Tip Leakage Flow: A Comparison between Small Axial and Radial Turbines

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ABSTRACT

A brief review is given on the nature of tip leakage flow in radial inflow turbines. A comparison is made of tip leakage loss in a single stage radial turbine and a two stage axial turbine. A simple tip leakage loss model is applied to both types of turbines for a given duty that is typical of micro turbine generators. For the axial turbine a loss model from the literature is applied based on discharge coefficients in the tip gap. For the radial turbine the same model is adapted to account of the scraping effect, which can dominate the tip leakage flow in this type of a turbine. The discharge coefficients, which govern the amount of leakage flow and the therefore the loss, are approximated from experiments. The analytic comparison shows that the scraping effect can reduce the tip leakage loss up to a factor of four between radial and axial turbines for the same duty.

NOMENCLATURE

$m$ mass flow rate
$p$ static pressure
$v$ velocity
tip gap height (z-axis)
$C_d$ discharge coefficient
$S_m$ meridional length
$U$ blade speed
$\rho$ density

Subscripts
3 rotor inlet
n blade normal
p passage
tf pressure driven tip leakage flow
sf scraping flow

INTRODUCTION

The operation of unshrouded steam and gas turbines requires a minimum of tip clearance between the rotating blades and the stationary casing. This gap gives rise to leakage flow that is driven by the pressure difference between the pressure side and the suction side. The tip leakage flow passes across the tip. It is largely unturned, so that no work is done. Consequently, it exits the tip gap with a magnitude of velocity similar to that of the mainstream flow relative to the rotor, but in a near perpendicular direction to the mainstream flow. The mixing of the mainstream and tip leakage flows gives rise to the loss of efficiency.
It is possible to change the effect of tip clearance by mounting a shroud on the tip of the blade. In a shrouded blade, the driving force of leakage through the seals is the pressure difference between rotor inlet and rotor exit as opposed to the pressure difference between the pressure and suction surfaces. Fitting a shroud to a rotating blade usually increases the aerodynamic efficiency of the rotor. It may also facilitate the active control of the tip gap, as the blade height varies with the expansion caused by temperature changes in the turbine. However, the weight of the shroud also requires the rotor to rotate at a lower speed. For a given expansion rate the shrouded turbine results therefore in less work output per stage than the unshrouded turbine, because, for a given stage loading coefficient, the specific work is proportional to the square of the blade speed. In radial turbines, a shroud also increases the complexity of manufacture of the rotor wheel.

PRELIMINARY STUDY

The technology of micro turbines has already been well studied in the field of turboshaft engines for aircraft applications. The continuing requirement for turboshaft engines is to reduce both the specific fuel consumption and the weight. If the engine is based on the simple cycle, the engine pressure ratio can be as high as 20:1, at a turbine inlet temperature of 1800 K or more. The weight target also leads to the requirement of a minimum number of turbomachinery stages with high efficiency. Hill (1989) investigated three candidates for a gas generator turbine. These are a two stage axial, a single stage axial and a single stage radial turbine. In a simple analysis, Hill assumed that the stage efficiency is a function of turbine loading alone. If optimal flow coefficients are used, this is indeed correct according to the Smith Chart. In order to calculate the aerodynamic efficiency, he made further assumptions about the blade speed, allowable metal temperature and cooling efficiency as a function of technology level. Figure 1 shows the improvement in efficiency and specific fuel consumption compared to the datum case. The assumptions underlying the analysis are summarised in the following table.

<table>
<thead>
<tr>
<th>Blade speed (axial)</th>
<th>Blade tip speed (radial)</th>
<th>Allowable metal temperature</th>
<th>Convective cooling efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>460 m/s</td>
<td>750 m/s</td>
<td>950 C</td>
<td>100%</td>
</tr>
</tbody>
</table>

Figure 1 shows that the two stage axial and the single stage radial turbine have the potential for a significant improvement in efficiency and specific fuel consumption for the given technology assumption. The radial turbine is a good match to the two stage axial turbine as it is operating at an optimum loading and requires lower cooling flows. The single stage axial turbine is operating at too high a loading in order to be competitive with the two other machines.

WHEEL MANUFACTURING

Many factors go into the choice of the wheel type for a given application. The brief analysis by Hill (1989) has shown that a single stage radial turbine has to be compared with a two stage axial turbine. The chosen wheel type and the method of manufacturing have many implications on the product quality, the cycle time and the
final cost. A single stage radial turbine has fewer parts than a two stage axial turbine and is usually more robust. Figure 2 shows, for example, how different manufacturing methods affect the designer’s choice of the maximum tip speed of a radial turbine. All these factors will also play a role in the final cost of the product. The manufacturing costs of a closed wheel that is milled from solid are, for example, three times higher than those of an open wheel that is milled from solid. In the following, the paper concentrates on the open wheel type. This may be investment cast or milled from solid.

