

Sensitivity of
Nozzle Guide Vane Flow Capacity
to Geometric Changes

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Thanks

Abstract

Contents

List of Figures	6
List of Tables	7
1 Flow capacity - definition and derivation	9
1.1 Mass flow rate through an NGV	10
1.2 NGV capacity in 1 dimension	11
2 Geometric throat area	12
2.1 1D vs 2D capacity uncertainty	13
2.2 2D vs 3D capacity uncertainty	18
3 Trailing edge	19
3.1 Definitions of loss for trailing edge performance evaluation . . .	19
3.2 Effects of trailing edge shape uncertainty on capacity predictability	21
3.3 Performance of an alternative trailing edge design	23
4 Leading edge	26
4.1 sensitivity of capacity to cooling holes on the leading suction side	26
4.2 Consideration of corrections necessary to interpret 2D result . .	29

Bibliography	30
Bibliography	31

List of Figures

2.1	Considerations for defining minimum area	13
2.2	Creation of 2D mid-span approximation of NGV geometry	15
2.3	Computational domain and boundaries	16
2.4	Mesh for Trent 900 smooth-vane capacity study	17
4.1	Details of mesh for XWB single cooling hole study	28
4.2	Total capacity percentage delta as a function of single cooling hole position	29

List of Tables

2.1	Boundary conditions at design pressure ratio for multi-vane capacity study	16
3.1	Boundary conditions for trailing edge suction-side flange shape uncertainty study	22
3.2	Boundary conditions for trailing edge centred-ejection design study	24
4.1	Boundary conditions for suction-side cooling hole position study .	27

Nomenclature

Romans

Greeks

Acronyms and Abbreviations

Subscripts

Chapter 1

Flow capacity - definition and derivation

The capacity of an internal combustion engine is an intuitive concept. It is simple to derive from the engine's geometry, it has tangible units of volume, and it provides an heuristic for the engine's size, performance, and air mass flow rate.

The design of turbomachinery invites an analagous concept to that of IC engine capacity, but a definition is not so obvious. Mass flow rate through an engine's nozzle guide vane is a function of the NGV's geometry and of its boundary conditions. If the mass flow rate can be quantified in a way which mitigates the boundary conditions, then the effects of an NGV's geometry on its mass flow rate may be isolated. A particular NGV will thus have a mass flow rate capacity, just as a particular IC cylinder has a volumetric capacity.

Defining a geometric mass flow rate capacity allows for experimental testing of NGVs without recreating the extreme boundary conditions to which real NGVs are exposed. Setting the correct pressure ratio is sufficient, as the subsequent derivation will show. This expedites the testing of different NGV geometries' effects on capacity.

The predictability of NGV capacity affects the design of every downstream turbine stage. If the NGV mass flow rate differs from its expected value due to a poor prediction, all subsequent turbine stages will be sized for an incorrect mass flow rate. The resulting errors in flow velocity and pressure will compound with each additional stage, leading to increasingly incorrect specification of turbine sizes and turning angles.

Andrea Guiffre' and Matteo Pini [1] performed numerical analysis to discuss scaleable guidelines for turbine stage design, validating their model using high-

fidelity CFD. Correct stage matching was found to be highly dependent on matching the *volumetric flow ratio*, defined by the authors as a stage's total-to-static density ratio. To accurately quantify this parameter within the analytical and computational methods discussed in this thesis, flow capacity would need to be predicted correctly.

A strong understanding of capacity predictability should allow engine-makers to pre-empt geometric changes that happen to NGVs during service, such as erosion and cooling hole blockage. It should also account for geometric uncertainties arising from the manufacturing process. While this study advocates for improved accuracy of capacity predictions, emphasis is placed on how these predictions are limited by the unpredictability of real-world manufacture and service.

1.1 Mass flow rate through an NGV

In one dimension, an engine nozzle may be modelled as a compressible flow from an upstream reservoir of total pressure p_0 , accelerating to velocity v and density ρ through a nozzle of cross-sectional area A . Mass flow rate through the nozzle is thus

$$\dot{m} = \rho A v \quad (1.1)$$

where density may be expressed as a function of *pressure ratio*, the ratio of the nozzle pressure to the total pressure

$$\rho = \frac{p_0}{RT_0} \left(\frac{p}{p_0} \right)^{\frac{1}{\gamma}} \quad (1.2)$$

and velocity is given by the compressible form of Bernoulli's equation as

$$v = \sqrt{2 \left(\frac{\gamma}{\gamma - 1} \right) \left[\frac{p_0}{\rho_0} - \frac{p}{\rho} \right]} \quad (1.3)$$

