Performance Prediction of Cross-Flow fans Using Loss Correlations

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ABSTRACT

Cross-flow fans (CFF) are double-stage cooling machines that have been invented in 1892 by Paul Mortier. Despite their early appearance, the CFFs did not get the same success as the axial or centrifugal fans, due to their weak efficiency and longwise form. However, they proved their usefulness in multiple domains such as air conditioners, and aircraft wings. The literature shows that many researchers attempted to predict their performances. Nevertheless, prior studies were limited to a few CFF designs, and the approaches could not be generalized. The main purpose of the present paper is to determine the characteristic curves of the pressure of the Eck CFF. It is a one-dimensional study that investigates several approaches of loss models inspired by those occurring in pumps, axial fans, or compressors, by using velocity triangles of the mean streamline. We will also reveal some parameters and hypotheses not mentioned in the literature. A basic apparatus was employed to collect the experimental results. The setup consists of a cross-flow fan of 24 forward curved blades and a surrounding casing. The Euler head and the pressure rise were extracted and compared with the theoretical results. The pressure coefficients obtained with the loss model corroborate the experience with good accuracy. Hence, the proposed methodology presents a good (first) approach to analyzing the performances of CFFs.

1. INTRODUCTION

The cross-flow fan also called a tangential fan, is an elongated machine with forward curved blades. The blade shape is usually a circular arc. A casing is designed to ensure the separation between the inlet and outlet airflows. The vortex wall and the rear wall are carefully chosen to optimize the performance and to stabilize the vortex inside the rotor. The clearance on both sides is fixed to reduce the noise and to decrease the turbulence in the edges. The main advantage of the CFF is its ability to increase the flow rate by increasing the rotor length, the fan will act in this case as a multitude of fans mounted in series.

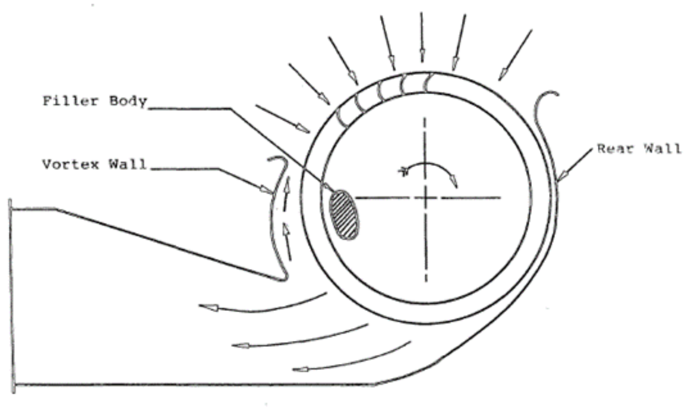

Figure 1. Mortier's cross-flow fan (US Pat. No .507. 455)

Since the appearance of the CFF, many scientists made changes in its design for several reasons; Firstly, they sought performance prediction with more facility; Secondly, they worked on optimizing global fan performance. Many casing shapes were proposed, and some singularities, as filler bodies, were placed within the impeller to arrange the airflow and stabilize the vortex. The experience showed that the changes made in the casing geometry might affect the global performance more than any changes made elsewhere in the rotor might do.

