# THE EFFECT OF FAN TIP CONFIGURATION ON AIR-COOLED CONDENSER AXIAL FLOW FAN PERFORMANCE

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## SUMMARY

Large diameter axial flow fans used in air-cooled condensers (ACC) are major power consumers in in ACC units, therefore it is desirable to make them as efficient as possible. It is well known that the blade tip region has a substantial influence on the performance of such fans. Several tests were performed on an axial flow fan which was specifically developed for ACC applications, in order to determine the effect of tip configuration on the performance of the fan. These included the testing of blade tip modifications as well as tip clearance. The results show that tip clearance is a major contributor to fan performance and that blade tip modifications can improve fan performance at larger tip clearances. An empirical model for tip clearance is also presented.

## INTRODUCTION

Large air-cooled condensers (AACs) and air-cooled heat exchangers have the advantage over wet cooled systems as they use air rather than water as a cooling medium. This makes these systems popular in countries such as South Africa where water is scarce and wet cooled technology is not always viable. A major component of an ACC system is an array of large diameter axial flow fans which is used to force air through the fin-tube bundles in the system. The fans used for these purposes are specifically designed to give stable operation when running in parallel, as in an ACC. They generally have very low angles of incidence and pressure steadily drops across them as the flow rate increases. South Africa currently operates the world’s largest direct dry-cooled power plant, the 3.6 $GW^e$ Matimba power station at Lephalale in the Limpopo Province. The ACC at the plant makes use of 288 large diameter axial flow fans[1], each with a diameter of 9.1 m. Two new 4.8 $GW^e$ dry-cooled power stations are currently under construction in South Africa, namely the Medupi and Kusile power stations. The ACC units at these plants will require close to 400 axial flow fans, driven by electric motors in the 200 kW range[2]. It is thus desirable to make the fans as efficient as possible in order to improve overall plant efficiency.

Research has indicated that the blade tip region has a profound influence on both fan noise and fan performance. Tip clearance tests performed by Kroger and Venter on the "V-fan" indicate that tip clearance has a major influence on fan performance[3]. Tests and numerical studies of blade tip modifications by Corsini, et al. indicate that modifications, such as endplates, at the fan tip can improve performance as well as reduce noise levels[4,5,6].

Tip clearance is an important factor in blade tip configuration with various studies showing a correlation between tip clearance and fan performance[3,7]. Wallis states that, "Provided the tip clearance does not exceed 1% of the blade height no adjustments to design pressure duty or fan efficiency are necessary."[7]. However Wallis concedes that designing a fan with such tip clearances is not always practical[7]. According to Venter and Kroger it can be shown that the power consumption of a fan can be reduced by ensuring the clearance between the tip and the casing is small, having obvious benefits for fan efficiency[3]. The flow mechanisms in the tip region are complex with interactions between flow over the casing wall and the fan tip leakage flow. A secondary vortex exists on the convex surface at the blade tip due to flow turning[7]. This flow has a radial direction, leaking over the blade tip through the gap between the blade and the duct wall. According to Wallis tip leakage flow opposes this motion restricting the secondary flow[7]. Venter and Kroger describes this as, "air leakage from the higher pressure fan outlet stream to the lower pressure inlet region around the tips of the fan blades."[3] .Thus the concept of an optimum tip clearance is well-founded and worth investigation as an improved, or part of an improved tip configuration.

Corsini, et al have conducted several highly detailed studies, employing extensive computational simulation and experimentation of the effects of blade tip endplates[4,5,6]. Their studies mainly focus on passive noise control, however the concepts generated seem to have an effect on fan performance[5,6]. In all of the research conducted the same test fan was used, operating at exactly the same conditions. Early concepts such as the TF (Tip Feature) and TFvte (Variable thickness endplate) endplates showed slight improvements in terms of both noise and efficiency over the datum fan[4]. This was attributed to the endplates' influence on the vortex and tip leakage structures at the blade tip[4]. Further research into the influence of the end plates on blade tip vortex structures confirmed this[5]. This research showed that the variable thickness endplate, the TFvte endplate, provides effective control over leakage vortex bursting, as well as improving fan efficiency[5]. The improvement in efficiency was attributed to the reduced three dimensional flow rearrangements as a result of the endplates. It was also found through spanwise flow analysis that the endplates resulted in beneficial aerodynamic behaviour along the blade span[6]. In their latest research Corsini, et al. developed a multi vortex breakdown endplate based on the work in the previous two papers[6]. This endplate aimed to correlate the evolution of tip vortex swirl level and the geometry of the endplate. This resulted in an endplate with an unusual shape that eliminated many of the tip leakage effects at the blade tip. Testing showed that this design was quieter and performed better than the TF or TFvte designs. Notably the fan produced a better pressure characteristic than both previous designs as well as the datum fan[6].

## VENTER AND KROGER'S TIP CLEARANCE MODEL

Venter and Kroger developed an empirical model for the effect of tip clearance on fan performance based on tip clearance test data from the V-fan. The model predicts the fan static pressure and volumetric flow rate ratios in terms of dimensionless tip clearance as shown in figure 1.