The above equations combine to express mass flow rate through the nozzle as

$$\dot{m} = A \frac{p_0}{RT_0} \left(\frac{p}{p_0} \right)^{\frac{1}{\gamma}} \sqrt{2 \left(\frac{\gamma}{\gamma - 1} \right) \left[\frac{p_0}{\left(\frac{p_0}{RT_0} \right)} - \frac{p}{\left(\frac{p_0}{RT_0} \right) \left(\frac{p}{p_0} \right)^{\frac{1}{\gamma}}} \right]} \quad (1.4)$$

which simplifies to

$$\dot{m} = A \frac{p_0}{RT_0} \left(\frac{p}{p_0} \right)^{\frac{1}{\gamma}} \sqrt{2 \left(\frac{\gamma}{\gamma - 1} \right) \left[RT_0 - RT_0 \left(\frac{p}{p_0} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (1.5)$$

$$\dot{m} = \frac{p_0}{\sqrt{T_0}} A \sqrt{\frac{\gamma}{R}} \left(\frac{p}{p_0}\right)^{\frac{1}{\gamma}} \sqrt{\left(\frac{2}{\gamma-1}\right) \left[1 - \left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}}\right]} \quad (1.6)$$

1.2 NGV capacity in 1 dimension

Capacity is defined as

$$\Gamma = \frac{\sqrt{T_0}}{p_0} \dot{m} \quad (1.7)$$

This provides an expression of mass flow rate independent of upstream total pressure p_0 and upstream total temperature T_0 . The expression is purely a function of throat area A and pressure ratio $\frac{p}{p_0}$:

$$\Gamma = A \sqrt{\frac{\gamma}{R}} \left(\frac{p}{p_0}\right)^{\frac{1}{\gamma}} \sqrt{\left(\frac{2}{\gamma-1}\right) \left[1 - \left(\frac{p}{p_0}\right)^{\frac{\gamma-1}{\gamma}}\right]} \quad (1.8)$$

A scale constant σ is defined as

$$\sigma = \sqrt{\frac{2\gamma}{R(\gamma-1)}} \quad (1.9)$$

for a compact expression of capacity as a function of throat area A and pressure ratio r :

$$\Gamma(A, r) = \sigma A \sqrt{r^{\frac{2}{\gamma}} \left(1 - r^{\frac{\gamma-1}{\gamma}}\right)} \quad (1.10)$$

This expression has its maximum value at the critical pressure ratio

$$r_c = \left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{1-\gamma}} \quad (1.11)$$

At lower ratios, the nozzle is choked and mass flow rate cannot increase further. Choked capacity is given by

$$\Gamma_c(A) = \sigma A \sqrt{\left(\frac{\gamma+1}{2}\right)^{\frac{2}{1-\gamma}} - \left(\frac{\gamma+1}{2}\right)^{\frac{1+\gamma}{1-\gamma}}} \quad (1.12)$$

It is shown that the capacity of one-dimensional nozzle flow is a function of only the flow's minimum area, provided the flow is choked and the ratio of specific heats is assumed constant.

The following chapters will analyse the capacity of two-dimensional and three-dimensional nozzle flows, presenting and discussing analytical techniques for applying the 1D capacity equation to 2D and 3D data. In such cases, capacity will be quantified using equation 1.7.

Chapter 2

Geometric throat area

A 1-dimensional supersonic nozzle is equivalent to a single streamline of variable cross-sectional area. It is possible to solve for the flow conditions throughout the nozzle, provided area is specified as a function of position along the nozzle. The mass flow rate capacity is a function of only the flow's minimum area, as in equation 1.12.

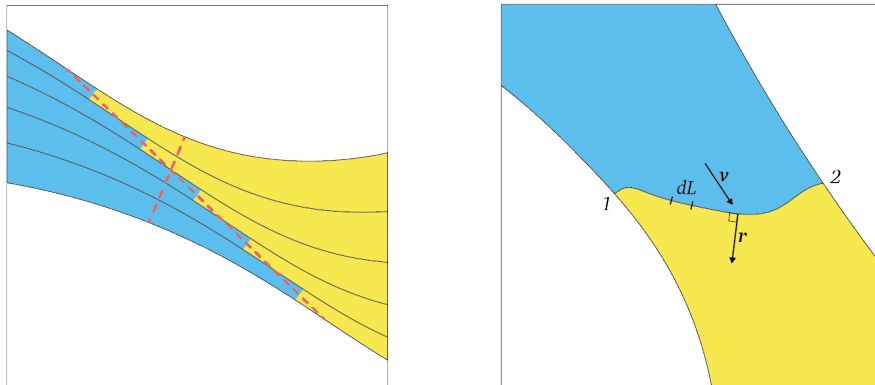
A 2-dimensional nozzle may be modelled as a group of adjacent streamlines, each of which may have dissimilar area functions. The streamlines' minimum areas may not correspond to a straight line across the narrowest part of the nozzle.

This is illustrated in Figure 2.1a by considering the division of a 2-dimensional flow into a finite number of streams of finite width. Each stream has an individual line of minimum width where sonic conditions exist. Collectively these lie on a line distinct from the overall passage line of minimum width. These two definitions of the effective throat line are marked.

