Noise prediction on the design of optimized fan by numerical simulation

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Abstract. The design of fan blade occasionally should be optimized in order to get maximum flow capacity of the air. The result of a blade design constructs to the fan with six components blade. During the operation, the fan generated noise because of turbulence flow around the rotor dynamics system. On the process optimized fan blade, we predicted that the noise was radiated to the environment by numerical simulation. The fan during operation magnitude of Sound Power Level (SPL) has various about 74 dB to 93 dB.

1. Introduction
Generating wind by a fan as well as an exhaust or a blower with certain specifications such as flow capacity per unit of time is very important in order to achieve the desired objectives. It is associated with a system known as the Heating Ventilation and Air Conditioning (HVAC). In order to develop HVAC, one of the activities is to design the impeller for the purpose of moving fluid from one place to another, although today many different types of fan can be found in the marketplace.

On the development of testing facility in the BBTA3, an open type wind tunnel test section with the size of 2m×1.5 m with a length of about 3 m designed with a maximum speed of 30 ms⁻¹ was built. It requires a fan with a diameter of almost 3 meters with a flow capacity of 90 cubic meters per second. In order to meet these needs, the authors designed a fan along with an optimization to obtain the maximum flow capacity.

However, the optimum fan that has been designed still generated noise [1][2]. Here we knew as a noise source to the environment [3][4][5][6][7][8][9]. How high is the level of noise generated need attention? The noise level of fan than was predicted by numerical simulations, and then the result was plotted against frequency spectrum of the power level. Furthermore, the optimized fan design is installed on the open type wind tunnel. During the operation, the open type wind tunnel in which the fan operated at a certain rotation per minute generated sound as a noise to the environment.

The open type Wind Tunnel Test (WTT) facility shown in figure 1 is part of testing facility belonging to BBTA3-BPPT. BBTA3 has also anechoic chamber test facility to support the noise testing and in the near future will be equipped as echoic wind tunnel facility.

2. Optimization on the fan of the open wind tunnel
Generally, a capacity of wind speed or flow generated by the fan blade is directly proportional to the radius of the fan, the input power and the total efficiency of the fan motor. The total efficiency of the
fan blade is an important factor in increasing the efficiency of fan motors. The diameter of the six-bladed fan design is 2.8258 meter.

**Figure 1.** Open type WTT facility.

**Figure 2.** RAF D airfoil geometry.

### 2.1. The design of the fan blade geometry

The fan blade geometry which has an efficient performance is the one with a maximal lift to drag ratio (L/D). Furthermore, with that consideration, the blade was designed using a geometry that has high L/D, i.e. a big RAF airfoil D geometry that has an L/D of 80.

Various studies show that a fan has a good efficiency when using blade geometry formed by an airfoil, which has a high L/D ratio. These can be achieved by using RAF airfoil D geometry shown in figure 2. RAF D airfoil’s aerodynamic characteristics vary with the Reynolds number, while the working area was defined at a speed of 30 m s\(^{-1}\), in which the Reynolds number is 2.0x10\(^6\). At the Reynolds number of 2.0x10\(^6\) aerodynamic characteristics of the airfoil RAF D is shown in figure 3 and figure 4.

Furthermore, the geometry design of the fan is shown in figure 5 and the discretization of space fluid flow into the computing space is shown in figure 6. Figure 5 is a physical space discretization
into a space computing to calculate characteristic aerodynamics related to the number of coordinates. The simulation result from the performance of the fan is shown in table 1.

Figure 3. The lift coefficients versus the angle of attack.

Figure 4. The lift coefficient versus drag coefficient.

2.2. Optimization of fan blade geometry
In the next stage the performance needs to be optimized by optimizing the fan blade parameters that have been set.

Figure 5. Fan geometry design.

Figure 6. Discretization computational space.

