necessitates the need to decrease firing temperature in order the unit to meet the agreed, nominal and expected operational interval.

Initially two curves and values were produced by this way, the nominal one (Clear Case) and the Constant Derate one. Further optimisation took place with the introduction of Dynamic Derate (DD) and then Deep Dynamic Derate (DDD) and Shallow Dynamic Derate (SDD). A particular real-life scenario was adopted, where an engine with approximately 20,000 OHs is faced with a degraded cooling air system issue. The operator is called to decide on the operational

strategy of the next 16 - 18 months of committed operation until the next major overhaul that will offer opportunity for repair. The model used the generated revenue algorithm in combination with Miner's equation. By this way, engine's operation in the form of firing temperature is adjusted on a daily basis following Spark Spread prices. As a result, the losses due to cooling air system degradation can be substantially trimmed while at the same time the lifing risk is effectively managed. In other words, an optimum balance between remaining lifing risk and commercial position can be achieved.

6 CONCLUSIONS AND SUMMARY

6.1 Project Overview

Cooling air system degradation is an issue that has been observed in a number of heavy-duty gas turbines, including GT26 engines. Performance of engine's low-pressure-turbine pre-swirl nozzles can be compromised, if the flow through the pre-swirl nozzles cavity inlet seals exceeds the design values. The outcome is an increase in total relative temperature of the cooling air that is routed to cool down low-pressure-turbine 1st stage blades. A plethora of references was found describing pre-swirl nozzle and their behaviour, under normal but also deterioration cases. However, no work was found in which the deterioration in pre-swirl nozzles performance is approached holistically, including the impact on downstream components, the resulted lifing risk and finally the mitigation measures.

As a starting point, a performance model for the engine was developed. The model was validated with the use of an in-house performance monitoring software. It offered insight to the boundary conditions of hot gas path flow, while it provided with output and thermal efficiency results on the derated engine cases.

The cooling air degradation mechanism was captured and quantified through an analytical solution, which had admittedly significant assumptions in place. A higher fidelity solution was pursued with the use of CFD. The CFD methodology was validated with the use of a Test Case, which had key similarities with GT26. The results showed that an increase of inlet seal flow by 5% rises cooling air temperature inside blade cooling passages by around 13K.

The results of the degraded cooling air system were fed in a blade cooling model. The model was built analytically, however Monte Carlo was used to capture the uncertainties around blade's internal configuration and hot gas path

flow conditions. The increase of cooling air temperature inside the blade incurred an increase of almost 9K in blade metal temperature.

The same combination of analytical method and Monte Carlo was used for the calculation of maximum stress that the low-pressure-turbine blade is subjected to. Centrifugal stress was expectably found to be the predominant stress contributor, accounting for more than 99% of the total stress. Gas bending and momentum stresses were found to have very low contribution to total stress levels and were consequently ignored.

The Monte Carlo results from blade metal temperature combined with OEM's given life of the component, and the results from blade stress calculations, were used to populate a Larson-Miller Parameter graph as actual operational points. In the same graph, historical Test Data points were mapped, showing creep conditions in which components of the same material fail. Due to the finite and rather limited number of Test Data points, Monte Carlo was used again to produce significant number of points all in statistical agreement with the limited number of the Test Data ones. By this way, the uncertainty around experimental results was captured and the comparison between actual operational points and Test Data involved same in size populations. The overlapping area between the actual points and the historical Test Data points was increased, when the degraded cooling air conditions were considered. The increase in overlapping area was quantified with the use of loading-strength graphs.

Three methods were identified as potential to reduce the increased risk that the Larson Miller graph illustrated. Stress reduction as a method was rejected as not applicable on land-based, constantly synchronised engines, however it may be a valid proposal worthy being reviewed, if the engine in discussion is an aero-engine one. Reducing operational time can protect the engine from catastrophic failures and forced outages, however shifting outage events is something extremely costly and hence unpopular in the electricity business for reasons that are discussed in detail in paragraph 5.2.2. Decrease in firing temperature is the most effective method to mitigate creep-induced lifing risks in

gas turbines, however it comes with significant performance penalties which in turn have to be managed in a commercially optimised manner.

The Technoeconomic Analysis charter started with a necessary overview of the electricity market in UK. Critical characteristics of the market such as seasonality and renewables intermittency were discussed. OEM related costs in hot gas path parts and services, as well as engine's depreciation - expressed with the term engine life usage – were discussed in detail as they constitute substantial cost factors. The notion of Spark Spread value was explained in detail as it consists an important metric for the electricity market but also for the economics of a power plant since it defines engine's dispatchability and consequently its profitability. The generated revenue model was presented in detail. The model is inclusive of all the key parameters that define a gas turbine's generated revenue, namely market's Spark Spread prices, ambient temperature and engine's efficiency, output, reliability, degradation and life usage. Ambient temperature together with Spark Spread pricing are predicted with Monte Carlo due to their stochastic and random nature. All the other parameters follow specific patterns, which are discussed to some details. The degraded cooling air scenario was adopted and the model was used to run cases with the engine being constantly and dynamically derated (CD or DD), the latter case having shallow and deep derate options (SDD and DDD). By this way the model achieved to show the difference in generated revenue between the two approaches. When Miner's lifing equation was added as an integrated function of the model, the model showed also the different life consumption, and critically the blade's remaining life, between the two approaches. In a summary, with the use of the model the operator can choose the optimum firing temperature for the engine (shallow derate or deep derate) following system's Spark Spread prices. By this way, the losses due to engine's degraded condition can be significantly trimmed while at the same time the remaining creep-induced lifing risk is effectively managed. In a medium risk approach, the operator can chose a Spark Spread price that offers a difference (or saving) between the two methods of ~£0.5m, while also the remaining operating hours are now considered as adequate, between 300 and 400 OHs.

