Mean Streamline Analysis for Performance Prediction of Cross-Flow Fans

Jae Won Kim*
Department of Mechanical Engineering, Sunmoon University, Asan City, Chung Nam, Korea
Hyoung Woo Oh
Department of Mechanical Engineering, Chungju National University, Chungju, Chungbuk, Korea

This paper presents the mean streamline analysis using the empirical loss correlations for performance prediction of cross-flow fans. Comparison of overall performance predictions with test data of a cross-flow fan system with a simplified vortex wall scroll casing and with the published experimental characteristics for a cross-flow fan has been carried out to demonstrate the accuracy of the proposed method. Predicted performance curves by the present mean streamline analysis agree well with experimental data for two different cross-flow fans over the normal operating conditions. The prediction method presented herein can be used efficiently as a tool for the preliminary design and performance analysis of general-purpose cross-flow fans.

Key Words : Cross-Flow Fan, Mean Streamline Analysis, Loss Correlation, Performance Prediction

Nomenclature

A : Cross-sectional area
C_f : Skin friction coefficient
D : Diameter
D_{hyd} : Impeller average hydraulic diameter
g : Gravitational acceleration
H : Total head
L : Longitudinal axial length of impeller
L_b : Impeller flow length
P_{in} : Input power
Q : Volume flow rate
S : Surface area
U : Tangential impeller speed
V : Absolute velocity
$\bar{V}$: Average velocity at measuring station
V_{slip} : Slip velocity
W_{ui} : Tangential component of impeller inlet relative velocity
$\bar{W}$: Average relative velocity
Z : Number of blades
α : Absolute flow angle from tangential direction
β : Relative flow angle from tangential direction
$\beta_{2o,b}$: Impeller exit blade angle at second stage
ΔH : Total head change
Δp_o : Total pressure change
Δp_s : Static pressure change
η_o : Total efficiency $=\Delta p_o Q/P_{in}$
ρ : Fluid density
$\sum\Delta H_L$: Summation of total head loss
ϕ : Flow coefficient $=Q/(U_{2o}D_{2o}L)$
ψ_o : Total pressure coefficient $=\Delta p_o/(0.5\rho U_{2o}^2)$
ψ_s : Static pressure coefficient $=\Delta p_s/(0.5\rho U_{2o}^2)$
ω : Impeller rotational speed

Subscripts

Euler : Euler work
enl : Enlargement
exit : Scroll casing exit

* Corresponding Author,
E-mail : jwk@email.sunmoon.ac.kr
TEL : +82-41-530-2337; **FAX** : +82-41-530-2986
Department of Mechanical Engineering, Sunmoon University, Asan City, Chung Nam, Korea. (Manuscript **Received** July 30, 2003; **Revised** May 5, 2004)

exp	: Expansion
id	: Ideal condition
inc	: Incidence
m	: Meridional direction
sc	: Scroll casing
sf	: Skin friction
th	: Throat of scroll casing
u	: Tangential direction
vsf	: Skin friction in scroll casing
1i	: First stage inlet
1o	: First stage outlet
2i	: Second stage inlet
2o	: Second stage outlet

1. Introduction

The cross-flow fans are not so widely used in industrial applications as axial-flow or centrifugal fans due to their low efficiency. However, their silent operations are particularly suitable in domestic applications such as air-conditioner and air-curtain systems. The cross-flow fan has a drum-type impeller with multiple forward-curved blades and can generate high discharge flow rate by simply increasing the longitudinal axial length of impeller. Fig. 1(a) represents the overall geometry of cross-flow fan running in the market.

