The Study of End Losses in a Three Dimensional Rectilinear Low Speed Axial Flow Compressor Cascade

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Abstract— A computational study has been conducted for a low-speed linear axial flow compressor cascade focusing mainly on analysis of the effect of roughness on the aerodynamic performance of the cascade. Measurement of total, profile & secondary loss was carried out for various values of roughnesses on the blades of cascade. The Gambit 2.4.6® was used for creating geometry and computational fluid dynamics (CFD) commercial software FLUENT 6.2.16 was used to solve the governing equations. Initially, both surfaces of the blades of the cascade are kept as smooth and various losses are analyzed. These losses are then compared with the corresponding losses on the blades on which roughnesses of 250, 500 & 750 µm are applied on suction (SSR) and pressure surfaces (PSR) individually as well as on both the surfaces together (BSR). The roughness effects for each of roughnesses are determined. The percentage of total, profile & secondary loss in the total loss for the same cascade & percentage of these losses in the total loss of smooth cascade in each case is compared to find the effect of roughness due to change in roughness of the blades and application of such roughnesses on the single or both sides of the blade. It is found that the percentage of mass-averaged total loss for the smooth cascade was found to be minimum. The total loss for the smooth cascade is found to be 24.48%. This loss increased marginally for each type of cascades with the increase in roughness. However, this increment was small in absolute terms with the increase in roughness for each of the cascades. This loss was maximum when roughness of 750 µm was applied on both the surfaces of blades. The total loss is segregated into secondary flow loss or end loss and profile loss for each roughness value. The secondary loss has contributed significantly to the total loss in each such case. The percentage of secondary loss in the total loss for the same cascade was 5.26, 5.13 & 5.06 respectively for the cascades when roughness of 250µm is applied on suction and pressure surfaces individually as well as on both the surfaces together. The percentage of secondary loss in the total loss of smooth cascade is found to be 5.33, 5.17 & 5.15 respectively for the above cascades with the given roughnesses. Percentage difference of secondary & profile loss for each of cascades comparing with smooth cascade is also found. The percentage difference of profile loss with rough blade as compared to profile loss of smooth cascade was found to be 1.08, 0.64 & 1.72 respectively for same roughnesses. The secondary loss is more pronounced in case of suction surfaces followed by pressure surfaces and then both Surfaces for the cascades for each of roughnesses for which losses have been measured. The secondary loss decreases with the increase of roughness at same location.

The percentage of secondary loss in cascade SSR, PSR and BSR is found to be decreasing with the increase in roughness compared to its value with smooth blades. The rate of decrement in case of SSR cascades is more pronounced followed by pressure surfaces and then both surfaces for the cascades for each of roughnesses of 250, 500 & 750 µm.

Keywords— Cascade, Energy Loss, Secondary Flow Loss, Profile Loss, Roughness.

1. INTRODUCTION

The performance of turbine and compressor in the gas turbine power plant has to be optimized and analyzed; therefore, methods for predicting losses through compressor and turbine components of these plants are greatly needed. One of the main reasons of performance deterioration in gas turbine power plant is the roughening of blades of compressor or turbine blades. The power produced by a Gas turbine is largely influenced by fouling of compressor or turbine blades. Power plant engineers strive hard to prevent the compressor & turbine from effects of roughness. This is important for keeping design as well as operation and maintenance cost of plant in control.

The axial-flow compressor compresses working fluid by a row of rotating blades called the rotor and then diffusing in a row of stationary blades to obtain a pressure increase. The diffusion in the stator converts the velocity increase gained in the rotor to a pressure increase.

Compressor fouling is defined as the deposition of airborne particles on to compressor blades which results from the adherence of dust or sand particles mixed with small oil droplets to compressor blade surfaces. Roughening of blade surface is also caused by erosion or pitting, individually or in tandem due to a variety of physical and chemical phenomena. It results in change in blade profile and roughness over the surface that varies along the blade span as well as from initial stage to final stage. The result is a reduction in compressor pressure ratio and an overall loss in mass flow, compressor efficiency and, therefore, overall power output.