Table 1. Simulation result.

| Parameter                           | Value    |
|-------------------------------------|----------|
| Rotation Speed                      | 26.1799  |
| Reference Diameter                  | 2.8258   |
| Volume Flow Rate                    | 87.6038  |
| Pressure Rise (IN-OUT)              | 302.0470 |
| Flow Coefficient                    | 0.3365   |
| Head Coefficient (IN-OUT)           | 0.0804   |
| Shaft Power                         | 55729.0000 |
| Power Coefficient                   | 0.0570   |
| Total Efficiency (IN-OUT) %         | 47.4806  |
| Static Efficiency (IN-OUT) %        | -43.1135 |
Fan blade parameters which influence the performance of the fan are the angle of the blade and the size of the chord. In this study, optimization method was done by using the following formulas:

\[ c = \frac{8\pi r}{BCL_d} (1 - \cos \varphi) = \frac{8\pi r}{BCL_d} (1 - \cos \varphi) \quad (1) \]

Blade setting angle:
\[ \beta = \varphi - \alpha = \varphi - \alpha \quad (2) \]

Flow angle:
\[ \varphi = \frac{2}{3} \arctan \frac{1}{\lambda_r} = \frac{2}{3} \arctan \frac{1}{\lambda_r} \quad (3) \]

Local design speed:
\[ \lambda_{rd} = \lambda_r \times \frac{r}{d} \quad (4) \]

Important parameters such as the number of blades \( B = 6 \), \( Cl_d = 0.86 \) and \( r = 1.42925 \text{ m} \) and \( \lambda_d = 2 \) are the input of calculation. Before optimization, the flow capacity of the fan is \( 87.6053 \text{ m}^3\text{s}^{-1} \) with power of \( 55.73 \text{ kW} \), which then increases to \( 91.7180 \text{ m}^3\text{s}^{-1} \) with power of \( 46.76 \text{ kW} \) after optimization.

3. The noise of optimized geometry fan

On the process of optimizing the fan blade using commercial CFD software, the noise of the fan could also be obtained. The prediction was done by using the Ffowcs-Williams and Hawkings (FW-H) equation \([10][11][12][13]\) that based on Lighthill’s acoustic analogy. The FW-H equation can be written as follows:

\[ \frac{1}{a_0^2} \frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = \frac{\partial^2}{\partial x_i \partial x_j} \left\{ \mathbf{T}_{ij} H(f) \right\} - \frac{\partial}{\partial x_i} \left\{ \rho P_{ij} n_j + \rho u_i (u_n - v_n) \delta (f) \right\} + \frac{\partial}{\partial t} \left\{ \rho_0 v_n + \rho (u_n - v_n) \right\} \delta (f) \]

where \( u_i = \text{fluid velocity component in the } x_i \text{ direction} \)

\( v_i = \text{surface velocity components in the } x_i \text{ direction} \)

\( n_i = \text{unit normal vector pointing toward the exterior region } u > 0 \), \( a_0 \) is the far-field sound speed, and \( T_i \) is the Lighthill stress tensor \([14]\). Furthermore this analysis supplied to assist in the evaluation of tonal noise levels generated by low-speed fans (Mach number less than 0.4). Noise input data are shown in table 2.

### Table 2. Noise input data.

| Number of Blades | 6 |
|------------------|---|
| Angular Velocity | 26.1799 [radian s⁻¹] |
| Number of Harmonics | 6 |
| Observer Location (radius) | 1.0000 [m] |
| Observer Location (theta) | 0.0000 [degree] |
| Loading Coefficient | 2.2000 |
| Reference Pressure | 2.0000e-05 [Pa] |
| Reference Power | 1.0000e-11 [W m⁻³] |
| Rotational Mach Number | 0.0826 |
4. Results and Discussions
The geometrical optimization of fan blades has been done. Based on the simulation result, the flow capacity increased and the power consumption was reduced. As far as we knew, the axial fan is one of noise sources to the environment, especially if the size of the fan of the wind tunnel is big enough. From the points of view of safety and comfort during operation, the noise of the fan should be studied.

4.1. Sound Pressure Level
Table 5 shows the noise prediction as Sound Pressure Level (SPL) at six harmonics frequency, called Blade Pass Frequency (BPF). SPL can be expressed as a logarithm of spectrum $S(f)$:

$$ SPL(f) = \log_{10} S(f) = \log_{10} S(f) $$

(6)

where $S(f)$ is computed using a fast Fourier transform (FFT) algorithm.