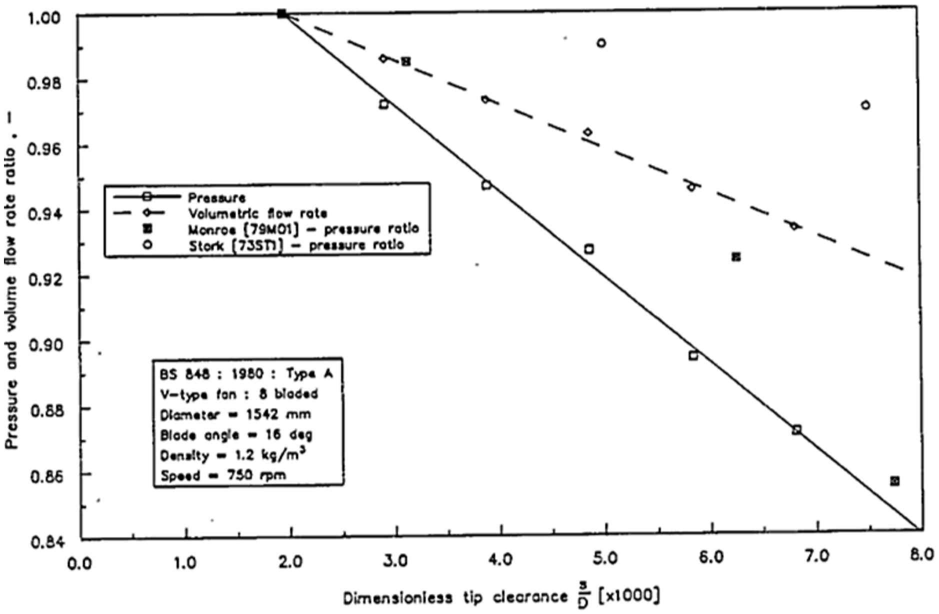



*Figure 1: Venter & Kroger "V-fan" empirical tip clearance model[3]*

The ratios are computed by comparing the ideal, or reference tip clearance, to larger, increased tip clearance values. The approximation used by Venter & Kroger is based on the assumption that the fan static pressure performance curve can be approximated by a linear equation[3]:

$$\Delta p_{sF} = m\dot{V}^2 + c \tag{1}$$

The model presented by Venter and Kroger requires a distinction to be made between parallel and non-parallel fan static pressure curves (for different tip clearances) such that parallel curves all have the same gradient m and non-parallel curves each have a unique gradient. Reference conditions are defined as the prevailing operating conditions of the fan with the smallest possible tip clearance and the reduced conditions are defined as the prevailing conditions at any larger tip clearance. System pressure loss coefficients are defined by Venter and Kroger as:

$$\Delta p_{loss} = K\frac{1}{2}\rho v^2 = k\dot{V}^2 \tag{2}$$

where

$$k = \frac{K\rho}{2A^2} \tag{3}$$

and

$$\dot{V} = vA \tag{4}$$

Venter and Kroger define the reduction in fan performance in terms of fan static pressure and volumetric flow rate ratios:

$$Fan\ static\ pressure\ ratio = \frac{(\Delta p_{sf})_{red}}{(\Delta p_{sf})_{ref}} \tag{5}$$

$$Volumetric\ flow\ rate\ ratio = \frac{\dot{V}_{red}}{\dot{V}_{ref}} \tag{6}$$

When designing a fan system the operating point of the fan is determined by obtaining the intercept on the fan curve between fan static pressure and system resistance. This means that the following relationship exists at the fan operating point for the reference curve:

$$(\Delta p_{sF})_{ref} = \Delta p_{loss} = k\dot{V}^2{}_{ref} \tag{7}$$

and for the reduced curve

$$(\Delta p_{sF})_{red} = k\dot{V}^2{}_{red} \tag{8}$$

Therefore

$$\frac{\dot{V}_{red}}{\dot{V}_{ref}} = \sqrt{\frac{(\Delta p_{sF})_{red}}{(\Delta p_{sF})_{ref}}} \tag{9}$$

The average fan static pressure ratio for the practical range of pressure loss coefficients in the application of a fan (say k varies between $k_1$ and $k_2$) is defined as follows:

$$\left[\frac{(\Delta p_{sF})_{red}}{(\Delta p_{sF})_{ref}}\right]_{avg} = \frac{\int_{k2}^{k1}\frac{(\Delta p_{sF})_{red}}{(\Delta p_{sF})_{ref}}dk}{\int_{k2}^{k1} dk} \tag{10}$$

This can be show to reduce to the following for non-parallel curves:

$$\left[\frac{(\Delta p_{sF})_{red}}{(\Delta p_{sF})_{ref}}\right]_{avg} = \frac{c_{red}}{c_{ref}}\left[1 + \frac{m_{red}-m_{ref}}{k_2-k_1}\right]\ln\left(\frac{k_2-m_{red}}{k_1-m_{red}}\right) = \left(\frac{\dot{V}_{red}}{\dot{V}_{ref}}\right)^2 \tag{11}$$

Which reduces to the following for parallel curves:

$$\left[\frac{(\Delta p_{sF})_{red}}{(\Delta p_{sF})_{ref}}\right]_{avg} = \frac{c_{red}}{c_{ref}} = \left(\frac{\dot{V}_{red}}{\dot{V}_{ref}}\right)^2 \tag{12}$$

This model will be applied to the B2-fan tip clearance data in order to test its applicability to the fan. These results, as well as the tip clearance and endplate test results are presented in this paper.

## TEST FAN

The fan selected for this project is a concept fan developed at Stellenbosch University, known as the B2-fan. This fan was specifically designed as a large rotor only fan for use large air cooled condensers[8]. It is suited to this project because it was proven to have good performance characteristics compared to other commercial fans in terms of efficiency and static pressure rise[8]. The fan is also similar in character to a free vortex fan which is useful for calculation purposes. The fan employs the NASA LS GAW 2 aerofoil profile and the specifications of the fan are listed in table 1 below.