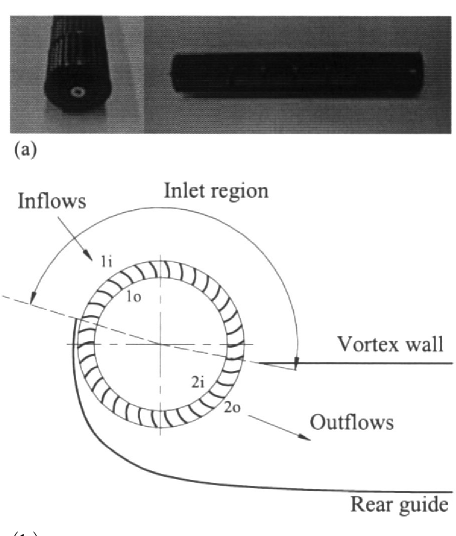

Fig. 1 Overall geometry and flow configuration of a cross-flow fan

Fig. 1(b) shows the two-stage traverse of inflows/outflows and the supplementary structure such as vortex wall (straight line in this figure) and rear guide. As shown in Fig. 1(b), the flows pass between the blades on one side of the impeller, through the internal space of the runner, and then through the blade passages for a second time to discharge on the other side of the impeller.

There have been a number of previous works for various kinds of performance characteristics of cross-flow fans in the open literature (Tanaka, 1993; Tsurusaki et al., 1997; Lazzaretto et al., 1997; Matsuki et al., 1988). Experimental investigations to verify the validity of similarity laws have been made by Tanaka and Murata (1994, 1995). They presented the most detailed analysis on similarity operation for cross-flow fans. It was found that performance curves for cross-flow fans are affected by flow viscosity for the blade Reynolds numbers lower than 10,000 to 15,000. Moreover, they observed that the impeller dimensions also influence fan performance curves. A larger impeller diameter results in an approximately proportional increase in the maximum flow coefficient.

Other studies in the literature (Murata and Nishihara, 1976a, 1976b; Porter and Markland, 1970; Martegani et al., 1999) have demonstrated that fan performance curves are strongly affected by the vortex shape and dimension, which are influenced by the shape and dimension of the casing walls for a given impeller. These findings suggest further studies of the effects on fan performance of Reynolds number and fan dimensions under varied casing geometrical characteristics.

Although the computational fluid dynamics (CFD) procedures have substantially progressed in the analysis of turbomachinery over the past several decades, owing to their expensive and time-consuming resource in calculating the unsteady three-dimensional rotating flow field through a turbomachine, the mean streamline performance analysis using empirical loss correlations has continued to play a key role in the most accurate and practical application of predicting the performance of turbomachinery.

The present study is aimed at finding proper loss correlations for more precise performance prediction of cross-flow fans. The predictions using the empirical loss models suggested here are compared with experimental data obtained from this study and also with the published data in the open literature (Porter and Markland, 1970).

2. Mean Streamline Analysis

The prediction method based on the mean streamline analysis using empirical loss correlations has been developed for performance evaluation of cross-flow fan and its predictive capabilities validated by comparing with experimental data obtained from the fan test rigs.

The measurement system (wind tunnel) that pertains to the ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) Standard (1985) has been utilized for the estimation of performance characteristics.

2.1 Empirical loss correlations

Mean streamline analysis adopted for performance prediction of cross-flow fans in the present study is a one-dimensional analysis, that the flow is considered to be uniformly distributed over the cross-section of flow passage, where only the overall characteristics of the component performance are modeled not with particular emphasis on understanding internal flow phenomena but with taking component losses into consideration.

In order to apply this prediction method to cross-flow fans, the velocity triangles (for the averaged velocity components) at the inlet and outlet parts have to be first determined. The averaged velocity can calculate the meridional velocity components over the impeller flow passage and the tangential velocity components evaluated from the meridional velocity and the impeller rotational speed. While the no-slip tangential flow condition has been employed for computing the tangential velocity components, the impeller exit flow condition, i.e. the tangential velocity component at the position *2o*, can be estimated by using the concept of a slip factor. Fig. 2 shows the typical inlet and outlet velocity triangles (Fig. 2(a) and Fig. 2(b), respectively) of a cross-flow fan.