**TIP LEAKAGE FLOW - A MAJOR LOSS SOURCE IN SMALL TURBINES**

The mixing of the tip leakage flow inside the gap and with the mainstream at the gap exit creates entropy (loss). This loss contributes up to a third of the loss of efficiency in a turbine. Typically, a clearance gap equal to one percent of the blade height is associated with a two or more percent loss of stage efficiency in an unshrouded, axial turbine (see Sjolander, 1997, for a brief but complete overview of the available literature on this subject). The penalty of a one percent clearance as percentage of the blade height in unshrouded radial turbines is about one percent of stage efficiency (see Japikse and Baines, 1994). A typical tip clearance in a medium-size turbine is about 1 to 2 percent of the span.

**THE NATURE OF TIP LEAKAGE FLOW IN A RADIAL INFLOW TURBINE**

Tip leakage flow in radial turbines appears to be driven by two main mechanisms (Dambach et al., 1998, Dambach and Hodson, 1999, Dambach, 1999). The first mechanism is the pressure difference over the rotor tip. The second mechanism is the blade normal component of the relative casing motion. The degree of interaction between these two mechanisms determines the nature of the flow in the gap region of a radial turbine.

The results of measurements in the tip gap region have shown three regions of tip leakage flow behaviour in a radial turbine, as schematically indicated in Figure 3 (Dambach, 1999). In the inducer region (labelled I), the nature of tip leakage flow is very different from that of axial turbines. The weak pressure difference over the tip of the inducer supports only a small amount of tip leakage flow. Scraping flow (fluid adjacent to the casing that is dragged by the strong relative casing motion) opposes the tip leakage flow on the suction side. A part of this scraping fluid is dragged through the gap to the pressure side of the blade (dragging effect). This dragging effect of scraping can dominate the flow inside the tip gap. It is also observed that the change of momentum of the scraping flow supports a tangential pressure gradient.

In the midsection (labelled II), the relative casing motion is weakened and the loading near the casing increases. A strong tip leakage flow accelerates into the gap and travels through the gap in near blade perpendicular direction. Because of the increased momentum of the tip leakage jet, most of the scraping fluid is blocked off near the suction side (blocking effect). This blocking effect causes a static pressure rise on the suction side. The part of scraping fluid that is still dragged into the gap is diverted, and little or no scraping flow exits the gap on the pressure side.

In the exducer (labelled III), the influence of the relative casing motion upon the tip leakage flow is negligible. The blocking effect takes place further away from the suction side and no scraping fluid is dragged through the gap. The tip leakage flow behaviour in the exducer is very similar to that in axial turbines.
Figure 4 shows secondary velocity vectors at 9\% S_m and at 58\% S_m for t = 1.2\% of span. It appears from Figure 4 that CFD (Denton, 1990) is able to resolve the local features of scraping flow in the gap region. Figure 4a shows that a small amount of tip leakage flow is turned immediately towards the hub, as it leaves the tip gap on the suction side. One part of the scraping flow is dragged into the gap by the dragging effect. The other part is scraped off the blade and deviated towards the hub and part of the fluid is seen to roll up near the suction side. Figure 4b displays that the CFD results capture a distinct blocking effect of scraping at the gap exit. It seems apparent that as a result of the blocking effect, the tip leakage flow turns sharply towards the hub.

A SIMPLE TIP LEAKAGE LOSS ANALYSIS

Several attempts have been made in the past to model the mixing processes involved with tip leakage flow in axial turbines. Endeavours to predict the tip leakage loss for radial turbines have mainly remained in the sphere of empirical loss estimates (see Baines, 1998). Although the empirical approach is no doubt useful to the designer, the aim of this paper is to apply a more physical model for tip leakage loss estimates. In so doing, it is hoped to bring loss models for radial turbines up to the standard of existing loss models for axial turbines.

If the tip leakage loss is to be predicted at an early design stage, an average tip gap mass flow as well as the velocity of the main flow on either side of the gap has to be approximated. According to Denton (1993), the loss of efficiency based on average quantities is calculated as follows

\[ \Delta \eta = \left( \frac{m_{tf}}{\rho v_{PS}} \right) \left( v_{PS}^2 - v_{SS}^2 \right) \]

where \( m_{tf} \) is the pressure driven tip leakage mass flow and \( v_{SS} \) and \( v_{PS} \) are the undisturbed components of the blade parallel velocity on either side of the blade sufficiently far away from the tip gap. Equation 1 assumes that the streamwise momentum of the tip leakage flow remains unchanged on its way to the suction side.

In order to demonstrate the applicability of a future correlation, a very simple Mach number distribution was chosen to approximate the velocity components \( v_{PS} \) and \( v_{SS} \). Figure 5 shows this idealised Mach number distribution in the relative frame. The velocity components were calculated as a function of the inlet and exit Mach number fixed by the velocity triangles and the assumed shape shown in Figure 5. The tip gap mass flow was estimated based upon the pressure difference over the tip and a tip gap discharge coefficient as follows

\[ m_{tf} = C_D \rho v_n = C_D \rho v \sqrt{v_{SS}^2 - v_{PS}^2} \]

where it is assumed that the leakage flow (velocity = \( v_n \)) exits the gap normal to the pressure surface. The result of Equation 2 was inserted into Equation 1.

