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WIND TURBINE TESTING

FINAL REPORT

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Wind Turbine Testing

by
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Introduction

Los Alamos National Laboratory (LANL) is investigating concepts for removing heat from nuclear material storage units. One method which has been suggested is the use of wind turbine ventilators on the roofs of the units. An advantage of these devices is that it would require no power to operate them, and in this sense the system would be fail safe. The internal heat in the storage units would naturally cause air to rise through the ventilators and the usually prevailing winds which average about 5 mph would combine with the buoyancy to exhaust the air from the units. However, it was not known whether these type of ventilators can deliver a sufficient air flow through the High Efficiency Particulate Air (HEPA) filters which must be an integral part of the ventilation system in a nuclear storage facility. Therefore, this study was undertaken to answer that question. In this report we will discuss the experiments which were run on two wind turbine ventilators, one with a one foot diameter inlet duct and one with a two foot diameter inlet duct. The wind turbines were supplied by LANL and tested in the New Mexico State University/LANL wind tunnel housed in the Mechanical Engineering Department on the Las Cruces Campus of NMSU. The experiments were run at a nominal wind speed of 5 mph, with and without heating below the ventilator inlet and with and without resistance elements at the ventilator inlet.

Experimental Methods

Roof ventilators (see Appendix for typical specifications) with one foot and two foot diameter inlet ducts (i.e. throat diameters) were tested in the NMSU/LANL 4 foot by 4 foot wind tunnel. The test set-up is shown in the photograph of Figure 1. Air was pulled through the tunnel by the 100 hp Joy Manufacturing multi-blade fan shown in the background of Figure 1. The variable pitch blades of this fan allowed the wind velocity past the ventilator rotor to be controlled. The rotational speed of the rotor was measured by a Strobotac frequency meter with a least count of 0.5 rpm. The tunnel wind velocity was measured using a pitot tube and a Dwyer No. 1425 Hook gage manometer with a least count of 0.001 inches of water (see Figure 2). The flowrate into the bottom of the ventilator (see Figure 3) was also measured with a pitot tube and a Hook gage. To simulate resistances which might be encountered in actual buildings, screens and filters were installed at the entrance to the ventilator (see Figure 3).

Figure 1: One Foot Diameter Ventilator Installed in Wind Tunnel.

Figure 2: Measurement of Wind Tunnel Velocity - Top View.

Experiments

The experiments run to determine the flowrates into roof ventilators under various conditions of wind velocity and inlet resistance are tabulated in Tables 1 and 2. Table 1 gives the experimental parameters for the one foot diameter ventilator and Table 2 gives the parameters for the two foot diameter ventilator. In these tables, the first column indicates the experiment number. The second through the fifth columns give the static pressure in inches of water, the dynamic pressure in inches of water, the air temperature in degrees Celcius, and the measured velocity in mph of the wind tunnel air at a point midway between the ventilator rotor and the wall of the wind tunnel (see Figure 2). Column six through ten present the ventilator parameters. These are: the static pressure in inches of water at the ventilator entrance (see Figure 3), the dynamic pressure at the entrance in inches of water, the air temperature at the entrance in degrees Celcius, the calculated volumetric flowrate in cfm through the ventilator, and the rotational speed of the ventilator rotor in rpm. Column eleven indicates if a restriction (resistance) was present at the ventilator inlet, column twelve indicates whether or not heat was added to the air entering the ventilator (see Figure 3), and column thirteen gives any comments pertinent to a specific experiment. In all, 12 experiments were performed with the one foot diameter ventilator and 21 experiments were performed with the two foot diameter ventilator. Before each experiment began the speed of the wind tunnel air approaching the ventilator was set at 5 mph by using a pitot tube placed at the centerline of the tunnel about 4' upstream of the ventilator.

Figure 3: Measurement of Wind Tunnel and Ventilator Velocity - Side View.

Table 1: Parameters and Results for One Foot Diameter Ventilator

Test No.	Tun. P_{sta} "H ₂ O	Tun. P_{dyn} "H ₂ O	Tun. T_{air} C	Tun. V_{air} mph	Vent. P_{sta} "H ₂ O	Vent. P_{dyn} "H ₂ O	Vent. T_{air} C	Vent. Q cfm	Vent. ω rpm	Vent. Restrictions*	Vent. Inlet Heat	Comments
1	-0.013	0.010	29.9	4.95	-0.017	0.009	28.4	324	67	None	No	set Vels.
2	-0.015	0.010	29.9	4.95	-0.015	0.002	28.4	153	50	Yes	No	1 FF
3	-0.010	0.011	25.3	5.15	-0.025	0.005	29.6	242	70	None	No	4-5mph wnd
4	-0.010	0.011	25.3	5.15	-0.014	0.013	29.6	390	76	None	No	Same as No.3
5	-0.013	0.011	25.3	5.19	-0.018	0.009	30.4	325	69/75	None	No	rpm chngd.
6	-0.013	0.012	23.2	5.36	-0.015	0.011	25.0	356	74	None	No	
7	-0.012	0.011	24.1	5.14	-0.013	0.008	24.5	303	69	Yes	No	1 WS
8	-0.014	0.012	25.5	5.38	-0.017	0.001	25.6	107	52	Yes	No	1 FF
9	-0.014	0.012	26.9	5.39	-0.017	0.001	26.9	108	52	Yes	No	2 FF
10	-0.014	0.014	27.0	5.82	-0.014	0.012	26.9	373	76	None	No	
11	-0.015	0.012	27.9	5.40	-0.019	0.001	26.9	108	53	Yes	No	1 PB
12	-0.015	0.012	28.4	5.40	-0.014	0.012	27.9	373	76	None	No	

