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# Optimization of sirocco fan blade to reduce noise of air purifier using a metamodel and evolutionary algorithm

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ABSTRACT

In this study, the flow inside an air purifier was visualized using computational fluid dynamics (CFD), and the noise caused by airflow was predicted using computational aero-acoustics (CAA). With the obtained results, the causes of blade passage frequency (BPF) noise and turbulent flow noise that dominate the operating noise of the air purifier were investigated. The relationship between design parameters and BPF noise was subsequently derived by the kriging metamodel, and the optimum fan blade dimensions that minimize operating noise were acquired by an evolutionary algorithm (EA). The relationship between BPF noise and design parameters mentioned was intended to be used as a design guide during the early stage of fan blade design. In addition, in this study was designed to be able to estimate the origin of noise intuitively by visualizing internal flow. Lastly, the low-noise performance of the air purifier was verified through analytical and experimental methods, including the fabrication and testing of a physical mockup. Compared to the initial model, the fan blade operating noise decreased by 4.5 dB(A) in the main range.



## 1. Introduction

In the past, air purifier designs focused on purifying performance only. As the technology gap among companies has narrowed, companies have begun to concentrate on other aspects such as dehumidifying and humidifying functions, design, material, finish, and noise. In particular, due to increasing interest in and consumer complaints about operating noise, there have been many efforts to reduce noise. Various methods to reduce noise have been studied, including optimization of duct structure to minimize flow resistance and active noise control, as well as conventional passive methods such as sound absorption. However, these methods use additional materials, increasing manufacturing cost and product weight. Accordingly, to meet consumer needs, there have been suggestions to control the source of the noise itself.

Recently, computational aero-acoustics (CAA) has become more useful as a means to estimate noise through computational fluid dynamics (CFD)-based 3D numerical analysis. Numerical CFD analysis enables 3D simulation of the flow inside the air purifier, including estimation of flow speed and noise distribution [1–4]. Additionally, there have been many studies [5–8] that used design of experiments (DOE), which maximizes the information gained through minimal testing by appropriately posting test points, as well as design optimization (DO), which aims to obtain an optimum solution by combining and approximating the estimated noise in each test point. Guo and Kim [9] identified the characteristics of internal flow through numerical analysis by applying the steady and unsteady 3D Reynolds-Averaged Navier–Stokes (RANS) equations to sirocco fans. Younsi et al. [10] analyzed the characteristics of the flow and noise arising from interactions between the impeller and the scroll of sirocco fans by using unsteady 3D RANS analysis.

As the first theoretical study on aerodynamic noise, Lighthill [11] derived a basic wave equation on jet noise, and Curle [12] made it possible to calculate aerodynamic noise when a still object exists. Furthermore, Ffowcs Williams and Hawkings [13] completed the Ffowcs Williams–Hawkings (FW–H) equation to enable the calculation of aerodynamic noise even when an object is still moving, for example, with rotational movement. Choi et al. [14] improved performance by determining an optimum scroll design; this was achieved by performing a steady flow numerical analysis

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on the scroll-related variables of the sirocco fan using DOE. Kim and Seo [15] enhanced efficiency for the reference shape of the sirocco fan through a numerical optimization design that combined the response surface method and RANS analysis. The kriging metamodel, a type of interpolation model, was used herein as a metamodel to optimize the blade, which can in turn minimize noise using the approximate virtual analyzer metamodel and DO. An evolutionary algorithm (EA) has been used as a DO algorithm.

The kriging metamodel was applied to the engineering sector as a model for computational experiments by Sacks [16]. In terms of estimation performance, it was superior to other models in predicting the performance of systems with strong nonlinearity and many design parameters. The EA was proposed by Rechenberg [17] and Schwefel [18,19] in the 1960s as a design optimization method; it mimics natural genetic variation and uses the principle of survival of the fittest, sequentially applying processes of selection, recombination, and mutation. Through these previous studies, the efficiency of CFD, CAA, experimental design, and DO has been evaluated in various fields. However, it has not been easy to find a study on the reduction of air purifier noise. To meet customers' needs for reduced noise while also minimizing development costs, it is important to develop an aerodynamic noise reduction process that applies 3D numerical analysis, kriging metamodel, and DO simultaneously.

In the present work, the relationship was derived between the design parameters that determine the fan blade shape and the resulting noise; the optimized blade dimensions to reduce noise were then derived using this relationship. To accomplish this, the flow inside the air purifier was visualized through CFD, and the sound pressure at a nominated location was estimated using CAA. Based on the values obtained, the two noise types that dominate air purifier operating noise (namely, BPF noise and turbulent flow noise) were elucidated. A kriging metamodel was created using the design parameters and BPF noise based on these findings. The optimum fan blade dimensions that minimize operating noise were subsequently acquired by coupling the kriging metamodel and EA. The relationship between BPF noise and design parameters reported herein is intended for use as a design guide in the early stage of designing fan blades. In addition, it is designed to provide an intuitive way to estimate the origin of noise by visualizing internal flow. Lastly, the optimal dimensions obtained by this process were confirmed to result in low-noise air purifier performance through analytical and experimental methods.

