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AN AERO-ACOUSTIC PERFORMANCE PREDICTION METHOD OF SIROCCO FAN

Chan Lee
Department of Mechanical Engineering,
University of Suwon
Hwaseong City, Gyeonggi, Korea

Hyun Gwon Kil
Department of Mechanical Engineering,
University of Suwon
Hwaseong City, Gyeonggi, Korea

ABSTRACT

A design-analysis method of Sirocco fan is developed for constructing 3-D impeller and scroll geometries, and for predicting both the aerodynamic performance and the noise characteristics of the designed fan. The aerodynamic blading design of fan is conducted by blade angle, camber line determinations and airfoil thickness distribution, and then the scroll geometry of fan is designed by using logarithmic spiral. The aerodynamic performance of designed fan is predicted by the mean line analysis with flow blockage, slip and pressure loss correlations. Based on the predicted performance data, fan noise is predicted by two models for cutoff frequency and broadband noise sources. The present predictions for the performance and the noise level of actual fans are well agreed with measurement results.

INTRODUCTION

Sirocco fans are widely used rotating machines in industrial, ventilation and air-conditioning, which produce the noise due to the pressure fluctuation on blade and scroll casing surfaces. However, the air-born fan noise is strongly related with the aerodynamic flow field and the performance of fan, so the noise control and reduction of fan must be attempted with the consideration of the interaction between aerodynamic and acoustic characteristics. For this reason, the actual fan design practice in industry calls for reliable analysis method for predicting both performance and noise level.

Therefore, the present study provides a simpler and less time consuming design-analysis method of Sirocco fan which is capable of predicting both performance and noise. The aerodynamic blading design of fan is conducted by blade angle, camber line determinations and airfoil thickness distribution, and then the scroll geometry of fan is designed by using logarithmic spiral. The aerodynamic performance of designed fan is predicted by the mean line analysis with flow blockage, slip and pressure loss correlations. Based on the predicted performance data, fan noise is predicted by two models for cutoff frequency and broadband noise sources. The present analysis method is applied to actual fan design practices, and their prediction results are compared with measured results to verify the prediction accuracy of the present method.

DESIGN AND ANALYSIS METHODS

The geometric shapes of impeller blade and scroll casing in Sirocco fan are designed by commonly available techniques for blade angle, camber line determinations and airfoil thickness distribution, and then the determination of the scroll geometry with logarithmic spiral. The key design variables of Sirocco fan are depicted in Fig.1.

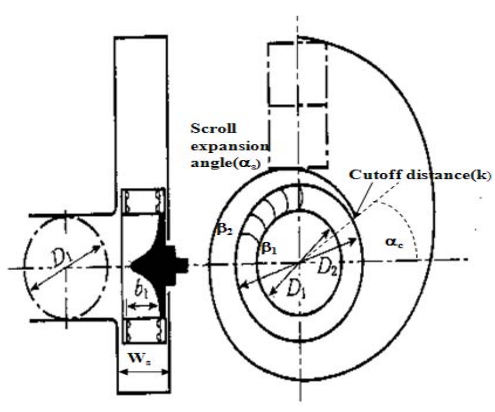

Fig. 1 Geometry and design variables of Sirocco fan

The performance and the efficiency of Sirocco fan are predicted by combining mean line analysis method with the correlation models for flow leakage, flow blockage, slip factor and pressure loss. The flow leakage, the flow blockage and the slip factor models used in the present study are as follows:

Flow blockage, leakage and slip coefficient

Flow blockage(B_F) and its effect on the effective impeller blade inlet width($_{b1,eff}$) are calculated by Yamazaki's correlation[1]

$$b_{1,eff} = b_1(1 - B_F) \tag{1}$$

$$B_F = 0.38 + a\left|\frac{b_1}{D_2} - 0.36\right|^3 + 0.62\left(\frac{D_1}{D_2} - 0.86\right) + 0.26(\Phi - 0.3)$$
$$a = \begin{bmatrix} 14.8 \ , \ b_1/D_2 < 0.36 \\ 3.0 \ , \ b_1/D_2 \geq 0.36 \end{bmatrix} \tag{2}$$

where Φ is flow coefficient.

Flow leakage($\triangle Q$) and slip coefficient(μ) can be predicted also by Yamazaki's correlation.

$$\triangle Q = \frac{(0.3266 - 0.0794\beta_2)}{\Phi} Q \tag{3}$$

$$\mu = 1 - \left\{1 + \frac{0.3\Phi}{\sin(\beta_2 - 20)}\right\} X \left\{1 - \frac{0.26}{\left(\frac{s_2}{c}\right)^{1/3} |\cos(\beta_2 - 20)|}\right\} \tag{4}$$

where Q is fan discharge flow rate and s_2 is the throat diameter at impeller blade outlet.

