PERFORMANCE TESTING OF A RETROFITTED ACC FAN

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SUMMARY

The MinWaterCSP project aims to reduce the water consumption of concentrating solar power (CSP) plants by 75 to 95% relative to wet cooling systems by introducing novel dry/wet cooling technology. As part of the project a large diameter axial flow fan was manufactured from a glass fibre reinforced polymer to demonstrate the ability of the consortium to manufacture and test an air-cooled condenser (ACC) fan. Once manufactured the 9.145 m fan was installed in the ACC of the Matimba power station for performance testing. The retrofitted fan was instrumented with strain gauges and its performance measured over a period of three days. Results showed that the retrofitted fan's shaft power was 140 kW for an average outlet air flow velocity of 3.5 m/s, indicating that it would be well suited as a full-scale ACC fan.

INTRODUCTION

Large scale industrial fans (typically ranging between 7 to 10 m in diameter) form an intrical part of air-cooled condensers (ACC). A complete ACC comprises of an array of ACC cells (typically between 1 to 400 fans) of which the forced draft configuration is depicted in Figure 1. In such condensers, steam that is exhausted from a turbine follows a sequence of ducts and is ultimately distributed into a series of heat exchanger bundles. Simultaneously, atmospheric air is drawn in by an axial fan and forced through the heat exchanger bundles, causing heat to be convected from the steam. The loss of heat causes the steam to condense and trickle into a collecting duct, where it is drained off.

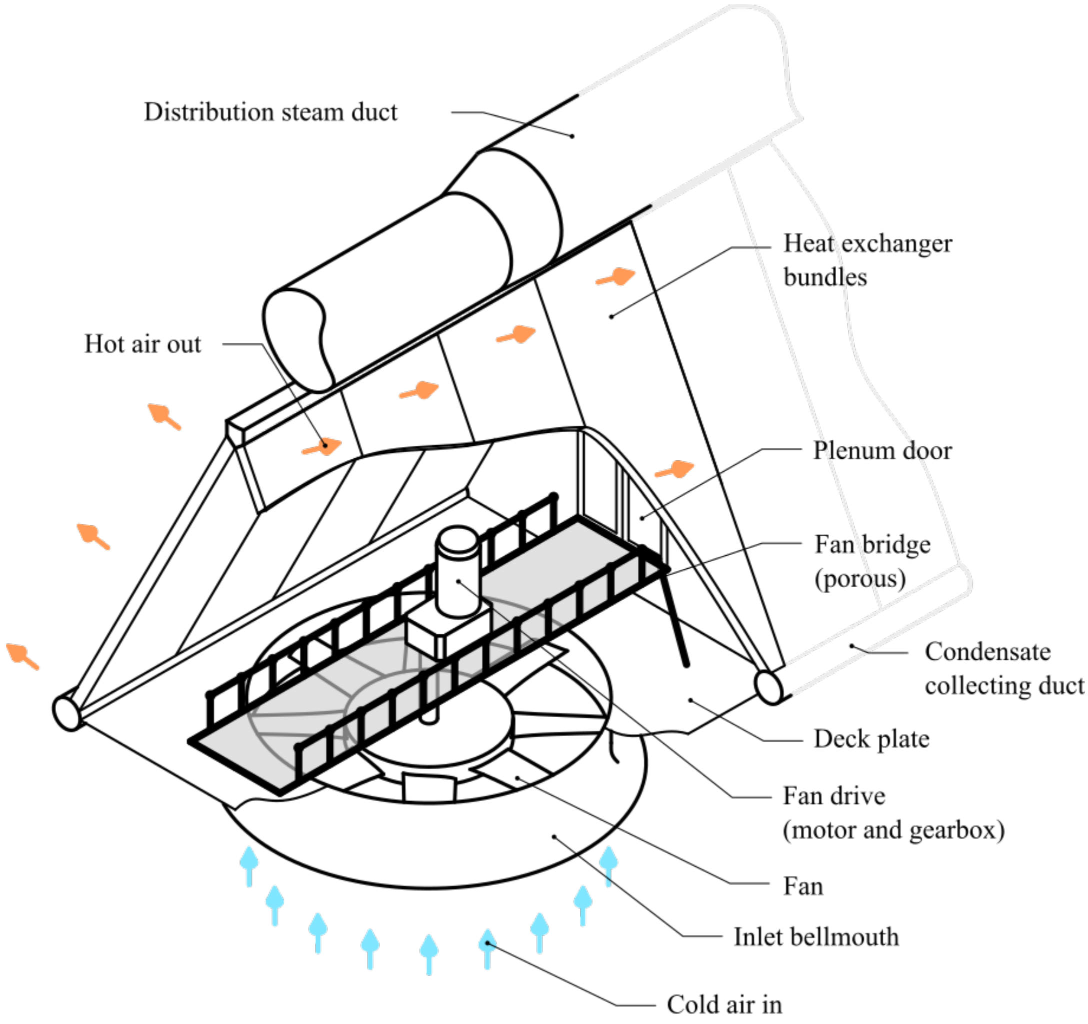


Figure 1 – A single typical ACC cell

In such systems, two areas have been identified for possible fan performance improvement:

Firstly aerodynamic fan performance characteristics, used during the design process of such systems, are normally selected from a catalogue of existing fans. Due to the relatively small market for large scale fans compared to smaller circulation fans (such as the heat, ventilation and cooling industry), the number of large fans available for selection is limited, often leaving the ACC system design engineer to specify fans that would operate under off-design conditions, causing the fan to operate at reduced efficiencies.

Secondly, due to the physical structure and operation of such a system, vibrations are inevitably excited. Blade vibrations too close to the natural system modes can cause increased fatigue rates as well as failure of blades and reduction gearboxes [1]. Since only a limited selection of fans are available, the vibrational characteristics of these fans remain uncontrolled and many times unknown to the ACC system design engineer. Thus, obtaining a system where the dynamic conditions during fan operation are favourable is often a result of coincidence.

Hence, it is foreseen that improvements are possible in axial fans both aerodynamically and structurally. Louw et al. [2], has shown that fan static efficiencies in the order of 60% can be achieved by conducting fan designs for given design conditions, being unique for every ACC system. With the focus on structures, Muiyser [1] has shown that typical frequency boundaries can be mapped, displaying favourable and troublesome fan operating zones for an ACC fan.

This is useful to the system design engineer to match the vibrational signature of the system with fans that would operate within frequency bands away from resonance frequencies (safe zones). Structural design improvements can be made by careful design of the blade composite lay-up, where different lay-ups would affect different blade resonance frequencies. In such a way the design can be manipulated to suit the safe operational zone for each ACC system.

One objective of the MinWaterCSP project was to show that a large diameter axial flow fan could be manufactured for installation in an ACC. The structural design of the fan blade was performed by one of the project partners and does not form part of this paper. The aerodynamic design of the fan is detailed by Louw et al [2]. Once designed, the fan was manufactured and retrofitted within an existing ACC for performance and structural testing. The purpose of this paper is to present the methodology and results of the tests that were conducted once the fan was installed to determine whether or not the retrofitted fan is fit for purpose.

INSTALLATION OF THE TEST FAN

To retrofitted fan, which will be referred to as the R-fan, was installed in unit 3 of Eskom's Matimba plant ACC. The location of the R-fan was determined by maintenance activities at the power plant and was close to the middle of the ACC, as shown in Figure 2. This location meant that the R-fan would be less exposed to the effects of wind than the fans situated around the edges of the ACC. High wind speeds typically result in reduced performance as well as increased dynamic blade loading due to the distorted inlet air flow conditions at the fan inlet [3]. In addition to the location of the R-fan, the adjacent comparison fan is also indicated in Figure 2. This fan was also instrumented to provide a relative indication of the R-fan's performance.