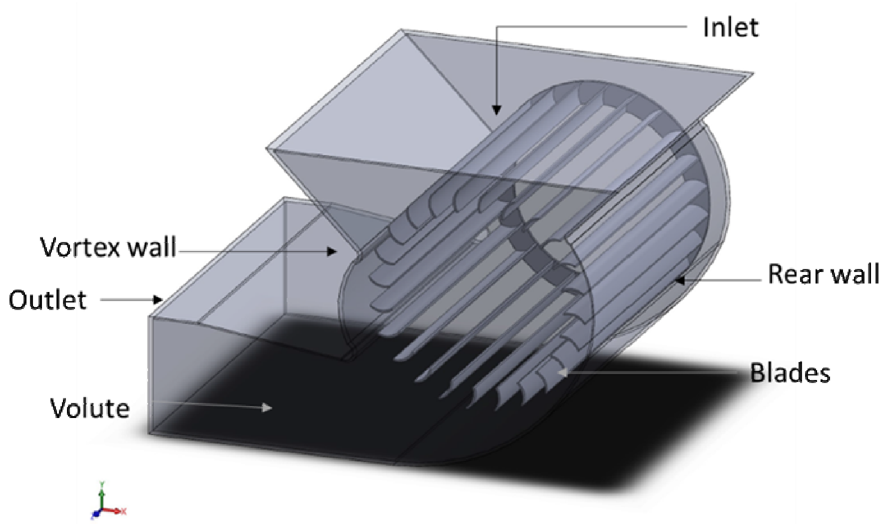

Figure 2. CAD of the Eck fan

The produced vortex inside the machine has been a subject of discussion. As noticed in the experience, that zone of recirculation controls the discharge arc. In the present paper we will suggest a geometrical approximation for the arc of the outlet flow, finally, we will employ suitable loss models in order to obtain the total pressure and the pressure rise specific to the Eck cross-flow fan.

2. THE EXPERIMENTAL SETUP

This study aims to predict the experimental results obtained in the work of Porter and Markland [1]. The Eck/Laing design is the most common CFF used in Europe, the reason why the authors employed it in their experience. In their paper, Porter and Markland gave a precise description of the fan geometry. The rotor has 24 forward curved blades and a length of 228.6 mm, the outer angle of the blade tip is equal to 52°, the inner one is equal to 20°, Rout = 76.2mm and Rin/Rout=0.8. Figure 3. illustrates the details of a cross-section of the rotor.

As the vortex regulates the outflow, we considered that the discharge arc is delimited between the beginning of the logarithmic part of the rear wall, and the tongue of the vortex wall. The net angle of the outflow arc is around 127°. This hypothesis gave satisfactory results, especially since the fan performance highly depends on the ratio between the suction and discharge arcs.

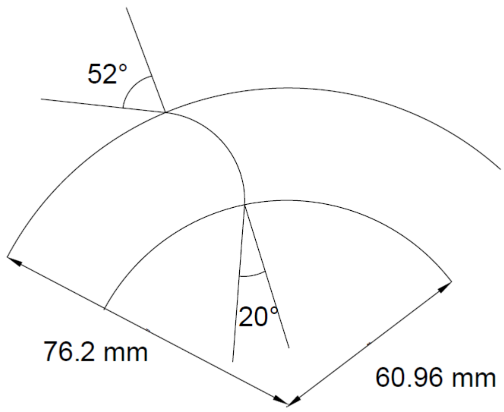

Figure 3. Eck rotor dimensions

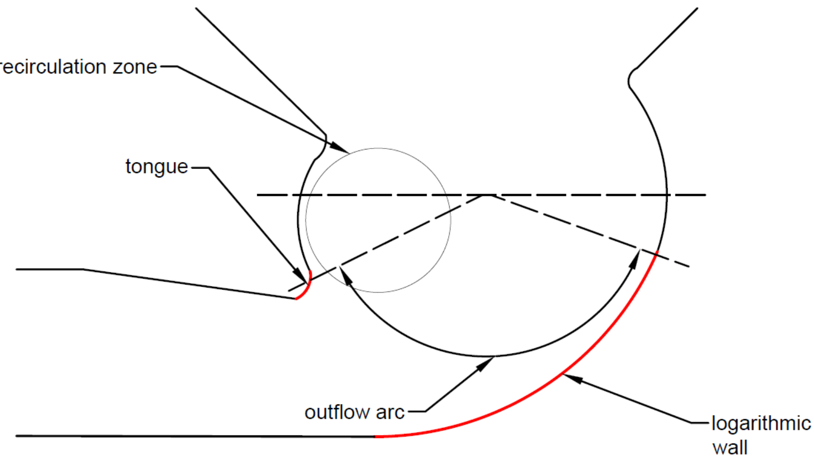

Figure 4. Discharge arc delimitation

3. VELOCITY TRIANGLES OF THE MEAN STREAMLINE

Many hypotheses are assumed in order to analyze the flow field inside the CFF, the air is treated as inviscid and incompressible. The relative velocity of the airflow *w* at the inner and outer rotor peripheries is considered tangent to the blade tip i.e., the air glides upon the blades without any deflection, e.g., β_1= 52° and β_2= 20°. *r* and *u* stand respectively for the radial and tangential components.