In modeling pressure loss, the present study considers the incidence, the diffusion, the mixing and the impulse losses within impeller blades, and the friction and the impulse losses in scroll[1,2] as follows:

Pressure losses in impeller blades

a. Incidence loss

$$\triangle p_{ii} = \zeta_{ii}\frac{\rho}{2}(\triangle W_{1u}\sin i)^2 \tag{5}$$

where $\triangle W_{1u}$, i and ζ_{ii} are the difference between ideal and actual tangential velocities, incidence angle and loss coefficient of 1.0.

b. Diffusion and friction losses

$$\triangle p_{id} = \zeta_{id}\frac{\rho}{2}\left[W_1^2 - W_2^2\right] \tag{6}$$

$$\triangle p_{if} = \zeta_{if}\frac{\left[2Zcb_m + \pi(r_2^2 - r_1^2)\right]}{W_1\sin\beta_1\pi D_1 b_1}\frac{\rho}{2}W_m^2 \tag{7}$$

where b_m , W_m are the mean values of the impeller inlet and outlet widths, the mean values of the relative velocities at blade inlet and outlet, and $\zeta_{if} = 0.0045$ and ζ_{id} =0 if W1 <W2 or ζ_{id}=0.2 if W1>W2.

c. Mixing loss

$$\triangle p_{im} = \zeta_{im}\frac{\rho}{2}W_r^2 \quad (9)$$
$$\zeta_{im} = \frac{Zt_B/\pi D_2\cos(180 - \beta_2)}{1 - Zt_B/\pi D_2\cos(180 - \beta_2)} \tag{8}$$

where t_B is the impeller blade thickness, W_2 is the relative flow velocity at impeller blade outlet.

d. Impulse loss

$$\triangle p_{is} = \zeta_{is}\frac{\rho}{2}U_2^2\frac{D_1}{D_2}\left[\frac{Q}{Q_d} - 1\right]^2 \tag{9}$$

where ζ_{is} is 0.7, U_2는 is the rotative speed at impeller blade outlet, Q_d is the design-point flow capacity.

Pressure losses in scroll

a. Impulse loss

$$\triangle p_{sr} = \zeta_{sr}\frac{\rho}{2}C_{m3}^2 \tag{10}$$

$$\triangle p_{st} = \zeta_{st}\frac{\rho}{2}\left(C_{u3} - \dot{C}_{u3}\right)^2 \tag{11}$$

where C_{m3}, C_{u3}, are the radial and the tangential velocities at the representative location of the angle, $\theta = 7\pi/4$ from cut-off, ζ_{sr}, ζ_{st} are 0.7, 1.0 respectively.

b. Friction loss

$$\triangle p_{sf} = \zeta_{sf}\frac{\rho}{2}C_3^2$$
$$\zeta_{sf} = 4.08/\left[\ln(Re^{2.6})(L_s/S_{es})\right] \tag{12}$$

where Re is the Reynolds number based on scroll arc length(Ls) and equivalent diameter(Ses).

Using the calculated flow leakage, flow blockage, slip coefficient and pressure losses by above-mentioned methods, the static pressure(p_{ac}), power(L_{ac}) and efficiency(η) can be computed as follows:

$$p_{ac} = p_{th} - (\Delta p_{ii} + \Delta p_{id} + \Delta p_{if} + \Delta p_{im} + \Delta p_{is}) - (\Delta p_{sr} + \Delta p_{st} + \Delta p_{sf}) \quad (13)$$

$$L_{ac} = (Q + \Delta Q) p_{th} \quad (14)$$

$$\eta = \frac{p_{ac} Q}{L_{ac}} \quad (15)$$

where p_{th} is the ideal pressure rise with no pressure loss, and Q is the discharge flow capacity of fan.

Based on the predicted performance data, the cutoff frequency noise components at blade passing frequency and its harmonics are predicted by the model of Ohta et al.[3] which is derived from fan noise similarity law, and the broadband noise of fan is predicted with the use of the models by Mugridge and Lowson[4,5] for the pressure fluctuation from blade surface boundary layer, wake vortex shedding and inlet flow turbulence. All the models used in the present performance and noise prediction methods contain various fan design variables and operating conditions to reflect their effects on aero-acoustic characteristics of Sirocco fan.