Table 2: Parameters and Results for Two Foot Diameter Ventilator

Test No.	Tun. P_{sta} "H ₂ O	Tun. P_{dyn} "H ₂ O	Tun. T_{air} C	Tun. V_{air} mph	Vent. P_{sta} "H ₂ O	Vent. P_{dyn} "H ₂ O	Vent. T_{air} C	Vent. Q cfm	Vent. ω rpm	Vent. Restrictions*	Vent. Inlet Heat	Comments [†]
13	-0.011	0.011		5.13					Rot.	None	No	set Velo.
14	-0.012	0.008		4.38					0.0	None	No	RS
15	-0.015	0.011	24.3	5.14					0.0	None	No	Reset Velo.
16	-0.019	0.018	23.5	6.57	-0.016	0.007	24.8	1134	70	None	Yes [‡]	
17	-0.018	0.017	24.2	6.39	-0.016	0.009	24.8	1286	61	Yes	Yes [‡]	1 WS
18	-0.018	0.017	24.2	6.57	-0.016	0.012	24.8	1485	61	Yes	Yes [‡]	1 WS, CM
19	-0.018	0.018	24.2	6.57	-0.016	0.015	25.4	1662	75	None	Yes [‡]	
20	-0.018	0.018	24.2	6.57	-0.017	0.007	25.4	1135	37	Yes	Yes [‡]	1 FF
21	-0.020	-0.018	26.9	6.60	-0.020	0.005	25.4	959	35	Yes	Yes [‡]	1 PB
22	-0.021	0.010	26.7	4.90	-0.020	0.005	25.4	959	0.0	None	No	RS
23	-0.0205	0.0195	26.4	6.84	-0.021	0.0125	27.0	1515	80	None	No	
24	-0.0205	0.0195	26.4	6.84	-0.020	0.018	44.0	1869	79	None	Yes	
25	NA	NA	26.4	0.0	0.000	0.0015	34.0	532	10.4	None	Yes	Tun. Off
26	-0.0175	0.010	24.4	4.89	-0.005	0.006	25.4	1048	0.0	None	No	RS
27	-0.017	0.017	25.8	6.39	-0.016	0.014	36.4	1630	71	None	Yes	
28	-0.017	0.017	25.8	6.39	-0.016	0.014	36.4	1630	71	None	Yes	CM
29	-0.016	0.015	28.9	6.03	-0.015	0.0075	40.0	1200	57	Yes	Yes	1 WS
30	-0.016	0.014	29.6	5.83	-0.016	0.011	36.0	1444	71	None	Yes	
31	-0.016	0.010	25.6	4.90	-0.007	0.005	25.7	958	0.0	None	Yes	RS
32	-0.021	0.019	27.6	6.77	-0.019	0.017	36.0	1796	79	None	Yes	
33	-0.023	0.019	28.5	6.78	-0.0175	0.004	37.0	873	39	Yes	Yes	1 FF

* WS=1/8" Hardware Screen, FF=1" Furnace Filter, PB=1mil Plastic Bag Material

† CM=Calibrated (Zeroed) Manometers, RS=Rotation of Ventilator Stopped

‡ Hot plate placement too low to generate buoyancy effects.

Results and Discussion

Pumping Effect of Ventilator Rotation

The effect of the rotation of the ventilator rotor upon the volumetric flowrate through the ventilator can be examined by comparing Expt. Nos. 16 and 22 in Table 2. For Expt. No. 16 the rotor was free to turn, but for Expt. No. 22 the rotor was clamped so that it could not rotate. For both of these experiments no resistance was added to the ventilator system nor was the air entering the bottom of the ventilator heated. The volumetric flowrate through the two foot diameter ventilator for Expt. No. 16 was 1134 cfm and for Expt. No. 22 it was 959 cfm, a drop in flowrate of 175 cfm. Hence, most of the flowrate through the ventilator appears to be caused by low pressure created by the exterior wind being deflected around the ventilator rotor. At an exterior wind approach velocity of 5 mph, the pumping effect of the spinning rotor is seen to be small compared to the overall flowrate through the ventilator.