If this concept is extended to infinitesimal streamlines (and the flow is isentropic) each streamline will experience sonic conditions at its point of minimum area, combining to form the 2-dimensional sonic line. The effective throat area of a 2-dimensional nozzle is thus the sum of its streamlines' throat areas. This is distinct from the sonic line length, and may be expressed by the integral

$$\int_1^2 \hat{\mathbf{v}} \cdot \hat{\mathbf{r}} dL \quad (2.1)$$

where $\hat{\mathbf{v}}$ is the unit vector of local flow velocity, $\hat{\mathbf{r}}$ is the unit vector perpendicular to the local sonic line, and the integral is performed on the scalar infinitesimal dL over the length of the sonic line, as illustrated in Figure 2.1b.



(a) Different minimum area points of dissimilar streams

(b) Integral for 2D nozzle equivalent throat area

Figure 2.1: Considerations for defining minimum area

–Review any definitions of minimum area from the literature.

Deepak Thirumurthy et al [2] discussed the challenges and uncertainties associated with the accurate matching of an aeroderivative gas generation turbine. The authors’ focus was on various means of predicting capacity to facilitate proper matching.

The authors noted that “the throat area of any blade or vane row has a strong influence on the overall capacity of the turbine. The vane or blade throat plane...is defined as the plane formed by the smallest passage area normal to the flow path. The effects are of first order when the throat area is changed for the first stage of the turbine.”

2.1 1D vs 2D capacity uncertainty

–Introduce section by talking about how 2D CFD is necessary because, unlike 1D where throat = min area, the throat surface in 2D and 3D is not known until the flow is already solved.

– A family of 2D NGVs were run in CFD. The point is to see how their capacity varies when plotted against various definitions of throat area. The usefulness of each definition will be discussed.

Geometric variation among Rolls-Royce Trent 900 NGVs has been subject to extensive measurement by Rolls-Royce. In one such set of studies, Terry Hall [3]

and Giulio Zamboni [4] provided geometric definitions of 6 NGVs. These studies considered 2 slightly different production standards within the Trent 900 NGV family, which are referred to by Rolls-Royce as the M-skew standard and the EP1 standard.

Rolls-Royce randomly selected three production NGVs from each standard and measured their surface geometries using the *white light/GOM* scanning method. This resulted in the geometric definition files which were provided to the present study by Hall and Zamboni. These files do not include film cooling holes or the trailing edge slot. The trailing edges are smoothly rounded.

–Need to talk about how the family of vanes show variations out of the factory, even though they have no cooling features. But I must check exactly what RR's definition of the minimum area was. Was it in fact affected by the TE slot?

– Present boundary conditions:

The 3D NGV surface shape was provided by Rolls-Royce as a set of co-ordinates which define 21 closed loops on the vane surface. Each loop consists of a pair of streamlines which diverge from a stagnation point on the vane's leading edge and converge at the trailing edge. The 11th loop was thus considered to be the closest approximation to a mid-span slice of the NGV.

MATLAB was used to process the 3D curve of the 11th loop, producing a 2D section via the following process.

1. Project the loop's coordinates onto a cylindrical surface whose radius is equal to the NGV annular radius at mid-span.
2. Periodically repeat the resulting shape in the circumferential direction to create an annulus of 2D vanes wrapped around the cylinder.
3. Transform the annulus into an infinite 2D linear cascade by taking the circumferential coordinate to be a vertical coordinate.

This process is illustrated in Figure 2.2.

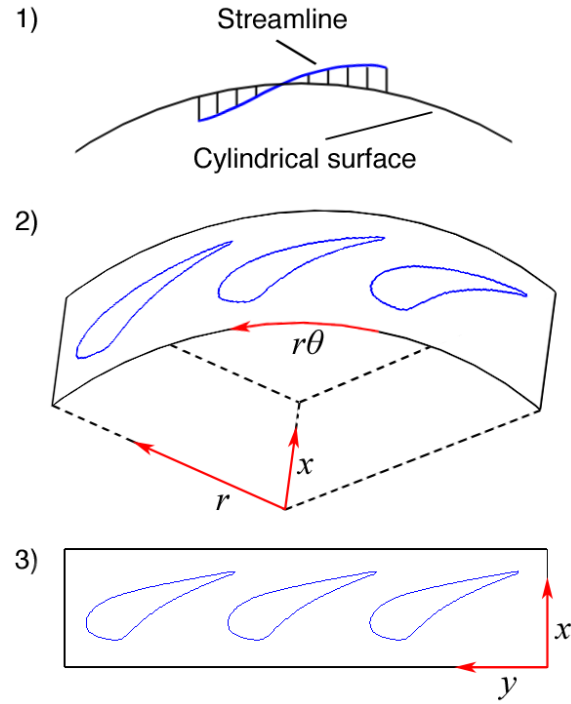


Figure 2.2: Creation of 2D mid-span approximation of NGV geometry

The resulting coordinates were connected to form a closed body. This body was placed in a computational domain consisting of a straight inlet section, a curved turning section, and a straight outlet section inclined at the NGV turning angle. The domain was modelled in Ansys Fluent, where the inlet boundary was specified to have total pressure p_{01} and the outlet was specified to have static pressure p_2 . The remaining domain boundaries were made periodic with one another to preserve the 2D linear cascade. The NGV surface boundary was specified as a solid wall with a no-slip boundary condition and no heat conduction. The resulting computational domain is illustrated in Figure 2.3.

