# TIP-LEAKAGE FLOWS IN AXIAL FLOW TURBINES

**(Term Paper No: 2)**

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## ABSTRACT

*The tip clearance, although essential to the turbine rotation, is responsible for one third of the total losses in a turbine stage. Mechanisms of formation of the tip leakage over rotor blades has been described in the paper. A brief overview of tip clearance flows and tip leakage vortex is taken in this paper. Different models defined for the study of tip leakage flows are also briefly defined in this pape. Different active and passive methods for control of tip leakage flows has been defined.*

## NOMENCLATURE

$Q_L$ –Leakage flow
$P_p$- pressure on pressure surface of blade
$P_L$- pressure on suction surface of blade
$\rho$ – density
$u_p$ -primary or inviscid velocity
$\Delta P$ – stagnation pressure loss
$C_L$- Lift coefficient
d- distance from the leading edge
$C_D$ - drag coefficient
L- total lift
C – chord length
S – Pitch
$\beta_p$- primary flow angle
$w_L$ – spanwise velocity

## 1. Introduction

Increasing the gas turbine efficiency could be achieved either by increasing the inlet temperature and pressure or by reducing the losses within the gas turbine. On the one hand, increasing the inlet temperature and pressure depends on the mechanical properties of the blade materials. On the other hand, reducing the losses depends on new efficient designs. One of the most crucial parts of the gas turbine and a source of multiple losses is the clearance between the blade tip and casing. Turbomachinery blades require clearance gaps between the rotating and stationary components and these lead to leakage paths. This clearance is known as "Tip clearance". The leakage flow is driven by the pressure difference between the pressure surface and suction surface. The flow usually separates from the pressure side corner of the blade tip, which leads to a contraction of the flow area available. If the blade thickness is small compared to the tip gap, the leakage flow may not reattach (as is the case in many compressors). However, in thicker blades a separation bubble forms. The leakage flow then emerges as a high velocity jet at the suction surface tip, almost perpendicular to the free stream flow. The shear between the leakage jet and the free stream flow generates a tip leakage vortex, with a rotation axis aligned with the streamwise direction. It accounts for more than one third of the losses inside the turbine stage, (Denton, 1993) & Booth (1985). Losses are generated through viscous shear in the clearance gap as well as shear and mixing of the leakage jet with the free stream. In addition, the leakage flow leads to blockage reducing the total mass flow and work transfer. In compressors, increased leakage significantly reduces the stability margin. Note that some turbomachinery, particularly axial turbines, use a shroud. This is a band that covers and connects the blade tips. This prevents the tip leakage flow described above. However, a leakage path still exists above the shroud from the high pressure end to the low-pressure end (upstream to downstream in a turbine). Shrouded blades are also significantly heavier with greater centrifugal stresses. Lattime and Steinetz, (2002), reported that improved leakage flow sealing in both the high pressure compressor (HPC) and high pressure turbine (HPT) can provide dramatic reductions in specific fuel consumption (SFC), compressor stall margin and engine efficiency as well as increased payload and mission range capabilities.

## 2. THEORY

### a) Necessity of Tip clearances

In order to allow the relative motion between the rotor and casing, a finite clearance is allowed between the blade tip and the casing. Also, it is needed to provide a clearance for centrifugal and thermal expansions. The tip clearance height is typically of fractions of millimetre; however, it has a very influential roll on the machine efficiency and life.

### b) The Leakage flow

The pressure difference between the blade pressure and suction sides drives the flow, at the vicinity of the tip pressure side, to leak from the pressure to the suction sides; therefore, it is called "leakage flow". Thus the leakage flow velocity distribution is a function of the blade loading and thus the blade loading is a major factor influencing the magnitude of the leakage flow. The flow field in turbine stage and especially around the tip is highly complicated. Vortical flow, mixing and separation are some characteristics of the near tip flow. Upon the leakage flow exit from the blade suction side it interacts with the mainstream forming the leakage vortex. The leakage vortex rotates in the opposite direction of the passage vortex. Figure 1 shows a 3D schematic of the leakage and secondary flow fields in an axial flow turbine.
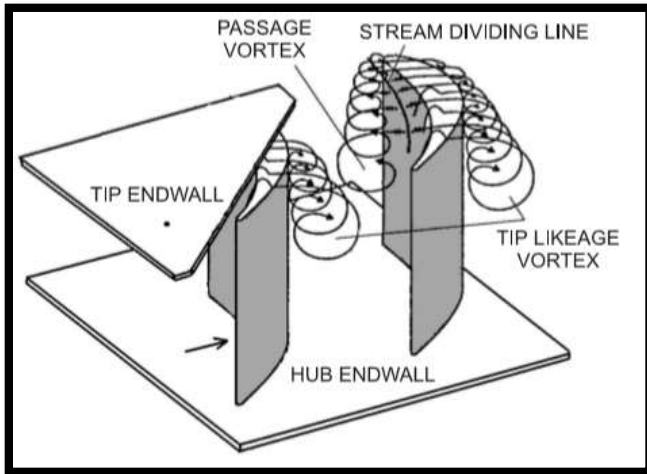