The viscous diffusion in the flow through the cascade results in the decrease in integrated flux of total pressure through the cascade.
Since this decrease in total pressure flux is related to the amount of kinetic and potential energy loss in the cascade, hence this pressure flux is termed as ‘total pressure loss’ or simply ‘loss’. This total pressure loss has significant effect on the efficiency of the cascade and hence it should be minimized. This total pressure loss has various components like annular loss, profile loss and last but not the least secondary losses. These losses of various kinds are summarized as under:

Profile Loss: The profile loss is the loss due to boundary layer on the blade surface and trailing edge thickness. When a fluid flows over a solid surface, the fluid velocity changes from zero at the wall (for stationary wall) to its free stream value. This fluid layer in the neighborhood of the solid boundary where the effect of fluid friction (viscous effects) is predominant is called the boundary layer. The flow outside the boundary layer can be considered frictionless or potential flow. The boundary layer separates near trailing edge as it has some definite thickness. The separation point shifts towards leading edge depending upon the extent of adverse pressure gradient [Samsher, 2002]. Build-up of various impurities and erosion introduces the roughness over the surface of the blades, which in turn greatly affect magnitude of profile loss.

End Wall or Secondary Losses: The Secondary loss contribution to the total loss is measured to enumerate effect of end-wall surfaces on the overall performance. The term secondary flows refers to the three-dimensional vortical flow structures that develop in blade passages due to turning of the flow and non-uniform inlet total pressure profiles. The efficiency is significantly affected due to the Secondary Flow. Secondary flows are superimposed upon the mean flow in transverse plane of the primary flow. Secondary flows cause to generate a non-uniform flow at exit of the blade row thereby efficiency of the blade row downstream gets reduced. Roughness over the surface of the blades also significantly affects the secondary losses.

Other than profile losses, secondary losses also contribute significantly to the total loss [Yahya, S. M., 1983], if the blade is short.

A. Boundary Layer Separation & Losses

The boundary layer is field of flow created around a solid body immersed in the fluid due to the viscous force of flowing fluid. When there is a relative motion between a solid body and a fluid, the fluid flow about the body can be divided into two regions: where frictional effects are significant and the other where these are negligible. The fluid flow in the vicinity of the solid body where frictional effects are significant is called boundary layer. The thickness of the boundary layer depends upon the size of the body and Reynolds number of the flow.

The velocity of the flow within the boundary layer varies from zero at the surface to maximum at the friction free flow region.

The boundary layer consists of three sections that are laminar, transition and turbulent flow regions. Initially the Boundary layer is consisting of laminar flow. Thereafter the laminar flow of boundary layer destabilises and changes to turbulent flow due to changing of limiting conditions of the flow. During the course of flow the laminar flow becomes turbulent and the process is called transition.

The transition from laminar to turbulent flow depends upon many factors. The effect of roughness of the surface of the blade on this transitional effect is also very significant. The fluid flow within the boundary layer is a very complex and so is this transition process. The flow during the transition process gets detached from the surface and a reverse flow starts happening due to Boundary layer separation.

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B. Laminar Sub Layer and Losses

In one of the cases laminar sublayer within the boundary layer is formed. The sublayer is also called the viscous sublayer.
This layer is very thin layer, which is formed next to the surface downstream of the laminar region and its velocity is such that the viscous forces dominate over the inertial forces. In some applications of the blade profile, the viscous drag force plays very vital role. For the very purpose of getting such application to be efficient the drag force has to be minimised. The drag force in the case of compressor cascade is very vital. In the case of compressor, it should be optimised so as to minimise the loss and separation of the flow. Figure 1 shows a 2D view of the formation of boundary layer and laminar sublayer within the boundary layer during the flow over the blades of a rectilinear compressor Cascade.

