POZNAŃ UNIVERSITY OF TECHNOLOGY

DOCTORAL THESIS

Method for direct noise analysis of transonic axial compressor blade

Author:
MSc. Eng. Jędrzej MOSIĘŻNY

Supervisor:
Prof. DSc. Eng. Michał CIAŁKOWSKI

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Declaration of Authorship

I, MSc. Eng. Jędrzej Mosiężny, declare that this thesis titled, 'Method for direct noise analysis of transonic axial compressor blade' and the work presented in it are my own. I confirm that:

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- Where I have consulted the published work of others, this is always clearly attributed.
- Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this thesis is entirely my own work.
- I have acknowledged all main sources of help.
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Abstract

This thesis proposes a method of assessing flow generated noise in transonic flows by direct formulation.

First a steady state Reynolds Averaged Navier-Stokes analysis of NASA R67 transonic axial compressor is performed as a validation study of the mesh and numerical setup. The result of the steady state analysis is then used as an initialization for transient DDES analysis performed on high quality, 11 million cells hexagonal mesh. The transient analysis covers 0.05s of physical flow time, which corresponds to about 800 revolutions of the rotor. Both steady state and transient simulations are performed on PL-Grid HPC infrastructure.

Transient results are analyzed with an in-house build program. The program uses information about static pressure, transient particle velocity and vorticity from each timestep. This data is then postprocessed into sound pressure levels, sound frequency and effective sound power level.

Information on generation of sound phenomena occurring in the blade passage are gathered from direct formulation and may be used as a validation case for FW-H or other computational aeroacoustic analogies dealing with flows in transonic regimes in rotating machinery.

Acknowledgements

In this place I would like to thank the Chair of Thermal Engineering of Poznań University of Technology, with special recognition to MSc. Eng. Bartosz Ziegler and PhD Eng. Przemysław Grzymisławski for thorough scientific and personal support during this project.

A big recognition goes to the owners and maintainers of the PLGRID - Polish HPC infrastructure, especially team in HPC Cyfronet center in AGH University of Science and Technology in Kraków. Being able to use the state of the art HPC clusters for analyses made this project possible.

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Abbreviations

CAA	**C**omputional **A**ero **A**coustics
CFD	**C**omputional **F**luid **D**ynamics
DDES	**D**elayed **D**etached **E**ddy **S**imulation
DES	**D**etached **E**ddy **S**imulation
HPC	**H**ight **P**ower **C**omputing
LES	**L**arge **E**ddy **S**imulation
N-S	**N**avier **S**tokes
SRS	**S**cale **R**resolving **S**imulation

Physical Constants

Speed of Light $c = 2.997\ 924\ 58 \times 10^{8}\ \mathrm{ms^{-S}}$ (exact)

Symbols

a	distance	m
P	power	W (Js^{-1})
ω	angular frequency	rads^{-1}

To my wife. For limitless patience...

Chapter 1

Introduction

1.1 Main Section 1

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Chapter 2

Current research on Computational Aeroacoustics

2.1 Main Section 1

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Chapter 3

Approach and direct formulation of noise analysis

3.1 Basic conservation equations in CFD

3.1.1 Momentum equations

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3.1.2 Continuity Equations

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3.1.3 Energy equation

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3.2 Resolving turbulence

3.2.1 RANS formulation of turbulent flow

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3.2.2 DDES Formulation of turbulence

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3.2.3 DDES Formulation of turbulence

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3.3 Direct formulation of noise signal

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3.3.1 Mesh size requirements

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Chapter 4

Test case

4.1 NASA Rotor 67 transonic axial compressor

The test specimen for given analysis is a NASA Rotor 67 (R67) transonic axial compressor. Originating as a first stage of two stage fan for evaluation of design procedures, validation of experimental facilities as well as meshing and CFD tools. Both stages were used in a multitude of studies for aerodynamics, geometry optimisation, noise analyses and structural analyses. Full design procedure can be found in references [2] and [3]. The CFD analysis and further post processing of the pressure signals shall be performed on a single passage of a first stage rotor of the compressor. The setup for the calculations (apart from the single passage constraint) is relevant do case described in [4], which was the main source for geometry and flowfield data.

Basic figures of the given rotor are, design pressure ratio of 1.63 at massflow of 33.25 kg/sec. The design rotational speed is 16 043 rpm, which yields a tip speed of 429 m/s and an inlet tip relative Mach number of 1.38. The rotor has 22 blades and an aspect ratio of 1.56 (based on average span/root axial chord). The inlet and exit tip diameters are 514 and 485 mm, respectively, and the inlet and exit hub/tip radius ratios are 0.375 and 0.478, respectively. A fillet radius of 1.78 mm is used at the airfoil-hub juncture. The square root of the mean square of the airfoil surface finish is 0.8 μm or better, the airfoil surface tolerance is $\pm$ 0.04 mm, and the running tip clearance is approximately 1.0 mm [4]. Surface roughness and some of the geometrical features are omitted during the preparation of the geometry and CFD mesh for reasons described in sections 4.2 and 4.3.