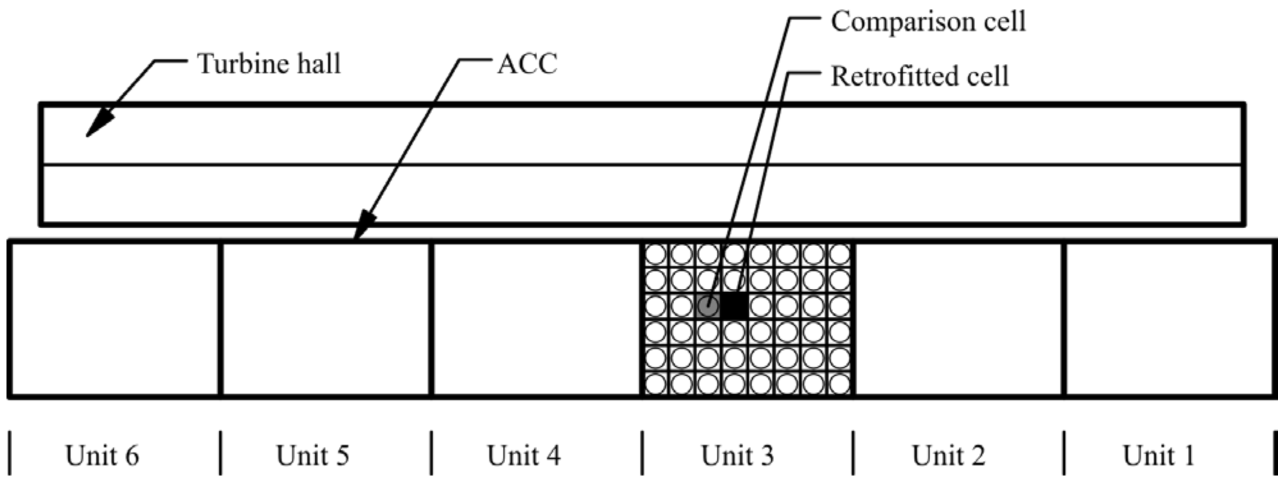


Figure 2 – Installed position of the R-fan in the Matimba ACC

The most noticeable difference between the R-fan and the fans currently installed in the Matimba ACC is the fact that the R-fan has a much larger hub-to-tip ratio. This meant that for the R-fan to be attached to the existing gearbox shaft, it needed to be installed with an additional aerodynamic hub that provided no structural support. In addition, the blades each needed to have a stem that extends past the aerodynamic surface of the blade. The installation of the R-fan within the ACC can be seen in Figure 3. The photo on the left was taken from below the ACC and shows the increased hub size of the R-fan when compared to the surrounding fans while the photo on the right shows the implementation of the aerodynamic hub with the long stem lengths of the R-fan blades.

(a) *(b)*

Figure 3 – The installed R-fan as seen from below the ACC (a) and from the deck in the plenum chamber (b)

INSTRUMENTATION OF THE TEST FAN

To determine the performance characteristics and structural response of the fan and blades, respectively, a large number of measurements were recorded over a period of three days. The measurements recorded on the R-fan are the same as those recorded by Muiyser et al. [4]. A summary of the sensors installed at the R-fan and the comparison cell are depicted in Figure 4:

1. The outlet air flow velocity was measured at four locations for the R-fan and comparison cell (V_R and V_C, respectively) using R.M. Young 27106T propeller anemometers. These anemometers were installed at the same side of each cell, as shown in Figure 4, to avoid measurement bias due to the prevailing wind direction. Alternatively, the anemometers may also be installed at the fan inlet. However, the work conducted by Muiyser et al. [4] has shown that the air flow exiting the heat exchanger bundles is better directed than the flow at the fan inlet. A single HBM QuantumX MX1601 was used to record these anemometer readings.
2. The current drawn by the three-phase motor driving the R-fan and comparison cell (P_R and P_C, respectively) was reported by the plant operator for the measurement period.
3. The rotational speed of the R-fan, N_R, was measured using an inductive proximity sensor installed in such a way that a single pulse would be generated each time the instrumented fan blade passes the downwind side of the fan bridge.
4. The torque of the reference fan, T_R, was measured at the gearbox output shaft using strain gauges in a full-bridge configuration.
5. The bending strain in the shaft of the fan blade, ε_R, was measured in two directions using strain gauges. A full-bridge configuration was used to compensate for temperature effects as well as axial loads introduced through centrifugal effects. These strain gauges and those attached to the fan shaft were connected to a LORD MicroStrain V-Link wireless bridge amplifier for data acquisition. The attachment of the strain gauges to the R-fan blade shaft is shown in Figure 5. The gauges were oriented to measure bending strain in the orthogonal flap- and lag-wise directions. Once installed the strain gauges were calibrated with a known bending moment to determine the relationship between applied load and measured strain.