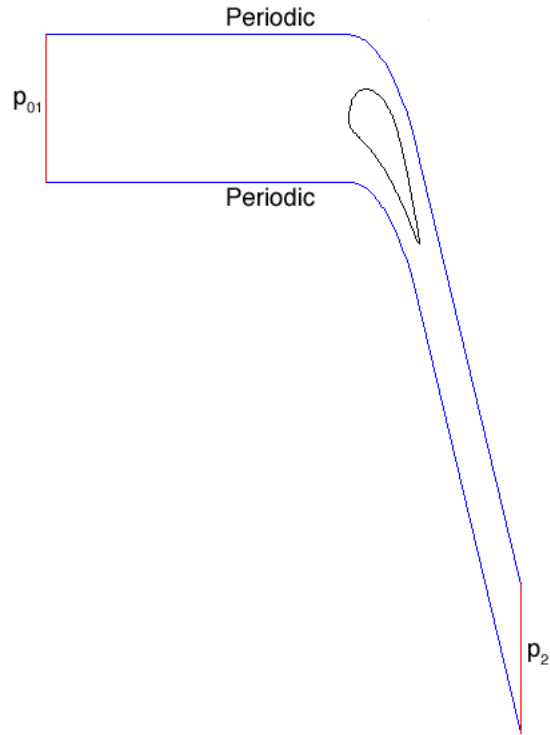


Figure 2.3: Computational domain and boundaries

This configuration has been the basis for all other 2D CFD analyses in the present study, where additional boundaries are added to account for cooling features as required. Values for the simulation were set according to Table 2.1.

Table 2.1: Boundary conditions at design pressure ratio for multi-vane capacity study

Parameter	Value
NGV series	Rolls-Royce Trent 900
NGV turning (degrees)	76.89
Design pressure ratio	1.79
Inlet total pressure (Pa)	4.33×10^6
Inlet total temperature (K)	300
Outlet static pressure (design) (Pa)	2.42×10^6
Outlet static temperature (Pa)	300
Solver type	Density-based
Cell count	35,000

- Present mesh

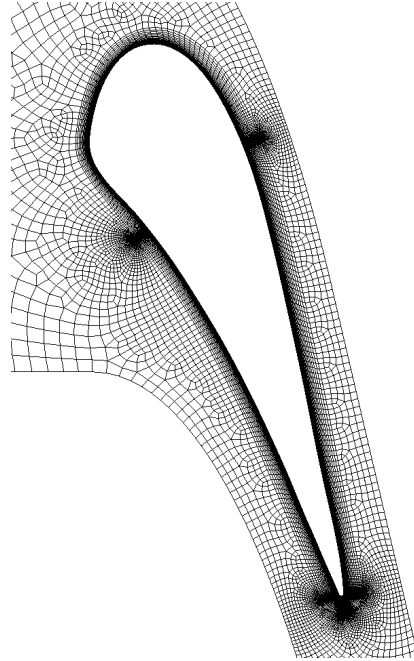


Figure 2.4: Mesh for Trent 900 smooth-vane capacity study

- When the "throat area" of the 2D family is plotted against their capacity at a choking PR, it shows a pretty good proportional relationship. This is despite some obvious differences in the shape of the M1 line.
- Why are there still small deviations? That is, can I present a better definition of throat area that fits even better? Possible topics:
 - Most likely it will follow the 1D rule if it is taken to be that streamline's M1 point. But not certain.
 - the relationship between the minA line and the M1 line
 - the total length of each M1 line
 - a plot of capacity vs M1 line length
- Why aren't there larger deviations? That is, can I derive a sensible geometric parameter and show that it isn't changing significantly?
- Is this all really at design PR? What if the M1 lines were obtained at more choked PRs but I forgot to say so? What does the correlation look like at other PRs?

2.2 2D vs 3D capacity uncertainty

- 3D CFD data was shared by Rolls-Royce for comparison with the 2D data. The point is to see why 3D is still so different from 2D, ie what are we failing so badly to predict with 2D?
- Discuss the most important ways in which the 3D capacity trends are different from the 2D ones. From a superficial look it appears that the 3D trends never choke. What could this mean about their having a M1 line at all?
- What does the correlation look like at other PRs?
- Review 3D effects from the literature.

Chapter 3

Trailing edge

3.1 Definitions of loss for trailing edge performance evaluation

Loss may refer to any process whereby the recoverable energy of a turbomachinery flow is reduced during transit through a turbine stage, other than by shaft work, with corresponding entropy creation. Minimisation of loss remains a primary objective of turbomachinery design, but there is not a universal consensus on how to quantify it.