4.1.1 Efficiency figures

4.1.2 LDA Validation results

4.2 3D geometry preparation

Geometry was prepared in Ansys ICEM 14.5 meshing software. Creating the geometry directly in the meshing software reduces the risk of creating flaws in the geometry, due to file translations. Mesh is created in millimeters.

The R67 blade coordinates for rotor and stator on both stages is given in references [2] and [3] and provide the blade elements in a Multiple-Circular-Arc fashion. In such approach the design blade elements lie on conical surfaces which approximate the actual stream flow surfaces. A blade-element-layout method is developed which preserves the constant-angle change characteristic of the circular-arc profile. More specifically, the mean camber line and the suction and pressure surface lines of a blade element are lines with a constant rate of angle change with path distance on a specified conical surface [5]. Although relatively comfortable for design purposes, such approach requires implementing a macro or script to desired CAD tool for creating the blade elements or transforming the MCA blade to Cartesian or cylindrical coordinates. Reference [5] provides an extended definition of MCA blade description as well as Fortran code for generating blade cross-section and geometric properties of the blade. Source [4] provides a list of coordinates for 14 profiles of the 1st stage rotor blade suction and pressure side, as well as coordinates for hub and casing int the meridional plane. These coordinates were used to create the geometry of the single passage of the subject blade. Coordinates are also available in Appendix C and project Github repository [6].

Geometry alignment is presented on figure 4.1. The coordinate system is a standard right-hand Cartesian CS. Rotation axis is set to Z-axis with flow in positive Z direction. The compressor rotation is set as in right-hand rule, the compressor rotates in clockwise direction when facing the blade leading edge. $Z = 0$ coordinate is defined by point number 1 on 1st blade design surface (see Appendix C for details).

Hub and casing flow path were created by importing formatted point data as a b-spline curve, followed by extrusion the curve to surface by rotating it by $\pm 60°$. Blade surfaces cylindrical coordinates were transformed to Cartesian coordinates using simple trigonometric calculations and imported as set of splines. Suction and pressure surface of the blade were created by lofting the surface along the imported splines. Leading and trailing edge radii were created in a similar manner with use of edge radius and edge tangency points given in [4]. Tip gap of the blade was created by offsetting the casing

Figure 4.1: Global coordinate system for geometry

surface by 1.016 mm in the normal direction towards the rotation axis and creating a section line between blade surfaces and the offset surface.

Due to the estimated mesh cell count, only one blade passage is created, therefore a set of periodic surfaces must be defined. ICEM software is capable of creating a midline as an average of coordinates of two given lines. A midline was created for every design profile and was manually extended beyond the blade leading and trailing edge. Midlines were lofted to create a midsurface which was later on copied with rotation by $\pm 0.5 \cdot \frac{360^\circ}{22}$ to create two identical periodic surfaces.

Aforementioned midlines were also rotated along Z-axis to create control surfaces for mesh stabilization and data acquisition down the process.

Reference [4] provides coordinates of hub and casing for the full experiment, however only a rotating part of the experimental rotor setup will be used. Two surfaces normal to Z direction at coordinates Z = -13.74 mm and Z = 93.65 mm are placed as inlet and outlet boundary conditions. Geometry was finished by necessary extrusions, trimming and other finishing operations to ensure high quality surface for meshing. Usually, the geometry must be watertight to ensure proper meshing process, however ICEM as patch-independent meshing software does not require that.

Physical experiment test compressor has a 1.78 mm fillet at airfoil-hub juncture. This feature was omitted as it would unnecessarily complicate the meshing process and increase the cell count.

Such approach allowed for creating a geometry for single blade passage with centered blade of 1st stage rotor of the test compressor (Fig 4.2).

FIGURE 4.2: Final single passage geometry

4.3 Meshing approach

Following requisites are posed to the mesh for the discussed case:

- Possibly low number of elements fulfilling the mesh sizing requirements stated in chapter 3.3,
- Mesh should be a fully structural mesh including the tip gap,
- The periodic boundary mesh must be identical/conforming for both boundaries,
- The mesh must have high quality metrics in terms of cell orthogonality and skew as defined by equations 4.1 & 4.2 respectively.
- The cell aspect ratio should be limited to 5000 as defined by equation 4.3

$$\dot{B} = \dot{m}(c_p\,(T - T_0) - T_0\,(c_p\,\frac{T}{T_0} - R\,ln\frac{p}{p_0})) \tag{4.1}$$

$$6^2 - 5 = 36 - 5 = 31 \tag{4.2}$$

$$6^2 - 5 = 36 - 5 = 31 \tag{4.3}$$

One of the initial mesh concepts was an unstructured mesh with triangular surface mesh extruded to prism boundary layer and mostly isotropic tetrahedra in the volume. This approach was quickly rejected for bad quality elements near the airfoil/hub junction

FIGURE 4.3: Mesh topology with conforming periodic boundaries

and tip gap, as well as element count in range of 4.5 million cells for sizing relevant for RANS analysis. This approach was quickly dropped.