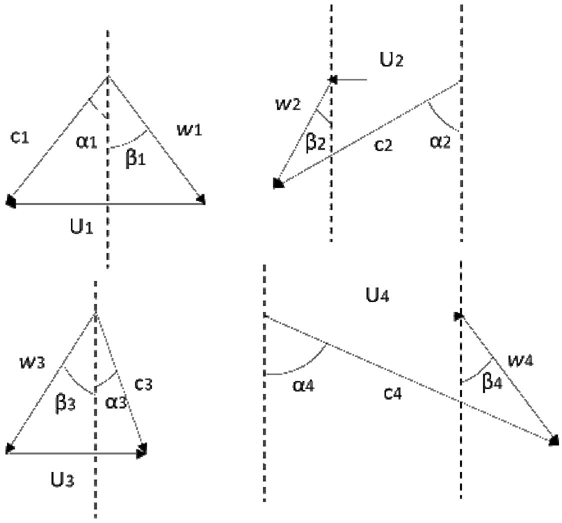

Figure 5. Velocity triangles in the CFF

The velocity components are determined as follows,

$$cr_1 = \frac{Q_v}{s_1\,L} \tag{1}$$

$$w_1 = \frac{cr_1}{\cos\beta_1} \tag{2}$$

$$wu_1 = \tan\beta_1 * cr_1 \tag{3}$$

$$cu_1 = U_1 - wu_1 \tag{4}$$

$$\alpha_1 = \mathrm{atan}\frac{cu_1}{cr_1} \tag{5}$$

$$c_1 = \sqrt{cu_1^2 + cr_1^2} \tag{6}$$

$$cr_2 = \frac{R_{out}}{R_{in}}\,cr_1 \tag{7}$$

$$cr_3 = \frac{s_4}{s_1}\frac{R_{out}}{R_{in}}\,cr_1 \tag{8}$$

$$cr_4 = \frac{s_4}{s_1}\,cr_1 \tag{9}$$

4. LOSS CORRELATIONS

The losses inside the CFF are inspired from centrifugal pumps and compressors correlations, owing to the similarities that share the fan with these turbomachines. The models presented in the paper of Oh and Yoon [3] and the paper of Oh and Chung [2] are employed in this work to predict the head losses inside the CFF.

4.1. Incidence loss

$$\Delta H_{inc} = 0.7\,\frac{wu_1^2}{2g} \tag{10}$$

4.2. Blades skin friction loss

$$\Delta H_{sf} = \frac{2C_f}{g}\frac{cu_{vol}}{D_{hyd}}\overline{\omega}^2 \tag{11}$$

$$\overline{\omega} = \frac{w_1 + w_{mid} + 2\,w_2}{4}$$

4.3. Enlargement loss

$$\Delta H_{enl} = \frac{(cr_{vol} - c_4)^2}{2g} \tag{12}$$

4.4. Expansion loss

$$\Delta H_{exp} = 0.75\,\frac{(cu_{vol} - c_4)^2}{2g} \tag{13}$$

4.5. Volute skin friction loss

$$\Delta H_{vol,sf} = 0.35\,C_{fv}\,\frac{S_{vol}}{A_{vol}}\,\frac{cr_{vol}^2}{2g} \tag{14}$$

5. RESULTS AND DISCUSSION

For this study, it was of interest to investigate the pressure coefficients, as shown in equations (17) and (18) as a function of the flow coefficient (equation (15)), using the calculated velocities and losses explained in the previous sections.

$$\varphi = \frac{Q_v}{D_{out}U_{out}L} = \frac{cr_1 s_1}{2\,R_{out}U_{out}} \tag{15}$$

$$\psi_{Euler} = \frac{(U_{in}cu_2 - U_{out}cu_1) + (U_{out}cu_4 - U_{in}cu_3)}{\frac{1}{2}\rho\,U_{0ut}^2} \tag{16}$$

$$\psi_{tot} = \psi_{Euler} - \frac{\sum H_{loss}}{\frac{1}{2}\,\rho\,U_{0ut}^2} \tag{17}$$

$$\psi_{rise} = \psi_{tot} - (\frac{c_{vol}}{U_{0ut}})^2 \tag{18}$$

Figure 6. illustrates the losses occurring within the fan. These findings prove that the incidence loss is the most dominant among the others, it is due to the accentuated curvature of the blade tip at the outlet periphery. On the other hand, figure 7. Reveals that the predicted curves of the total pressure and the pressure rise, coming from our loss models, are in good agreement with the data collected from the experience.