Main body of the flow ignore the roughness of surface if it is smaller than laminar sublayer. The flow is then assumed to be passing through hydraulically smooth surface. The flow is defined as hydraulically rough in case roughness elements are larger than the laminar sublayer. The hydraulically rough surfaces perturb the flow passing over to them. However, the results of the CFD of turbine cascade showed that excess of the roughness do not increase the loss predominantly [Vinod Kumar Singoria et al., 2012] over the moderately smooth surfaces. However, the loss increases more rapidly with the increase of roughness at the smaller values of roughness. In the compressors, the contribution of adverse pressure gradient to the overall losses is more, as compared to that of surface roughness.

The laminar boundary layer formed initially at upstream of a blade profile gets separated due to dropping of surface pressure as it marches downstream from the leading edge to trailing edge. There could be a zone of an adverse pressure gradient depending on the turning on the surface of the blade. Thus, the boundary layer could grow rapidly or even separate in such a region. The laminar boundary layer formed initially at upstream has more chances of separation as it moves downstream on the Suction surface of blade. The boundary layer for the turbine flows is lesser prone to separate as the fluid flow in the direction of pressure drop. Because of the generally falling pressure in turbine flow passages, much more turning in a given blade row is possible without danger of flow separation than in an axial compressor blade row. In turbine the leading edge is thicker than a compressor.

The compressors differ in this regard as fluid undergoes a pressure rise in the compressor. The chances of separation of boundary layer in case of Compressor is more because adverse pressure gradient retard the fluid in the Boundary layer & ultimately brings the fluid in to rest thereby the outer layer of stagnant fluid separates from this fluid. There are various other factors which largely affect the transition of boundary layer are velocity of fluid flow over the blades of cascade, surface roughness, Reynolds number and curvature of the blade surface etc.,

Even a small roughness may cause the thin viscous inner layer to break up. This leads to increase the wall friction and heat transfer coefficient. Blade roughness effect therefore is considered to be one of the main sources of pressure loss.

C. Formation of Vortex and Secondary Flow Phenomenon

The formation of vortex in a cascade largely affects the secondary loss. The incoming boundary layer separates upstream of the leading edge, forming a horseshoe vortex. This vortex is consisted of two legs. The leg formed at the pressure side and suction side of the blade are called pressure vortex and suction vortex respectively. The legs so formed differ from each other. The suction side leg is affected mainly by curvature of the suction surface. Whereas the pressure side leg is affected by curvature and pressure difference between pressure side and suction side of adjacent blade of the corresponding flow channel. The pressure side vortex leg usually increases towards the exit of the cascade. The formation of a suction and pressure side leg and passage vortex in a wind tunnel experiment is shown in figure 2.

Langston et al. [1977] was among the first to study the evolution of secondary flows using hot wire and flow visualization techniques to qualitatively assess flow patterns at boundary layer, near the end wall region of a cascade. According to Langston, the incoming inlet boundary layer splits into two streams, one moves towards the pressure surface and other towards suction surface. Adverse pressure gradient is resulted in 3-D flow separation and horsehoe vortex formation. Passage vortex is formed due to boundary layer and pressure gradient across the blade passage and rotates in anticklockwise direction. Cross flow is observed at end wall, because of the blade to blade pressure gradient. Suction surface leg rotates in opposite direction of pressure surface leg and consequently termed as counter vortex.

The pressure surface leg of horseshoe vortex merges and strengthens the passage vortex. Later, different experiments were performed by Marchal et al. [1977], Sieverding et al., [1983], Wang et al. [1997] and Sharma et al. [1987] which complied with the conclusions of 1997 and Sharma et al. [1987] which complied with the conclusions of Langston et al. [1977]. Wang et al. [1997] concluded that pressure side vortex moves towards the suction side and merge with passage vortex at approximately one fourth of the distance from the leading edge.