Effect of Heat Addition upon Ventilator Flowrate

The effect of the buoyancy of heated air entering the bottom of the ventilator upon the flowrate through the ventilator can be seen by examining Expt. Nos. 16, 25, and 32 in Table 2. For Expt. No. 16 no heat was added, but for Expt. No. 25 enough heat was added to increase the air temperature by about 7.6 C and the wind tunnel was not running. The ventilator flowrate for Expt. No. 16 was 1134 cfm and for Expt. No. 25 it was 532 cfm. Thus, it appears that heating causes an increase in the flowrate through the two foot diameter ventilator of 532 cfm, i.e. the combined flowrate due to a nominal 5 mph wind and a 7.6 C increase in air temperature entering the bottom of the ventilator should be $1134 + 532 = 1666$ cfm. Notice that Expt. No. 32, which had an 8.4 C increase in air temperature and for which the tunnel was running, showed a measured flowrate of 1796 cfm. If we consider the increase in flowrate per degree C to be

$$\frac{\Delta Q}{\Delta T} = \frac{532}{7.6} = 70 \text{ cfm/C} ,$$

then for a 8.4 C increase in temperature the increase in flowrate should be

$$70 \times 8.4 = 588 \text{ cfm} .$$

Thus, using the flowrate from Expt. No. 16 and adding 588 cfm gives

$$1134 + 588 = 1722 \text{ cfm}$$

which is quite close to the value of 1796 cfm found in Expt. No. 32. Since only a few experiments with significant temperature changes were

performed, these results should be used with caution. Furthermore, scaling was not examined for the temperature effect, so the results should be considered valid only for the two foot diameter roof ventilator.

Effect of Overall Headloss

The relation between headloss and ventilator flowrate will be examined by considering nearly all the data in Tables 1 and 2. When we speak of overall headloss we mean the static pressure loss which occurs between the bottom inlet of the ventilator and the air exiting the rotor of the ventilator, as shown in the sketch of Figure 3. Bernoulli's equation⁽¹⁾ written for this situation would be

$$\frac{p_{in}}{\gamma_{air}} + \frac{V_{in}^2}{2g} = \frac{p_{ex}}{\gamma_{air}} + \frac{V_{ex}^2}{2g} + h_{\ell} \quad , \quad (1)$$

in which p is the static pressure, V is velocity, γ is specific weight, h_{ℓ} is headloss, and g is the acceleration of gravity. We take the inlet velocity, V_{in} , to be zero and the inlet pressure, p_{in} , to be the atmospheric pressure, p_{atm} , then

$$h_{\ell} = -\left[\frac{p_{ex} - p_{atm}}{\gamma_{air}} + \frac{V_{ex}^2}{2g}\right] \quad . \quad (2)$$

The value

$$\frac{p_{ex} - p_{atm}}{\gamma_{air}}$$

is the static gage pressure of the air in the windtunnel, which in Tables 1 and 2 is called Tun. P_{sta} and is given in units of inches of water ($"H_2O$), i.e.

$$P_{sta_{tun}} = \frac{p_{ex} - p_{atm}}{\gamma_{air}} \quad , \quad "H_2O \quad . \quad (3)$$

The value

$$\frac{V_{ex}^2}{2g}$$

is the dynamic pressure of the air exiting the ventilator and we define it as

$$P_{dyn_{ex}} = \frac{V_{ex}^2}{2g} \quad , \quad "H_2O \quad . \quad (4)$$

$P_{dyn_{ex}}$ is not a value found in Tables 1 and 2, but it can be calculated from the measured dynamic pressure, Vent. P_{dyn} , found in these tables, as we shall see.

The dynamic pressure of the air flowing through the inlet duct of the ventilator is measured by a pitot tube positioned as shown in Figure 3. The

velocity at the exit can be calculated from the measured flowrate passing through the ventilator by using the continuity equation, i.e.

$$V_{ex} A_{ex} = Q ,$$

so

$$V_{ex} = \frac{Q}{A_{ex}} . \quad (5)$$

Now the dynamic pressure is

$$p_{dyn} = \frac{V^2}{2g} = \frac{Q^2/A^2}{2g} .$$

Thus,

$$P_{dyn_{ex}} = \frac{Q^2/A_{ex}^2}{2g}$$

and

$$P_{dyn_{vent}} = \frac{Q^2/A_{vent}^2}{2g} ,$$

so

$$P_{dyn_{ex}} = P_{dyn_{vent}} \frac{A_{vent}^2}{A_{ex}^2} . \quad (6)$$

Since $P_{dyn_{vent}}$ is given in Tables 1 and 2, then $P_{dyn_{ex}}$ can be calculated from Eq.6 and a knowledge of the area ratio , A_{vent}/A_{ex} .

The exit area of the ventilator rotor can be calculated by assuming the rotor to be a sphere. The bottom and top of the sphere are blocked by a cap and and the entrance duct, respectively, as shown in the sketch of Figure 4.

Figure 4: Rotor Configuration for Calculating A_{ex} .

The formula⁽²⁾ for the outflow area is

$$A_{ex} = 4\pi R_s [R_s - h] , \quad (7)$$

in which R_s is the radius of the sphere and h is the vertical distance between the poles of the sphere and the cord planes, as shown. For the one foot diameter ventilator, $R_s = 8.75''$ and $h = 3.8125''$, therefore $A_{ex_1} = 1.2\pi \text{ ft}^2$. Similarly, for the two foot diameter ventilator, $R_s = 15.5''$, $h = 7.125''$ and, therefore, $A_{ex_2} = 3.766\pi \text{ ft}^2$. Since the ventilator is challenged by a wind at right angles to its axis, we assume that the ventilator flow exits only the leeward half of the outflow area, i.e. the actual exit area is only half the calculated A_{ex} .