*Table 1: B2 Fan specifications*[8]

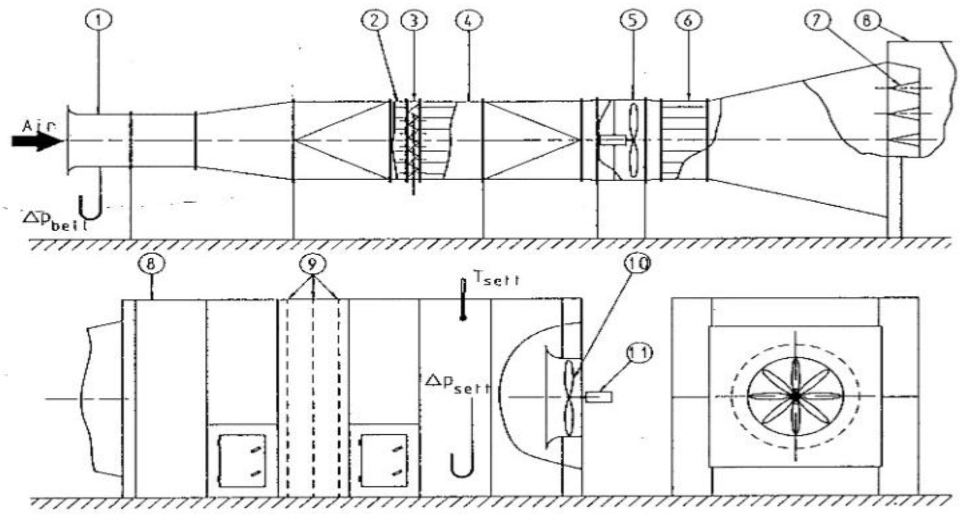
| | |
|---|---|
| Shroud diameter [m] | 1.542 |
| Blade number | 8 |
| Chord hub [m] | 0.184 |
| Chord tip [m] | 0.153 |
| Fan diameter [m] | 1.536 |
| Hub/tip ratio | 0.4 |
| Hub diameter [m] | 0.6144 |

In order to accommodate blade tip modifications the fan blades had to be modified. This involved the manufacture of two stainless steel mounting points. Two holes were drilled into the tip of each blade along the camber line and the mounting points were fixed into the blade tip by means of epoxy resin. The mounting points allowed modifications to be fixed to the blade by means of two M3 countersunk screws. A photograph of the mounting points in the blade tip is shown in figure 2 below.

*Figure 2: Blade tip mounting points at the blade tip*

# FAN TEST FACILITY

All tests were undertaken on the large diameter fan test facility at Stellenbosch University. This facility is classified as a BS848 type A (Free inlet free outlet) test facility, meaning that it has been designed to the standards prescribed in BS848[9]. A schematic drawing of the facility is shown in figure 3.




1- Inlet bellmouth
2- Flow straightener
3- Flow control louvres
4- Duct
5- Booster fan
6- Flow straighteners
7- Flow straighteners
8- Plenum chamber
9- Mesh screens
10- Test fan within duct
11- Electric motor

*Figure 3: The BS 848 type A fan test facility at Stellenbosch University[3]*

Ambient conditions are monitored using a mercury thermometer for temperature, which is accurate to 0.5 $^{o}$C and pressure using a mercury barometer accurate to 0.005 mmHg.

A calibrated bellmouth is used to determine the volumetric flow rate through the system. A static pressure measurement is taken at the bellmouth, allowing the volumetric flow rate through the system to be calculated by the following equation, were αε is a known calibration constant.

$$\dot{V} = \alpha\varepsilon \frac{\pi d_{bell}^2}{4}\sqrt{\frac{2\Delta p_{bell}}{\rho_{atm}}} \tag{13}$$

Downstream of the bellmouth is a series of louvres which can be adjusted in order to vary the flow rate. Downstream of these is an auxiliary fan which is used to make up for the pressure and frictional losses in the system. After this fan there are several flow straighteners to eliminate the whirl introduced into the system by the auxiliary fan. The air then enters a large plenum chamber. In this chamber static pressure is again measured. From this measurement one can deduce the fan static pressure by the following equation.

$$\Delta p_{sF} = -(\Delta p_{plen} + p_{d_{settling}}) \tag{14}$$

where

$$p_{d_{settling}} = \frac{1}{2}\rho\left(\frac{\dot{V}}{A_{plen}}\right)^2 \tag{15}$$

where $A_{settling}$ is the known cross sectional area of the chamber.

Attached to the shaft is a magnetic pickup which is used to measure the rotational speed of the shaft. A torque transducer is also fixed to the shaft in order to measure the fan power. Using the following equations shaft power and fan static efficiency can be calculated.

$$P = T\left(\frac{2\pi N}{60}\right) \tag{16}$$

$$\eta_{sF} = \frac{\Delta p_{sF}\dot{V}}{P} \tag{17}$$

The final step of the calculation procedure is to scale the results to the representative pressure and density using the following equations:

$$\dot{V}' = \dot{V}\left(\frac{N'}{N}\right) \tag{18}$$

$$\Delta p'_{sF} = \Delta p_{sF}\left(\frac{N'}{N}\right)^2\left(\frac{\rho'}{\rho_{plen}}\right) \tag{19}$$

$$P' = P\left(\frac{N'}{N}\right)^3\left(\frac{\rho'}{\rho_{plen}}\right) \tag{20}$$

$$\eta'_{sF} = \eta_{sF} \tag{21}$$

where

- $\rho'$ = 1.2 kg/m$^3$
- $N'$ = 750 rpm

These equations and measurements cover all the data that is needed to assess the performance of both the standard and modified fan.