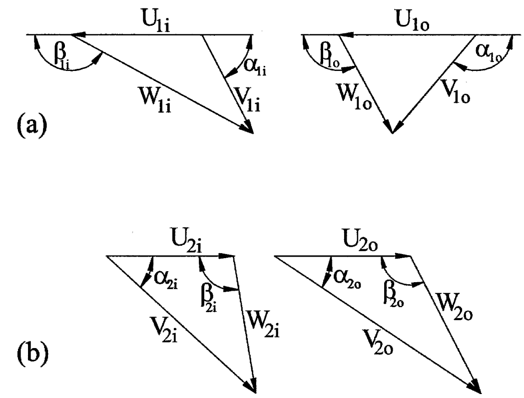

Fig. 2 Velocity triangles of a cross-flow fan

Even though the flow between the impeller blades is assumed to be under ideal frictionless conditions, the relative eddies in the impeller blade passage can make the impeller exit relative flow angle differ from the impeller exit blade angle. A measure of deviation can be calculated using the concept of a slip factor. The present study applies the slip factor formulation of Stodola (Eck, 1973), without any modification (Eq. (1)), to calculate the tangential absolute velocity component at the impeller outlet. Figure 3 shows the impeller exit velocity triangle considering the slip velocity (V_{slip}).

$$V_{slip}=\frac{\pi U_{2o}\sin\beta_{2o,b}}{Z} \qquad (1)$$

Euler input work of cross-flow fans can then be calculated as follows :

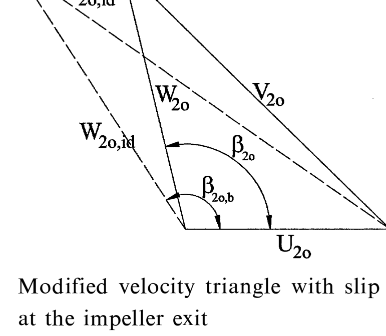

Fig. 3 Modified velocity triangle with slip velocity at the impeller exit

$$H_{Euler}=[U_{2o}V_{u2o}+U_{1i}V_{u1i}]/g \tag{2}$$

where the plus sign is due to the opposite direction of U_{1i} and V_{u1i}. However, in most cases, the performance calculation of cross-flow fans can be achieved with acceptable accuracy under the condition of no inlet prewhirl ($V_{u1i}\cong 0$).

The pressure rise performance of cross-flow fans can be evaluated by the following equations:

$$\Delta p_o=\rho g[H_{Euler}-\sum\Delta H_L] \tag{3}$$

$$\Delta p_s=\Delta p_o-\frac{1}{2}p\overline{V}^2 \tag{4}$$

where $\sum\Delta H_L$ represents the sum of all internal total pressure head losses in cross-flow fan. The present study adopts the recommended loss models for centrifugal pumps (Oh and Chung, 1999) to predict the performance curves of cross-flow fans. The dynamic pressure velocity in this study is an average velocity based on the area of a large plenum chamber on which the static pressure taps are fitted.

Total internal aerodynamic losses consist of losses in the impeller and those in the scroll casing. Many investigators have studied losses generated within the turbomachinery component. Their specific effects and contributions to the entropy increase in turbomachines are given in the literature (Whitfield and Baines, 1990; Denton, 1993). The representative distributions of specific loss mechanisms in turbomachines are given in Thanapandi and Prasad (1990). The loss mechanisms in the impeller are assumed to be made up of incidence and skin friction losses and the scroll casing loss includes expansion, enlargement and skin friction losses.

Incidence loss, ΔH_{inc} The direction of the relative flow no longer coincides with the blade angle as the flow rate is varied. As a result of any variation of incidence at the leading edges of the blades, a loss called an incidence loss arises and can be modeled as follows:

$$\Delta H_{inc}=\frac{W_{ui}^2}{2g} \tag{5}$$

Skin friction loss, ΔH_{sf} The skin friction loss is defined as the total pressure loss due to shear forces exerted on the fluid within the impeller channels. This loss is modeled using pipe flow correlations as

$$\Delta H_{sf}=2C_f\frac{L_b}{D_{hyd}}\frac{\overline{W}^2}{g} \tag{6}$$

Expansion loss, ΔH_{exp} This loss is due to sudden expansion of flows from impeller exit to scroll casing, which can be estimated as the following equation:

$$\Delta H_{exp}=0.75\frac{(V_{u2o}-V_{th})^2+V_{m2o}^2}{2g} \tag{7}$$

Enlargement loss, ΔH_{enl} Enlargement loss from scroll casing throat to fan discharge is usually expressed as