Daniel Back da Trindade et al [5] characterised sources of loss as shock loss, profile losses, tip leakage, endwall losses, and cooling losses. Tip leakage results from flow through the clearance space between the vane tips and the casing. It will not be considered by the present study, since the present study is of fixed NGVs. Endwall losses result from 3-dimensional secondary flows, and will not be considered in the present study which is of 2-dimensional phenomena.

Trindade et al characterised shock loss as resulting from entropy creation across the shock wave during supersonic flow. The present study will consider this mechanism during discussion of CFD results at choking pressure ratios.

Profile loss was characterised by entropy creation by viscosity in the boundary layer on the blade surface. Entropy generated near the blade's trailing edge was considered as profile loss due to the high entropy creation in the wake region following flow separation at the trailing edge. The authors extended this definition to include the entropy generated as the flow turns tightly around the trailing edge curvature and forms expansion waves.

The authors also defined the category of cooling losses to refer to the aerodynamic penalties incurred by cooling in general, namely “thicker blade profiles from coolant holes, interaction of coolant film with the blade boundary layer, mixing losses between coolant and main flow and endwall losses.” The present study seeks to isolate the mechanisms whereby film cooling causes boundary layer changes and mixing effects, discussing film cooling in Chapter 4.

The present study seeks to define a coefficient of NGV total pressure loss whose arguments are flow measurements or numerically predicted parameters. Giel et al surveyed the total pressure wake profiles of a blade at mid-span, defining a *total pressure coefficient* as

$$Cp_t = \frac{p_{01} - p_{02}}{p_{01} - p_2} \quad (3.1)$$

where p_{01} and p_{02} are the inlet and wake total pressures, and p_2 is the wake static pressure. This definition is to be used for the total pressure wake profiles in this thesis. The authors also defined a *kinetic energy loss coefficient* as

$$e_2 = \frac{\left(\frac{p_{01}}{p_{02}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{p_{01}}{p_2}\right)^{\frac{\gamma-1}{\gamma}} - 1} \quad (3.2)$$

Giel et al’s results will be compared to those of this study, considering wake profiles and loss as a function of trailing edge thickness.

Jie Gao et al [6] defined a slightly different *total pressure loss coefficient* as

$$C_{pt} = \frac{p_{01} - p_{02}}{p_{02} - p_2} \quad (3.3)$$

–Discuss these two slightly different definitions

–Saha paper has its own loss definitions too - seems to be doing a definition that considers the coolant/mainstream mass flow ratio.

$$\zeta = 1 - \frac{(1 + Y) \left(1 - \left(\frac{p_2}{p_{02}}\right)^{\frac{\gamma-1}{\gamma}}\right)}{\left(1 - \left(\frac{p_2}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}}\right) + Y \left(1 - \left(\frac{p_2}{p_{0c}}\right)^{\frac{\gamma-1}{\gamma}}\right)} \quad (3.4)$$

where Y is the coolant-to-mainstream mass flux ratio.

In the case of no coolant flow this becomes

$$\zeta = 1 - \frac{1 - \left(\frac{p_2}{p_{02}}\right)^{\frac{\gamma-1}{\gamma}}}{1 - \left(\frac{p_2}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}}} \quad (3.5)$$

3.2 Effects of trailing edge shape uncertainty on capacity predictability

If nozzle guide vane flow capacity is known to be a function of effective throat area, we must survey the practical reasons for which effective throat area might vary. In a real engine, NGV effective throat area might depart from its design value if the NGVs depart from their design shape, either because of the finite precision of manufacturing, or because of erosion while in service.

A goal of the present study is to help ascertain how much these off-design conditions might limit the predictability of NGV flow capacity, and to comment on ways to mitigate this limitation. The NGV trailing edge is an area of focus because it is particularly susceptible to both finite-precision and in-service departures from its design shape.

The trailing edge is susceptible to finite-precision departures from its design shape because of how it is manufactured. Vane pairs are cast to a finite precision as discussed in Section 2.1, where significant geometric variation appears in vanes even without cooling features. The vanes are then subject to machining to finish off the trailing edge shape. The finishing creates the most aerodynamic trailing edge possible, with a thin suction-side flange that cleanly meets the endwalls, and a flange chord length that is consistent across the span of the trailing edge. The creation of a repeatable minimum geometric area cannot be said to be a primary objective.

The trailing edge is susceptible to in-service departures from its design shape because it is the most fragile part of the vane. The NGV remains largely intact over a lifetime of extremely hot supersonic flow, but the trailing edge flange may be subject to significant erosion. If there is a blockage in any of the plena which feed the trailing edge slot, part of its span will receive reduced coolant flow, accelerating the degradation of the trailing edge flange near that area. Such cooling failures are not uncommon in sandy environments.

This study will consider the potential worst case for in-service erosion of the suction-side flange, namely its complete deletion. This worst-case erosion is assumed to take place at the NGV mid-span because this area is exposed to the greatest flow speeds and temperatures. This justifies 2-dimensional simulation of the mid-span flow. The goal is to parameterise changes to the flange and comment on how these changes alter the effective minimum area and the NGV flow capacity.