The instruments used in the experiment are:

- Two HBM PD1 inductive pressure transducers to measure the bellmouth and settling chamber static pressures. They have a range between -1000 and 1000 N/m$^2$.
- A HBM T22 torque transducer with a measurement range of -100 Nm.
- A magnetic pickup to measure rotational speed.

The signal from the speed transducer is sent through a frequency to voltage converter, which converts the frequency reading from the pickup into a voltage reading which is the sent, along with the signal from the pressure transducers and torque transducer to the data acquisition system. The data acquisition system consists of an HBM PMX data acquisition system and computer. Communication between the computer and PMX box is two way, allowing excitation voltages and calibration constants to be set via the CATMAN AP user interface on the computer. Data acquired is read into the computer through the CATMAN software and saved for analysis.

# TIP CLEARANCE TESTS

The first tests performed were the tip clearance tests. The fan was mounted with the blades set at the design angle of 31 degrees. Test runs were performed for tip clearances between 2 mm and 7 mm from the duct wall. The results of these tests are presented below.

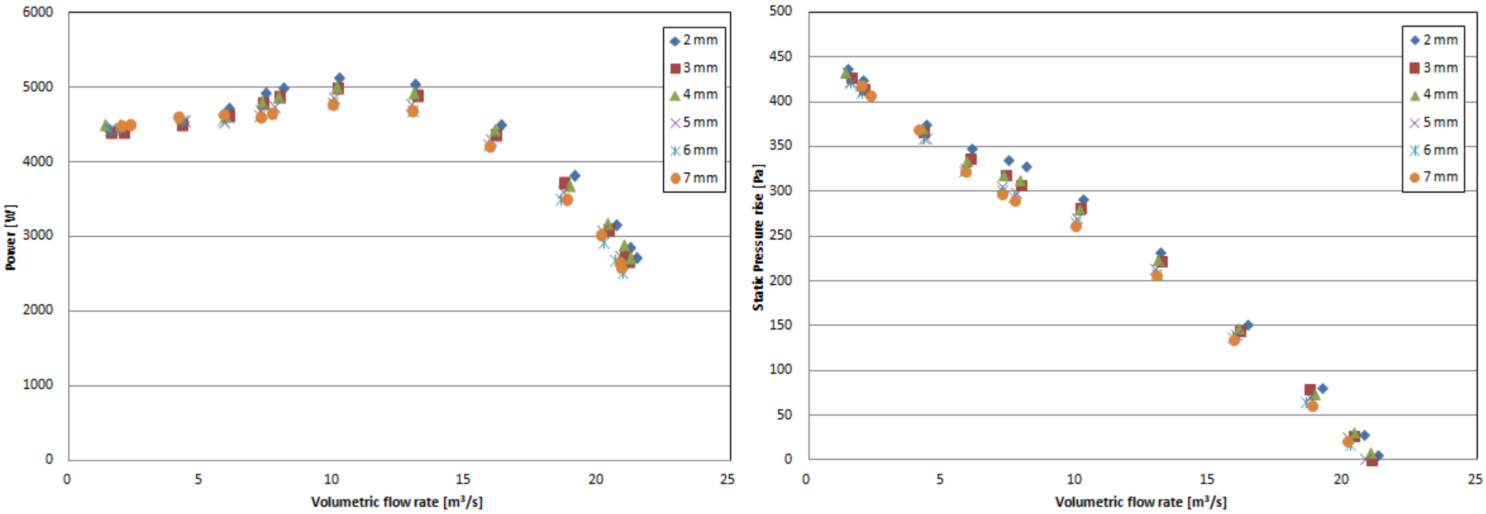



*Figure 4: Fan power (L) and fan static pressure (R) for the B2 fan at various tip clearances*