The ratios $(A_{vent}/A_{ex})^2$ needed to determine $P_{dyn_{ex}}$ are, therefore

$$\left(\frac{A_{vent}}{A_{ex}}\right)_1^2 = \left(\frac{\pi 1^2}{4} / \frac{1.2\pi}{2}\right)^2 = \frac{1}{5.76}$$

for the one foot diameter ventilator, and

$$\left(\frac{A_{vent}}{A_{ex}}\right)_2^2 = \left(\frac{\pi 1.9583^2}{4} / \frac{3.766\pi}{2}\right)^2 = \frac{1}{3.858}$$

for the two foot diameter ventilator and in which the subscript 1 indicates the one foot diameter ventilator and the subscript 2 the two foot diameter ventilator. By using these ratios the dynamic pressure at the exit can be calculated from the ventilator dynamic pressure given in Tables 1 and 2 through Eq.6. Knowing $P_{dyn_{ex}}$ allows the calculation of the total headloss across the ventilator from Eq.2, assuming that the measured tunnel P_{sta} is

$$P_{statun} = \frac{p_{ex} - p_{atm}}{\gamma_{air}}$$

and

$$P_{dyn_{ex}} = \frac{V_{ex}^2}{2g}$$

that is,

$$h_\ell = -[P_{statun} + P_{dyn_{ex}}] \text{ , "H}_2\text{O.} \quad (8)$$

Since headloss can also be thought of as a resistance to flow, let us define

$$R = h_\ell \text{ .} \quad (9)$$

From Tables 1 and 2 the overall resistance R has been calculated from Eq.8 and listed in Tables 3 and 4 as a function of measured flowrate Q .

Table 3: Overall Resistance for One Foot Diameter Ventilator

Expt. No.	$R, "H_2O$	Q, cfm
1	0.008	324
2	0.013	153
3	.020	242
4	0.001	390
5	0.008	325
6	0.004	356
7	0.005	303
8	0.016	107
9	0.016	108
10	0.002	373
11	0.018	108
12	0.002	373

Table 4: Overall Resistance for Two Foot Diameter Ventilator

Expt. No.	$R, "H_2O$	Q, cfm
16	0.0172	1134
17	0.0157	1286
18	0.0149	1485
19	0.0141	1662
20	0.0162	1135
21	0.0187	959
23	0.0173	1515
24	0.0158	1869
27	0.0134	1630
28	0.0134	1630
29	0.0141	1200
30	0.0131	1444
32	0.0166	1796
33	0.0220	873

If we assume a linear relation between flowrate Q and the resistance R , then from Table 3 a linear least squares fit⁽³⁾ to the data yields

$$Q_1 = 696.6 - 36272.4R_1 \text{ cfm}, \quad (10)$$

in which the subscript 1 indicates the one foot diameter ventilator. The standard deviation σ is

$$\sigma_1 = 82.2 \text{ cfm}. \quad (11)$$

Hence, the probable error⁽²⁾ ϕ is approximately

$$\phi_1 = \pm 0.675\sigma_1 = \pm 55.5 \text{ cfm}. \quad (12)$$

Similarly, from Table 4 a linear least squares fit to the data gives

$$Q_2 = 2631.1 - 77384.6 R_2 \text{ cfm}, \quad (13)$$

in which the subscript 2 indicates the two foot diameter ventilator. The standard deviation is

$$\sigma_2 = 245.6 \text{ cfm} \quad (14)$$

and the probable error is

$$\phi_2 = \pm 0.675 \sigma_2 = \pm 165.8 \text{ cfm}. \quad (15)$$

The scalability of the results is of interest. The tunnel approach velocity was maintained at 5 mph = 7.33 ft/sec throughout the testing, hence a characteristic flowrate is taken to be $Q_c = 7.33 A_{vent}$. Thus, the non-dimensional flowrate is

$$\hat{Q} = \frac{Q}{Q_c} = \frac{Q}{7.33 A_{vent}} \quad (16)$$

A characteristic resistance can be taken to be the potential head from the inlet to the exit of the ventilator, i.e.

$$R_c = D \frac{\gamma_{air}}{\gamma_{H_2O}}, \quad (17)$$

where we assume D is the ventilator diameter since the two ventilators are geometrically similar (their diameter to height ratios are equal). The terms γ_{air} and γ_{H_2O} are the specific weight of air and water, respectively. The non-dimensional resistance is the resistance R divided by R_c , i.e

$$\hat{R} = \frac{R \gamma_{H_2O}}{D \gamma_{air}}, \quad (18)$$

where we assume that the specific weights are constant at the same values for all the experiments ($\gamma_{H_2O} = 62.4 \text{ lb/ft}^3$, $\gamma_{air} = 0.066 \text{ lb/ft}^3$).

Non-dimensionalizing Eqs. 10 and 13 using the characteristic flowrate and characteristic resistance gives

$$\hat{Q}_1 = 121.0 - 6.68 \hat{R}_1, \quad \hat{\phi}_1 = \pm 14.3 \quad (19)$$

and

$$\hat{Q}_2 = 119.2 - 7.44 \hat{R}_2, \quad \hat{\phi}_2 = \pm 11.1 \quad (20)$$

Thus, it appears that the flowrate versus resistance results for the one foot ventilator and the two foot ventilator scale to within the probable error of the experimental data.