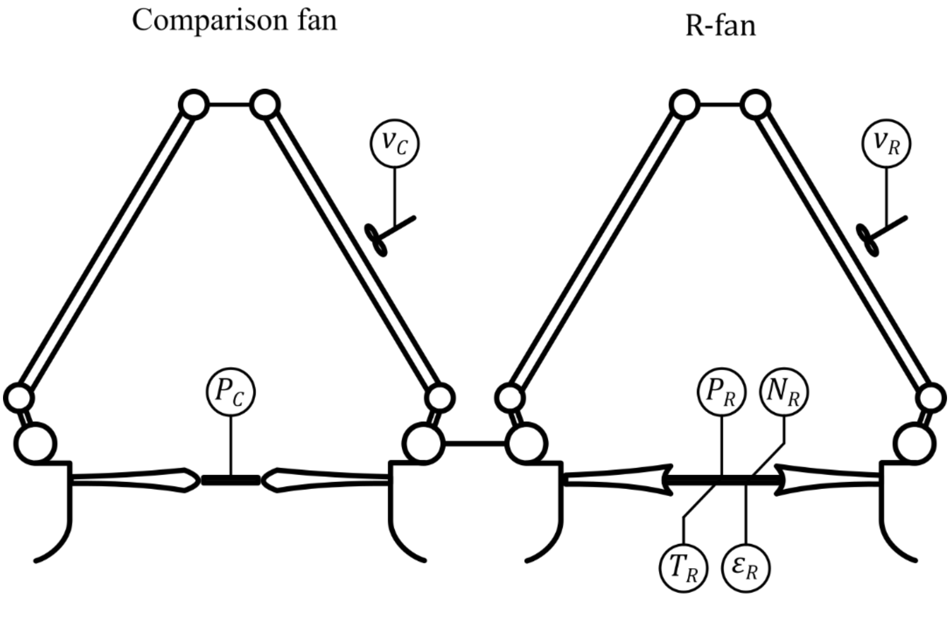
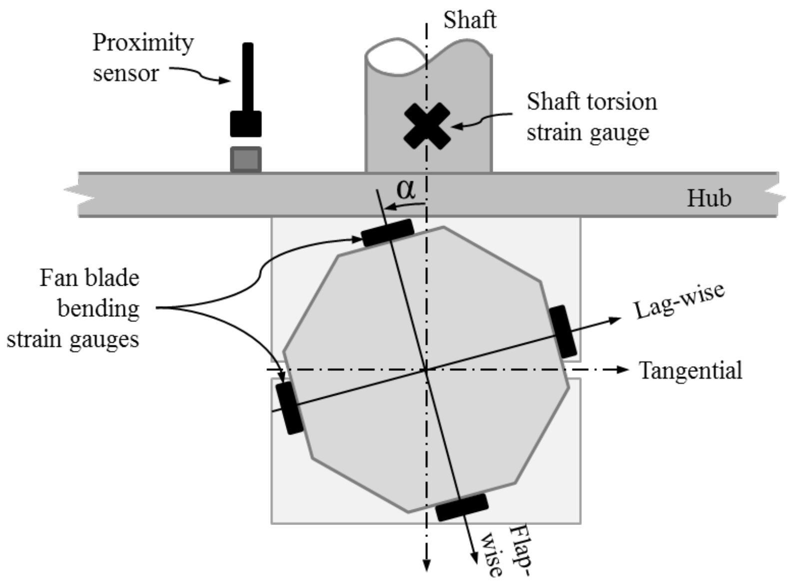


Figure 4 – Summary of the measurements recorded at the R-fan and adjacent comparison cell



(a) *(b)*

Figure 5 – Diagrammatic representation of strain gauge installation (a) and photo of the strain gauges attached to the fan blade (b).

FAN PERFORMANCE

Tests were conducted over a period of three days in October and November of 2016. Figure 6 shows the weather conditions for the test period. As there was no on-site weather mast the data had to be obtained online [5]. The times when measurements were recorded are highlighted in the figure and indicate that the wind speed was lower on 2016-10-27 than on the other days. Additionally, it can be seen that the wind direction was mostly North-Westerly across all of the days that measurements were recorded and that the air temperature was higher on 2016-10-27 and 2016-11-01 than on 2016-11-02, which resulted in differing air densities.