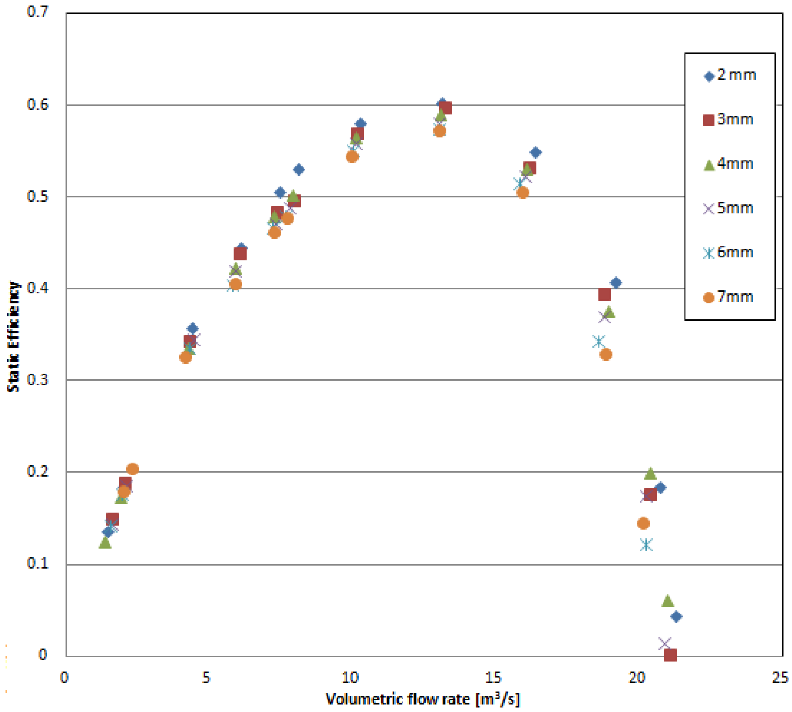



*Figure 5: Fan static efficiency at various tip clearances*

Based on this data the empirical model developed by Kroger and Venter was applied. After applying the assumption to linearise the data it was seen that the curve fits that would be used to compute the model did not correlate well with low flow rate data. This meant that the model would provide inaccurate forecasts for the influence of tip clearance on fan performance. For this reason it was decided to generate a piecewise model by applying the correlation to data for flow rates above and below 6 $m^3/s$ separately. 6 $m^3/s$ was chosen as the split point as the fan static efficiency data starts to spread out beyond this flow rate, as can be seen in figure 5. This resulted in the curve fits shown in figure 6 and the correlations shown in figure 7.

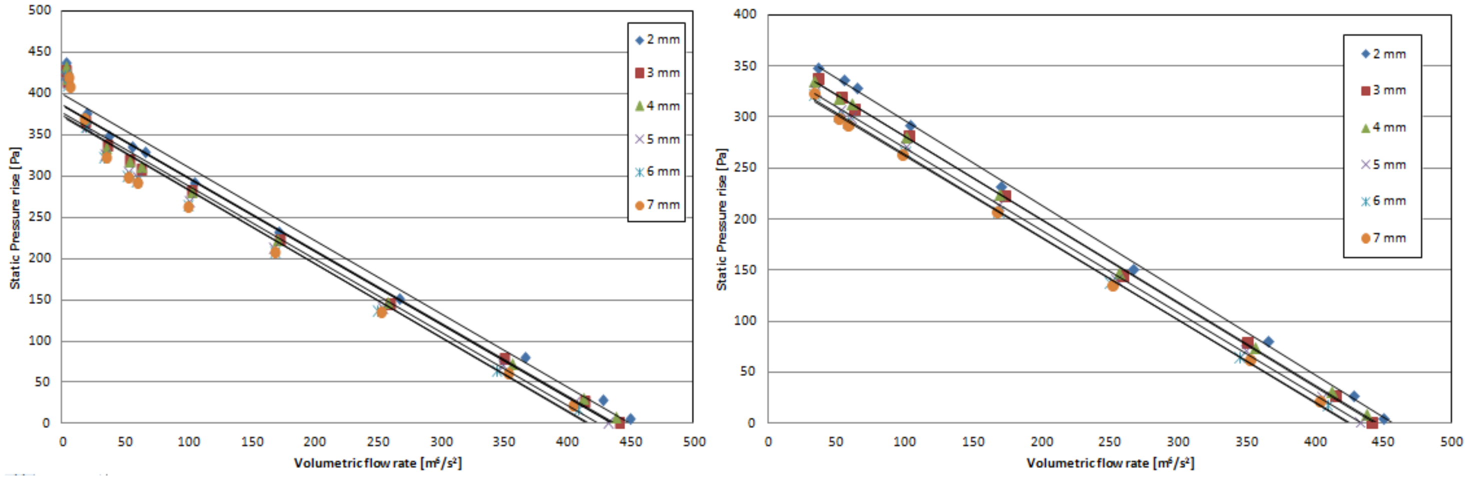



*Figure 6: Empirical curve fits for static pressures at flow rates above 6$m^3/s$ (L) and below 6$m^3/s$ (R)*

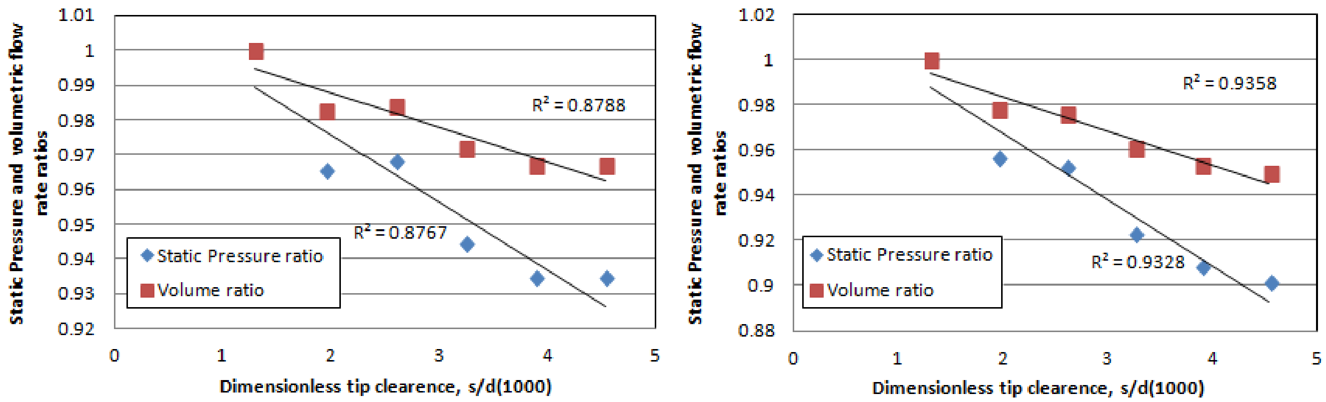



*Figure 7: Empirical model for the influence of tip clearance at flow rates above 6 $m^3/s$ (L) and below 6 $m^3/s$ (R)*