There are two main designs for leading edge geometry: the fillet and the bulb for reducing secondary flow loss. Young J. Moon et al. [2000] analyzed the effect of end wall fencing for reducing the secondary flow using k-ε turbulence model.
They also justified the optimized positioning of the end-wall fencing for reducing the secondary flow losses, because the end wall fencing prevents the merging of pressure side horse shoe vortex with the passage vortex and hence total pressure loss decreases.

Arun K. Saha et al. [2008] analyzed the turbulent flow through a three dimensional non-axisymmetric blade passage and observed that end-wall contouring reduces the pitch wise pressure gradient near the end-wall which reduces the chances of flow separation. Sonoda Toyotaka et al. [2009] use axis-symmetrical end wall contouring method for reducing the secondary losses in high pressure turbine having low aspect ratio. They investigated the effect of three types of end wall contouring: 1) only hub contour, 2) only tip contour and 3) hub and tip contour and observed that hub contouring, the tip contouring and the hub and tip contouring all reduce the mass averaged overall loss by 4%, 5%, and 10%, respectively, as compared to the base line.

Brear et al. in [2010] strived to reduce the pressure surface separation by modifying the leading edge geometry. They observed that increasing the blade thickness at the pressure surface decrease the strength of secondary flow by increasing the momentum near the wall. Shih et al. [2003] observed effects of leading-edge airfoil fillet on the flow in a turbine. The increased size of the stagnation zones on the end-walls about the airfoil’s leading edge lowers the flow speed and velocity gradients there, which in turn reduces turbulence production. G. I. Mahmood et al. [2007] studied the secondary flow structure in a blade passage with and without leading edge fillet and observed that the size and strength of the passage vortex become smaller with the fillets. T. Korakianitis et al. [2010] has proposed a direct design method based on specifying blade surface-curvature distributions so as to minimize the chances of flow separation. Qi Lei et al. [2011] analyzed the effect of leading edge modification on the secondary loss. They used vortex generator for introducing counter rotating vortex which oppose the passage vortex and hence reduce the secondary flow losses.

Much work has been done to understand the occurrence and modeling of secondary flow and end loss phenomenon. Moreover researchers had tried to reduce the secondary loss in any cascade in order to get higher aerodynamic efficiency of the power plant. It is a well-established fact that roughness over the blade surface increases the profile loss and corresponding total loss for a given cascade. But effect of roughness on the secondary flow and corresponding losses has not been studied much. In this paper results of computational study of flow through Three Dimensional Rectilinear low-speed axial flow Compressor Cascade in general and secondary loss and its dependence on roughness of the blades of the Cascade in particular is presented.

The secondary flow is an important and complex flow phenomenon in an axial-flow turbine and compressor. Vinod Kumar Singoria et al., [2012] had conducted computational study on effect of roughness on secondary flow through the axial-flow turbine. However, the research on the secondary flow through axial-flow compressor has been very scarce. This paper is an attempt to present report of such investigation of ‘profile loss’ and ‘end loss’ or secondary loss both. The paper also present an overview of total, profile & secondary loss for a compressor cascade in which air enters with low inlet velocity.

II. MODELLING

The friction losses and the secondary losses are generated as the fluid flows through blades passages in the compressor. The roughness on aerofoil surfaces affects these losses greatly. The present computational study for finding End Losses in a Three Dimensional Rectilinear Compressor Cascade has been carried out using CFD. The brief of CFD software used and boundary conditions is presented here.

A. CFD Simulation

CFD is a computational tool that solve the fundamental nonlinear differential equations (mentioned below) that describe fluid flow (Navier-Stokes and allied equations), for predefined geometries and boundary conditions.

i. Conservation of momentum (Navier-Stokes equation)

$$\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial p}{\partial x_j} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i + F_i$$  

(1)

ii. Conservation of mass (Continuity equation)

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = S_m$$  

(2)

FLUENT 6.2.16 ® converts unsolvable governing equations (Navier-Stokes equations) to a solvable set of algebraic equations for a finite set of points within the space under consideration. FLUENT 6.2.16 ® uses iteration technique for finding detailed information with respect to velocity, pressure, temperature, and chemical species etc.