The plot of the SPL versus frequency is shown in figure 7. BPF knows the highest noise of the axial fan. A noise prediction at this BPF is very important.

| Harmonic | Frequency [Hz] | Sound Pressure Level - Lp [dB] |
|----------|----------------|--------------------------------|
| 1        | 25,000         | 91.2921                        |
| 2        | 50,000         | 84.0674                        |
| 3        | 74,9999        | 79.8412                        |
| 4        | 99,999         | 76.8426                        |
| 5        | 124,9999       | 74.5168                        |
| 6        | 149,9999       | 72.6164                        |

4.2. Sound Power Level
Although Sound Pressure Level and Sound Power Level have the same unity, i.e. dB, essentially they are slightly different: they are based on pressure and power, respectively. Table 4 shows the noise prediction as Sound Power Level (SPL) at six harmonics frequency, usually called Blade Pass Frequency (BPF). The SPL will decrease if the harmonics frequency increase. Furthermore, the plot of the SPL versus frequency is shown in figure 8. BPF knows the highest noise of the axial fan.
4.3. Directivity
The direction of sound propagation can be seen in figure 9 as SPL versus angle theta (in radian). The highest SPL occurs at theta 0, π and 2π and the lowest occurs at theta 0.5π and 1.5π.

4.4. Overall Noise
The total energy contained in the spectrum express as;

$$E = \int_0^\infty S(f)df$$  \hspace{1cm} (7)

Where we integrate over all resolved frequencies. The Overall Sound Pressure/Power Level (OASPL) is

$$OASPL = \log_{10} E_{ASPL} = \log_{10} E$$  \hspace{1cm} (8)

The OASPL calculation result is shown in table 5.

4.5. Broadband Noise
The broadband noise model is derived from Proudman’s formula (see expression Proudman Sound Power in equation (8)) which predicts overall sound power. Associated variable (Proudman Sound Power) is evaluated on the entire domain, allowing visualization of isosurfaces that can be used to locate the portion of the flow that is responsible for noise generation [15].

$$P_A = \alpha \rho_0 \left( \frac{u^2}{T} \right) \left( \frac{u}{a_0} \right)^5$$  \hspace{1cm} (9)

The value of broadband noise power in dB is calculated by this equation:

$$L_P = \log_{10} \left( \frac{P_A}{P_{ref}} \right)$$  \hspace{1cm} (10)

where $P_{ref}$ is the reference acoustic power ($P_{ref}=10^{-12}$W m$^{-3}$). Broadband noise is shown in table 6 and figures 10.

| Table 4. Sound Power Level. |
|----------------------------|
| No | BPF [Hz] | Sound Power Level - $L_P$ [dB] |
|----|----------|-------------------------------|
| 1  | 25.0000  | 92.2528                       |
| 2  | 50.0000  | 85.0281                       |
| 3  | 74.9999  | 80.8019                       |
| 4  | 99.9999  | 77.8034                       |
| 5  | 124.9999 | 75.4775                       |
| 6  | 149.9999 | 73.5772                       |

| Table 5. Overall noise. |
|-------------------------|
| Sound Pressure Level [dB] | 92.5356 |
| Sound Power Level [dB]    | 93.4963 |
4.6. Noise Source
Monopole, Dipole, and Quadrupole noise sources, derived from Ffowcs Williams and Hawkings (FW-H) equations. These sources can be compared with each other and with the broadband noise to determine the dominant noise source in the design [15].

Table 7, Table 8, figure 11, figure 12 and figure 13 show the kinds of noise source.

| Tabel 6. Broadband noise. |
|---------------------------|
| Minimum       | 0.0 [dB] |
| Maximum       | 93.7 [dB] |
| Average       | 17.9 [dB] |
| Total Power   | 1.9674e-07 [W] |

Figure 10. Isosurface at 95% of Proudman Sound Power.

5. Conclusions
The design of the impeller both axial and centrifugal needs to be optimized in order to obtain the maximum flow capacity and efficiency. Although it is known that the form of the fan impeller is a source of noise, it is important to know the noise level, whether it is still within the range of threshold or it needs to be specially treated so it does not affect the environment.
The axial fan is very widely used in the HVAC system. The HVAC system certainly requires an efficient fan with a tolerable noise level. In this study, a six-blade fan with the diameter of 2.8258 m which is suitable with the flow capacity of an open type wind tunnel was designed. The result of the simulation indicates noise levels that can be tolerated by the SPL range from 74 dB to 93 dB.

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