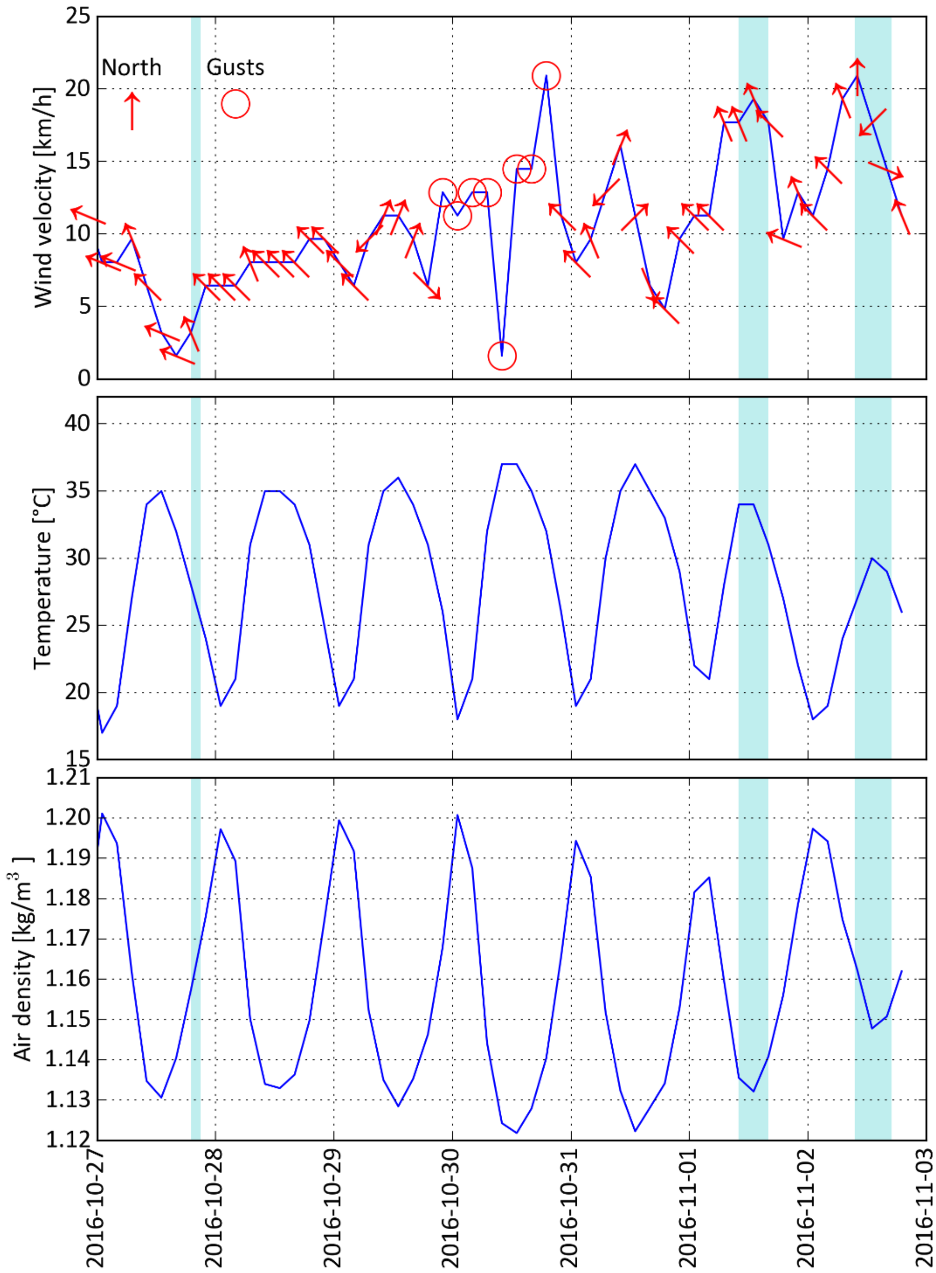


Figure 6 – Summary of weather conditions for the measurement period

The average bundle outlet air flow velocities of the retrofitted and comparison cells on 2016-11-01 and 2016-11-02 are provided in table 1. On each of these days the testing was conducted for two periods, denoted by (a) and (b). The results show that the R-fan is providing a flow rate that is at most 3.8% lower than that of the comparison cell. However, this flow rate was achieved at the initial blade angle, which, as shown in Figure 7, resulted in the fan drawing approximately 10% less current than the comparison fan. As such, the blade angle of the R-fan could be increased to achieve the same bundle outlet air flow velocity as the comparison fan.

Table 1: Measured averaged outlet velocities

Day		**Bundle outlet velocity**	
		Reference cell [m/s]	Comparison cell [m/s]
2016-11-01	(a)	3.50	3.52
	(b)	3.36	3.45
2016-11-02	(a)	3.54	3.51
	(b)	3.57	3.71