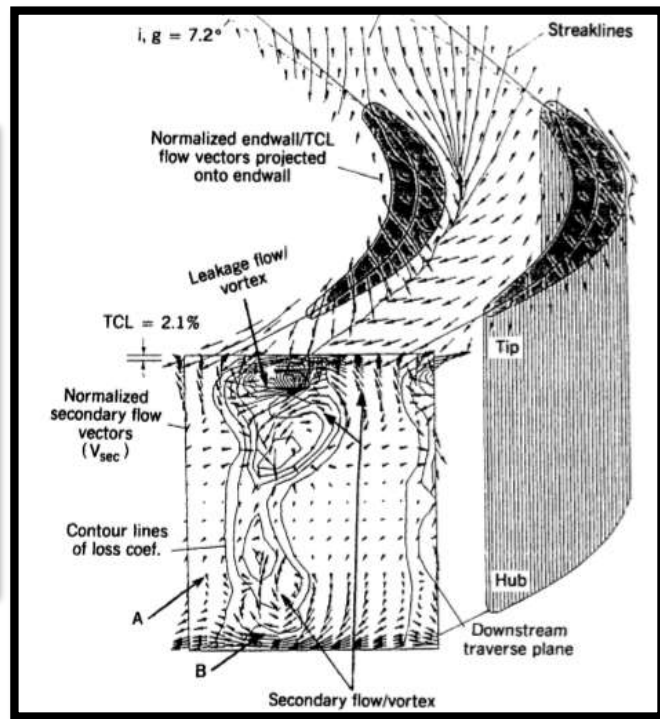Figure 1 Leakage and secondary flow fields of an axial turbine row, Sjolander (1997)



Figure 2 TIp clearance effect in turbine cascade (Yamamoto, 1989) [TCL stands for tip clearence, normalized secondary flow vectors are based on mass averaged flow direction]

With the inviscid and incompressible flow assumption, the leakage flow across the gap is given by:

$$Q_L = \sqrt{\frac{(p_p - p_L)}{\rho}} \tag{1}$$

One of the assumption in this approximation is the total pressure across the gap is same and leakage flow arises due to static pressure difference across the gap. Thus the local velocity depends on local pressure difference, and thus the blade loading is a major parameter influencing the magnitude of leakage flow. The mass flow through this leakage gap, which doesn't participate in energy conversion

process, depends upon the gap height. Furthermore, when the viscous effects are present, both in gap and the endwall region, the height of gap play an important role. Blade clearance height and blade loading (local pressure difference between pressure and the suction surface) are the two most important parameters, which controls the leakage flow.

The total velocity with in the gap in relative frame of reference can be considered as resultant of primary or inviscid velocity ($u_p$) near the blade in the absence of gap and the leakage $Q_L$. The leakage flow tends to roll up in a vortex. Because the leakage jet and the main flow are at different angle, a flow discontinuity exists. This flow discontinuity eventually roll up into the core of the vortex. The viscous effects in the gap may result in flow separation on the blade tip and suction side as shown in figure 2(b). Further complication may arise when there is secondary flow, scarping vortex, coolant injection etc. are there. The secondary flow and leakage flow oppose each other figure 2(c). The leakage flow has a beneficial effect when the corner separation is present. The leakage flow jet tends to wash out the separated region, thus improving the performance figure 2(d). The relative motion has a substantial influence on the magnitude of leakage flow, strength, and location of the leakage vortex. The rotation in compressor tends to augment the leakage flow and move the leakage jet closer to the pressure side figure 2(e). The rotation has opposite effect on turbines figure 2(f). Furthermore, the blade loading is much higher for turbine. Hence, the leakage flow velocity for a turbine tends to much higher than encountered in the turbine.
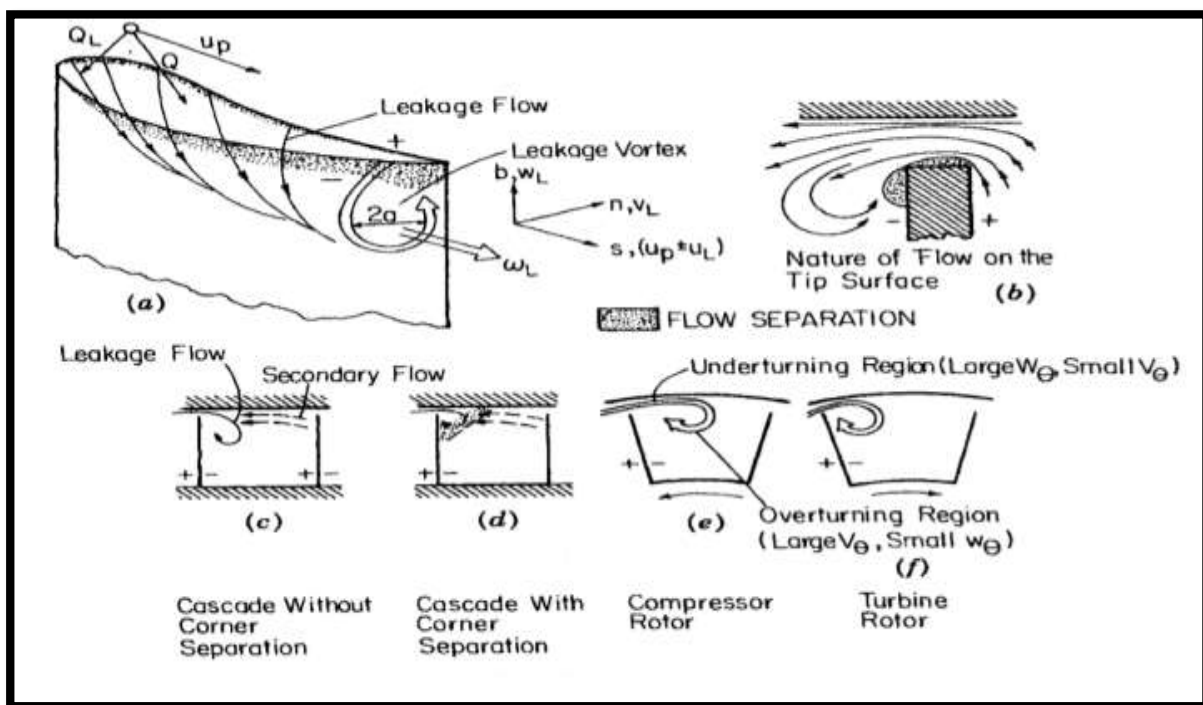