Effects of Added Resistance

In order to determine the relationship between flow through the ventilator and additional loss elements, such as filters, experiments were run with three different types of elements: wire hardware screen with $1/8'' \times 1/8''$ mesh, fiberglass furnace filter material $1''$ thick, and a porous plastic membrane made from 1 mil plastic bag material. The static and dynamic ventilator pressures given in Tables 1 and 2 were taken downstream of the location of the added resistive element. Thus, the headloss through the resistive element can be calculated as follows. First, using the experiments in which no added resistive element was present (Expt. Nos. 1, 3, 4, 5, 6, 10 and 12 for the one foot diameter ventilator and Expt. Nos. 16, 19, 23, 24, 27, 30 and 32 for the two foot diameter ventilator), a loss coefficient K based upon the square of the flowrate was calculated:

$$h_{\ell_o} = KQ^2, \text{ ''H}_2\text{O.} \quad (21)$$

The headloss and flowrate used to calculate K from Eq.21 were the average of the headlosses ($h_{\ell} = -[P_{sta} + P_{dyn}]_{vent}$) and the average of the ventilator flowrate given either in Table 1 or Table 2. The results were

$$K_1 = 5.562 \times 10^{-8}, \text{ ''H}_2\text{O} \quad (22)$$

and

$$K_2 = 1.6912 \times 10^{-9}, \text{ ''H}_2\text{O} \quad (23)$$

in which the subscript 1 refers to the one foot diameter ventilator and the subscript 2 to the two foot diameter ventilator.

The headloss due to the resistive element was found by taking the headloss obtained from Tables 1 and 2 and subtracting KQ^2 from it. Since for some of the resistive elements more than one experiment was run, the final headloss value for that element was taken to be the average for these experiments. We call the headloss through the resistive element R_f . The results are given in Tables 5 and 6 for the one foot and two foot diameter ventilators, respectively.

Table 5: Headloss due to Resistive Elements-One Foot Diameter Ventilator

Resistive Element*	Flowrate Q, cfm	Headloss R_f , ''H ₂ O
None	340	0.0
1 WS	303	≈ 0.0
1 FF	130	0.0135
2 FF	108	0.01535
1 PB	108	0.01735

* WS = $1/8'' \times 1/8''$ Hardware Screen, FF = $1''$ Thick Furnace Filter, PB = 1 mil Plastic Bag Material

Table 6: Headloss due to Resistive Elements-Two Foot Diameter Ventilator

Resistive Element*	Flowrate Q, cfm	Headloss R_f , "H ₂ O
None	1739	0.0
1 WS	1323.7	0.00514
1 FF	1004	0.01155
1 PB	959	0.01500

* WS = 1/8" × 1/8" Hardware Screen, FF = 1" Thick Furnace Filter, PB = 1 mill Plastic Bag Material

Since we need to know how the added resistance will affect the flowrate, we fit the data of Tables 5 and 6 with a linear least squares fit for the resistance, i.e.

$$R_{f_1} = 0.02355 - 0.00007Q_1, \quad \phi_1 = \pm 0.000585, \quad \text{"H}_2\text{O} \quad (24)$$

and

$$R_{f_2} = 0.02992 - 0.00001762Q_2, \quad \phi_2 = \pm 0.000894, \quad \text{"H}_2\text{O} \quad (25)$$

The loss through a filter is usually proportional to velocity, i.e. it is diffusional in nature. We assume that the characteristic headloss R_{cle} for the loss elements is, therefore

$$R_{cle} = KV_{vent}$$

where V_{vent} is the maximum velocity measured, i.e. the velocity through the ventilators when no added resistance is present. Now

$$V_{vent} = \frac{Q}{A_{vent}} = \frac{Q}{\frac{\pi D^2}{4}}.$$

Hence,

$$R_{cle} = \frac{4KQ}{\pi D^2} = K^* \frac{Q}{D^2}.$$

Thus, for the one foot diameter ventilator

$$R_{cle_1} = K^*Q_1 = 340K^*$$

and for the two foot diameter ventilator

$$R_{cle_2} = K^* \frac{Q_2}{(1.9583)^2} = K^* \frac{Q_2}{3.835} = 453.45K^*,$$

where we have taken $Q_1 = 340$ cfm from the first row in Table 5 and $Q_2 = 1739$ cfm from the first row in Table 6. Since we know the resistive elements used in the experiments for both sizes of ventilator were similar, we have assumed that K^* is the same for both sizes of ventilator. Furthermore, we

know the resistances in Tables 5 and 6 are on the order of 0.01 "H₂O, so we assume that $K^* = 0.0001$ for both sizes of ventilator. Hence, characteristic resistances for the two sizes of ventilator are

$$R_{cle_1} = 0.034 \text{ , "H}_2\text{O} \quad (26)$$

and

$$R_{cle_2} = 0.0453 \text{ , "H}_2\text{O}, \quad (27)$$

respectively. The characteristic flow is again $Q_c = 7.33A_{vent}$. The non-dimensional forms of Eqs. 24 and 25 are found by using R_{cle} and Q_c . The results are

$$\hat{R}_{f_1} = 0.6926 - 0.01185\hat{Q}_1 \text{ , } \hat{\phi}_1 = \pm 0.0172 \quad (28)$$

and

$$\hat{R}_{f_2} = 0.6605 - 0.00896\hat{Q}_2 \text{ , } \hat{\phi}_2 = \pm 0.0198 \quad (29)$$

Thus, it appears that the relation between added resistive element loss to ventilator flow rate scales to within the probable errors of the experimental data.