6.2 Conclusions and Impact

The most important conclusions through this research project are cited below:

- Cooling air system degradation issues can expose hot gas path components to higher cooling air temperatures. The increase in cooling air temperature can be calculated analytically and by this way, the user can obtain an approximate order of magnitude of the temperature change. CFD model can be used to offer higher fidelity and more accurate results, which are necessary for the downstream in the process calculation of blade metal temperature.
- A critical observation, which came early in the process, is that the above issue and risk can survive totally undetected and cause failures and forced outages, which most probably will be catastrophic in magnitude and consequences. It is important to be stated that the above-described failure mechanism leaves unvaried main operational parameters of the engine such as exhaust temperature, which are closely monitored for engine's health purposes. The operator is unaware of a serious, underlying and accelerated failure mechanism.
- A repair of the cooling air system (namely to re-instate seals clearance in pre-swirl nozzles cavity) involves full-engine opening, a task which is expensive, arduous and lengthy, hence it is scheduled to take place strictly on a cycle of 4 to 5 years. So even in the cases that the issue is detected and understood, the operator most probably will be unable to intervene drastically in the form of a forced outage as interim outages incur excessive commercial losses.
- Blade metal temperature can be calculated with satisfactory accuracy when detailed analytical models are combined with Monte Carlo, which allows for all the uncertainties to be captured. The same approach applies on blade stress calculations.

- The risk of increased blade metal temperature can be quantitatively assessed with the use of overlapping diagram and calculation of safety margin between design and actual Larson-Miller Parameter characteristics.
- Blade metal temperature can be controlled, as expected, with the use of engine's firing temperature. The analytical method, that was used to calculate blade metal temperature in the case of degraded cooling air system above, is used again to quantify the effect of decreased firing temperature on blade metal temperature.
- The operator can choose to derate drastically the engine so blade metal temperature is brought down to design levels, aiming to maintain the nominal cycle of 5 years; the derate in this case has to be deep and the operator will be burdened significant cumulative losses due to drop in generated volume and efficiency. As expected, decrease of firing temperature brings a drop in engine's thermal efficiency and output whose exact extent is calculated with the use of engine's performance model.
- Alternatively, If the derate is mild the losses can be trimmed; however, in this case the risk of a blade failure remains and has to be quantified, as an in-service failure is an event which has to be avoided at any cost, due to the extended consequential damages and very large generation-lost costs that will be incurred.
- Optimisation is necessary as minimum risk involves deep derating of the engine and significant losses, while a mild derate of the engine still carries considerable lifing risk. Hence the importance of the techno-economic model which is able to optimise engine's operation and performance and mitigate any risk of in-service failure, by taking into consideration remaining life and electricity market conditions.

The above conclusions lead to the two-dimensional impact of the current project. One dimension is towards a more effective risk management and one towards the commercial optimisation of heavy-duty gas turbines operation.

More specifically:

- ❖ on the risk management dimension, the proposed method detects and assess the impact of degraded cooling air system in gas turbines, while the increased life consumption and the associated risk are quantified. It is important to be stated that in similar cooling air degradation cases, engines operate phenomenically in healthy state with the operators usually being unaware of the underlying issues.
- ❖ On the commercial optimisation dimension, the proposed model calculates engine's future generated value by integrating engine performance characteristics (for various degrees of derate), predicted ambient conditions, economic value of engine's life usage, general engine degradation, general engine reliability and, more importantly, electricity market conditions in the form of Spark Spread values. By this way the operator can choose the optimum dynamic and variable derate of the engine and the timing for the repair, based on attractive or not market conditions and expected generated value. The same model takes into consideration the lifing aspects, so the risk for an unplanned outage is managed.

In summary, this work proposes a holistic approach to a serious, often-observed and usually undetected integrity issue. The user can detect, assess, quantify the risk and manage the engine effectively and optimally from both commercial and lifing perspectives. The proposed model of this work has the potential to reduce significantly the costs associated with a lifing issue, by trimming the losses from derated operation and by maintaining the original planned overhauls schedule, while the risk of unplanned failure is also effectively managed. This is important for author's sponsoring company but also in a wider

context for a number of heavy-duty gas turbine owners, who operate in a more than even volatile and competitive global electricity market.

6.3 Future Work

Finally and as already mentioned above, through the course of this research work, a number of interesting for future research topics or just areas where improvements can be made have been identified. The author recognises that these areas, though some of them interact closely with the current project, fall somehow outside his scope. However, recommendations for future work and improvements always can and have to be made.