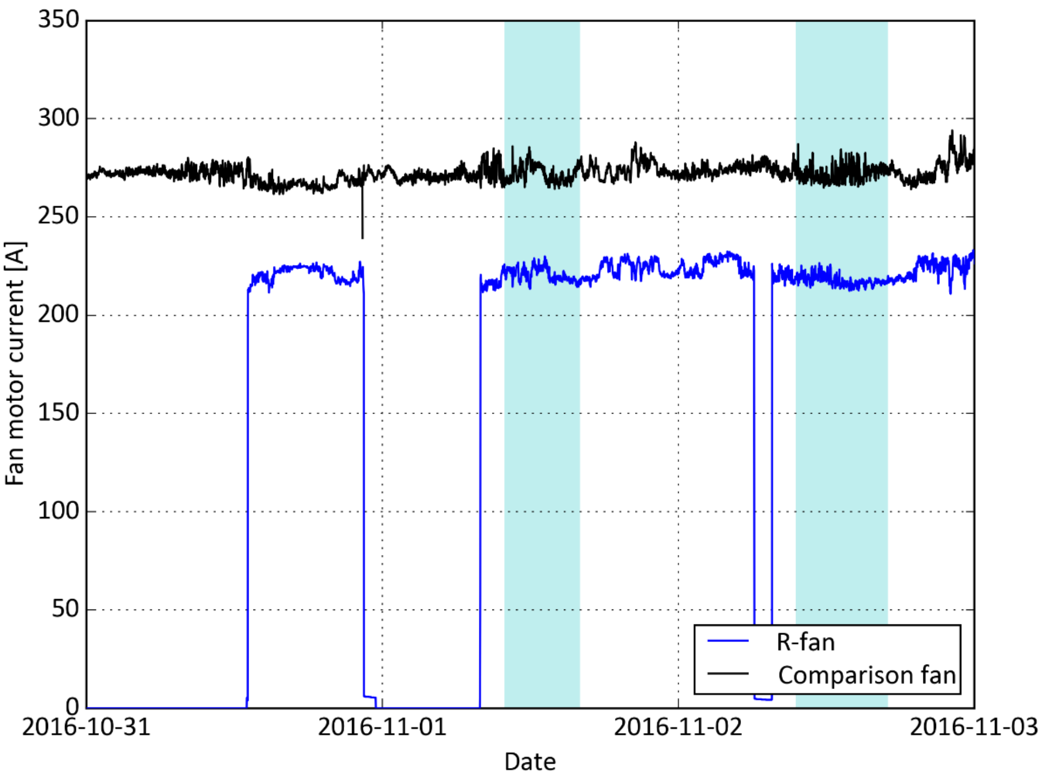
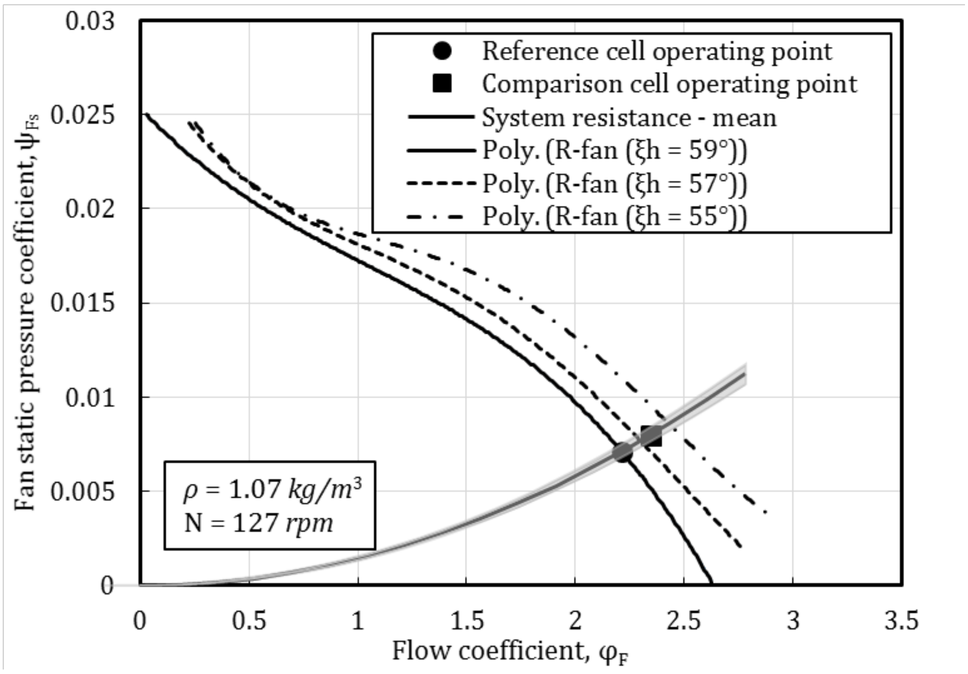


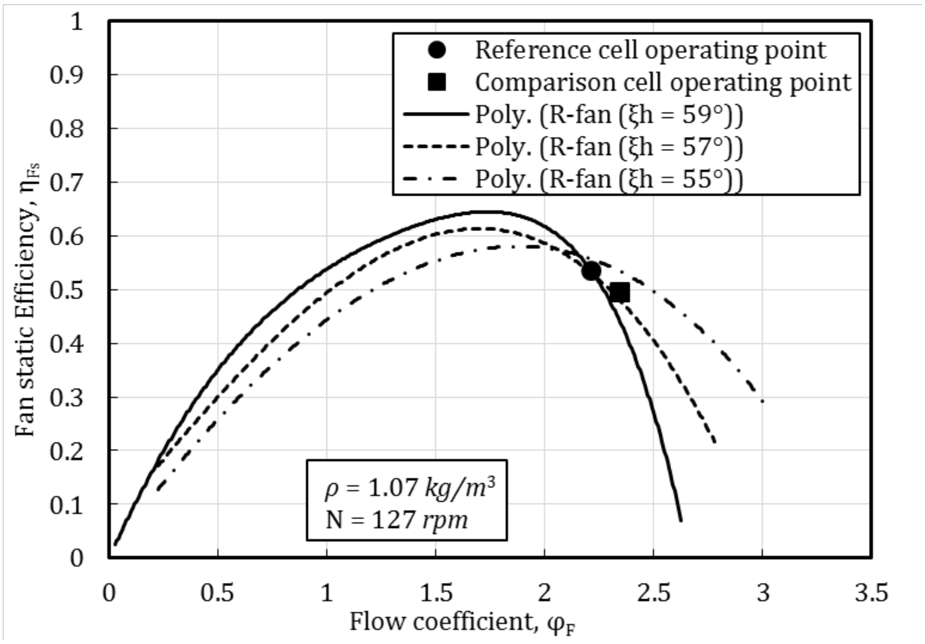
Figure 7 – Amperages for the R-fan and the comparison fan