## ENDPLATE TESTS

Several basic endplates were tested at tip clearances of 6 mm, 5 mm and 4 mm to gauge the effect of these modifications. Inspiration for the use of endplates came from the aircraft industry where wingtip devices are regularly employed to reduce tip vortices and improve the lift generated by the wing. A similar effect is desired for the fan blades. Winglets used in the aircraft industry tend to be single or double sided, depending on the manufacturer, so it was decided to test both types in this research. Corsini, et al. showed that the use of a single sided endplate that protruded from the pressure side of the blade had some benefit in terms of fan performance[5,6,7]. Corsini however did not test double sided endplates that protruded on both the suction and pressure sides of the fan blades, thus it would be of interest to test if the double sided winglet has any added benefits in terms of fan performance.

The single sided and double sided endplates were manufactured in four different heights, a 10 mm, 20 mm, 30mm and 40 mm version. The heights were measured from the center of the leading edge of the blade, towards the pressure side for the singe sided endplate and in both directions on the double sided endplate. The endplates were manufactured from 0.9 mm thick stainless steel and were fixed in place on the blade tips of the test fan using the mounting points described earlier. All endplate test results are compared to the results of the fan running with an unmodified tip at a 2 mm tip clearance.

The first round of tests covered the single sided 10 mm (S10) endplate. The results of the tests are shown in the figure below.

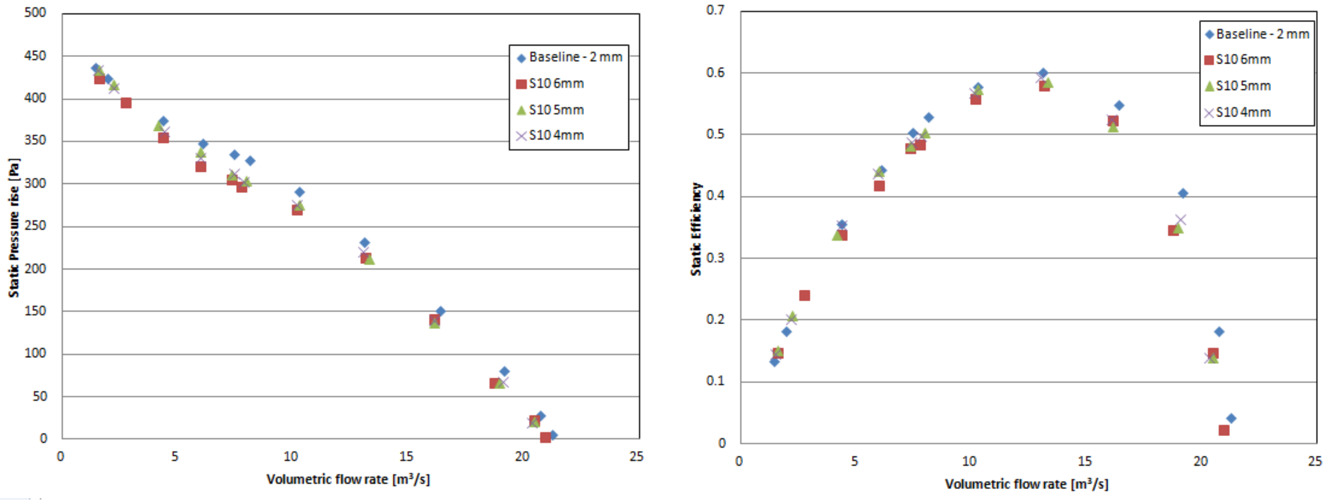



*Figure 8: S10 endplate fan static pressure (L) and fan static efficiency (R)*

The next tests were on the double sided 10 mm endplate, the results are shown below.