More specifically, some of the interesting areas that have been identified for further research are:

- *Cooling air system degradation case, due to erosion in pre-swirl nozzles inner passages;*

degradation of pre-swirl nozzles performance through increase of their inner surface roughness is considered as credible mechanism and actually one with favourable for development flow conditions in place. As current research work has explored the impact of degradation on pre-swirl nozzles cavity inlet seals, a research path to explore failure mechanisms on pre-swirl nozzles themselves is highly recommended as future work. It can be in the form of developing models that describe relation between nozzle's roughness and nozzle's discharge coefficient, and conclusively between nozzle's roughness and system's overall effectiveness. Cases in aero-engines or land-based gas turbines where air inlet filtering configuration is not stringent can also be explored, as increased dust content and humidity in the ingested ambient air can further promote favourable conditions for erosive flows inside pre-swirl nozzles. In addition, domestically produced particles (DOD) are not unusual findings in gas turbines. In particular for GT26, the long 22-staged compressor is featured with abradable coating tip-end blades. Particles of these coatings can easily travel through cooling air system and through pre-swirl nozzles. Clearly an increase in roughness inside

nozzles can be expected and this probably happens in the early weeks or months of engine operation, when compressor blades rub on compressor casing to form their final operating tip profile. Increase in roughness will increase friction and ΔP across pre-swirl nozzles. This will decrease total pressure downstream of nozzles, decrease static pressure in same area and consequently affect nozzle's exit velocity. Discharge coefficient of nozzle will also be affected, as flow conditions will further deviate from the ideal ones. In a similar study, Alexiou et al. (3) presented an equation for discharge coefficient C_d correction, by taking into account specific geometrical characteristics of pre-swirl nozzles as well as surface roughness values inside them.

- *CFD and FEA for blade cooling and blade stress;*
the modularised approach of the current work means that every module or objective can be treated separately and higher fidelity solutions can be pursued for every aspect of it. For example a conjugate CFD model for the blade which will provide more accurate blade metal temperature results with details on the localised temperature distribution, as opposed to the mean blade metal temperature of the current work. Similarly, high fidelity blade stress results can be produced through FEA analysis. In the same context, CFD and FEA tools can also be used for the validation of analytical combined with Monte Carlo methods.
- *Transient analysis of an engine with degraded cooling air system;*
current work investigates steady state conditions only, both in clear case and degraded cooling air system case. Transient analysis can examine the impact of engine's degraded cooling air system on start-ups, shut-downs and steep load changes. In this way, the impact of the degraded cooling air system on failure mechanisms such as thermo-mechanical fatigue can be better understood and quantified.

- Advanced Monte Carlo algorithms;*

current work used Monte Carlo algorithms extensively, however in their simplest available form - details can be found in bibliography ref. (XIV). More sophisticated Monte Carlo algorithms, like Markov Chain Monte Carlo (MCMC) or Pseudo-Random Number Generators (PRNG) algorithms are available within MATLAB or other specialised software. Stochastic modelling is of great importance, especially in the field of electricity prices prediction, where the inheritably random nature of the market precludes any other form of deterministic or analytical approach. Any improvement on market prognostics models with use of Monte Carlo or similar algorithms is clearly a promising-value research path.
- Increase of compressor mass flow to mitigate losses due to engine's derate;*

following blade stress calculation results, it was concluded that gas bending and momentum derived stresses contribute trivially on blade's total stress levels, hence they were excluded from being blade stress mitigation options. However, by making use of this outcome, options can be explored to offer higher mass flow in the compressor and expectably higher output, through further opening of the VIGVs. Engine's performance model estimates that a 1kg/sec increase in compressor mass flow can increase engine's output by 1-1.5MW. This option can be further investigated and the output increase to be offered to engines, which are subjected to constant or dynamic derate as a counter measure to their losses.
- Spark Spread – firing temperature automation;*

The proposed model in section 5.4 relies on human intervention to adjust, following Spark Spread variations, firing temperature accordingly. The aim, as discussed, is to optimise commercial position and to manage effectively lifing risks. A level up in automation and sophistication of the whole process could potentially integrate in the same module a machine-

learning algorithm, which reads and interprets market's Spark Spread prices, through available on-line data (81), and then interacts with engine's controller, which is responsible for firing temperature set points. By this way, operator's intervention and human factor potential errors and omissions can be drastically minimised. However, critical features of this automated system must be the maximum thermal gradient that the engine can tolerate as well as the time hysteresis between Spark Spread values and engine's new firing temperature set points.

- *Loading – strength diagrams on Larson-Miller graphs;*
the use of loading-strength diagrams on Larson-Miller graphs was discussed in paragraph 4.8.2. It was recognised that no available references were found in the public domain on the combined use of these two methods for blade materials. The author of this work is confident that further elaboration can take place on this interesting area, aiming to improve the quantification process of the visual effect that thousands Monte Carlo points can have on a Larson-Miller graph. Choudhury's work (33) can be used as a starting point to this direction.