A non-trivial topology with fully conforming periodic boundaries was introduced (fig 4.3). This topology fulfills all the prerequisites stated apart from possibility to mesh a structural tip gap. Such approach makes it impossible from topological standpoint to place a structural mesh in this area. A RANS sufficient mesh without tip gap area (blade was extended to the casing surface) was created. The cell count for this mesh is below 0.5 million cells with better skewness and orthogonal quality. This mesh was utilized for numerical setup and data acquisition testing as it was faster to converge.

Final topology was a standard h-grid topology for airfoil 4.4. Although it is impossible to create a conforming periodic interface with such mesh topology, a fully structural tip gap was implemented. Omitting the blade-hub juncture fillet simplified the mesh. Such topology eradicates the necessity of placing 5-way topology points.

Mesh was created in Ansys ICEM software using structural blocking method. The topology was sliced and associated to internal surfaces mentioned in the above section. This enforces mesh layering along design streamlines and provides high quality mesh on internal surfaces for flow field data acquisition further in the process. Blade wall boundary condition is distributed among five separate parts: blade pressure side, blade suction side, leading and trailing edges and tip surface.

Element sizing in volume and in tangent direction to the blade is limited to 3 mm, with 5 mm at inlet and outlet boundary conditions. The mesh sizing requirements are described in chapter 3. Blade boundary layer is produced by creating an o-grid around blade geometry. Hub and casing boundary layers are created by changing the sizing on the blocks adjacent to the geometry. Sizing of the first element is estimated with y+

FIGURE 4.4: Mesh h-topology

FIGURE 4.5: Mesh non-orthogonality histogram

parameter as described in equation: 4.4. First element thickness in on the blade surfaces ranges from is $2/mum$ on tip airfoil and $10/mum$ on hub airfoil. This corresponds to $y+ \approx 2$ calculated by streamline velocity values given in [4].

$$6^2 - 5 = 36 - 5 = 31 \tag{4.4}$$

Figures 4.5, 4.6 and 4.7 provide overview of mesh quality defined by figures 4.1, 4.2 and 4.3 respectively. Created mesh is of high quality and is sufficient for both RANS (chapter 5) and DDES (chapter 6) analyses. Final mesh reached roughly 11 million cell count.

FIGURE 4.6: Mesh skew histogram

FIGURE 4.7: Mesh aspect ratio histogram

4.4 Case preprocessing

CFD analyses for generating raw pressure field data are performed in ANSYS Fluent 17.2 software on Prometheus HPC cluster located in Kraków. Access for this infrastructure was granted by PLGrid infrastructure. Calculations were run on 5 nodes of 24CPU cores and 128GB RAM each [7], which resulted in decomposition to 120 cores, resulting in allocating around 110 thousand cells to a single HPC core.

Following is the numerical setup for all of the performed analyses. The setup corresponds to "peak efficiency conditions" of the experimental compressor.

4.4.1 General settings and material properties

Analysis was resolved using implicit density based solver. The theory behind the solver is described in chapter 3. Material used in the anaysis resembles standard air modeled as ideal gas as in equation with following properties described in table 4.1

TABLE 4.1: Standard air properties

C_p	1006.43	$\frac{J}{kg \cdot K}$
λ	0.0242	$\frac{W}{m \cdot K}$
μ	1.7894e-05	$\frac{kg}{m \cdot s}$
M	28.966	$\frac{kg}{kmol}$

$$pV = nRT \tag{4.5}$$

Analysis operating pressure is set to 0 Pascal, which is a standard practice in compressible flow CFD.

Fluid zone is set as "frozen rotor" - rotating reference frame. Although no mesh motion is implied, the effect of Coriolis accelerations and centrifugal acceleration will be taken into account by adding respective acceleration components to momentum equations as described in chapter 3. The rotational velocity is set to 1680 rad/s.

4.4.2 Boundary conditions

Following boundary conditions were applied to the mesh boundaries. Following setup is again typical for compressible flow CFD.

TABLE 4.2: Test case boundary conditions

boundary	**condition type**	**value**	
Inlet	Pressure Inlet	101350	Pa
Outlet	Pressure Outlet	10200	Pa
Hub	Movig wall	1680	rad/s
Blade	Moving wall	1680	rad/s
Casing	Stationary wall	-	-
Internal profiles	Internal	-	
Periodic boundaries	Interface	-	

Moving wall boundaries represent the rotational velocity of the compressor blade and are necessary for stationary reference frame formulation. Boundary condition setup has no differences to usual setups of analyses of such kind.

4.4.3 Numerical scheme

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4.5 Data acquisition

Chapter 5

RANS Analysis

5.1 Main Section 1

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5.1.1 Subsection 1

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5.1.2 Subsection 2

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5.2 Main Section 2

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Chapter 6

DDES Analysis

6.1 Main Section 1

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Chapter 7

Results of flow field noise analysis

7.1 Main Section 1

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Chapter 8

Conclusions & Further work

8.1 Main Section 1

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Appendix A

Code for direct formulation of noise analysis

Write your Appendix content here.

Appendix B

Code for discrete Fourier analysis

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Appendix C

Blade design surface coordinates

Write your Appendix content here. [8] [9] [10] [1] [5] [2] [3] [4]

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