$$\Delta H_{enl}=\frac{(V_{th}-V_{exit})^2}{2g} \tag{8}$$

Skin friction loss in scroll casing, ΔH_{vsf} The skin friction loss in scroll casing is also taken to be equivalent to skin friction loss experienced by a fully developed flow in a pipe. This loss is modified, using experimental coefficient ξ (taken as 0.35 in this study), as follows:

$$\Delta H_{vsf}=\xi C_f\frac{S_{sc}}{A_{th}}\frac{V_{th}^2}{2g},\ \xi=0.2-0.5 \tag{9}$$

2.2 Experimental apparatus

A test setup was designed and built following the ASHRAE Standard (1985). The overall layout of the experimental apparatus is shown in Fig. 4. With such arrangement, the facility could be used to measure the fan performance curves. Basic experimental measurements including fan static pressure, discharge flow rate, and impeller rotational speed are taken in the present study. Wall static pressures in a plenum chamber are averaged in a manifold with four duct taps. The bulk total pressure, delivered total head, is derived from the static pressure measurements and the assumption of uniform velocity at the measuring station. The fan discharge flow rate is

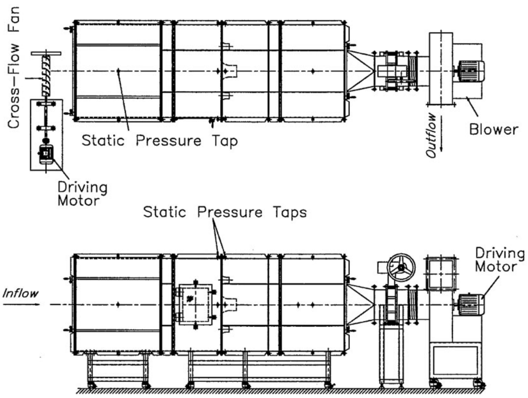

Fig. 4 Schematic diagram of test setup

calculated from the pressure difference across nozzles in a settling chamber. A discharging blower achieves airflow regulation with a butterfly valve in the fan tester. The performance measurements are made at constant speeds of 1, 015rpm through 1,450rpm (with a step of 145rpm) by a direct current electric motor. Uncertainties are evaluated for flow rate and pressure rise. Flow variables for flow rate and pressure rise utilize electric signals from pressure transducers whose accuracy is equal to ±0.1 percent. The overall uncertainty level of flow rate is in the range of ±0.04658 percent to ±0.10691 percent because of the flow rate coefficient for the present venturi meter in the fan tester.

3. Results and Discussion

Performance predictions by the proposed method herein are compared with the test results of two cross-flow fans. Fig. 5 shows a comparison of predictions (denoted by lines) and test data (represented by symbols) for the static pressure rise characteristic curves of the prototype whose impeller is composed of 34-circular arc forward-curved blades with outer diameter 95mm and longitudinal axial length 848mm. It is found that the agreement between calculations based on the mean streamline using the empirical loss correlations and experimental results is quite well over the normal operating flow regions at higher rotational speeds.

Although the present mean streamline (one-

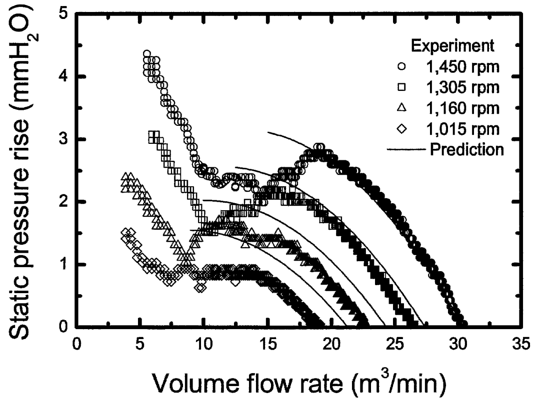

Fig. 5 Comparison of predicted and experimental static pressure rise performances for a cross-flow fan

dimensional) analysis has proved to be inadequate in the low flow rate region, where the flow is strongly unsteady, unstable and three-dimensional, the overall predictive capability provides an indirect confirmation of the accuracy of the employed loss correlations in the normal operating conditions. The calculated distributions of specific losses concerned in this analysis program are presented in Fig. 6 (e.g. operating condition for 1,450rpm). The overall distributions of losses indicate that the scroll casing and discharge losses are major losses over the operating flow region comparing to the impeller losses. As a reference, in Fig. 6, the discharge loss as an additional loss in the present study is defined as the total pressure