For the axial turbines, the discharge coefficient defined by Heyes and Hodson (1993) was used. They assumed that partial mixing of the tip leakage jet after the vena contracta occurs between 1.5t and 6t of the blade width inside the tip gap. They introduced a partial mixing factor \( M \) defined as the fraction of tip gap mass flow rate
that mixes inside the gap. Assuming a linear variation of the partial mixing inside the
gap with $\lambda$, where $\lambda$ is the tip gap height to width ratio ($t/w$), $M$ takes the following
values

\[
\begin{align*}
M &= 0 & \text{for } \lambda > 2/3 & \text{no mixing} \\
M &= 4/3 - 2*\lambda & \text{for } 1/6 > \lambda > 2/3 & \text{partial mixing} \\
M &= 1 & \text{for } \lambda < 1/6 & \text{complete mixing}
\end{align*}
\]

Using $M$ and the contraction coefficient $\sigma$ at the vena contracta, the momentum
equation in blade normal direction for the case of tip leakage flow without casing
rotation yields the discharge coefficient as follows

**Equation 3**

\[
C_D = \frac{\sigma}{\sqrt{1 + \frac{2\sigma M(\sigma - 1)}{1 - \sigma + \sigma M}}}
\]

McGreehan and Schotsch (1988) presented an empirical technique for calculating the
contraction coefficient $\sigma$ based upon the Reynolds number and the ratio of the
pressure side corner radius at the blade tip divided by the local gap height ($r/t$). For
the present work the same values were taken for the axial turbine as measured on the
test rig of the radial turbine. Those were $r/t = 0.2$ for gap A0, $r/t = 0.1$ for gap B0 and
$r/t$ for gap B8 (Dambach, 1999). Gap A0 corresponds to a gap of 0.6% of span, gap
B0 to a gap of 1.2% of span and gap B8 to an average value of 2.5% of span. The
above technique was applied and the contraction coefficient $\sigma$ was fed into Equation 3
to obtain a discharge coefficient $C_D$.

For the radial turbine the discharge coefficient $C_D$ was approximated from
experimentally determined discharge coefficients measured by Dambach (1999). The
measurements together with the approximations are shown in Figure 6. The discharge
coefficient is defined as the ratio of the actual pressure driven mass flow at gap exit
divided by the ideal tip gap mass flow as follows

**Equation 4**

\[
C_D = \frac{\rho \int_0^{z_0} v_n \, dz}{t \sqrt{2\rho \Delta p}}
\]

where $z = 0$ is at the blade tip surface and $z = z_0$ is the height where the blade normal
velocity is equal to zero. The results in Figure 6 show the effect of scraping upon the
tip gap mass flow in a radial turbine.

Figure 7 shows the tip leakage loss coefficient for a single stage radial turbine and a
two stage axial turbine based on Equation 1. Both turbines were designed with a
mean-line analysis for a given pressure ratio of 2.5 and inlet temperature of 1500 K.
The generated power of 200 kW is typical for micro-turbine applications. Figure 7
shows that the effect of scraping significantly reduces the tip leakage loss in a radial
turbine. For small clearances in the radial turbine the model indicates approximately a
one percent decrease of efficiency for a one percent clearance height per span, which
corresponds to experimental findings documented in the literature (Baines, 1998). For
small clearances in the axial turbine the model indicates about a two percent loss of 
efficiency for a one percent clearance height per span, which also corresponds to 
values found in the literature (Sjolander, 1997). The loss of efficiency due to tip 
leakage flow in axial turbines is then doubled because of the two stages.

CONCLUSION

This paper has shown how a tip leakage loss model for axial turbines (Denton, 1993) 
can be applied to radial turbines. In order to model the nature of tip leakage flow in 
radial turbines, the scraping effect is taken into account via a discharge coefficient. 
The result of the simple analysis shows that a radial turbine suffers less from tip 
leakage flow than axial turbine. For a given duty in the range of micro-turbine 
generators the model shows that tip leakage loss can be about four times worse than 
for a radial turbine for typical clearances. The difference is even be accentuated for 
smaller clearances because of the scraping effect in the inducer region of a radial 
turbine.

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FIGURES

Figure 1  Improvement of efficiency and SFC compared to datum case (Inlet temperature 1800 K and engine pressure ratio 20:1)

Figure 2  Allowable tip speed as a function of the manufacturing type of a radial turbine wheel
Figure 3  Schematic of the three regions of different tip leakage flow in a radial turbine
Scraping flow Dragging effect

a) Predicted Secondary velocity vectors at 9% $S_m$

Scraping flow Blocking effect Tip leakage flow

b) Predicted Secondary velocity vectors at 58% $S_m$

Figure 4  Predicted Secondary velocity vectors for a square tip CFD model

Figure 5  Idealised isentropic Mach number in the relative frame for an axial and a radial turbine
Figure 6  Measured and approximated discharge coefficients in a radial turbine for different gap heights (A0 is 0.6% gap/span, B0 is 1.2% gap/span and B8 is on average approximately 2.5% gap/span)

Figure 7  Comparison of tip leakage loss between a single stage radial turbine and a two stage axial turbine (Inlet total temperature = 1500 K, Pressure ratio = 2.2, massflow = 0.5 kg/s)