2D CFD has been used to model the effects of incremental removal of material from the trailing-edge suction-side flange of an NGV.

–Present boundary conditions

–Geometry made by Fig 2.2

–Domain is Fig 2.3

–BCs are:

Table 3.1: Boundary conditions for trailing edge suction-side flange shape uncertainty study

Parameter	Value
NGV series	Rolls-Royce Trent 900
NGV turning (degrees)	76.89
Design pressure ratio	1.79
Inlet total pressure (Pa)	4.33×10^6
Inlet total temperature (K)	300
Outlet static pressure (design) (Pa)	2.42×10^6
Outlet static temperature (Pa)	300
Solver type	Density-based
Cell count	35,000

–Present mesh with larger and better pictures, and show how the mesh is altered as erosion amount varies –Plan of action - figure out how to save a specific viewpoint in Fluent, and how to take good pictures. Make sure this viewpoint is suitable for both the SS and PS pictures.

–Show capacity trend for all the data as an initial summary, then show capacity vs erosion amount for a couple of relevant PRs (perhaps design and fully choked)

–2D CFD predicts a maximum 2 percent change in capacity (when the flange is completely gone)

–What is changing in the flow as a result of the erosion, and is it driving capacity change via the expected route of throat area changes (whatever we’ve decided is the best definition of throat area in the previous chapter)

–Cite the state of the art for measuring the capacity of a ring of real parts - Burdett: –Daniel Burdett - Analysis of ultra-low uncertainty gas turbine flow capacity measurement techniques [7] –Arguing: Experimental capacity measurement of a ring of NGVs is now excellent and getting better. CFD still has trouble with small geometric changes because they are so tiny compared with even small deviations in the mesh. The TE is especially difficult to model with CFD because

it involves so many different flows meeting and mixing, with unpredictable geometry –Raw data: Experimental data showing continued improvements to the accuracy of the unsteady capacity measurement technique –I discuss: The TE is indeed particularly hard to predict. Having established an extremely accurate measurement technique for an annulus of NGVs, how could such ideas be adapted to focussing on small geometric changes at the TE? –I also cite: The history of this technique at Oxford, showing it has embodied a continual evolution in measurement accuracy and sensitivity to boundary conditions, but has been relatively underused for the study of geometric effects

–Jie Gao - Experimental and numerical investigations of trailing edge injection in a transonic turbine cascade [6] –Arguing: TE coolant ejection is good for reducing the wake size and reducing shock effects on the exit angle –Raw data: Their experimental and CFD linear cascade, five-hole probe data and surface pressure plots, downstream loss profile –I discuss: Does it match my blowing rate study? Do my turning angles agree with theirs as Mach number changes? Do I see the same reduction of shocks? Do I broadly agree about the benefits of TE ejection? Remember badly-predicted turning angles are like bad capacity predictions –I also cite: Whoever’s loss definition they use - why didn’t they compare it to others? –My raw data: A lot of my own TE CFD

Gao et al experimentally and numerically investigated the external flow field near the trailing edge of a linear cascade with trailing edge injection. The authors’ focus was on the effects of the trailing edge coolant flow on the vane’s loss mechanisms and flow exit angle. Without trailing edge injection, increased exit isentropic Mach number caused the appearance of trailing edge suction side shock waves which altered the flow angle. With trailing edge injection, there was a slight reduction in this shock effect but a strong reduction in the shock wave near the trailing edge pressure side.

3.3 Performance of an alternative trailing edge design

Improved materials and manufacturing techniques may necessitate alternative trailing edge geometries. Paul W. Giel et al [8] noted that “in the pursuit of higher turbine inlet temperatures for reduced fuel burn and emissions consistent with NASA’s goals [9], Ceramic Matrix Composite (CMC) materials are now being implemented in gas turbine engines...They enable higher turbine inlet temperatures, thus enabling higher overall pressure ratios (OPRs) for the engine and higher thermal efficiency.”

The authors used a linear cascade to measure the aerodynamic performance of a set of blades representing the geometric constraints of the CMC manufacturing

method. Of main concern was the constraint that “the trailing edge thicknesses of CMC blades are anticipated to be significantly larger than those of current state-of-the-art metallic blades,” which may be expected to cause increased loss.

–discuss other justifications for studying this design - is it more predictable because it suffers less erosion?

–Introduce my alternative design: 2D CFD was done on an alternative design of TE featuring centred coolant ejection. This shape of TE was also incrementally cut back on the pressure side so as to more resemble the existing TE design.