Matching Ventilators with HEPA Filters as Resistive Elements

High Efficiency Particulate Air (HEPA) filters must be installed in the ventilation systems of nuclear facilities. The size 5 HEPA filter is a standard filter used in many of these systems. It has a cross-sectional area through which the air flows of $2' \times 2'$ or $4ft^2$ and a nominal pressure drop of about 1"H₂O for a flowrate of 1000 cfm through that area. Since the flowrate through HEPA filters is diffusional in nature, we can assume the flowrate Q_{HEPA} is directly proportional to the filter headloss, R_{HEPA} , i.e.

$$Q_{HEPA} = 1000R_{HEPA} \quad (30)$$

in which the units of R_{HEPA} are in inches of water and the units of Q are in cfm. We assume that the ventilation system has a bank of n parallel size 5 HEPA filters. All the air passing through these n HEPA filters is discharged through m roof ventilators to the atmosphere. If the system is to operate, then the added resistance of the HEPA filters must match the allowable added resistance of the ventilators, i.e.

$$R_{f_1} = R_{HEPA}$$

or

$$0.02355 - 0.00007 \frac{Q_{tot}}{m} = \frac{Q_{tot}}{1000n} \quad (31)$$

for the one foot diameter ventilator, and

$$R_{f_2} = R_{HEPA}$$

or

$$0.02992 - 0.00001762 \frac{Q_{tot}}{m} = \frac{Q_{tot}}{1000n} \quad (32)$$

for the two foot diameter ventilator.

When we examine the data in tables 5 and 6, we see that for the one foot diameter ventilator a reasonable attainable flowrate is $Q_1 = 130$ cfm and for the two foot diameter ventilator, $Q_2 = 1000$ cfm. Hence, for the one foot diameter ventilator assume $Q_{tot}/m = 130$ cfm and for the two foot diameter ventilator assume $Q_2/m = 1000$ cfm. Using $Q_{tot}/m = 130$ cfm in Eq. 31 gives

$$R_{f1} = 0.02355 - 0.00007(130) = \frac{130m}{1000n} ,$$

or

$$\left(\frac{n}{m}\right)_1 \approx 9 . \quad (33)$$

That is, for each one foot diameter ventilator we would need 9 size 5 HEPA filters in parallel to keep the resistance through the HEPA filters small enough for the ventilator to overcome in an exterior wind of 5 mph.

Similarly, for the two foot diameter ventilator, using $Q_{tot}/m = 1000$ cfm in Eq. 32 gives

$$R_{f2} = 0.02992 - 0.00001762(1000) = \frac{1000m}{1000n} ,$$

or

$$\left(\frac{n}{m}\right)_2 \approx 81 . \quad (34)$$

That is, for each two foot diameter ventilator we would need 81 size 5 HEPA filters in parallel to keep the resistance through the HEPA filters small enough for the ventilator to overcome in an exterior wind of 5 mph.

Suppose a nuclear storage vault requires a flowrate of 10,000 cfm to keep it cool. Then the number of roof ventilators one foot in diameter or two foot in diameter to provide 10,000 cfm in a 5 mph wind would be:

$$m_1 = \frac{10000}{130} \approx 77 \text{ one foot diameter ventilators} \quad (35)$$

or

$$m_2 = \frac{10000}{1000} = 10 \text{ two foot diameter ventilators.} \quad (36)$$

So, the number of size 5 HEPA filters required in parallel would be:

$$n_1 = 9m_1 = 693 \text{ HEPA filters} \quad (37)$$

or

$$n_2 = 81m_2 = 810 \text{ HEPA filters.} \quad (38)$$

If we assume the filter banks are square, then the length of a side would be approximately

$$L_1 = \sqrt{693} \times 2' \approx 53' \quad (39)$$

and

$$L_2 = \sqrt{810} \times 2' \approx 57' \quad (40)$$

that is, the ventilation system plenum chamber housing the HEPA filters would be very large.

Here we have assumed that only a single bank of HEPA filters is required. Often in nuclear facility ventilation systems several banks of roughing filters, pre-filters, and HEPA filters are required in series. Since the resistances of the banks would add, then the total number of filters in each bank would need to be increased to keep the total resistance at a level which could be handled by roof ventilators operating in a 5 mph wind. Because of the very large number of HEPA filters which would be required for use with wind ventilators, it does not appear that it would be feasible to use roof ventilators for cooling nuclear storage vaults.