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APPENDICES

Appendix A : Gas Turbine Parts Description

Main parts of the gas turbine are:

- Compressor;

compressor is axial, subsonic with 22 stages. The first three stages of vanes (or stationary blades) are of variable angle in order to control air mass flow. The first stage is called Variable Inlet Guide Vanes (VIGV) and the second and third stages are called Variable Guide Vanes (VGV1 and VGV2). Bleed points from various stages of compressor feed the cooling air system; these are shown in more details in appendix B, figure B.1.

- EnViromental (EV) combustor;

EV combustor is located between the compressor and the high pressure turbine. Combustor's arrangement is annular and it has 24 retractable burners evenly distributed in an annular ring. EV burners are designed like 2 half-cones, which are slightly offset sideways to form 2 slots of constant width. Aim of this shape is to induce a vortex flow inside the cone of the burner.

- High pressure turbine (HPT);

High pressure turbine is a single stage turbine. The blades are fixed with axial fir-tree grooves in rotor's high-pressure-turbine disc and secured in position with axial locks. Cooling of the blades is achieved with a combination of convective and film cooling.

- Sequential EnViromental (SEV) combustor;

SEV combustor re-heats exhaust gasses coming from the high pressure turbine. The design of the combustor is also annular. There are 24 burners distributed evenly in a ring, followed by an annular combustion zone, which is surrounded by transpiration and convection-cooled walls.

- Low pressure turbine (LPT);

Low pressure turbine is located downstream of SEV combustor. It has 4 stages, which are internally air cooled apart from the last one. Cooling method for the first stage of rotating blades is a combination of convective, impingement and film cooling methods. All blades are fixed with axial fir-tree grooves in the rotor.

- Exhaust gas system (EGS);

The exhaust gas system directs hot exhaust gasses from the outlet of low pressure turbine to the heat recovery steam generator (HRSG). An important part of the exhaust system is the diffuser, whose role is to decelerate gas velocity and to guide them to the heat recovery steam generator.

Appendix B : Gas Turbine Cooling Air System Schematic and Description

Cooling air system in GT26 performs the three following tasks:

- I. protects hot gas path (HGP) parts and rotor from overheating,
- II. seals both low pressure and high pressure turbine sections to prevent entry of exhaust gas,
- III. cools down the hot end of compressor.

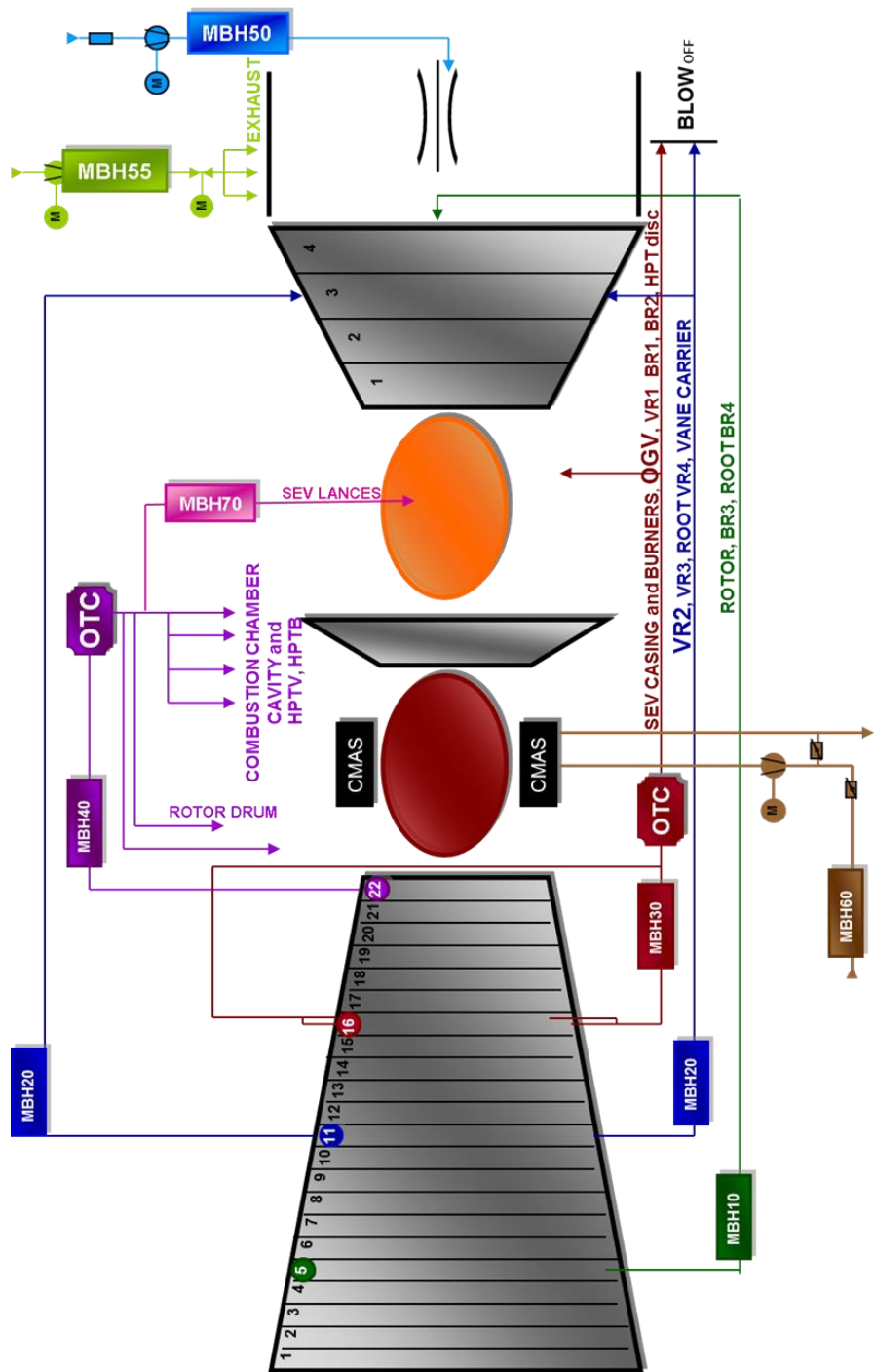
Main sub-systems of the cooling air system are:

- rotor cooling system
- cooling air system for turbine vane carrier
- low pressure cooling air system
- high pressure cooling air system
- exhaust bearing area ventilation
- exhaust gas housing cooling system
- casing mantling air system (CMAS).

Another two very important components of the cooling air system, which are not shown above, are the low pressure and high pressure *Once Through Coolers* (OTCs). These are external to the engine heat exchangers that cool down low and high pressure bleeds from the compressor (MBH30 and MBH40 systems accordingly), before they re-direct the cooling air back to high and low pressure turbines accordingly. This is an important characteristic for GT26's cooling air system and another unique feature, when compared with the other three OEMs' approach in this area.

The in-depth understanding of details and particularities of GT26's cooling air system is of great importance for the project, as this knowledge is essential for

the development of a realistic performance model for the engine, a topic which is discussed section 4.2.



Figure_Apx B-1: GT26 cooling air system schematic

A brief description of GT26 cooling air system, based on the above schematic is cited below:

- i. Compressor bleed MBH10³⁸
from: compressor's 5th stage,
cools: via rotor's central bore, 3rd stage low pressure turbine blades, 4th stage blades roots and 3rd and 4th stages low pressure turbine heat shields.
- ii. Compressor bleed MBH20
from: compressor's 11th stage
cools: 2nd and 3rd low pressure turbine vanes, vane carrier, 2nd, 3rd, 4th stages low pressure turbine heat shields and 4th stage low pressure turbine vane roots.
- iii. Compressor bleed MBH30
from: compressor's 16th stage
cools: 1st and 2nd stage LPT blades, rotor and heat shields between low pressure turbine inlet and 3rd stage low pressure turbine blades.
- iv. Compressor bleed MBH40
from: compressor 22nd stage
cools: HPT vanes and blades, HPT disc and rotor drum.

³⁸ Note on the use of KKS:

KKS (Kraftwerks Kennzeichen System) identification system for power stations is a technical and logical classification and nomenclature system for power stations. With KKS all plant sections and components are identified on a system-related basis and by this way a common language amongst manufactures, plant owners and operators can be utilized. Alstom, like most of the European turbine manufactures, use exclusively KKS system for their products. KKS system is also used in this project for the description of GT26 parts. KKS letters MBH (MBH10 – MBH70) refer to gas turbine cooling air systems. MBH30 in particular is the system that supplies cooling air to LPT pre-swirl nozzles.

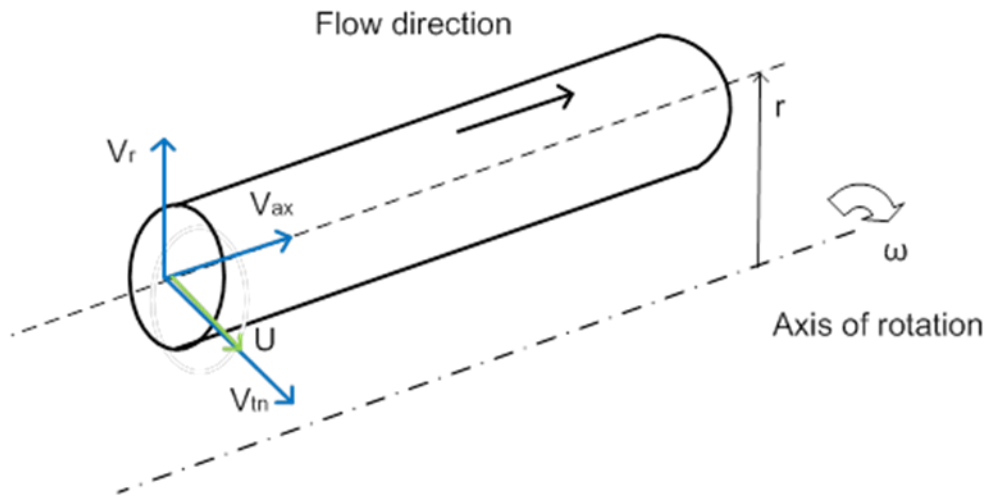
- v. External supply³⁹ MBH50
from: external fan
cools: bearing 1 area (casing, support, cables and instrumentation in the area).
- vi. External supply MBH55
from: external fan
cools: struts and exhaust housing.
- vii. External supply MBH60
from: external fans
cools: casing mantling system (CMAS).
- viii. Compressor bleed MBH70
from: compressor 22nd stage
cools: SEV lances.

