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# An investigation on the effect of pitchwise endwall design in a turbine cascade at different incidence angles

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ABSTRACT

This paper describes the effects of non-axisymmetric endwall profiling on the aerodynamic performance of a linear turbine cascade at different incidence angles. The sinusoidal profiling is carried out with constant profile curvature along the mean streamline path. Three different profiles, with varying hump to dip height, are analyzed numerically and the performances are compared with the planar profile. Reynolds Averaged Navier Stokes (RANS) equations are solved in their conservative form using Finite Volume Method with SST turbulence model. The calculated results indicate that the profiled endwall minimizes the lateral movement of weaker boundary layer fluid from the hub-pressure side corner. In comparison with planar case, the flow deviations are largely contained with endwall profiling but closer to the endwall it enhances the overturning and secondary flow kinetic energy. The reduction in loss coefficient is estimated to be 1.3%, 8.7% and 38% for incidence angles of −10°, nominal and +15° respectively. The sinusoidal profiling has brought down the pitch averaged flow deviation and secondary flow kinetic energy at nominal and positive incidence angles but the impact is insignificant at negative incidence. Profiling minimizes the rolling up of the passage vortex and makes the passage vortex to migrate closer to the endwall. This flow modification brings down the losses in the core flow but enhances the losses near the endwall.

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## 1. Introduction

In any gas turbine about 33% of the total losses are attributed to secondary flow losses [1]. Secondary flows refer to three-dimensional, vortical flow structures that develop in turbine vane and blade passages due to high turning of the flow and non-uniform inlet total pressure profiles. Sieverding [2] gives the comprehensive description of experimental secondary flow features and effect of boundary layer in a straight turbine cascade. Numerous experimental and theoretical works have been reported and the behavior of secondary flows and its influence on the turbine performance are almost well understood. Out of various methods (filleting the leading edge, fencing the blade passage, endwall profiling, blade lean and blade thickening) suggested for reducing the secondary flow losses, endwall profiling emerged as a viable and effective mechanism.

In general, the endwall profiling is achieved either by axial profiling along the passage with no pitchwise variation or non-axisymmetric profiling along the passage with profile variations in both the axial and pitchwise directions. The profiling is aimed either to accelerate the boundary layer fluid at the endwall or to reduce the pitchwise pressure gradient at the endwall. Harvey et al. [3] carried out non-axisymmetric endwall profiling to reduce the passage vortex strength which causes an increase in exit angle deviation. Normally underturning occurs away from the wall while overturning appears near the endwall. By controlling the lateral pressure gradient, overturning near the endwall can be minimized and this is made possible by providing streamline curvature. The profiled endwall gives less rolling up of boundary layer and convection of high energy fluid when compared with planar endwall [4]. Also it gives significantly less angle variation at exit and reduces secondary kinetic energy. The profiling reduces the cross passage pressure gradient in the blade row by the action of streamline curvature. It was found that the inlet boundary layer locally separates on the suction surface of passage, generating extra losses which feeds directly into the core of passage vortex [5]. Ingram et al. [5] found 31% reduction in the secondary loss for profiled endwall in comparison with planar case. Though they observed performance enhancement via experimental methods the computational results failed to predict the loss prediction qualitatively and quantitatively.

For designing of the endwall several procedures have been put into practice. Rose [6] used a streamline curvature in the radial

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## Nomenclature

| | | |
|---|---|---|
| $A$ | Profile amplitude | m |
| $C_{ax}$ | Axial chord | m |
| $C_{P0}$ | Total pressure loss coefficient | |
| $C_P$ | Static pressure loss coefficient | |
| $H_p$ | Pressure side leg horseshoe vortex | |
| $i$ | Incidence angle | Deg |
| $P$ | Static pressure | N/m$^2$ |
| $P_0$ | Total pressure | N/m$^2$ |
| $Re$ | Reynolds number | |
| $S$ | Blade span | m |
| $S_p$ | Suction side leg horseshoe vortex | |
| $T$ | Transverse blade pitch | |
| $V$ | Velocity | m/s |
| $V_u$ | Axial velocity | m/s |
| $V_v$ | Tangential velocity | m/s |
| $V_w$ | Radial velocity | m/s |

*Symbols*

| | | |
|---|---|---|
| $\alpha$ | Exit flow angle | degrees |
| $\rho$ | Density | kg/m$^3$ |
| $\xi$ | Zeta | |