Conclusions

The following conclusions can be drawn:

1. For both the one foot and two foot diameter ventilators operating in a 5 mph wind, the largest portion of the flowrate through the ventilator is caused by the reduced pressure in the exterior air flowing around the rotor of the ventilator.
2. For the two foot diameter ventilator a flowrate of about 70 cfm per degree Celcius above ambient temperature is induced by the buoyant effect.
3. The volumetric flowrate through the ventilators as a function of overall resistance appears to scale between the two sizes of ventilator tested.
4. Because of the large number of HEPA filters which will be necessary in a ventilation cooling system for nuclear storage systems which utilizes roof ventilators as the sole source of moving the air, it is probably not feasible to pursue this concept.

Acknowledgements

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References

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APPENDIX

ROMLAIR

...free as the breeze!

Turbo Ventilators

WHY TURBO VENTILATORS-The Romlair Turbo Ventilator is an extremely popular and universally adaptable exhauster. This popularity is due to its clean, low silhouette rugged construction and high capacity. In operation the rotative movement imparted by the wind velocity creates exhaust by mechanical action similar to a backward curved blower wheel. This centrifugal action counteracts resistance and boosts the natural flow of air. Momentum, created by the high peaks of wind velocity, assures a higher, more consistent exhaust pattern than is possible with other gravity ventilators. Turbo ventilators are efficient, impressive and psychologically appealing due to their effortless spinning motion.

BASES-ACCESSORIES

Square to round bases for any type of roof are shown on page 5. Manually operated disc dampers, where required, are an integral part of the Romlair Model RR-12 through RR-48. Unit disc dampers and the Model RDF Duct Fan designed to convert existing round gravity ventilators to power are also shown on page 5. Bird and insect screens are not required in Romlair rotary ventilators.

FEATURES

First cost is last cost when Romlair ventilators are installed. Engineered construction provides rugged trouble free ventilators that last for years. Ball bearings are sized to withstand many times the actual thrust loads carried, and are lubrication packed for years of service. Special retainers provide for easy relubrication and exclude foreign matter from the bearing area. Ventilators to 14 inch throat, employ virtually indestructible bearings of long wearing nylon. The resilient nature of this material provides silent, almost frictionless operation without need for relubrication. Fabrication of all hot dipped galvanized sheet material guarantees long life without premature rust or corrosion under all weather conditions. Protection is afforded by grey enamel.

SMOOTH AND QUIET-Romlair Turbo Ventilators are weatherproof and run "free as the breeze" without sound. Careful factory balance and alignment assures true, vibrationless operation when installed according to factory recommendation.

SPECIFICATIONS

Ventilators shall provide mechanical ventilation through spinning action as manufactured by Romla Ventilator Co., Gardena, California, and shall be mounted on manufacturer's square to round bases. Ventilators shall be weatherproof and stormproof under all normal operating conditions. Ventilators up to 14" throat shall have bearings of self-lubricating long wearing nylon, and sizes 16" through 48" ventilators shall have heavy duty ball bearings. Ventilators shall have a sealed bearing housing to exclude dust, grit, and foreign matter, a double cap for additional strength, and a locked on rotor mechanism to prevent loss or damage in high winds. Capacities shall be based on ratings based on formulas outlined in the ASHRE guide for natural ventilation.

DIMENSIONS							
TURBO VENTILATORS							
MODEL	GALV. GAUGE	WIDTH	HEIGHT	MODEL	GALV. GAUGE	WIDTH	HEIGHT
RR6	26	10	11 - 3/4	RR20	24	26 - 3/8	24 - 1/2
RR8	26	12 - 1/2	13	RR24	24	32	29
RR10	26	14 - 1/2	14 - 1/2	RR30	22	39	32
RR12	24	16 - 3/8	15 - 1/2	RR36	22	46	38
RR14	24	18 - 5/8	17 - 1/2	RR42	20	52	40
RR16	24	21 - 3/4	20	RR48	20	58	41
RR18	24	23 - 3/4	22 - 1/2				

the ventilator with all the *plus* features

1. Heavy duty ball bearings, sizes 16 to 48, self-lubricating nylon on smaller sizes.
2. Double cap assures permanent alignment and stability under severest conditions.
3. Locked on rotor mechanism prevents loss or damage in high winds, easily removable.
4. Corrugated die pressed vanes provide more strength, stability and uniformity.
5. Oil impregnated sintered radial bearings assure quiet operations and longer life.
6. Sealed bearing housing excludes dust, grit and foreign matter, and retains lubricant.
7. Hot dipped galvanized sheet per federal specification number QQ1716-2, class DI.
8. Finish coat of enamel provides eye appeal plus greater protection.

HOW TO SPECIFY SIZE AND NUMBER

1. Figure cubic feet in space to be ventilated.
2. Determine rate of air exchange (See below).
3. Divide cubic feet of air space by number of minutes required for air change (see below). Result is cubic feet of air to be exhausted per minute.
4. Recommended ventilator spacing is 20 ft. or in industrial buildings one or more ventilators per bay, depending on building width.
5. Divide results of #3 by number of ventilators you propose to use. Then turn to capacity data to determine size ventilator to specify.