–Present boundary conditions

Table 3.2: Boundary conditions for trailing edge centred-ejection design study

Parameter	Value
NGV series	Rolls-Royce Trent 900
NGV turning (degrees)	76.89
Design pressure ratio	1.79
Inlet total pressure (Pa)	4.33×10^6
Inlet total temperature (K)	300
Outlet static pressure (design) (Pa)	2.42×10^6
Outlet static temperature (Pa)	300
Solver type	Density-based
Cell count	35,000

–Present mesh with larger and better pictures, and show how the mesh is altered as the design is altered

–Plot capacity trends for the centred-ejection designs vs the baseline

–Is there better capacity predictability because the effects of erosion are less pronounced due to the thicker flanges?

–Discuss whether this loss may be worse or better if a centred-ejection design is used, still perhaps plotting more than one definition of loss

–When using this configuration, is there an optimal blowing rate that might re-energise the base region? Plot blowing rate vs loss, remembering that the graph I currently have is erroneously for the other type of TE, not for the centred-ejection kind. Should I try to re-run just this one thing?

–Compare my centred-ejection TE with a "naturally-formed by erosion" centred-

ejection TE from the literature

To be filed:

–Ranjan Saha - SHOWER HEAD AND TRAILING EDGE COOLING INFLUENCE ON TRANSONIC VANE AERO PERFORMANCE [10] –Arguing: Loss turns out very differently depending on which loss equation you use, and some formulations might be better than others –Raw data: Their annular sector rig which tested NGVs with and without showerhead and TE coolant to quantify loss –I discuss: How do their definitions of loss compare to mine and others', and is there an answer to which one is best in general? –I also cite: All the other possible loss definitions, going back to Raffel and Kost and further if necessary –My raw data: Any of my CFD data can yield loss according to any preferred definition, but this is most pertinent to my T900 TE cutbacks discussion

Chapter 4

Leading edge

–It is sometimes necessary to add extra cooling holes on the suction-side at the LE

–Coolant introduced into this part of the flow has complex interactions with the mainstream, affecting NGV capacity

4.1 sensitivity of capacity to cooling holes on the leading suction side

–Adding cooling holes causes complex effects - here discussed as loss, so I need citations for capacity too: –Jie Gao - Experimental and numerical investigations of hole injection on the suction side throat of transonic turbine vanes in a cascade with trailing edge injection [11] –Arguing: Like their above paper but with a SS cooling slot. It causes passage blockage, a thicker wake, and more loss. But when there are shock waves, the film coolant enhances the TE PS shock but reduces the TE SS shock, which could be used to greatly reduce the TE SS shock if the injection is upstream of the throat –Raw data: Their experimental and CFD linear cascade, with strong emphasis placed on the loss definitions and techniques established by Denton and Xu, Mee et al, and Schobieri –I discuss: Ways in which my SS injection might interact with the flow (including shockwaves) as they suggest. Will I see it without having TE injection too? –I also cite: Denton and Xu, Mee et al, and Schobieri for established precedent on loss quantification –My raw data: My moving coolant row study

–2D CFD was done on an XWB NGV with no coolant features apart from the addition of a single cooling row on the upstream suction side. The position of

this feature was varied, thus varying whereabouts in the flow the coolant was introduced.

–Present boundary conditions

Table 4.1: Boundary conditions for suction-side cooling hole position study

Parameter	Value
NGV series	Rolls-Royce Trent XWB 84K
NGV turning (degrees)	73.97
NGV throat width (m)	1.25×10^{-2}
Design pressure ratio	1.65
Inlet total pressure (Pa)	4.33×10^6
Inlet total temperature (K)	300
Outlet static pressure (Pa)	2.63×10^6
Outlet static temperature (Pa)	300
Coolant/mainstream mass flow ratio	0.003
Solver type	Density-based
Cell count	40,000