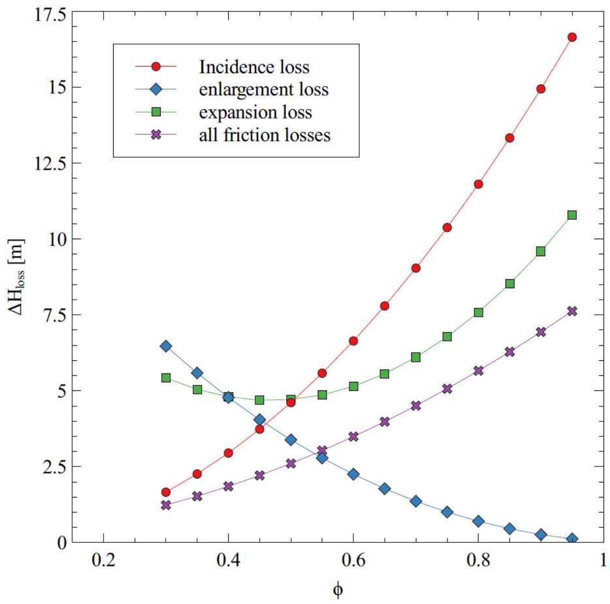

Figure 6. Predicted losses through the CFF for 2250 rpm

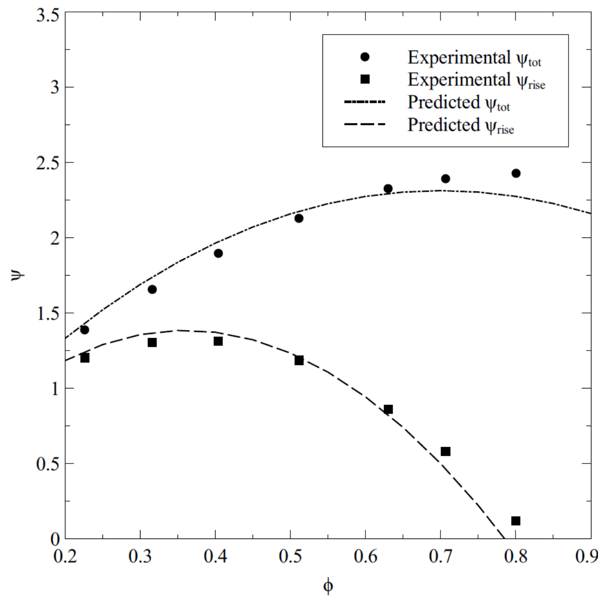

Figure 7. Comparison of the predicted and experimental pressure coefficients for 2250 rpm

6. CONCLUSIONS

As many investigations failed to determine the pressure curves for the Eck cross-flow fan, the most satisfactory conclusion that we can come to is that the loss model we suggested, applied to this kind of fan represents a good alternative when it comes to predicting the curves of performance characteristics. As can be seen, the theoretical results confirm the experience fulfilled by Porter and Markland with acceptable precision, and this in normal operating conditions. The main advantage of the streamline analysis compared to previous studies is the simplicity of determining the components of the correlations.

Hence, this method casts a new light on predicting the pressure curves of any tangential fan and improve its design.

SYMBOLS

Q_v	Volumetric flow rate
α	Flow angle
β	Relative angle
φ	Flow coefficient
ψ	Pressure coefficient
c	Absolute velocity
w	Relative velocity
U	Rotational speed
r	Radial component
u	Tangential component
L	Rotor length
R	Rotor radius
s	Passage arc
S	Volute internal surface
A	Passage area
H	Head
ρ	Air density

Subscripts

in	inner
out	outer
1	Suction stage inlet
2	Suction stage outlet
3	Discharge stage inlet
4	Discharge stage outlet
mid	Blade midpoint
vol	Volute
hyd	Hydraulic
tot	total
exp	Experimental

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