Cutoff frequency noise

$$SPL = 10\log\left[\frac{A_2}{4\pi}\left(\frac{\pi}{Z}\right)^{\gamma}\right] + 10\log He^{\gamma} - 10\log St^{\gamma} + \left[166 + 0.269\frac{r_c}{D_2} - \left(162 + 4.24\frac{r_c}{D_2}\right)\left(\frac{k}{D_2}\right)\right] + 17.2 - 20\log\left(\frac{r}{1.621}\right) \quad (16)$$

where A_2, γ and $He = fD_2/a$ are the cross section area of impeller blade outlet, 6 and Helmohltz number, and f is frequency and a is sound speed of air. Also, for Strouhal number, St =1, 2, 3 represent fundamental frequency, 2nd, 3rd harmonics respectively, and r is the noise measuring distance from fan.

Broadband noise

a. Noise due to turbulent boundary layer and wake

$$PWL(f) = 47 + 25\log p_s + 7\log Q + 10\log X + F_1(f)$$

$$X = \left[\frac{1-\eta_s}{\eta_s}\right]\left[\frac{\Phi^2 + 1 - \Psi + \Psi^2/2}{\Psi}\right]^{3/2} \quad (17)$$

where p_s, Φ, Ψ and η_s are fan static pressure[mmAq], flow coefficient, pressure rise coefficient and fan static efficiency, and $F_1(f)$ in eqn.(17) is expressed in Fig. 2.

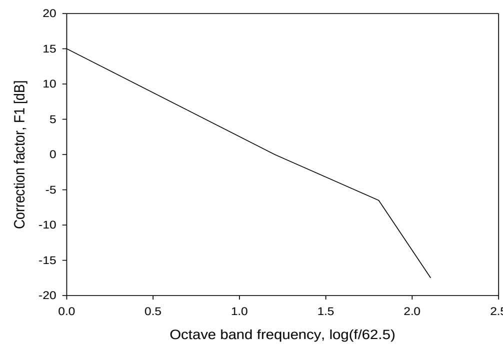

Fig. 2 Correction factor(F_1)

b. Noise due to inflow turbulence

$$SPL = 10\log\left[\rho_o^2 a_o^2 l \frac{\Delta L}{r^2} M^3 I^2 \kappa^3 (1+\kappa^2)^{-7/3}\right] + 10\log\left[\frac{1+K_c}{K_c}\right]$$

$$\kappa = \frac{\pi f c}{V} \quad K_c = \frac{10 S^2 M \kappa^2}{1-M^2},$$

$$S^2 = \left[\frac{2\pi\kappa}{1-M^2} + \left(1 + \frac{2.4\kappa}{1-M^2}\right)^{-1}\right]^{-1} \quad (18)$$

where l, I, ΔL, M, ρ_o and a_o are inflow turbulence length scale, turbulence intensity, half blade span, Mach number, air density and sound speed of air.

Table 1 Design specifications of 5 fan models

Fan model	A	B	C	D	E
Blade tip diameter[m]	0.150	0.320	0.154	0.460	0.200
Blade hub diameter[m]	0.123	0.285	0.118	0.405	0.170
Blade width[m]	0.060	0.120	0.073	0.217	0.080
Blade inlet angle[deg]	83	66	60	83	65
Blade outlet angle[deg]	136	160	155	168	150
Number of Blades	33	61	41	42	78
Scroll width[m]	0.072	0.168	0.085	0.370	0.120
Cutoff distance[m]	0.010	0.032	0.010	0.046	0.016
RPM	1550	660	2960	850	1000

ANALYSIS RESULTS AND DISCUSSIONS

In order to verify the reliability of the present method, the present design-analysis method is applied to 5 Sirocco fan models used in air-conditioning, ventilation, industrial and home appliance applications as shown in Table 1. Figs. 3 and 4 show the geometric shapes of impeller blades and scroll of fan model C.