Using Gambit 2.4.6® as preprocessor, three-dimensional model of a typical Compressor blade cascade geometry has been made [Seung Chul Back et.al., 2010]. Virtual prototype is prepared allocating proper boundary conditions representing the actual flow behavior. Thereafter the computational software FLUENT 6.2.16 ® is used as solver & post processor for flow simulation.
In turbomachinery flow is affected by rotation, three-dimensionality, curvature, separation, free stream turbulence, compressibility, large scale unsteadiness, heat transfer and other effects. Fluid flows of practical relevance are mostly turbulent. Turbulence models approximate these transport processes in terms of mean flow field by empirical formulations. Therefore, in view to obtain a better result, turbulence modeling is chosen due to its being realistic. Turbulence models modify the original unsteady Navier Stokes equations by introduction of mass averaged fluctuating components to produce Reynolds Mass averaged Navier Stokes (RANS) equations.

The most widely used models for turbomachinery application is the $\kappa$ - $\varepsilon$ model [Shih et al., 2003]. In this model, the turbulent kinetic energy ($\kappa$) and the energy dissipation rate ($\varepsilon$) are considered as the properties, which govern the turbulent flow phenomena. The Realizable $\kappa$ - $\varepsilon$ turbulence model of Shih et al. [2003] has been selected for solution of present problem. This model is expected to provide more accurate results since it contains additional terms in the transport equations for $\kappa$ and $\varepsilon$ that are more suitable for stagnation flows and flows with high streamline curvature.

B. Description of Computational Domain

A computational study of flow through three Dimensional Rectilinear Turbine Cascade for smooth profile and profile having roughness of 500 $\mu$m was carried out [Vinod Kumar Singoria et al., 2012] results of which were compared with values of percentage loss coefficients measured along the pitch by Samsher [2002]. Results of computational study and experiment were found to be in agreement as shown in Figure 2. The same inlet and outlet boundary conditions are taken as compared in experimental setup. Subsequently study was conducted with the aim to find percentage loss coefficients along the pitch, applying roughness values of 250, 750 and 1000 $\mu$m separately for each roughness value on suction and pressure surfaces individually as well as on both the surfaces together [Vinod Kumar Singoria et al., 2013]. The atmospheric temperature is assumed to be constant at 27 °C which was the mid value of temperatures range of experiments conducted by Samsher [2002]. The patterns of losses of various kinds as obtained by Vinod Kumar Singoria et al. [2012] were in agreement with results of Samsher [2002] as shown in figure 3.

The roughnesses of 250, 500 & 750 $\mu$m are applied over the blades of Compressor Cascade for the present study. The study is conducted for a compressor cascade for following conditions:

i. both surfaces of the blade of the cascade are smooth
ii. roughness of 250, 500 & 750 $\mu$m is applied separately over:
   a. Entire pressure surfaces of all blades (PSR )
   b. Entire suction surfaces of all blades (SSR )
   c. Both surfaces (suction & pressure sides) of all blades (BSR).

The Cascade of four blades is created. The blades of compressor cascade have chord length of 37 mm and arranged linearly at a pitch of 0.84 times of the chord length. The span of the blades is 37.5 mm. For the present analysis air is taken as the fluid. Air with an inlet velocity of 30 m/s is passed through the cascade. The cascade is open to atmosphere at the exit. Keeping in view the limitation of computational facility the rectilinear cascade consisted of three flow channels using four test blades in the test section, choosing chord, pitch, and inlet fluid flow angle and stagger angle appropriately. A three dimensional model of the profile is designed with the help of Gambit 2.4.6© and the dimensions of the model were kept similar to the experimental setup of Seung Chul Back et.al (2010). In this experiment upstream stagnation temperature and pressure have been measured at one chord length upstream of the leading edge and the midspan height for reference purposes. Static pressure has been measured on the blade pressure and suction surfaces to obtain loss coefficients downstream at 30% chord lengths from the trailing edge of blades at the midspan height [Seung Chul Back et.al, 2010]. In the present paper results of a three-dimensional study has been presented for which results for a span wise loss coefficients were obtained instead of values of mid-span of the setup of Seung Chul Back et.al (2010). Specification of compressor cascade model for the present study is shown in Table 1.