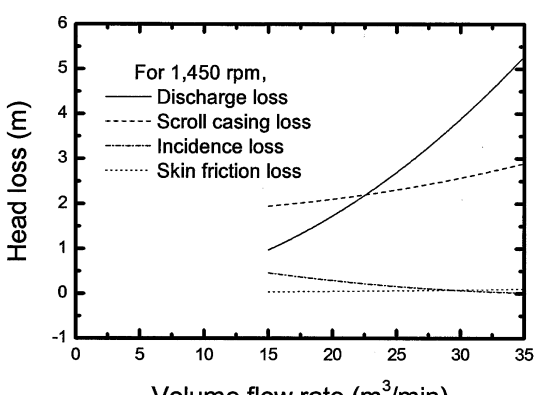

Fig. 6 Loss distributions in a cross-flow fan over the normal operating condition

loss which is attributable to fluid momentum loss through sudden area change from the scroll casing discharge to the large plenum chamber with the static pressure taps fitted and can be expressed as a reduction in the scroll casing discharge dynamic pressure with expansion loss coefficient (Streeter, 1961).

As shown in Fig. 5, the characteristic curve of cross-flow fans shows a dip at low flow rates, i.e. the pressure head first decreases with increasing flow rate to form a local minimum and then increases to reach a local maximum. The eccentric vortex motion inside a rotating fan in this low capacity range causes this wavy variation. The eccentric vortex motion reduces flow rates as an obstacle in a cross-flow fan (Lazzaretto et al., 1997 ; Murata and Nishihara, 1976a, 1976b ; Porter and Markland, 1970; Eck, 1973). This phenomena lead to the noticeable discrepancy between calculations based on the mean streamline analysis and measured data under the condition of low flow rate and rotational speed.

The prediction procedures based on the same numerical scheme have also been tested against Porter and Markland's work (1970) for analysis validation of the present method. A comparison between the calculated and experimental characteristic curves is illustrated in Fig. 7. Although the present analysis method using the empirical loss models suggested in this paper is quite simple, it can be seen from the results in Fig. 7 that the overall agreement of the prediction performance

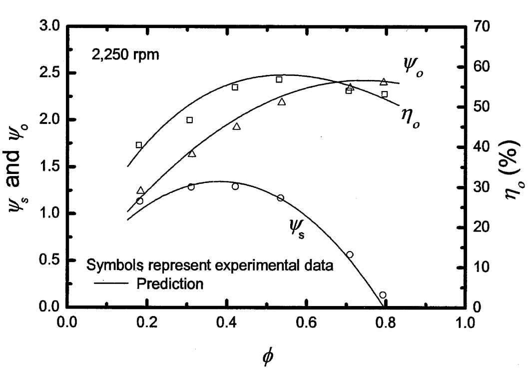

Fig. 7 Comparison of calculated and measured performances for a cross-flow fan (Porter and Markland, 1970)

with the experimental data is fairly good over the operating range.

4. Concluding Remarks

The mean streamline analysis procedure has been utilized for prediction of the performance characteristics of cross-flow fans. The set of loss correlations suggested by this study is found to predict the performance curves of cross-flow fans with acceptable accuracy. Although the present analysis method could not properly handle the low flow characteristics peculiar to cross-flow fans in the low flow region, the predictions agree very satisfactorily with the measured performance curves over the normal operating conditions. The predictive procedure developed throughout this study can be used efficiently as a conceptual design tool in the preliminary design phase of cross-flow fans and also assist the understanding of the operational characteristics of cross-flow fans.

Acknowledgments

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