Figure 3 Nature of leakage flows, Laxminarayan Budugur

## c) Effects of leakage flows
The effects of leakage flow and vortex are many; the important effects are,
- Leakage flow is a important cause of three-directionality in the flow.
- The dissipation and mixing of leakage flow and vortex introduces aerodynamic losses and inefficiency.
- The blade unloading at the tip which further affects the surge and stall margin of the compressor.
- The leakage flow and vortex perceived by the further rotor cause unsteadiness in the subsequent blade row. Unsteady pressure, unsteady boundary layer, unsteady transition and noise generation.
- Leakage flow and vortices cause change in heat transfer and turbine cooling flow requirements.
- Leakage flow impinging on subsequent blade row cause vibration, higher blade stresses, and flutter.
- The tip vortices, with lower pressure inside the core cause cavitation, which leads to lower the efficiency.

## d) Drawbacks of leakage flow
The existence of the leakage flow is responsible for certain losses and increases some others. Booth (1985) breakdowns the losses in some turbine stages as shown in Figure 30. He estimated that the leakage flow is responsible for one thirds of the total loss in a turbine

stage. He also shows that increasing the tip clearance by 1% span results in a 1% reduction in the stage efficiency. The drawbacks of the leakage flow in the turbine rotor can be summarized as follows:

- Losses:

The losses related to the leakage flow can be classified as internal and external losses. Internal losses are these losses which take place inside the tip clearance due to viscous effect in the boundary layers and mixing process. External losses are these losses which take place outside the tip clearance due to mixing with the mainstream.

- Upturning:

The leakage flow does not participate or partially participate in the energy conversion process, depend on their entrance location to the tip clearance, Tallman (2000).

- Unloading:

The leakage flow migration over the tip from the pressure side to the suction side would unload not only the tip section but also significant percentage of the blade span, Baskharone (2006). This reduces the power extraction from the turning flow.

- Blockage:

The leakage vortex occupies part of the passage area which reduces the available area for the mainstream thus reducing the main stream mass flow rate and hence the output power, Tallman (2000).

- Three-Dimensionality:

The leakage flow introduces large three-dimensionality to the flow field through mixing process, entrainment process, vortex formation, diffusion, and convection phenomena. It is not only confined to the vicinity of the tip but spreads inward. Its spread depends on the type of turbomachinery, aspect ratio, leakage flow strength, etc., Lakshminarayana (1996).

- Unsteadiness:

The leakage flow and the leakage vortex are sources for boundary layer, pressure and velocity fluctuation. Its effect not only affects the current stage but also extended to the subsequent stage. The resulting fluctuation in the pressure and velocity fields results in flatter and noise problems as well as unsteadiness in the output power, Lakshminarayan (1996) and Tallman (2000).
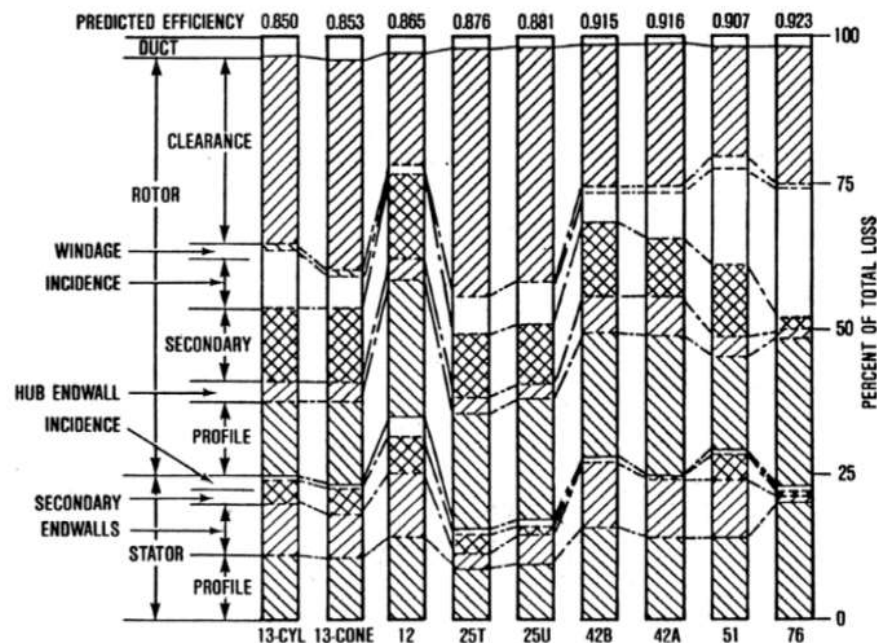
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Figure 4 Losses breakdown in some turbine stages, Booth (1985)

- Complex Heat Transfer Pattern:

The leakage flow and the leakage vortex cause a considerable change to the heat transfer pattern and hence change the cooling requirements. It makes the blade tip to be the most frequent part for repair (Bunker, 2001).

**e) Benefits of Reduced Leakage Flow**

Turbine performance, efficiency, and life are substantially influenced by leakage flow. The benefits of reduced leakage flow are manifested in the following items, Lattime and Steinetz, (2002):

1.  Fuel saving

Engine SFC is directly related to HPT blade leakage flow. Reducing the leakage flow even with a small percentage would permit the engine to meet performance and desired output power with less fuel burn.