To simulate a compressor rotor four blades are placed in a row to form a cascade. There are various terms involved in a cascade such as blade angle, camber, camber angle, stagger angle, aspect ratio and solidity etc. Blade angles describe the blade and its orientation. Blade angle are measured with respect to the shaft or axis. The aspect ratio is the ratio of the blade length to the chord length. The simulation of given cascade needs that the cascade model for the present study is shown in Table 1.
Secondary loss at the end walls includes loss from the boundary wall on the end wall wetted surface, loss due to flow separation and diffusion of passage secondary vortex. Energy loss coefficients represent total energy loss while fluid flow is taking place along the cascade from inlet to outlet. Energy loss coefficient at mid span where flow is 2-dimensional represents profile loss, whereas near the walls, it represents total loss i.e. profile loss plus secondary flow loss. To segregate secondary loss at the end wall, profile loss at the mid space was subtracted from the total loss. The value of energy loss coefficient is calculated using equation-3.

\[
\xi = \frac{\int_{0}^{1} \xi \rho V_{ax} dy}{\int_{0}^{1} \rho V_{ax} dy}
\]

To calculate a single value of energy loss coefficient, the mass average value of loss coefficient was calculated using the relation as shown in equation 4 [Yahya, S. M., 1983].

\[
\xi = \sqrt{\frac{\int_{0}^{1} \xi \rho V_{ax} dy}{\int_{0}^{1} \rho V_{ax} dy}}
\]

Where \( \xi \) is the mass average loss coefficient, \( V_{ax} \) is the axial velocity, \( \rho \) is the density of air, \( S \) is the pitch distance and \( dy \) is the elemental length in pitch wise direction.

III. RESULT AND DISCUSSION

The pressure and velocity vector were analyzed at appropriate locations in order to understand behavior of the flow through cascade. Various values of pressure (total as well as static pressure) at inlet & exit are taken along the whole blade span. Mass averaged total loss coefficients were computed along the complete blade span starting from end-wall surface at zero mm to other end-wall surface at 37.5 mm height. In order to visualize the flow near the end wall, the measurements near the end-walls were taken at small distances & secondary losses were computed. For first 1.5 mm height from bottom end wall, measuring points are 0.5 mm apart from each other.
Thereafter, it was computed at every 2 mm interval till 15.5 mm blade height. Thereafter the loss coefficient for mid span height at 18.75 mm is computed. The computations for the loss coefficient are similarly repeated for upper half section of the blade till the top end wall at height of 37.5mm. Table 4 shows the various intervals at which total loss coefficients were found.

The total (combined) losses in a blade cascade are estimated by the energy loss coefficient \( \zeta \), which is essentially the sum of profile loss coefficient & end loss coefficient as given by Kostyuk and Frolov (1988) in equation 5.

\[
\zeta \text{ (total)} = \zeta \text{ (pr)} + \zeta \text{ (sec)} \tag{5}
\]

A. Effect of Roughness on Total Energy Loss Coefficients

Mass Averaged Pressure Loss Coefficients have been determined using values of total and static pressures at exit and total pressure at inlet. These coefficients were computed along the complete blade span. A plot of total energy loss coefficients for cascade PSR with a roughness of 250µm is shown in Figure 5. End losses or secondary losses are obtained by subtracting mid-span value of profile loss from individual mass averaged loss value along the blade height.

It is expected that total energy loss increases till both end of the blade compared to its value at mid span. The loss coefficient in case of smooth, Both surfaces Rough (BSR), Pressure Surface Rough (PSR), Suction Surface Rough (SSR) Cascades for roughness 250, 500 & 750 µm has been shown in Table 5.