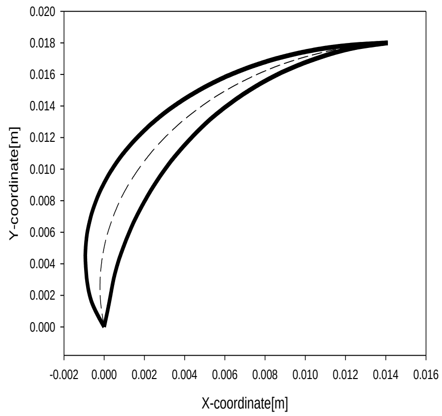

Fig. 3 Impeller blade section design of fan model C

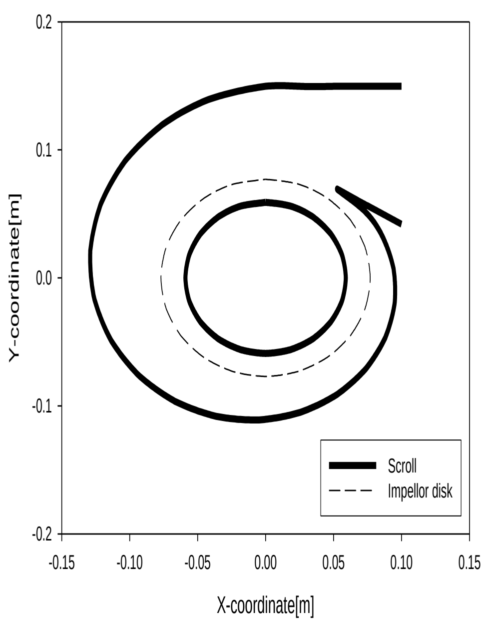

Fig. 4 Scroll section design of fan model C

Fig. 5 shows the performance comparisons between the prediction and the measurement for 5 Sirocco fans. The present prediction for fan static pressure is well agreed with the measurement(6) within 8% relative error.

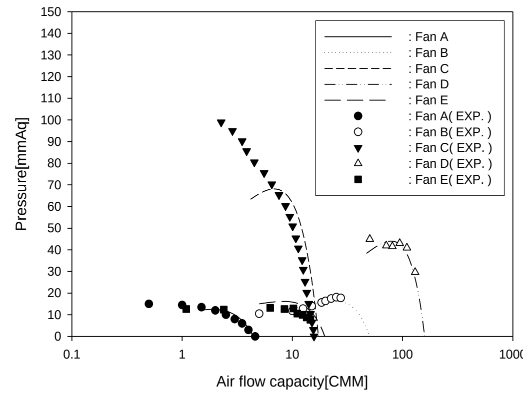

Fig. 5 Performance comparison between the prediction and the measurement of 5 fan models

Table 2 summarizes the comparison between the prediction and the measurement(6) for the overall noise levels of 5 fan models at design flow condition. As shown in Table 1, the predicted fan noise levels are well agrees with the measurement within 4% relative error.

Table 1 Fan noise level comparisons

Fan model	Prediction [dB, dB(A)]	Measurement [dB, dB(A)]	Relative error (%)
A	51.78	54.00	-4.11
B	24.59	24.00	+2.46
C	48.79	47.55	+2.61
D	80.68	80.00	+0.85
E	39.38	39.54	-0.40

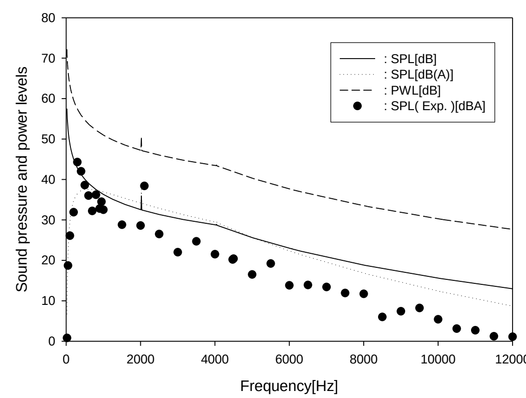

Fig. 2 Noise spectrum of fan model C

Fig. 6 also represents the noise spectrum of the fan model C under design flow condition, and the comparison between the prediction and the measurement shows that the present analysis method is very suitable for predicting fan noise spectrum.

CONCLUSION

A design-analysis method of Sirocco fan is proposed for constructing 3-D impeller blade and scroll geometry and for predicting both the performance and the noise characteristics of designed fan. Using the present method, 5 fan models with different design specifications are designed, and their performance and noise characteristics are predicted and compared with the measurement within a few percent relative error. Therefore, the present method is expected to be used as a basic design tool for high efficiency and low noise Sirocco fan development.

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REFERENCES

(1) Yamazaki, S. and Satoh, R., 1987, "An Experimental Study on the Aerodynamic Performance of Multi-Blade Blowers(2nd report)," JSME Trans. (B), vol.53 no. 485, pp.108-113

(2) Yamazaki, S. and Satoh, R., 1987, "An Experimental Study on the Aerodynamic Performance of Multi-Blade Blowers(3rd report)," JSME Trans. (B), vol.53 no.490, pp.1730-1735

(3) Ohta, Y., Outa, E. and Tajima, K., 1991, "An Acoustic Model for Prediction of Centrifugal Blower Noise Induced by Blade Discharge Flow around the Scroll Cutoff," Yokohama International Gas Turbine Congress Proceedings, pp. 101-108

(4) Mugridge, B.D., 1976, " Noise Characteristics of Axial and Centrifugal Fans as Used in Industry," Shock and Vibration Digest, vol. 45, no.3

(5) Lowson, M.V., 1993, "Assessment and Prediction Model for Wind Turbine Noise," Flow Solution Report 93/06, pp.1-46

(6) Lee, C., 2010, “Design and aero-acoustic performance analysis method of fan,” High Efficiency and Low Noise Fan Technology Development Corporation workshop proceedings, pp. 71-76

