QUANTIFYING THE PERFORMANCE OF A TOP-DOWN NATURAL VENTILATION WINDCATCHER

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ABSTRACT

Measurements and smoke tests show that the quadrants of a Windcatcher with a positive pressure across them act as supply ducts, while those with a negative pressure across them act as exhaust ducts. However, analysis of the side and leeward C_p values shows that they do not necessarily balance mass flow in and out of the Windcatcher, indicating that either the pressure in the supplied room drops or there is an amount of infiltration through the building fabric initiated by the Windcatcher.

In order to better understand Windcatcher performance, a simple analytic model is developed that utilises experimental data to estimate the losses in the system. Two different scenarios are considered for the room adjoining the Windcatcher: (i) this room is perfectly sealed; and (ii) air infiltration is allowed into the room so that the pressure in the room remains atmospheric. Here, it is observed that, for those values of C_p reported for a square Windcatcher in the literature, the overall volume flow rate of air out of the room always exceeds that coming into the room. Based on this data, the analytic model may be used to estimate the losses in the Windcatcher, from which it is then straightforward to derive a simple relationship between the overall area of the Windcatcher and the volume flow rates into and out of the Windcatcher in order to predict Windcatcher performance for a given application.

1. INTRODUCTION

The key to correctly sizing natural ventilation elements is the ability to predict flow rates through them. Current literature (CIBSE 2005) provides simple explicit calculation methods for envelope flow models that calculate flow rates for all natural ventilation principles using the orifice equation with a coefficient of discharge that is fixed by the shape of each opening (Etheridge 2004).

The modern Windcatcher natural ventilation element is relatively complex and consists of many parts that may vary according to its application. Therefore, to maintain flexibility, a derived figure for the total system losses expressed as the sum of the losses in individual sections is more useful than using a single discharge coefficient for the Windcatcher.

Both approaches are dependant upon knowledge of the external pressure distribution around the ventilation element which usually originates from empirical testing. Empirical and theoretical studies have been conducted (Elmualim et al. 2002; Elmualim 2006; Parker et al. 2004) to determine the average coefficient of pressure (C_p) on each Windcatcher face, with respect to wind direction, wind speed and duct volumetric flow rates. A further empirical study (Parker et al. 2004) has determined the pressure drop across the upper louvered section with respect to duct velocity. This data is important because it can be used to demonstrate system performance with respect to wind speed and direction, and so give an indication of system losses. The calculation of total system losses is fundamentally important to the successful prediction of volumetric flow rates through the Windcatcher. Thus, empirical data can be used to calculate a value for an overall system loss coefficient for the Windcatcher which, when combined with empirically derived values for C_p, can be used to predict flow rates in the Windcatcher ducts for different wind velocities.

This paper will review empirically and theoretically based literature that quantifies the performance of Windcatchers. A model is then proposed that determines the total losses in the Windcatcher system with the aim of predicting volumetric flow rates for a wide range of Windcatcher geometries. Finally, an equation to calculate the cross sectional area of a Windcatcher duct required to supply a specified volumetric flow rates is developed.

2. PREVIOUS RESEARCH

Wind tunnel test data exists for a Windcatcher and CFD calculations have also been generated in order to corroborate empirical findings. Two tests have been undertaken: the first by Elmualim (Elmualim et al. 2002) used two pressure tappings on the centre line of each face and one on the centre line of the trunk, while the second by Parker (Parker et al. 2004) used nine pressure tappings on each face that covered the centre line and each of the vertical edges. Both tests measured a 500mm square Windcatcher located in an open wind tunnel and Elmualim's Windcatcher (Elmualim et al. 2002) supplied air to a small sealed room with a volume of 15.25m^3. These tests provide data for C_p values on each face of the Windcatcher, the flow rates through the Windcatcher's ducts relative to a wind speed (u_w), and an indication of total losses. Both tests agree on two generalities: firstly, Windcatcher quadrants that have a positive pressure across them act as supply ducts, while those with a negative pressure act as exhausts. This is also confirmed by observation using smoke and by measurement (Elmualim et al. 2002).