It is observed from the Table 5 that the loss coefficient is high at hub and casing due to the end-wall boundary layers. It is seen that the localized loss coefficients for all types of application of roughness are also marginally increased with the increase in magnitude of roughness. The profile loss is lowest for smooth blade and increases when roughness is applied over pressure, suction surface separately and together in the same order. The trend of energy loss coefficient for other roughnesses i.e. 250, 500 and 750 µm is same as summarized in Table 5. When the roughness of 250µm was applied over pressure and suction surface separately and together, the profile loss coefficient increases to 23.41, 23.52 and 23.66% respectively compared to 23.26 % in case of smooth. The increment of profile loss due to increase in roughness in the present study is not much because roughness Reynolds Number \( k' \) is such that the roughness regime is fully rough for all type of location of roughness. It can be concluded that roughness of a given magnitude is not much affecting the total loss contrary to the turbine cascade. The difference between data of profile loss is minimal for the smooth surface and rough surface.

Compressor cascade showed total losses amounting to very higher side because separation of boundary layer in case of Compressor is more because of adverse pressure gradient in the direction of flow. The profile loss contributes to very large extent in the total loss as the Boundary layer formed is thicker and more prone to separate during flow through such cascades. The profile loss contribution in the total loss ranges from 95% to 97% as shown in Table 5.

B. Effect of Roughness on Secondary Loss

It is evident from results shown in Table 5 that difference in percentage values of total energy loss & Profile loss gives percentage of secondary loss in total energy. In case of smooth blade secondary loss is 4.99 %. The secondary loss is in a very small proportion in the total loss as the profile loss contributes to very large extent in it. Contribution of this loss ranges merely from 4.99 % to 5.33 % in the total loss as shown in Table 5.

In fact, results showed that the increase in roughness causes secondary loss to reduce for each of PSR, SSR and BSR cascades separately. The dependence of secondary loss over the roughness does not have a direct relationship for the obvious reason as mentioned in preceding section. Moreover this loss is also affected due to complexity of the fluid flow within the boundary layer and transition process of the boundary layer. There are various other factors such as Reynolds number and curvature of the blade surface, velocity of fluid flow over the blades of cascade etc. which largely affect the transition of boundary layer. Presence of roughness over the different part of blade also affects the secondary loss differently. The Pressure side leg of horse shoe vortex is affected by curvature and pressure difference between pressure side and suction side of adjacent blade of the corresponding flow passage thereby contribute more to the Secondary losses. The compressor blades have lesser curvature therefore presence of roughness over pressure surface would not contribute much in increasing of secondary losses contrary to turbine cascades. The curvature over suction side of blades of cascade however affects secondary loss differently. The counter vortex which initiates from suction side of blades mixes with the passage vortex and reduces the effect of passage vortex. Therefore, the secondary losses are lowered in this case. Increase in roughness further enhance counter vortex and hence secondary losses are further reduced.

The results of computational study [Vinod Kumar Singoria et al., 2013] showed that the total energy loss is increased with application of roughness in case of turbine cascade, in spite of absolute change in secondary loss. Therefore, when the absolute change in secondary loss is non-dimensionalised with total loss with the same roughness the same is not truly reflected in the percentage change in secondary loss.
Therefore, they hypothesized that percentage of secondary loss in total energy loss should be non-dimensionalised with the total loss in case of smooth blades. The percentage of secondary loss in total energy loss, thus calculated, is shown in Table 5.

IV. CONCLUSIONS

• The pattern of variation of energy loss coefficient in span-wise direction (y/s) is same for smooth as well as rough blades. Moreover the energy loss coefficient is least for smooth blades and it reaches the maximum value in case the roughness is introduced on pressure as well as suction surface together.