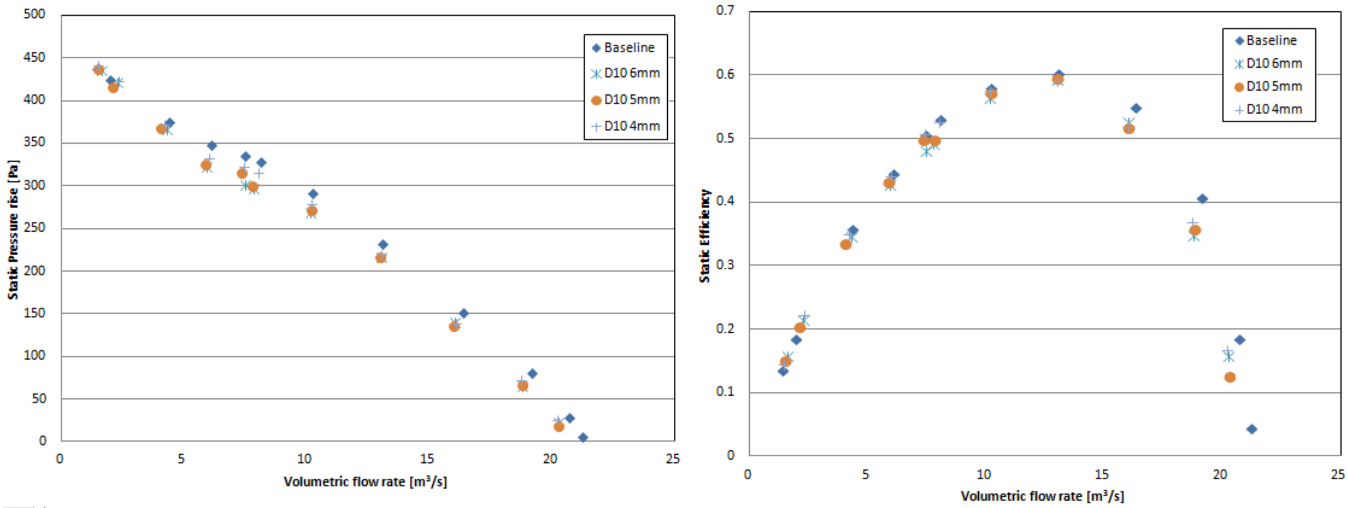



*Figure 9: D10 endplate fan static pressure (L) and fan static efficiency (R)*

These results showed no drastic difference compared to the single sided plates, so no further double sided plates were tested. The results for the single sided 20 mm (S20) endplate are shown below

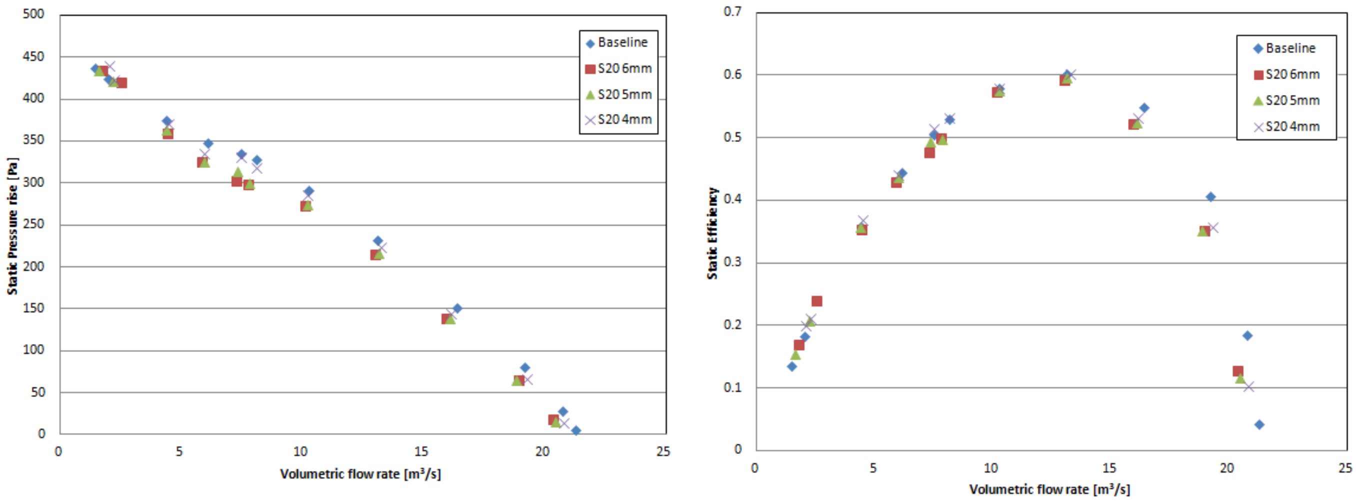



*Figure 10: S20 endplate fan static pressure (L) and fan static efficiency (R)*

Testing of the single sided 30 mm endplate was attempted, however it was found that the endplates touched the casing, even at the largest tip clearance possible (6 mm). This was due to the length that the endplates protruded from the pressure side of the blade, amplifying the effects of vibration and tip displacement in the blades resulting in the tips of the endplates scraping the casing. For this reason the test of the S30 and single sided 40 mm (S40) endplates were abandoned to prevent damage to the test facility. Instead the S20 endplate was modified by cutting the back third of it off as shown in figure 10, resulting in the S20mod endplate.

*Figure 11: S20mod endplate*

The results of the S20mod endplate tests are shown below in figure 11.