–Present mesh, showing how the variable location coolant row is introduced

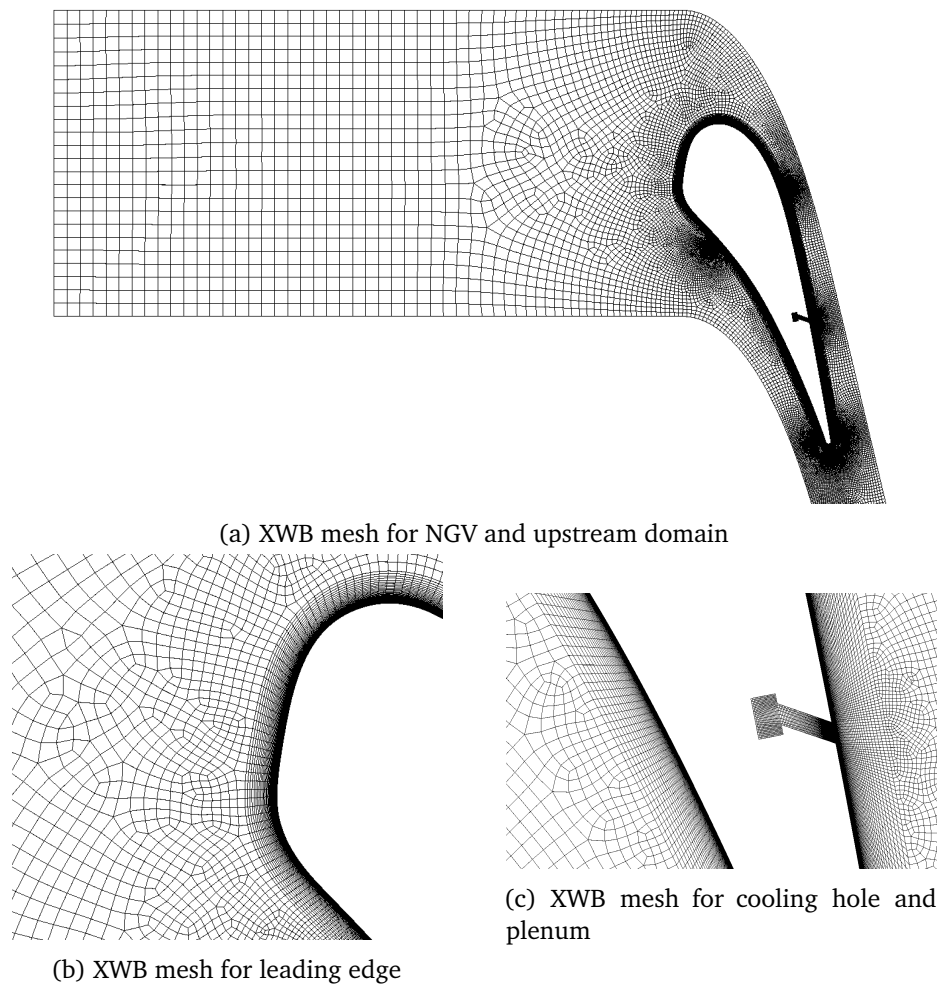


Figure 4.1: Details of mesh for XWB single cooling hole study

–Review the literature for why correctly modelling turbulence is important for capturing the mainstream/coolant mixing, and thus why two turbulence models are compared.

–Plot capacity changes versus hole position and discuss its causes, mentioning the literature reviewed in the chapter intro (plus the two definitions of capacity where coolant is concerned - which one is used in earlier chapters??) A maximum 0.5 percent change in capacity resulted from an unrealistically drastic shifting of a 2D cooling slot along the vane’s suction side.

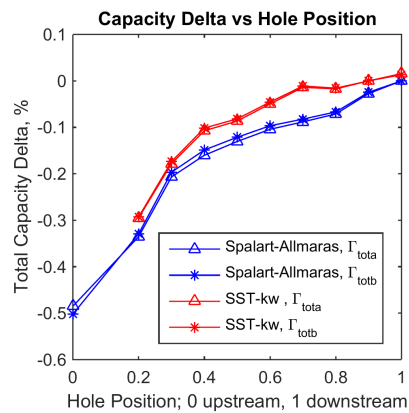


Figure 4.2: Total capacity percentage delta as a function of single cooling hole position

4.2 Consideration of corrections necessary to interpret 2D result

–Different:

–Mass flow rate between a 2D slot and a 3D row of holes

–Mixing between coolant and mainstream

–Angle of ejection

–Papers about matching the conditions when simulating a cooling row:

–S. Ravelli - NUMERICAL ASSESSMENT OF DENSITY RATIO AND MAINSTREAM TURBULENCE EFFECTS ON LEADING EDGE FILM COOLING: HEAT AND MASS TRANSFER METHODS [12] –Arguing: In a CFD simulation of four film cooling hole rows, the usual parameters (DR, BR, MFR, TuI) aren't enough to ensure you have matched the engine conditions –Raw data: A previous experimental study (PSP) to validate their CFD, and their CFD –I discuss: Criticise my own single cooling slot work - is it really representative enough for its capacity effects to be predictable –I also cite: Other CFD on film cooling –My raw data: My single cooling row data, emphasis on the CFD parameters and setup

–Giovanna Barigozzi - Experimental investigation of the interaction between showerhead coolant jets and main flow [13] –Arguing: When film coolant is ejected upstream, observed high turbulence and velocity fluctuations suggest

the mixing is a random process without coherent structures, and unsteady, and 3D –Raw data: Their experimental study, including pressure-sensitive paint and particle-image velocimetry –I discuss: Do we know less than we thought about how much film coolant is actually getting to the LE SS? Reason why you might want an extra row there? Also, clear support for my arguing that proper CFD investigations of SS LE film cooling should not be a 2D slot and should not be steady

–Wei He - Film cooling and aerodynamic performances of a turbine nozzle guide vane with trenched cooling holes [14] –Arguing: A novel design for film cooling holes, using a zigzag-shaped trench, has some advantages but is only really suitable for the middle PS –Raw data: Their 3D CFD on a single-vane linear cascade, where the trench moves around various positions –I discuss: They mention BR etc, relevant to me. Their movable trench is a bit like my movable 2D slot, except it is mainly confined to the PS, whereas mine is on the SS. But my quasi-3D study could be easily adapted to repeat their study of the novel trench shape –My raw data: My Q3D study

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