The C_p on a Windcatcher face is calculated by:

$$C_p = \frac{\Delta p}{\frac{1}{2} \rho u_w^2} \qquad (1)$$

where ρ is the density of the air, Δp is the difference between the static pressures on the face and in the free stream and u_w is the wind velocity. The CFD results of Elmualim (Elmualim 2006) confirm the findings of the two wind tunnel studies (Elmualim et al. 2002; Parker et al. 2004) and provide a comparison of C_p values (table 1) with the wind direction perpendicular to a single Windcatcher face (defined as α=0°). This indicates that the measurements and predictions are reasonably consistent, especially for the windward side, but with the greatest discrepancy for the side and leeward faces.

Losses through the elements of the Windcatcher system were also investigated by Parker (Parker et al. 2004) who used a fan to suck/blow air through the louvered and duct sections of the Windcatcher in order to calculate the losses. Parker plotted the pressure drop (Δp) against duct volume flow rate ($\dot{Q}$) and later used this data when estimating the volumetric flow rates delivered by the Windcatcher. It is, however, far more useful to express these losses in terms of a loss factor (K) which may be written as

$$\Delta p = \frac{1}{2} \rho u^2 K \qquad (2)$$

where u is the duct velocity.

Table 1: C_p values for a Windcatcher with α=0°

Windcatcher Face	Wind Tunnel (Parker et al. 2004)	Wind Tunnel (Elmualim et al. 2002)	CFD (Elmualim 2006)
Windward	0.853	0.830	0.840
Side	-0.348	-0.330	-0.550
Leeward	-0.116	-0.100	-0.440

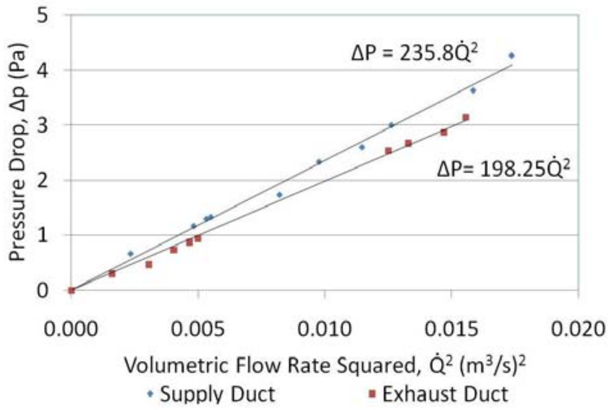

Figure 1: Losses in the Top Section of a Windcatcher (Parker et al. 2004)

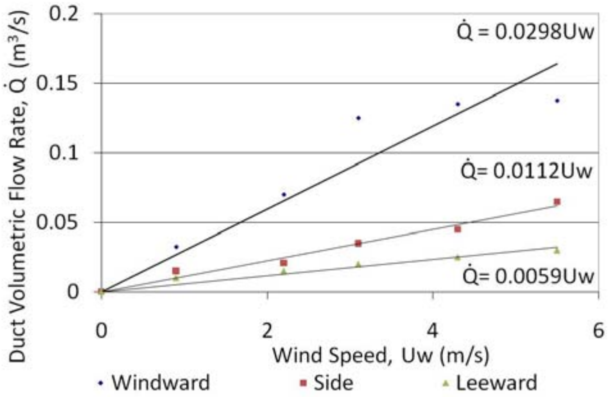

Figure 2: Duct Volumetric Flow Rates with Relation to u_w (Elmualim et al. 2002)

Values for K may be extrapolated from Parker's data by plotting Δp against $\dot{Q}^2$ (see Figure 1) Here,

$$\Delta p = \frac{\rho K}{2A^2}\dot{Q}^2\,, \tag{3}$$

where A is the cross sectional area of a duct. After curve fitting Parker's data (assuming that each line goes through the origin) values of K = 1.5 and K = 1.32 are obtained for the flow into and out of the Windcatcher respectively. This indicates that for the upper section of the Windcatcher losses are greater in the supply duct.