*Subscripts and superscripts*

| | |
|---|---|
| 1 | Inlet of cascade |
| 2 | Outlet of cascade |
| $-$ | Pitchwise mass averaged quantity |
| $=$ | Mass averaged quantity |

direction to counteract accelerations in the circumferential direction and observed a 70% reduction in the non-uniformities. Harvey et al. [3] designed the endwall using b-spline curves that run through circumferential and axial directions and optimized the performance in terms of secondary kinetic energy and overturning near the endwall. Brennan et al. [7] redesigned the HP turbine of Rolls-Royce Trent 500 engine using a forward and inverse 3D design method. The profile shape was determined by six control points fixed at different axial locations. A sinusoidal shape may be generated at these locations and a surface is fitted over six b-splines. The predicted reduction in secondary loss was by 0.24% of stage efficiency for nozzle and by 0.16% for the rotor. More recent developments shows the three dimensional endwall designs via automatic optimization process for maximizing stage efficiency [8] and [9]. The combination of endwall contouring and leading edge filleting was tested in a turbine NGV by Turgut and Camci [10,11].

Much of the optimized profile designs were based on analysis made with a certain set of operating parameters. The suitability of the designed endwall profile at different operating conditions have been overlooked in many cases. Ideally, the designed endwall must satisfactorily work for various operating conditions of the turbine. The objective of the present work is to quantify the effect of endwall profiling on secondary flow losses in a linear turbine cascade under various incidence angles. Variations in the exit flow deviation as well as the secondary flow kinetic energy are also brought under the scope of this investigation.

## 2. Methodology

In the present investigation a linear turbine cascade (Durham Cascade) has been chosen. The Durham cascade is a low speed, large scale linear cascade for a high pressure rotor design. The blades are designed to have an aerodynamic similarity same as real machines rather than geometrical similarity [12]. It operates at a Reynolds number several times lower than a real turbine. The blade profile and geometrical details are given in Table 1. The Durham cascade has already been subjected to numerous study on endwall contouring and filleting in the past. Flow separations are more dominant in highly cambered blades, which is the reason behind selecting this cascade.

## 3. Geometrical modeling and meshing

The geometrical modeling and meshing of the computational domain is carried out using ICEM CFD. The inlet of the fluid domain is 1.5 times the axial chord distance ($C_{ax}$) away from the leading edge of the blade. The outlet plane is kept at a distance of two times the axial chord distance away from the trailing edge of the blade. In order to reduce the computational effort and time, only half of the span is modeled by specifying symmetric wall condition at the midspan region. Along the transverse direction, translational periodicity is set at one pitch length (Fig. 1a). The mesh around the blade and hub surface are fully structured and remaining flow domain is unstructured in axial and tangential direction. The maximum grid element size for whole domain is 3.5 mm with scale factor of 1. Fine prism layers are attached around the blade and hub surface to take care of the boundary layer effects (Fig. 1b). The first cell height normal to the surface is 0.35 mm with exponential height ratio of 1.15 up to 12 layers.

**Table 1**
Cascade blade details.

| | |
|---|---|
| Blade inlet angle | 47.6° |
| Blade exit angle | −68.0° |
| Stagger angle | −36.1° |
| Blade chord | 224 mm |
| Axial chord | 181 mm |
| Blade pitch, $B$ | 191 mm |
| Blade span, $S$ | 400 mm |
| Reynolds number<br>(based on axial chord and exit velocity) | $4.3 \times 10^5$ |
| Exit Mach number | 0.11 |

## 4. Solver details

The simulations are carried out with Reynolds Averaged Navier Stokes (RANS) equations using ANSYS-CFX. The RANS equations are solved in their conservative form using the Finite Volume Method. The inlet boundary conditions are as follows; total pressure profile is given at the inlet as shown in Fig. 2 with a turbulent intensity of 5%. The working fluid is air, as an ideal gas, which enters the domain with a static temperature of 292.15 K. At the outlet, fixed mass flow is specified for all the cases. The domain walls are specified with no slip condition and assumed to be perfectly insulated. High-resolution discretization scheme was chosen for all simulations and convergence criteria is set to $10^{-6}$.

## 5. Design of endwall

In the present investigation three different sinusoidal endwall profile has been used and they are compared with planar case. Endwall profiles are derived from the following equation.

$$P_n = A * \sin\left[\frac{\pi(T - tc)}{w}\right] \tag{1}$$

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