³⁹ As external to the engine, not through compressor bleed(s)

Appendix C : Rotational and Stationary Reference Frame Theory

Figure C.1 shows a typical rotating passage. The passage is rotating with rotational speed ω and the passage wall velocity is (82), (57), (58):

$$U = \omega r \quad (C- 1)$$



Figure_Apx C-1: Typical example of rotating passage

In the stationary reference frame (out of the rotating passage), an observer sees a fluid which is characterised by axial velocity V_{ax} , radial velocity V_r and tangential velocity V_{tn} . Fluid's static temperature is T_s , which is invariant and does not change with the reference frame. Fluid's total temperature is given by equation:

$$T_{tot} = T_s + \frac{V_{tn}^2 + V_r^2 + V_{ax}^2}{2 C_p} \quad (C- 2)$$

In the rotational reference frame (inside the rotating passage), an observer sees the same axial V_{ax} and radial velocity V_r , however the tangential velocity is now $V_t - U$. Consequently, the total temperature in the rotational reference frame $T_{tot, rel}$ is given by equation:

$$T_{\text{tot,rel}} = T_s + \frac{(V_{\text{tn}} - U)^2 + V_r^2 + V_{ax}^2}{2 C_p} \quad (\text{C- } 3)$$

By deducting C-3 from C-2:

$$T_{\text{tot}} - T_{\text{tot,rel}} = \frac{2 U V_{\text{tn}} - U^2}{2 C_p} \quad (\text{C- } 4)$$

This can also be re-written as:

$$V_{\text{tn}} = \frac{(T_{\text{tot}} - T_{\text{tot,rel}}) 2 C_p + U^2}{2 U} \quad (\text{C- } 5)$$

Appendix D : Mesh Quality Metrics

ANSYS mesher uses a number of different mesh quality metrics. The two most important are *skewness* and *orthogonal quality* (69).

Skewness is defined as:

$$\text{Skewness} = (\text{optimal cell size} - \text{cell size}) / \text{optimal cell size}$$

Orthogonal quality (OQ) is the other important quality characteristic for ANSYS mesher. It is defined as the minimum of the below ratios for every face i . Figure D.1 illustrates the vectors that are used to determine orthogonal quality of a cell.

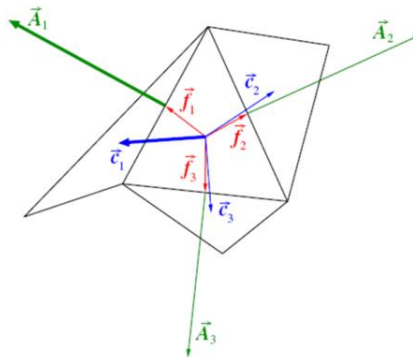
Equations are:

$$A_i f_i / |\vec{A_i}| |\vec{f_i}|$$

$$A_i c_i / |\vec{A_i}| |\vec{c_i}|$$

Where,

$ \vec{A_i} $	Face normal vector
$ \vec{f_i} $	Vector from centroid of cell to centroid of that face
$ \vec{c_i} $	Vector from centroid of cell to centroid of adjacent cell that shares face



Figure_Apx D-1: Vectors used to compute Orthogonal Quality for a cell

Apart from mesher's quality metrics (skewness and orthogonal quality), CFX solver uses also its own quality criteria for the imported mesh. This is explained by the fact that CFX can import mesh files from different mesh generation algorithms (for example ICEM) so maintaining a separate independent mesh quality assessment process is important. CFX's mesh quality metrics are the *orthogonal angle*, *expansion factor* and *aspect ratio*.

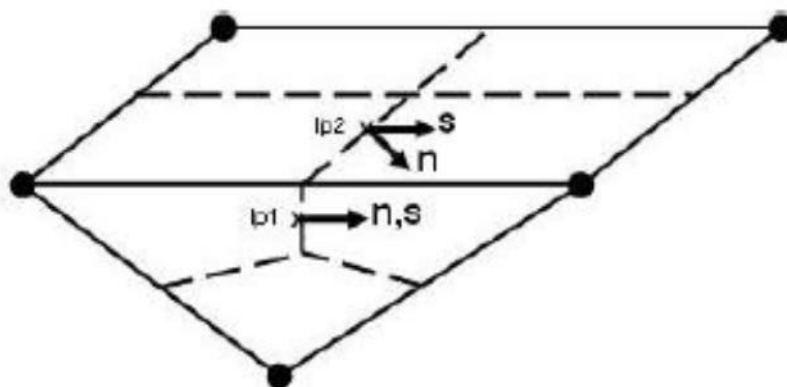
Starting from *orthogonal angle*, the defining equation is:

$$\text{Orthogonal angle} = 90^\circ - \arccos(n \cdot s)$$

Where,

n ip-face normal vector

s node-to-node vector



Figure_Apx D-2: Orthogonal angle vectors

Expansion factor measures how well the nodal position corresponds to control volume centroid of a cell. As a definition, expansion factor is ratio of largest to smallest element volumes surrounding a node.