2.  Reduced Emissions

Modern aero engines emissions are made up of over 71% $CO_2$ with 28% water ($H_2O$) and 0.3% nitrogen oxide ($NO_2$) along with trace amounts of carbon monoxide (CO), sulfur dioxide ($SO_2$), etc. Air transport alone accounts for 2.5% (600 million tons) of the world's $CO_2$ Production. Obviously a reduction in fuel consumption shall significantly reduce engine emissions.

3.  Extended service life

Reduced leakage flow increases engine efficiency and permits the engine to meet the performance and desired output power with lower rotor inlet temperature. As the turbine runs at lower temperatures, while keeping the same power, hot section components suffer from less thermal loads. This increased component and hence engine service life and increases the intervals between overhauls.

## 3.  Models for tip leakage flow

The tip leakage flow has been analyzed by many investigators. These analyses can be broadly classified in:

### 3.1 Leakage flow analyses:

In this technique the leakage flow velocity is computed from the theoretical or approximated pressure distribution. The kinetic energy associated with leakage flow is calculated to derive an expression for functional dependency of losses and efficiency.

### 3.2 Potential tip vortex model:

In this model, the leakage vortex is represented by potential vortex located at trailing edge of the blade tip, including the tip vortices shed out from all the blades. The wall effect is included via image vortices. Hence the model consists of two infinite row of tip vortices, spaced at $2\tau$ apart in the spanwise gap. The induced velocity due to this system of vortices is calculated to find out this induced drag. Because the vortex is a point vortex. This analysis is useful in capturing the gross effects such as efficiency and losses.

### 3.3 Modified potential tip vortex model:

To allow for local chord wise effects of the leakage vortex on pressure distribution. Scientist modeled the tip vortex by assuming that the vortex leaves at an angle of the tip chord and moves in the spanwise direction. The strength of the vortex is assumed zero at quarter-chord point (where the leakage flow is observed to originate) and attains maximum strength at trailing edge. The location of the center of the vortex near the suction side provides a peak in suction peak as well as quantitative trend in the change in $C_p$ distribution observed. Induced flow field is calculated from Bio-Savart's law.

### 3.4 Combined tip vortex model for flow prediction:

This model recognizes the feature in a real flow, the presence of core rotating with a solid body rotation surrounded by a potential vortex.

### 3.5 Three dimensional viscous flow analysis:

There have been several attempts made to include viscosity and solve the exact form of Navier-Stokes equation in the gap region.

One of the earliest analysis is due to Rains (1954). He used Eq.(1) for calculating energy loss due to leakage flow. He was also one of the first to analyze the roll-up of leakage flow into vortex. Considering the deformation of the leakage flow sheet due to velocity induced by vortex. Considering the leakage of the vortex flow sheet due to velocity induced by vortex, and applying the Bernoulli's equation across the gap he proved that radius of the vortex (a) is

$$\frac{a}{\zeta} = 0.14 \left[ \frac{d}{\zeta} C_L^{1/2} \right]^{0.85} \tag{2}$$

**Combined vortex model for flow prediction-**

A potential or a point vortex, is not valid for accurate prediction of blade to blade flow in the clearance region. The detailed flow pattern does not resemble that of the potential vortex which explains it. The potential vortex model fails to predict the loss core near the clearance region and velocities.

Leakage flow originating from the tip all along the chord form a core of rotating fluid which lies below the suction surface and inboard of the blade tip. A theoretical model which takes into account the presence of vortex core with solid body rotation was developed by Laxminarayana (1970). The analysis is valid for incompressible and inviscid flows, and inviscid flow, and implicit in the analysis of vortex formation. If the flow is turbulent vortex may diffuse rapidly; the theory is not valid in such a case.

This model is represented in figure. Space between rows of the two vortices is given by 2(ζ+a). The inner region behaves as forced vortex and the outer region behaves as free vortex.

Radius of the vortex core (a): Calculate by using eq. (2)

Angular rotation of vortex core (ω) : Angular rotation of the vortex can be determine if the radius and strength of the shed vortex (ω) are established.

$$\omega = \frac{(1-K)\Gamma}{2\pi a^2} \tag{3}$$

Where, $1 - K = 0.23 + 7.45\left(\frac{\zeta}{s}\right)$

For, $0.01 < \frac{\zeta}{s} < 0.1$

Location of the vortex core in the passage-

The induced flow of the vortex flow tends to move the vortices away from the suction surface. If we know the induction flow field due to image vortices ad assume that the vortices originated at the leading edge of the blade, we can compute the distance b.

$$v_L = \frac{(1-K)\Gamma}{2S}\left[\frac{\sin hM}{\cos hM - \cos 2\pi\frac{(y-b)}{S}} - \frac{\sin hN}{\cos hN - \cos 2\pi\frac{(y-b)}{S}}\right] \tag{4}$$

$$w_L = \frac{(1-K)\Gamma}{2S}\left[\frac{\sin 2\pi\frac{(y-b)}{S}}{\cos M - \cos 2\pi\frac{(y-b)}{S}} - \frac{\sin 2\pi\frac{(y-b)}{S}}{\cos N - \cos 2\pi\frac{(y-b)}{S}}\right] \tag{5}$$

Where,

$$M = \frac{2\pi(Z-a-r)}{S} \quad and, N = \frac{2\pi(Z+a+r)}{S}$$

*Note – For further relations, graphs and comparison refer appendix.*

## 4. Passive and active control
### 4.1 Passive control

Passive control methods, are static processes, which mainly uses geometry modification in order to control the leakage flow. The passive control methods cannot be changed after manufacturing. The geometrical modification can be applied to the blade or casing. The following are examples of these methods:

**Casing treatment**

Different designs were applied to the casing such as honeycomb (Moffitt and Whitney, 1983) and casing trenching

**Blade treatment:**

Modifying the blade tip is usually done by inserting some obstacles to the leakage flow motion in the cross-tip direction. These
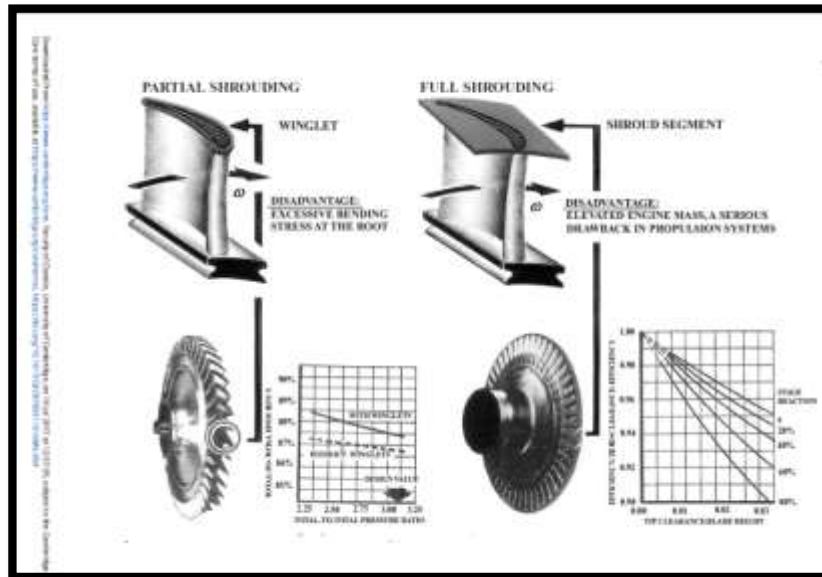


Figure 5 Partial(a) and full shrouding(b) for indirect tip-leakage control [Shroud System, Baskharone (2006)]

obstacles may take the form of shroud, squealers, pins, etc.

In the shroud system, there are two configurations of the shrouds: Full and partial shrouds. The more common one is full shroud shown in Fig. 5(a). It is a cylindrical sheet that is wrapped around all tips of the rotor blade. In spite of the efficiency of the full shroud in preventing leakage flow, it has a negative effect on the blade aerodynamics due to blockage of main stream which reduces the blade efficiency. Moreover, the shroud increases the engine weight, which is very important for jet engines, and consequently increases the root stresses. The other type of the shrouds is the partial shroud (winglet) shown at Fig. 5(b). It was basically devised to reduce the weight of the full shroud but it did not eliminate it completely. The partial shroud may attach to: the pressure side, suction side and pressure and suction sides (double sided) Squealers.

**Squealers**

The squealer is an obstacle in the shape of strip (cantilever) mounted to the tip. Squealer tips are gaining popularity among manufactures because it reduces the weight of machine and its aerothermal performance. The squealers usually placed along the pressure side, suction side and camber line as see from Fig. 6. Single squealer or combination of two squealers is commonly used as shown in Fig. 1.9.
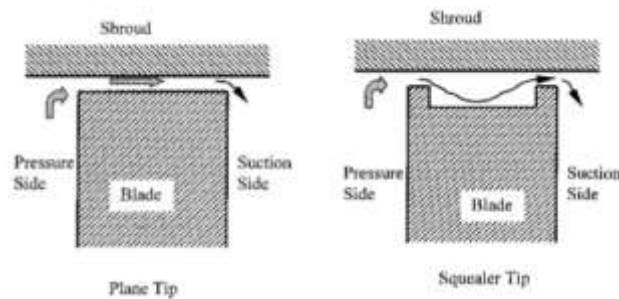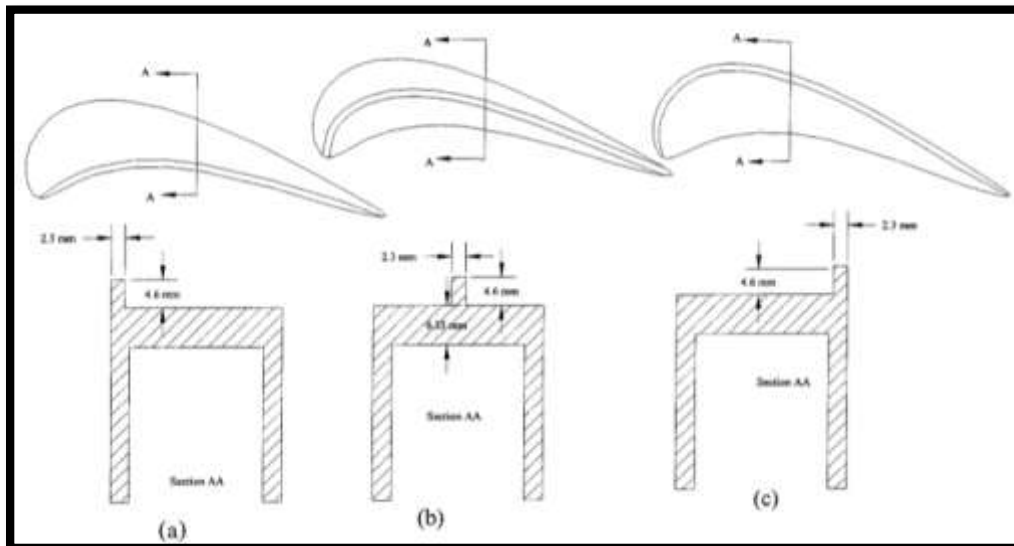