Elmualim (Elmualim et al. 2002) measured $\dot{Q}$ with respect to u_w (see Figure 2) for the louvered and duct sections of a Windcatcher in a wind tunnel and found that mass did not necessarily balance through the Windcatcher system. This is demonstrated by comparing the sums of the supply and extract gradients in figure 2 which shows that the difference is $\dot{Q} = 0.0015u_w$. Thus, either the pressure in the room changes or, perhaps more likely, there is an amount of infiltration through the building fabric of the supplied room.

3. THEORETICAL MODEL

A theoretical model is developed here in order to quantify the ventilation performance of a Windcatcher. Empirical evidence shows that mass through a Windcatcher system does not necessarily balance, see Figure 2. Therefore, if the supply room is sealed then the pressure in the room must change in order to maintain mass continuity; if the supply room is not sealed then one may expect air infiltration and the pressure in the room to remain at atmospheric pressure. Both sealed and unsealed rooms are modelled by first specifying a control volume (see the dashed line in figure 3), which represents the limits over which the governing equations are applied. Here, the control volume extends sufficiently to ensure that u_i=u_w and u_r=0 assuming that u_w is perpendicular to a single face so α=0°. The energy equation is then applied assuming that the flow is steady and incompressible. Height differences are ignored and p_i=0 (gauge). Flow is assumed to enter though quadrant 1 and exhaust through quadrants 2, 3 and 4. Quadrants 2 and 3 are assumed to be identical so that $\dot{Q}_2 = \dot{Q}_3$ since C_{p2}=C_{p3}, and u_2=u_3.

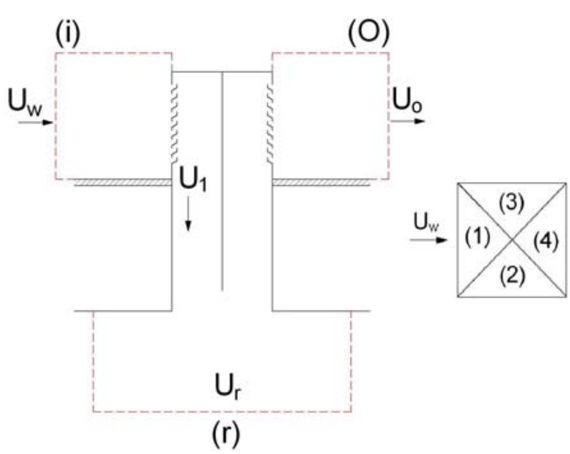

Figure 3: Windcatcher with Control Volume and Plan

3.1 Sealed Room

To begin with, it is assumed that the supply room is sealed and so application of the energy equation (Munson et al. 2004) between points *i* and *r* gives:

$$\frac{1}{2}\rho u_w^2 - p_r = \frac{1}{2}\rho u_1^2 K_1 \quad (4)$$

where u_1 is the flow velocity in duct 1. Application of the energy equation between points *r* and *O* gives

$$p_r - \frac{1}{2}\rho u_w^2 C_{p_o} = \frac{1}{2}\rho u_o^2 K_o + \frac{1}{2}\rho u_o^2 \quad (5)$$

Because u_2=u_3 and C_{p2}=C_{p3}, in general we have

$$p_r = \frac{1}{2}\rho u_1^2 K_1 - \frac{1}{2}\rho u_w^2 \quad (6)$$

$$p_r - \frac{1}{2}\rho u_w^2 C_{p2} = \frac{1}{2}\rho u_2^2 K_2 \quad (7)$$

$$p_r - \frac{1}{2}\rho u_w^2 C_{p4} = \frac{1}{2}\rho u_4^2 K_4 \quad (8)$$

Finally, mass continuity gives

$$\dot{Q}_1 = \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 . \quad (9)$$

Equations (6)–(9) may be solved simultaneously for velocities u_1, u_2, and u_4, and for the pressure p_r, provided that one knows the values for K. It is convenient here to combine equations (6), (7) and (9) to give,

$$u_4^2 + 4\left(\frac{A_2}{A_4}\right)u_2 u_4 + a_2 u_2^2 - b_2 = 0 \quad (10)$$

and equations (6), (7) and (8) to give,

$$u_4^2 + \left(\frac{A_2}{A_4}\right)u_2 u_4 + a_4 u_4^2 - b_4 = 0 . \quad (11)$$