Aspect ratio measures how stretched a control volume is. As a definition, aspect ratio is ratio of largest to smallest face areas surrounding a node.

All three indicators (orthogonal angle, expansion factor and aspect ratio) are quite important for the solution as deviation from recommended values can lead

to serious discretisation errors. Table D.1 lists thresholds for the three indicators. Annotations that solver provides during solution are OK (good), ok (acceptable) and ! (poor). In addition, the solver provides a percentage of number of elements with good, acceptable and poor quality characteristics during run. As a general advice, when percentage of poor (!) elements is less than 1% no further improvements are required (65).

Table_Apx 1: ANSYS Solver mesh quality thresholds

Maximum Aspect Ratio (double precision)	OK <10000
	ok 10000<100000
	! >100000
Maximum Expansion Factor	OK <5.0
	ok 5.0<20.0
	! >20.0
Minimum Orthogonal Angle	OK >50°
	ok 50°>20°
	! <20°

Appendix E : Advection Scheme Theory for CFX

In CFX the advection term requires the integration point values of a property ϕ to be approximated in terms of its nodal values. The equation that is used is (69):

$$\phi_{ip} = \phi_{up} + \beta \nabla \phi \cdot \Delta \vec{r}$$

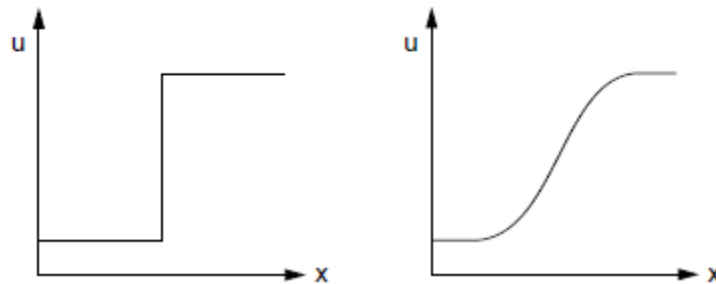
Where,

- ϕ_{ip} value of the property ϕ at the integration point
- ϕ_{up} value of the property ϕ at the upwind node
- β control parameter that allow different choices for schemes as described below
- $\vec{r}$ vector from the upwind node to the integration point

For β CFX offers three choices:

I. 1st Order Upwind

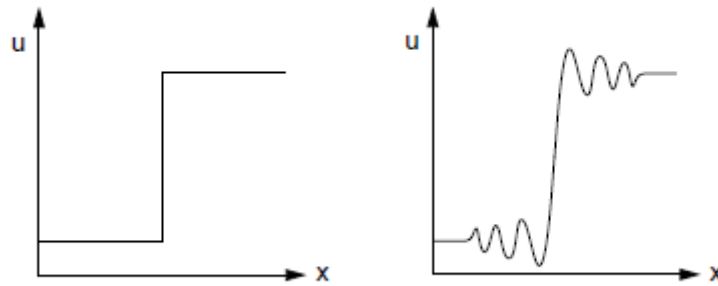
β becomes zero ($\beta=0$), so equation C.1 becomes very simplified. This scheme is quite robust but unavoidably discretisation errors are introduced that tend to smear steep spatial gradients as shown in figure C-1.



Figure_Apx E-1: CFX theory, 1st Order Upwind

II. Specified Blend Factor

β takes values between 0 and 1. The resulting discretisation is more accurate than 1st Order Upwind and can reproduce steep spatial gradients that 1st Order misses. However, it is an unbounded technique that can cause non-physical oscillations to the property values and introduce by this way dispersive discretisation errors (figure C-2).



Figure_Apx E-2: CFX theory, Specified Blend Factor

III. High Resolution Scheme

The High Resolution Scheme uses a special technique for β at each node, computed to be as close to unity (1) as possible but without the introduction of new extrema. This technique computes $\phi_{\min}$ and $\phi_{\max}$ at each node and ensures that β does not overshoot $\phi_{\max}$ or undershoot $\phi_{\min}$. Though admittedly High Resolution Scheme offers the most accurate discretisation method, the disadvantage is that the solver sometimes encounters robustness issues.