A further aerodynamic comparison is shown in Figure 8, where the higher operating point (higher flow and fan static pressure) of the Comparison cell relative to the Reference cell can be observed. The displayed R-fan performance curves for different blade angles were obtained through experimental testing at scale level (1.542 m diameter fans) in an ISO 5801 test facility [6]. From Figure 8b, the high efficiency capability of the R-fan is displayed in the fact that the blade angle can be adjusted by approximately 3° from the design root stagger angle (ξ_h = 59°) to match the

Comparison cell's performance (where the fan consumed approximately 206 kW), while only consuming 191 kW (7 % less power). This illustrates the possibility to obtain a higher performance when the fan is purposefully designed for the ACC operating point. Moving to lower flow rates, the static efficiency of the R-fan reaches a maximum of 64 %, showing that a further efficiency improvement is attainable, keeping the fan motors from tripping under adverse conditions such as high windy periods, when flow through an ACC tends to reduce.



(a)



(b)

Figure 8 – Performance characteristics comparing different blade angles of the R-fan performances for (a) fan static pressure rise and (b) fan static efficiency

SHAFT TORQUE AND FAN BLADE LOADING

The fan shaft torque as well as the blade loading that was measured is presented in table 2. In addition to the average values the RMS of the dynamic component of the data is also shown. As shown in Figure 6, the wind speed was higher on 2016-11-01 and 2016-11-02 than on 2016-10-27. This resulted in higher dynamic blade loading in the flap- and lag-wise directions due to increased levels of distorted inlet air flow. There is a similar increase in the dynamic component of shaft torque while there was no discernible trend in the average values of the measurements.

The dependence of dynamic blade loading on wind speed, shown in table 2, is also illustrated in Figure 8. In this figure the bending load measured in the flap-wise direction of the blade is presented over a 10 minute period for 2016-10-27 and 2016-11-01. The blade loading measured on 2016-10-01 has a higher amplitude due to the increased wind speed.

Table 2: Measured operational R-fan input shaft and blade loads

		Shaft Torsion†		**Shaft Power**	**Flap-wise Bending**		**Lag-wise Bending**	
		[kN.m]		[kW]	[N.m]		[N.m]	
		Avg	RMS	Avg	Avg	RMS	Avg	RMS
2016-10-27	(a)	11.05	0.213	146.97	3526	312	918	157
	(b)	10.83	0.213	144.03	3529	303	869	129
2016-11-01	(a)	10.54	0.289	140.26	3849	509	872	299
	(b)	10.94	0.380	145.48	4052	494	946	270
2016-11-02	(a)	10.76	0.304	143.11	3428	389	893	247
	(b)	10.94	0.266	145.56	3891	463	902	262

† Shaft torsion calculated with E = 207 GPa

As shown by Muiyser et al [1], the amplitude of blade vibration is elevated when the natural frequency of the fan blade is equal to a harmonic of the fan’s rotational speed. As such, a bump test was performed once the fan was installed to determine the first natural frequency of the fan blades. The bump test showed that the first natural frequency of the fan blades was equal to 4.9 Hz, which is not equal to any harmonic of the 2.1 Hz rotational speed of the fan.

Figure 9 shows fast Fourier transforms (FFTs) of the same measurements presented in Figure 8. As expected, the measurements recorded on 2016-11-01 exhibit a much larger peak at the fan rotational speed of 2.1 Hz due to higher inlet air flow distortion as a result of the increased wind speed. Additionally, it is important to note that the natural frequency of the fan blade at 4.9 Hz is not excited by the harmonics of the fan’s rotational speed.