Figure 6 Squealers



Figure 7 Various combinations of two squealers: (a) pressure and suction side(double squealer), (b) pressure side and camber line and (c) camber line and suction , Azad et al. (2003)
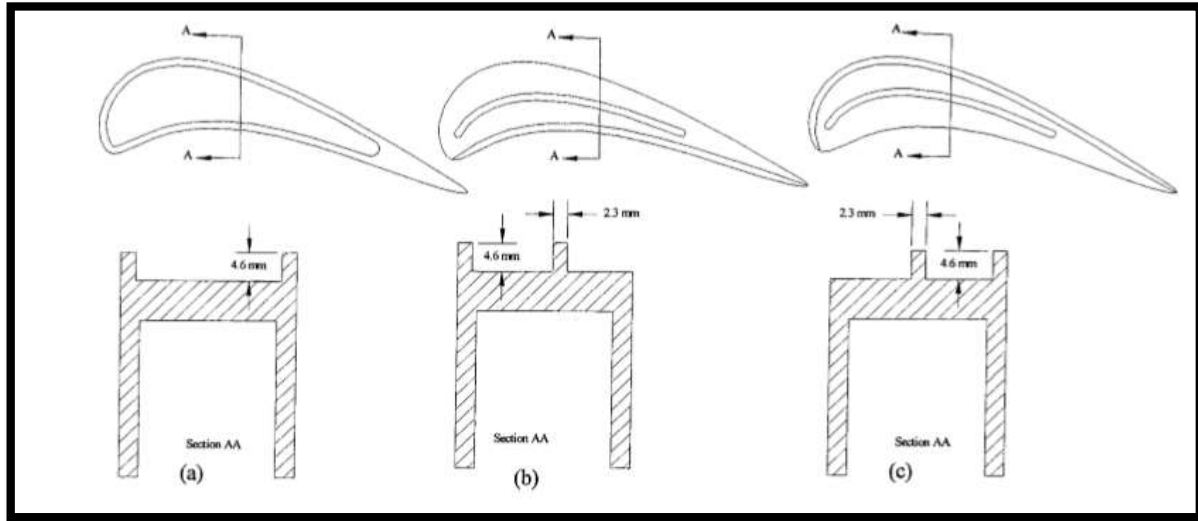
Figure 8 Various positions of single squealer: (a) pressure side, (b) camber line and
(c) suction side, Azad et al. (2003).

## 4.2 Active control

Active control methods are dynamic processes, usually an external energy sources are used, to control the leakage flow. It can be turned on/off at any time. Plasma jet and air suction/injection are examples. In Plasma jet method, electrical energy is supplied to Plasma actuator to generate Plasma jet, Van Ness II (2009). Figure 1.3 shows a schematic of a Plasma actuator. The actuator is placed on the tip as seen in Fig. 1.4. In air injection method, compressed air is injected through certain holes to counter the momentum of the leakage flow, Dey (2001), as shown in Fig. 1.5. The air flow may be normal or inclined. The injection holes may be located on the tip or on the casing.
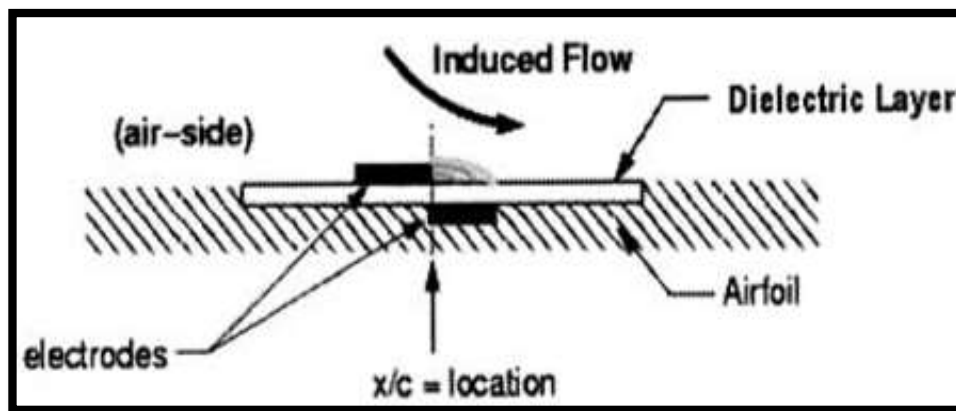


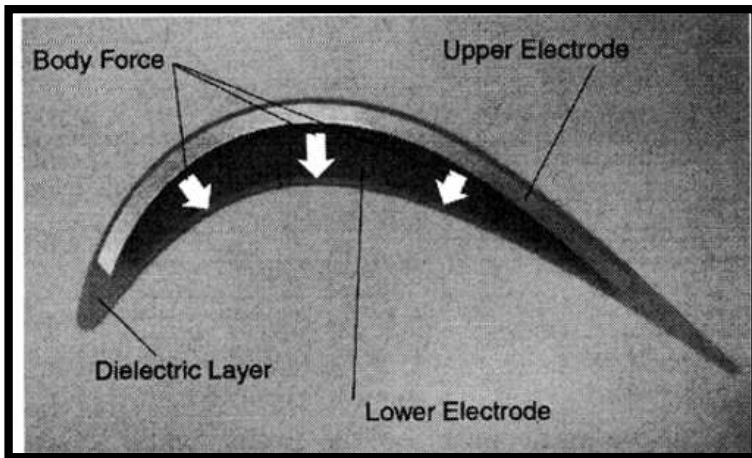Figure 8 Schematic of Plasma actuator, Van Ness II (2009).