## 2. Measurement of the air purifier noise for the initial model and identification of the noise source

An air purifier can be divided into the upper duct, lower duct, and sirocco fan (Fig. 1). The central motor is fixed to the lower duct, while the sirocco fan is anchored to the rotating shaft. The lower and upper ducts are fixed to the upper and lower housings, respectively, and these housings are tightened with bolts. Regarding the air flow path, air is supplied through a filter in the inlet by the rotation of the sirocco fan. After rotating along the surface of the fan blade, air is discharged toward the outlet along the duct walls. The elements of the air purifier were arranged to measure noise, as shown in Fig. 2. Far-field noise was measured 1 m away from the front of the air purifier. A test was performed in an anechoic room to simultaneously measure free-field noise. To minimize the local characteristics of sound pressure, the sound pressure was measured for 10 s and then linearly averaged. After repeating the measurement 10 times, the arithmetic mean was calculated. For noise analysis, B&K's pulse analyzer was used. To perform frequency analysis on the sound source as measured using a microphone, the time-domain data was converted into frequency-domain data. After

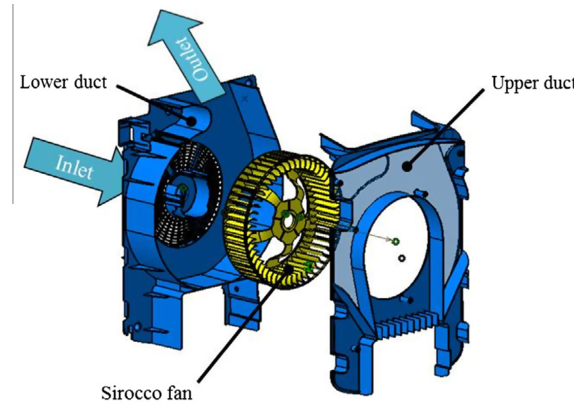


**Fig. 1.** Configuration of the air purifier.

setting the sampling time to 0.078 ms, values from 1 to 6400 Hz were analyzed for every 1-Hz interval. An A-weighting network, which had the best correlation with the amount of subjective sense, was applied to the frequency-converted data. Several constant percentage bandwidth (CPB) methods are available for analyzing the noise in the auditory perceptive region; herein, a 1/6-octave analyzer was used. The 1/3-octave method is probably the most widely used; however, this method analyzes frequency ranges that are too broad to evaluate the effect of narrow-band BPF noise. Therefore, the 1/6-octave analyzer was used.

The noise generated by the rotating fan blade can be divided into broadband noise and narrowband noise (or discrete frequency noise). Fig. 3 provides the specifications of the sirocco fan and shows the blade passage frequency (BPF) that was detected during operation. As shown in this figure, the sirocco fan generates two different BPFs. Five ribs connecting the central unit to the motor and the outer blades generate a lower BPF, while the 51 fan blades generate a higher BPF on the outside. These BPFs produce harmonic elements in proportion to their BPF constants. To determine the effect of each BPF and its harmonic elements on the overall noise and the noise performance of an initial model, noise was measured in a test, and frequency was analyzed using a fast Fourier transform (FFT).

Fig. 4 shows the results of the frequency analysis on the measured noise in the initial model. It was confirmed that the noise was composed of small and large discrete noises generated by BPF noise and its harmonic elements, and broadband noise, which connects discrete noise. In the low-frequency region, lower BPF noise, attributed to the ribs, dominated. Closer to the high-frequency region, higher BPF noise, attributed to the fan blades, and broadband noise began to dominate. In terms of BPF noise generated by the ribs, the 2nd, 4th, and 5th harmonic elements (proportional to the constant) were present, starting at 67.5 Hz. These harmonic elements were relatively small, however, and had little effect on overall noise.

The maximum discrete noise was observed at 688.5 Hz, which is identical to the fan blade BPF and was therefore attributed to fan blade BPF noise. According to the 1/6 octave analysis, the major frequency band that includes fan blade noise was the highest in terms of sound pressure, and the sound pressure of the approximate rear band dominated the overall noise. This result was attributed to turbulent flow noise when BPF noise was created, and was associated with changes in status of the adjacent flow by the fan blade rotation. It was confirmed that total sound pressure from 649.8 Hz (the lower frequency of the 1/6 octave band with central frequency of 688.5 Hz) to 1151.9 Hz (the upper frequency of the four adjacent upper bands) was 58.8 dB(A), while the total sound pressure in other regions was 58.6 dB(A). It can thus be concluded