Appendix F : Mesh Sensitivity Analysis

For mesh (grid) sensitivity study, Richardson's Extrapolation method is used. Chaplin's work (71) was used as reference for Richardson's method. Definition of Grid Refinement Ratio is:

$$r_{\text{eff fine, med}} = (N_{\text{fine}} / N_{\text{med}})^{1/3} \quad \text{C- 1}$$

$$r_{\text{eff med, coar}} = (N_{\text{med}} / N_{\text{coar}})^{1/3} \quad \text{C- 2}$$

Where,

N_{fine} number of elements for fine mesh

N_{med} number of elements for medium mesh

N_{coar} number of elements for coarse mesh

Definition for Observed Order of Convergence:

$$P_{\text{con}} = \ln \left(\frac{C_{x, \text{coarse}} - C_{x, \text{med}}}{C_{x, \text{med}} - C_{x, \text{fine}}} \right) / \ln r \quad \text{C- 3}$$

Definition for Richardson's Extrapolation:

$$C_{x, 0} = C_{x, \text{fine}} + (C_{x, \text{fine}} - C_{x, \text{med}}) / (r_{\text{eff}}^{P_{\text{con}}} - 1) \quad \text{C- 4}$$

Definition for Grid Convergence Index:

$$GCI_{\text{med, coar}} = \frac{F_s \left| \frac{C_{x, \text{med}} - C_{x, \text{coar}}}{C_{x, \text{med}}} \right|}{r_{\text{eff}}^{P_{\text{con}}} - 1} \quad \text{C- 5}$$

$$GCI_{\text{fine, med}} = \frac{F_s \left| \frac{C_{x, \text{fine}} - C_{x, \text{med}}}{C_{x, \text{fine}}} \right|}{r_{\text{eff}}^{P_{\text{con}}} - 1} \quad \text{C- 6}$$

Finally, Asymptotic Range results from equation:

$$AR = GCI_{med, coar} / (GCI_{fine, med} r_{eff}^{P_{con}})$$
C- 7

The Asymptotic Range should give a result close to unity (~1).

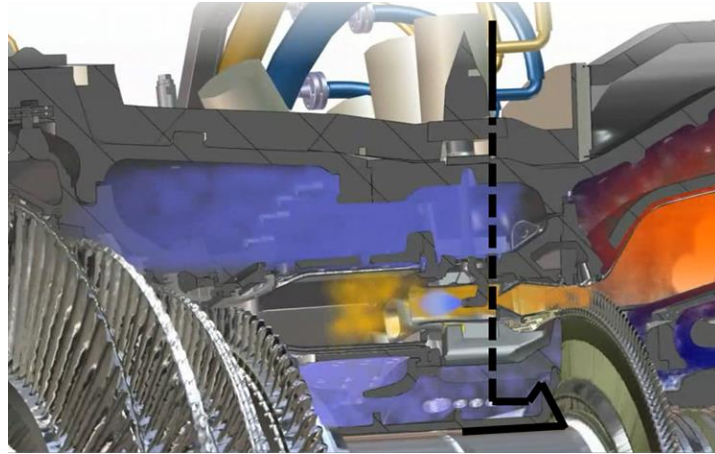
Appendix G : Borescope Inspection Path to Pre-swirl Nozzles Cavity

As the root cause of cooling air system degradation is the relaxation of the pre-swirl nozzles cavity inlet seals, it is useful to show the borescope path that enables the maintenance personnel to detect and quantify the issue. The location of the seals is shown in Figure G.1. As they cannot be repaired in a closed-engine event, such as A and B Inspections, their repair requires an open-engine overhaul event, namely a C Inspection (paragraph 3.2.3). C Inspections take place every 4 to 5 years; hence, it is important to manage the lifing and commercial position of the engine upon identification of the problem.

The seals can be remotely inspected with the use of borescope. This is extremely important as by this way the need for open-engine inspection is removed. Figure G.1 illustrates the exact path for the borescope, through an SEV lance bore. Borescope images are able to offer a dimensioned view of the seals. Comparing the as found condition of the seals with the as-build (design), the operator should be in position to calculate the new clearance between rotor and casing.

Flow through the seals can be calculated with a plethora of 1D CFD software, among others the author has used FlowMaster (82). The geometry is known, while Optigen provides the boundary conditions upstream of the seals. The results with the increased flow through the seals feed to the CFD model in paragraph 4.4.3. As there was no validation case for this calculation (though experiment or high fidelity CFD etc) this piece of work was deemed as more appropriate to be cited in the appendices rather in the main body of the thesis.

— — — Borescope Path



Figure_Apx G-1: Borescope path for PSNs cavity inlet seals

Appendix H : Gas Turbine Maintenance Events

As part of the contractual operational and maintenance (O&M) support, OEM prescribes three types of scheduled maintenance events:

- **A Inspection;**

It includes primarily a visual through borescope inspection of the engine. A selected number of combustion lances (EV and SEV) are pulled out, so visual inspection of high and low-pressure-turbine vanes and blades is feasible. In addition, through purpose-made plugs on compressor casing, visual inspection of some of compressor stages is also feasible.

- **B Inspection;**

It is very similar to an A Inspection, but includes also calibration of critical instruments and sensors, for example bearings' vibration sensors, combustion pulsation sensors and others. It also defines to an extent scope of work for C Inspection.

- **C Inspection;**

It is the major overhaul event of the engine. It includes removal of all upper compressor and turbine casings, removal of rotor, detailed inspection of compressor with its blades with use of non-destructive tests (NDT) and replacement of all hot gas path parts. Hot gas path parts that have one cycle life (36,000 EOH or 32,000OH) are scraped, while parts that are designed for 2 cycles (72,000 EOH) are being re-conditioned (rejuvenated) and re-used.