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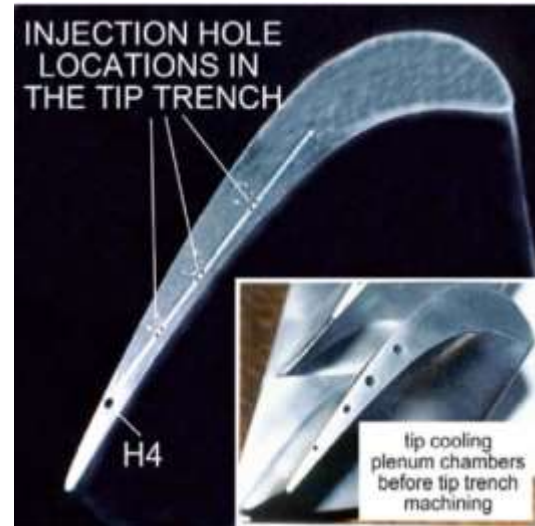Figure 10 Plasma actuator placed on a blade tip, Van Ness II (2009).



Figure 9 Air injection through blade tip, Rao et al. (2004).

## 5. CONCLUSIONS

- The tip clearance loss accounts for up to one-third of the total losses in a turbine stage.
- The reason for generation of tip clearance effect and tip-leakage vortex formation were investigated.
- There are different type of models by which tip-leakage flow and vortex can be investigated. With some limitations, each model can be applied for different cases. Combined vortex model for flow prediction is found to be best to apply in case of viscous flow consideration.
- Different desensitization methods for flow control has been defined in this paper. passive and active control methods have been discussed elaborately.

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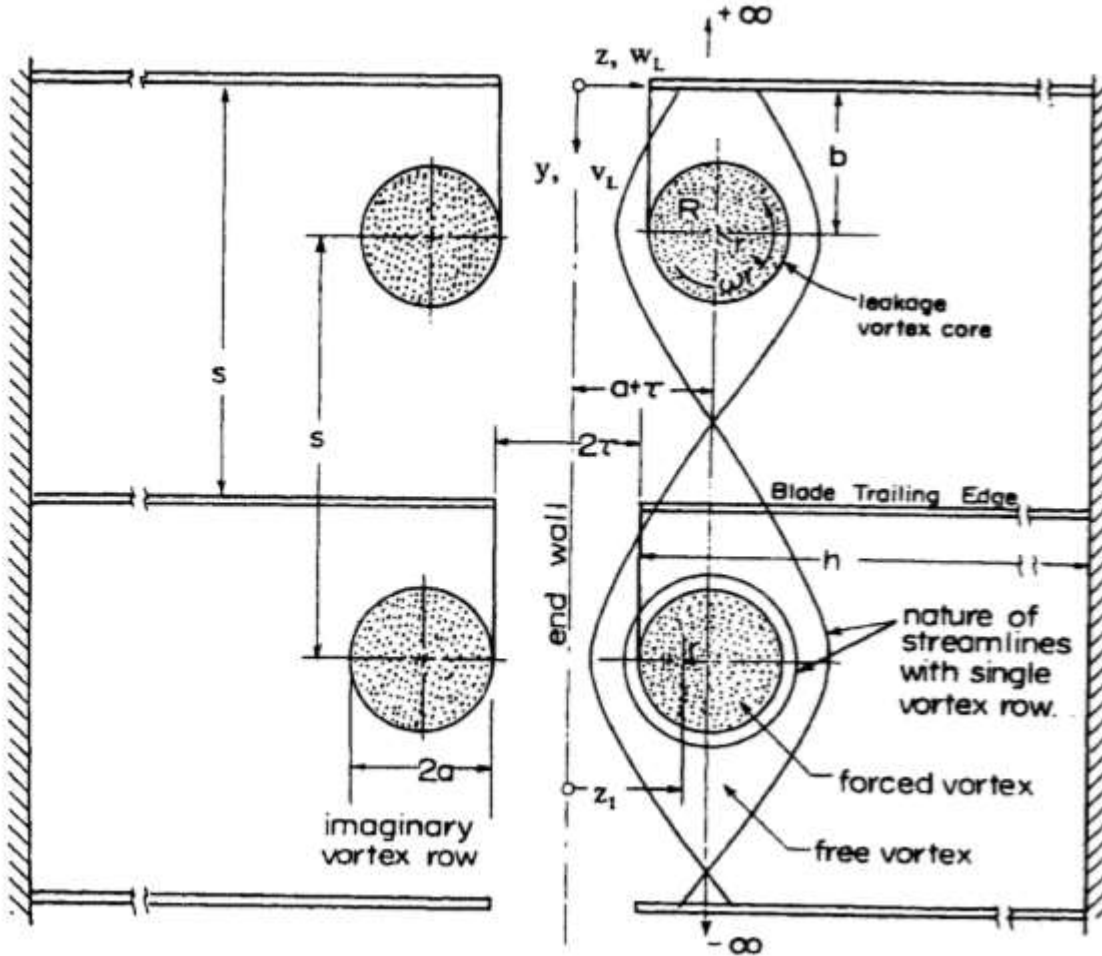
# Appendix-



Figure 11 Modified clearance flow model showing the location of vortices in the tangential plane of the cascade