Here

$$a_2 = 4\left(\frac{A_2}{A_3}\right)^2 - \frac{K_2}{K_1}\left(\frac{A_1}{A_4}\right)^2 \quad (12)$$

$$b_2 = \left(\frac{A_1}{A_4}\right)^2 \left(\frac{1-C_{p2}}{K_1}\right)u_w^2 \quad (13)$$

$$a_4 = \frac{1}{4}\left[\left(\frac{A_4}{A_2}\right)^2 - \frac{K_4}{K_1}\left(\frac{A_1}{A_2}\right)^2\right] \quad (14)$$

$$b_4 = \left(\frac{A_1}{A_2}\right)^2 \left(\frac{1-C_{p4}}{4K_1}\right)u_w^2 \quad (15)$$

Equations (10) and (11) are then solved for u_2 and u_4. using the Newton Raphson method It is then straightforward to substitute these values back into equations (9) and (6) to give values for u_1 and p_r respectively.

3.2 Unsealed Room

If infiltration, $\dot{Q}_r$ is permitted in the supply room and this is assumed to make up the mass short fall (so that p_r =0), then equations (6) to (9) simplify to give

$$\frac{1}{2}\rho u_w^2 = \frac{1}{2}\rho u_1^2 K_1 \quad (16)$$

$$-\frac{1}{2}\rho u_w^2 C_{p2} = \frac{1}{2}\rho u_2^2 K_2 \quad (17)$$

$$-\frac{1}{2}\rho u_w^2 C_{p4} = \frac{1}{2}\rho u_4^2 K_4 \quad (18)$$

$$\dot{Q}_r = 2u_2 A_2 + u_4 A_4 - u_1 A_1 \quad (19)$$

Table 2: Pressure Loss Factors, *K*

Component\Duct	Supply	Exhaust
Upper louvered section	1.50	1.32
Inlet	1.00	1.00
Outlet	1.00	1.00
Frictional Loss	0.06L/d_H	0.06L/d_H
Transition Loss	0.46	2.24
Total	**3.96+0.06L/d_H**	**5.56+0.06L/d_H**

So now it is easy to write

$$\dot{Q}_1 = \frac{u_w A_1}{\sqrt{K_1}} \quad (20)$$

$$\dot{Q}_2 = u_w A_2 \sqrt{\frac{-C_{p2}}{K_2}} \quad (21)$$

$$\dot{Q}_4 = u_w A_2 \sqrt{\frac{-C_{p4}}{K_4}} \quad (22)$$

Using these equations it is now simple to obtain quick estimates of duct volumetric flow rates with relation to u_w.

4. RESULTS AND DISCUSSION

The analytic model is now used to estimate the velocity of air in the Windcatcher. Before doing this we must first estimate values for K used in the equations above. It is assumed here that K is constant regardless of wind speed/direction and that different Windcatcher designs may be accommodated by breaking down K into individual components. Accordingly, values of K for individual elements of a Windcatcher are estimated from empirical data reported by from Parker (2004) and from standard values (see CIBSE 2001). The K values estimated for different components of the Windcatcher are reported in Table 2, which also includes an estimation of the frictional losses imparted by the duct walls. Here, it is difficult to quantify the frictional losses accurately because the empirical data indicates that flow patterns are likely either to be transitional or in early turbulence.