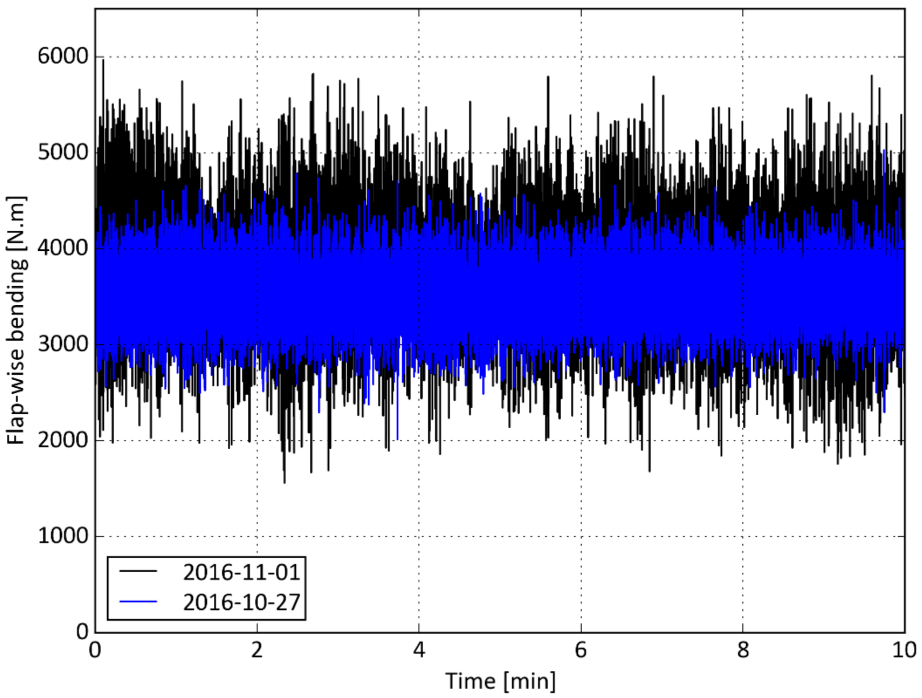


Figure 8 – R-fan flap-wise bending measured on 2016-10-27 (top) and 2016-11-01(bottom)

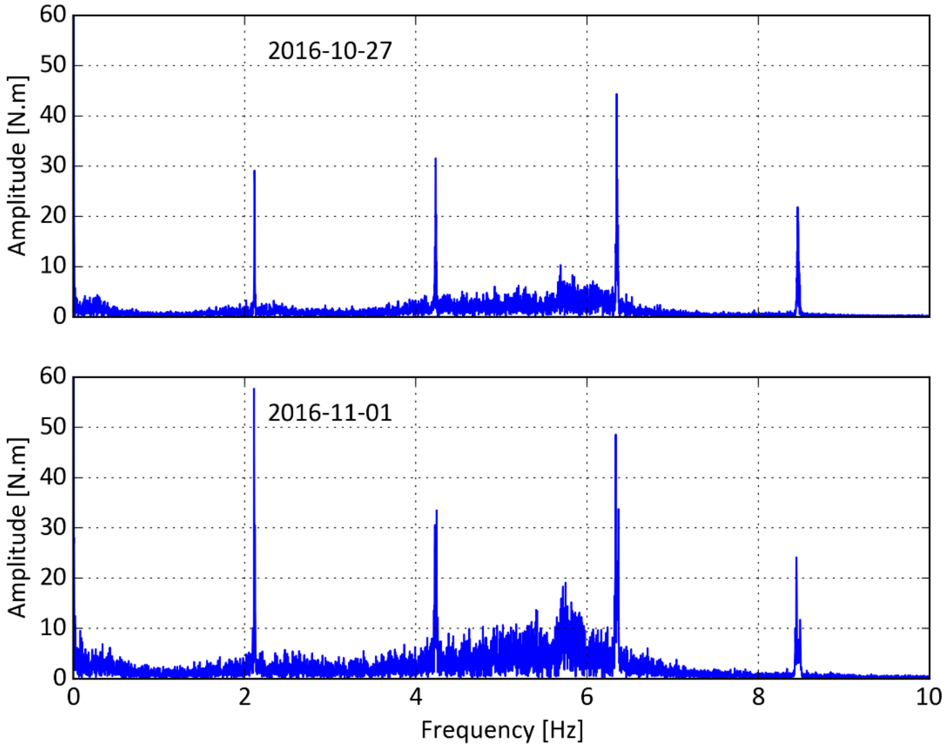


Figure 9 – FFTs of the flap-wise bending measured on 2016-10-27 (top) and 2016-11-01(bottom)

CONCLUSION

A large axial flow fan was manufactured for installation in an ACC as part of the MinWaterCSP project. The fan was installed in the Matimba ACC where subsequent performance testing was conducted. These tests showed that the fan provided approximately 4% lower air flow than the currently installed fans, but drew 10% less electric motor current. Comparing the same operating points, the R-fan proved to consume 7 % less power than the Comparison case for the same volumetric air flow displacement, displaying the possible advantage of constructing a customized fan design for a specific ACC operating point compared to typical multi-purpose designs. Additionally, strain gauge measurements recorded from a fan blade showed that none of the blade's natural frequencies are excited during operation. As such, it has been shown that the retrofitted fan is fit for purpose.

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