Inside the vortex core deviation from flow outlet angle at any spamwise location $z=z_1$ is given by

$$\tan \Delta\beta_2 = \frac{\omega r}{w_m} = \frac{(1-K)C_L C}{4\pi a}\left(\frac{r}{a}\right) \qquad (6)$$

Where, $r = a + \tau - z_1$

Flow is overturned when $z<(a+\tau)$ and underturned when $z>(a+\tau)$. Outside the vortex core, the change in outlet angle is given by,

$$\tan \Delta\beta_2 = \frac{\omega r}{w_m} = \frac{(1-K)C_L C}{4S}\left[\frac{\sin hM}{\cos hM - \cos 2\pi \frac{(y-b)}{s}} - \frac{\sin hN}{\cos hN - \cos 2\pi \frac{(y-b)}{s}}\right] \qquad (7)$$

If we know the $\Delta\beta_2$ indicates underturning and negative values indicates overturning. We can calculate the outlet angle $\beta_2 = \beta_{2p} + \Delta\beta_2$

At the spanwise location (z) kinetic energy

$$\frac{w_L^2 + v_L^2}{V_1^2} = \left(\frac{(1-K)C_l C V_m}{4SV_1}\right)^2 [I_1 + I_2] \qquad (7)$$

Where,

$$I_1 = \int \left[ \frac{\sin hM}{\cos hM - \cos 2\pi \frac{(y-b)}{s}} - \frac{\sin hN}{\cos hN - \cos 2\pi \frac{(y-b)}{s}} \right]^2 d\left(\frac{y}{S}\right) \tag{8}$$

$$I_2 = \int \left[ \frac{\sin 2\pi \frac{(y-b)}{s}}{\cos M - \cos 2\pi \frac{(y-b)}{s}} - \frac{\sin 2\pi \frac{(y-b)}{s}}{\cos N - \cos 2\pi \frac{(y-b)}{s}} \right]^2 d\left(\frac{y}{S}\right) \tag{9}$$

Limitations of the model-
- The analysis has not been validated for centrifugal compressor.
- It is not valid for transonic rotors with shock.
- If the viscous and turbulence effect are large, inviscid flow assumption is not valid.
- In such situations where flow separates at the corner on the blade surface near the trailing edge. The theory is not accurate for such case.
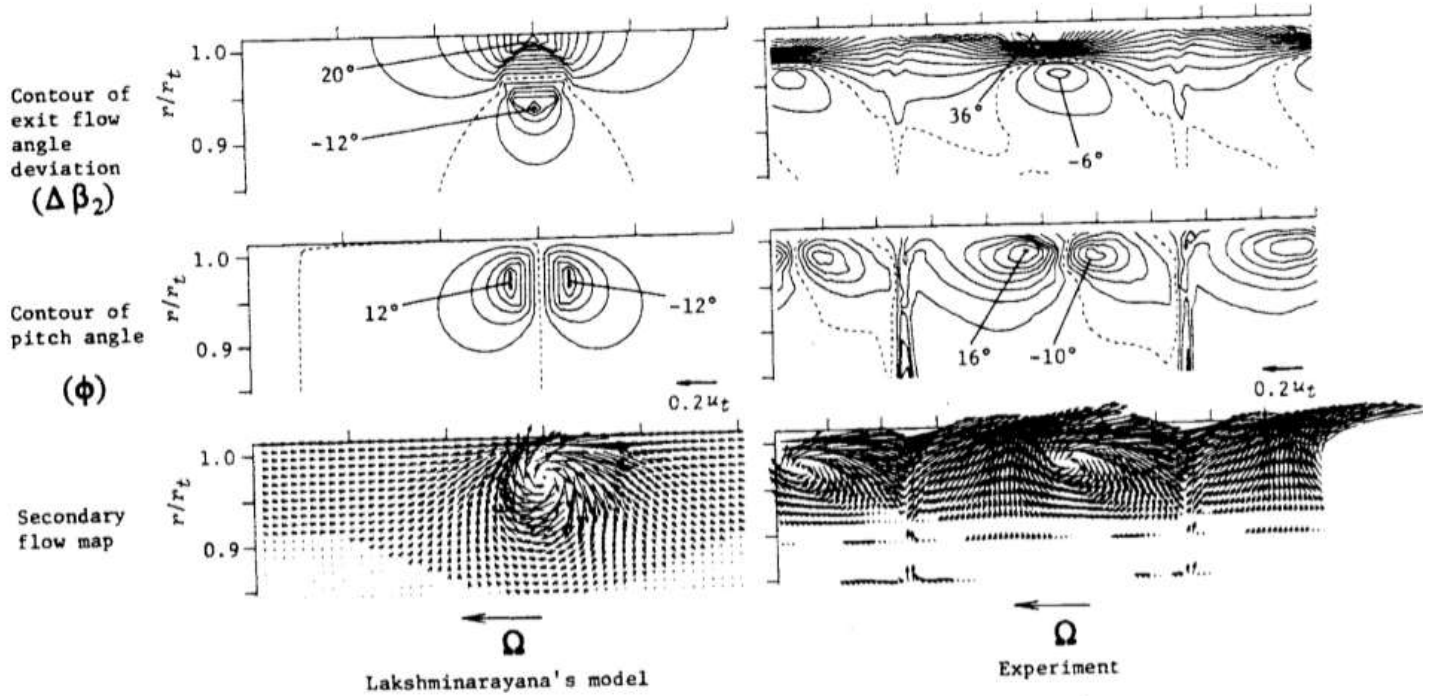


Figure 12 Comparison of exit flow angle deviation, pitch angle, and leakage flow velocity between experiment and Laxminarayan model for low speed axial flow compressor