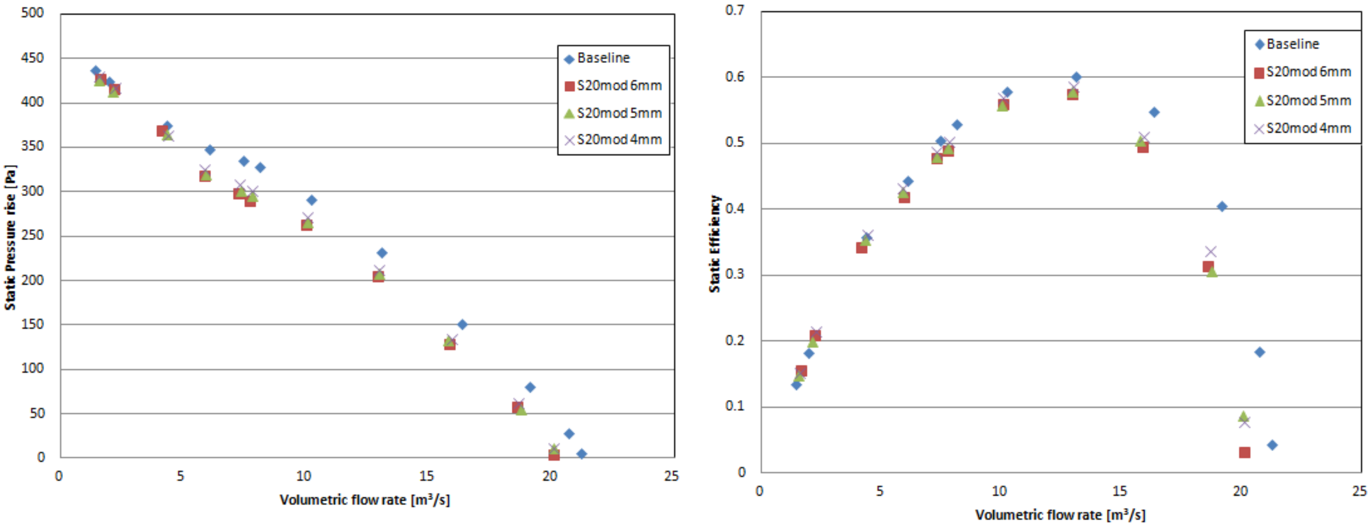



*Figure 12: S20mod fan static pressure (L) and fan static efficiency (R)*

# DISCUSSION

The tip clearance tests confirm that tip clearance has a profound impact on fan performance. Smaller tip clearances result in less tip leakage flow due to the smaller area available for the flow to leak through. This results in improved fan performance both in terms of fan static efficiency and fan static pressure rise, as can be seen in figures 4 and 5. It is also interesting to note that the performance of the fan is improved across all flow rates and that the fan is able to operate at slightly higher flow rates with a small tip clearance. The optimum tip clearance given by these tests was 2 mm, which resulted in a peak fan static efficiency of 60.3% as well as giving the best static pressure rise. This result is in line with the findings of Wallis and Kroger and Venter[3,7].

The application of Kroger and Venter's empirical model to the B2 fan tip clearance data has provided a good correlation as can be seen in figure 7. The correlation is significantly better at high flow rates above 6 $m^3/s$ with an $R^2$ value of 0.93 for both the pressure and volume flow ratios. This provides a good guideline for designers of fan systems when it comes to deciding a tip clearance for an axial fan. At low flow rates below 6 $m^3/s$ the correlation is not as strong, however the model is still more accurate than a lump model of all flow rates as the gradients of the linearized performance curves at low flow rates are significantly different to those at high flow rates, as seen in figure 6. This provides a reasonable guide for the design of low flow rate systems, which is of interest in the design of ACC fan systems. The entire piecewise model is especially powerful as it can be easily scaled using the fan laws and applied to a fan of any size.

The endplate tests show that the endplate modifications at the blade tip do have an effect on fan performance. The S10 endplate showed an improvement in performance at the 4 mm tip clearance. Although it did not reach the same efficiency levels as the clean tip running at 2 mm, it did improve on the clean tip running at a 4 mm tip clearance, reaching a peak static efficiency of 59.4%. This is due to the endplate blocking some of the tip leakage flow, improving performance in the same way that a reduction of tip clearance does. The D10 endplate also resulted in a slight increase in performance, however it did not perform as well as the single sided endplate. It can thus be concluded that the protrusion of the endplate on the suction side of the blade does not have much of an effect.

The S20 endplate was the top performing endplate with a peak static efficiency of 60.3% as seen in figure 10, the same as the clean tip at a 2 mm tip clearance. This is because the 20 mm endplate blocks more of the tip leakage flow, resulting in fewer loses and better fan performance. This is of interest as the fan achieved this at a tip clearance of 4 mm, significantly larger than the 2 mm clearance which the clean fan achieves this efficiency at. In large ACC systems (fan diameters of approximately 10m) it is often not practical to run fans at tight tip clearances as the shrouds have to be very precisely machined and the clearances accurately set, increasing costs. The endplates are a simple design that can easily be manufactured using inexpensive methods such as laser cutting, thus could providing a potential alternative for improving ACC performance without decreasing tip clearances.

It must however be noted that the fan static pressure rises achieved by the fan with endplates was always lower than that of the clean fan and that performance dropped off more sharply at high flow rates as can be seen in figures 9, 10, 11 and 12. The reduction in fan pressure rise indicates that the high efficiencies are attained because the endplates reduce the amount of shaft power required.

The S20mod endplate performed very poorly delivering low efficiencies and fan static pressure rises. This indicates that the location of the endplate over the trailing edge of the blade is important when it comes to the design of endplates. The end plate must extend across the entire pressure side of the blade in order to reduce leakage flow and improve performance.

The endplates used in this study were relatively crude with square corners and straight edges. However based on these results it is clear that they do have an effect and further research could result in better designed endplates that further improve fan performance.

## CONCLUSION

Tip clearance has a profound impact on fan performance with smaller tip clearances resulting in improved fan performance. The B2-fan running at a 2 mm tip clearance resulted in the best possible fan performance resulting in high fan static efficiencies and good static pressure rise as can be seen in figure 10. It also showed the best fan performance over all flow rates.

The S20 endplate attained a similar peak efficiency, however its static pressure rise capability was less than the clean fan at the 2 mm clearance. The performance of the fan also dropped off more steeply at high flow rates. However the use of the endplate allowed a high design point efficiency to be achieved at a larger tip clearance, which is of use in industry where large axial flow fans generally run with fairly crude tip clearances. It also has to be noted that the conclusions found in this paper are specifically for ACC fans and may not be valid for other industrial fans which have to reach much higher pressures and can be operated at much smaller tolerances.

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