RECOMMENDED PERIODS OF A+R CHANGE FOR VARIOUS TYPES OF BUILDINGS

TYPES OF BUILDINGS	MINUTE AIR CHANGES
ASSEMBLY HALLS	2-10
ATTICS	2-4
AUDITORIUMS	4-15
BAKERIES	1-5
BANKS	3-10
BARS	1-5
BEAUTY PARLORS	4-10
BOILER ROOMS	1-4
BOWLING ALLEYS	2-6
BOX ANNEALING	2-5
BREWERIES	2-8
CHURCHES	4-15
CLASSROOMS	3-8
CORRIDORS	3-8
DRY CLEANERS	2-5
DINING ROOMS	1-5
ENGINE ROOMS	10-15
FACTORIES (LT MFG.)	1-2
FACTORIES (HVT MFG.)	2-8
FORGE SHOPS	2-6
FOUNDRIES	5-8
GARAGES	1/2-1
GLASS PLANTS	1-3
GYMNASIUMS	2-3
HEAT TREATING	3-6
KITCHENS	2-8
LAUNDRIES	2-5
MACHINE SHOPS	1-6
OFFICES	1-2
PACKING HOUSES	1-2

ROMLAIR ROTARY TURBINE SELECTION

In order to determine the exhaust capacity of any model ROMLAIR Rotary Ventilator under varying conditions of wind velocity, temperature difference between inside and outside air, and height of ventilator above intake (windows, doors, etc.) the table below may be used.

To determine exhaust capacity of various sizes under conditions considered to be average, merely use 5 MPH column, 10 degree temperature difference, and 30' stack height. All capacities are based on formula outlined in the ASHRE Guide.

C. F. M. - RATINGS															
STACK HEIGHT	WIND SPEED	TEMP. DIFF.	MODEL RR THROAT DIAMETERS												
			6	8	10	12	14	16	18	20	24	30	36	42	48
10	4	10	81	144	224	323	440	575	727	896	1290	2018	2905	3945	5178
		20	83	149	234	334	455	595	752	926	1334	2086	3004	4080	5355
		30	88	157	244	352	479	627	793	977	1406	2200	3167	4300	5644
	5	10	96	172	268	385	525	687	870	1070	1541	2410	3471	4713	6186
		20	98	174	271	390	533	697	881	1085	1564	2445	3520	4781	6275
		30	104	186	289	410	567	742	938	1155	1664	2602	3747	5088	6678
	8	10	144	257	400	576	786	1028	1299	1600	2305	3603	5189	7046	9248
		20	146	261	407	586	798	1044	1320	1626	2342	3663	5274	7162	9400
		30	148	264	410	591	806	1054	1333	1642	2364	3677	5324	7229	9488
	10	10	178	317	494	711	970	1268	1604	1975	2845	4448	6405	8690	11416
		20	182	325	506	729	994	1301	1644	2025	2817	4561	6568	8918	11705
		30	186	332	517	746	1016	1326	1678	2071	2983	4655	6715	9139	11875
20	4	10	83	149	232	334	455	595	752	926	1334	2086	3004	4080	5355
		20	97	173	269	388	529	692	874	1077	1551	2426	3493	4742	6224
		30	99	177	276	398	543	710	897	1105	1592	2489	3584	4867	6388
	5	10	98	174	271	390	533	697	881	1085	1564	2445	3520	4781	6275
		20	101	180	281	404	551	721	912	1123	1617	2529	3641	4944	6489
		30	105	188	293	422	576	753	952	1172	1689	2641	3804	5165	6779
	8	10	146	261	407	586	798	1044	1320	1626	2342	3663	5274	7162	9400
		20	151	270	420	604	824	1078	1363	1678	2418	3781	5444	7392	9702
		30	164	293	456	656	895	1170	1480	1822	2625	4105	5911	8026	10534
	10	10	182	325	506	729	994	1301	1644	2025	2817	4561	6568	8918	11705
		20	187	334	520	749	1021	1336	1689	2080	2996	4684	6745	9158	12020
		30	194	347	540	777	1059	1386	1752	2158	3109	4861	7000	9504	12474
30	4	10	88	157	244	352	479	627	793	977	1406	2200	3167	4300	5644
		20	99	177	276	318	543	710	897	1105	1592	2489	3584	4867	6388
		30	111	198	308	444	606	792	1002	1234	1777	2779	4002	5434	7132
	5	10	104	186	289	416	567	742	938	1155	1664	2602	3747	5088	6678
		20	105	188	293	422	576	753	952	1172	1689	2640	3804	5165	6779
		30	122	218	339	488	666	871	1101	1356	1953	3054	4398	5971	7837
	8	10	148	264	410	591	806	1054	1333	1642	2364	3697	5324	7229	9488
		20	164	293	456	656	895	1170	1480	1822	2625	4105	5911	8026	10534
		30	170	303	472	680	927	1212	1533	1888	2719	4252	6123	8314	10912
	10	10	186	332	517	746	1016	1326	1678	2072	2983	4655	6715	9139	11875
		20	194	347	540	777	1059	1386	1752	2158	3109	4861	7000	9504	12474
		30	197	351	547	787	1074	1404	1775	2189	3149	4925	7091	9629	12638