• It is observed that applying roughness on blade surface definitely increases the profile loss as well as total energy loss coefficient. But the mass averaged secondary loss is lower than the secondary loss in a smooth cascade in case of PSR, SSR & BSR compared to smooth blade case.

• Reduction of secondary losses due to increase in roughness on SSR cascade is more pronounced because of more flow separation of boundary layer and thus mixing of passage vortex with suction side counter vortex takes place resulting into least secondary losses.

• Secondary Loss in BSR is more than SSR because the roughness over pressure surface promotes passage vortex, whereas roughness over suction surface strengthen counter vortex present as suction surface leg. Therefore, we can say that secondary losses in the case of pressure surface are more that of suction surface for the same roughness level.

• Due to the end wall boundary layers the loss coefficient is high at hub and casing.

Nomenclature

\( \rho \) Density
\( u_i \) Velocity vector
\( S_m \) Momentum Source Term
\( P \) Static Pressure
\( \rho g_z \) Gravitational Body Force
\( F_i \) External Body Force
\( \tau_y \) Stress Tensor
\( K_{eff} \) Effective Thermal Conductivity
\( J_f \) Diffusion Flux
\( S_S \) Source term includes heat of chemical reaction
\( T \) Temperature
\( E \) Energy term
\( h \) Enthalpy
\( P_{st} \) Static pressure at outlet
\( P_{ps} \) Total pressure at inlet
\( P_{sl} \) Total pressure at outlet
\( \gamma \) Ratio of specific heats for air
\( \zeta_y \) Local energy loss coefficient

REFERENCES

V. FIGURES AND GRAPHS

![Figure 1: Formation of boundary layer and laminar sublayer within the boundary layer](image1)

![Figure 2: Vortex formation due to separation of incoming boundary layer at leading edge of the blade of cascade](image2)

![Figure 3: Validation of computational result with the experimental result obtained by Samsher [2022]](image3)

![Figure 4: Schematic diagram of cascade](image4)

![Figure 5: Total energy loss coefficients with respect to non-dimensional distance from bottom to top end-wall for roughness of 250 \( \mu \text{m} \) over pressure surface](image5)

### Table 1
Specification of compressor cascade model

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<td>1</td>
<td>Aspect ratio (c/h)</td>
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<td>2</td>
<td>Chord length, c</td>
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<td>3</td>
<td>Pitch of the cascade (s)</td>
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<td>4</td>
<td>Solidity (c/s)</td>
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<td>5</td>
<td>Inlet velocity</td>
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<td>Reynolds No.</td>
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<td>7</td>
<td>Viscous model used</td>
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<td>Software used</td>
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<td>Type of analysis</td>
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### Table 2
Roughness characteristics of test blades

<table>
<thead>
<tr>
<th>Sand grain roughness (( \tilde{k}_z )) in ( \mu \text{m} )</th>
<th>Dimensionless Equivalent roughness ( \frac{k_z}{c} )</th>
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<tr>
<td>250</td>
<td>0.0067</td>
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Table 3
Determination of Roughness regime based on roughness Reynolds Number $k^+$ for test blades (Seung Chul Back et al, 2010)

<table>
<thead>
<tr>
<th>S. N</th>
<th>Distance from bottom wall (mm) (a)</th>
<th>Dimensionless Distance (b) = (a)/37.5</th>
<th>Roughness regime</th>
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<td>0.04</td>
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<tr>
<td>5</td>
<td>3.5</td>
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<td>18.75</td>
<td>0.50</td>
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<table>
<thead>
<tr>
<th>S. N</th>
<th>Distance from bottom wall (mm) (a)</th>
<th>Dimensionless Distance (b) = (a)/37.5</th>
<th>Roughness regime</th>
</tr>
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Table 4
Non dimensional distance on blade span

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<th>Dimensionless Distance (b) = (a)/37.5</th>
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</thead>
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<td>Roughness levels</td>
<td>Percentage of Total energy loss in total energy of the air (A)</td>
<td>Percentage of Profile loss in total energy of the air (B)</td>
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