The sum of the losses estimated using Parker's data (Parker et al. 2004) and standard values gives, for a 500mm square Windcatcher with a 1m duct, a value of K_{supply}=3.79 and $K_{extract}$=3.61. These values can then be compared with losses measured for the whole Windcatcher system by applying them to the sealed and unsealed models to give duct flow rates with respect to u_w that may be compared against Elumalim's data measured in a wind tunnel (see Figure 2). This method shows the initial K values to be too low and so by iterating to produce flow rates that closely match Elmualim's data, the total system losses are revised to be K_{supply}=4.25 and $K_{extract}$=5.85 for both the sealed and unsealed room scenarios. Clearly these values are higher than those initially estimated and show that losses are greater in the extract duct. We believe that this discrepancy is caused by the flow regime in the actual Windcatcher (transition or very early turbulence) being different from that assumed in standard texts (fully developed turbulence). Accordingly, an extra loss factor is added here to compensate for this discrepancy and is included in Table 2 as a "Transition Loss". For each model/data source and duct, expressions for $\dot{Q}_{duct}$ with respect to u_w are presented in table 3. They show that the K values have been generated to give the best agreement of flow rates for the windward ducts and that it has been difficult to get good agreement for the extract ducts. These differences may be explained by empirical measurement inaccuracies. Infiltration for the unsealed model is found to be $\dot{Q}_r$=0.0089u_w with the flow into the room, and is the equivalent of the flow rate through the leeward duct. For the sealed room model, the pressure change is found to be $p_r = -0.555u_w^2$ showing that the pressure in the supply room falls in order to balance mass through the system.

The losses presented here are based upon a Windcatcher functioning under ideal conditions and are consistent for the sealed and unsealed models and must be considered our best possible estimate. Their calculation uses Parker's C_p values that have been corroborated by two other studies and so should be considered reliable. The results for the unsealed room show that for Parker's C_p values, the overall volume flow rate of air out of the supplied room always exceeds that coming in. Therefore, either the pressure in the room drops or there is an amount of infiltration through the building fabric initiated by the Windcatcher. With this knowledge it is now possible to use the models to estimate duct volumetric flow rates to be compared against other data in order to test the robustness of the models and the K values. Furthermore, the unsealed model may readily be used to estimate the duct area required to provide a known flow rate $\dot{Q}_{out}$ where

$$\dot{Q}_{out} = 2\dot{Q}_2 + \dot{Q}_4 \quad (23)$$

Thus for a square Windcatcher in which each segment has the same cross sectional area, A, an expression may be derived from equations (20)-(22) to give

$$A = \frac{\dot{Q}_{out}}{u_w \left\{ 2\sqrt{\frac{-C_{p2}}{K_2}} + \sqrt{\frac{-C_{p4}}{K_4}} \right\}} \quad (24)$$

This explicit and flexible approach enables quick and easy calculation of the required cross sectional area for Windcatchers with a variety of constituent parts. This model will be refined in the future by representing the flow velocities in more detail, adding different wind directions and the effects of natural buoyancy and its robustness will be tested through comparison with additional theoretical and empirical data.

5. CONCLUSIONS

This paper presents estimated system losses for the supply and extract ducts of a square Windcatcher that are consistent for sealed and unsealed supply room scenarios based upon function under ideal conditions. Analysis of the empirically derived C_p values shows that they do not necessarily balance mass flow in and out of the Windcatcher indicating that either the pressure in the supplied room drops or there is an amount of infiltration through the building fabric initiated by the Windcatcher. With this knowledge it is now possible to predict Windcatcher performance for a given application.

Table 3: $\dot{Q}_{duct}$ Generated per m/s of u_w, α=0°

Duct\Source	Model: Sealed Room	Model: Unsealed Room	Wind Tunnel (Elmualim 2006)
Windward	0.0295	0.0303	0.0298
Side	0.0133	0.0152	0.0112
Leeward	0.0030	0.0088	0.0059

REFERENCES

CIBSE (2001). Guide C - Reference Data.

CIBSE (2005). Natural Ventilation in Non-Domestic Buildings AM10.

Elmualim, A. A. (2006). "Dynamic modelling of a wind catcher/tower turret for natural ventilation." Building Services Engineering Research and Technology 27(3): 165-182.

Elmualim, A. A. and Teekaram A. J. H. (2002). Natural Ventilation Testing Carried out for: Monodraught Ltd., BSRIA: 18.

Etheridge, D. W. (2004). "Natural Ventilation through Large Openings - Measurements at Model Scale and Envelope Flow Theory." International Journal of Ventilation 2(4): 325-342.

Munson, B., Young D. and Okiishi T. (1998). Fundamentals of Fluid Mechanics, John Wiley and Sons.

Parker, J. and Teekeram A. (2004). Design and Application Guide for Roof Mounted Natural Ventilation Systems. Bracknell, Berkshire, BSRIA: 75.