Analysis on hub vortex and the improvement of hub shape for noise reduction in an axial flow fan

Jung, Jae Hyuk[1]
Air Solution Lab., LG Electronics Co. Ltd.
School of Mechanical Engineering, Yonsei University
51 Gasan-digital-1-ro, Geumcheon-gu, Seoul, Republic of Korea (South Korea)

Joo, Won-Gu[2]
School of Mechanical Engineering, Yonsei University
50 Yonsei-ro Seodaemun-gu, Seoul, 03722, Republic of Korea (South Korea)

ABSTRACT

This study aimed to reduce the flow induced noise in an axial flow fan installed in the outdoor unit of an air conditioner. This study found the reduced fan efficiency due to a vortex produced by the flow separated from the edge of the hub entrance via affecting the fan blade. The vortex increased the turbulent intensity in the fan's wake, which is a major noise source of the fan. This study proposes a new shape for a hub to reduce the hub vortex. The size of the new hub is extremely small, which is 13.6% of the size of the existing hub. The axial fan with the new hub was named "Hubless fan" in this study. Computational fluid dynamics (CFD) analysis on the flow field revealed that the vortex was significantly reduced in the Hubless fan and this resulted in reduction of the turbulent intensity and vorticity in the wake. By measuring the noise of the Hubless fan with microphones in an anechoic room for experimental verification, reduction of the broadband noise at the certain frequencies on the noise spectrum and reduction of the overall sound pressure level by 1.2 dB compared to existing fans were confirmed.

Keywords: Axial flow fan, Noise, Air conditioner
I-INCE Classification of Subject Number: 30
http://i-ince.org/files/data/classification.pdf

1. INTRODUCTION

Axial flow fans are one of the fluid machinery that are the most commonly used in the field of electronics. In the field of air conditioning, in particular, an axial flow fan is used to cool down a condenser in the outdoor unit of an air conditioner. Numerous strudies on an axial flow fan for outdoor units are ongoing in order to improve the fan performance and efficiency and to reduce the noise. With the development of the experimental techniques, computational fluid dynamics, and computiotional aeroacoustics, researches after 2000s have involved detailes of the flow field and analysis of the noise source. Amongst all, the tip leakage vortex, which is one of the

[1] jaeh@yonsei.ac.kr
[2] joo_wg@yonsei.ac.kr

major noise sources in an axial flow fan, has been studied extensively. Jang el al. studied the tip leakage vortex by using laser doppler velocimetry (LDV) measurement and large eddy simulation (LES) and proved that the periodical movement of the tip leakage vortex was related to the blade passing frequency (BPF) noise[1,2]. While studying the tip leakage flow, the authors found a strong pressure fluctuation occuring near the hub of the leading edge and they determined it as an influence of the tip leakage vortex as well.

The tip leakage vortex and flow noise of an axial flow fan for the outdoor units of an air conditioner have been studied recently through experimens and numerical aalysis by a number of researchers. Wang et al. used Reynolds-averaged Navier–Stokes (RANS) model to study the changes in shape of the tip leakage vortex and efficiency of the fan according to the shape of the bell-mouth (shroud) shape in an axial flow fan of the up-flow outdoor unit; however, consequent changes in flow noise were not analysed in the study[3]. Tian el al. conducted experimental and numerical research on the aerodynamic performance and noise according to the shape of the discharge grille of the top dischrging outdoor unit of an air conditioner[4]. The authors analysed the mechanism in which the shape of a grille influences the flow rate and broadband noise by using the vortex shedding noise model. Zhao el al. adopted numerical analysis to calculate the flow noise depending on the presence and absence of the winglet in an axial flow fan for the outdoor unit presenting the similar front discharge with our study[5]. The authors confirmed noise reduction by approximately 2.1 dB(A) with the presence of the winglet; however, how the winglet changed the tip leakage flow and how the changed tip leakage flow reduced the noise were not analyzed in that study.

In our previous study, the relationship between the changes in the tip leakage vortex according to the major shape of the axial flow fan and fan efficiency was studied[6]. In addition, the influence of the vortex occurring in the entrance side of the hub on the end-wall loss on the hub side and fan efficiency was studied in an axial flow fan with the hub in general shape, which has the entrance and exit surface in blunt cylindrical shape. In the study, the hub vortex, which was periodically shed from the hub edge, interfering the leading edge of the blade was found and such movement of the vortex was revealed to affect the fan efficiency[7]. Therefore, designing the fan hub with consideration for the production and path of the vortex is recognized to be one of the pivotal elements for designing a high-efficiency and low-noise fan. Nevertheless, there are only few researches studied on the relationship between the vortex movement according to the shape of the hub in an axial fan and the flow induced noise. In this study, a fan with improved hub shape was designed in order to improve the fan efficiency by minimizing the occurrence of the hub vortex and influence on the blade and to reduce the noise; and the performance of the fan was numerically and experimentally examined.

2. APPROACH

2.1 The model fan

The detailed geometry of the axial flow fan is shown in Figure 1. The height of the shroud was 30% of the fan's axial length and the shroud was divided into round-shaped inlet, straight line part, and diffuser part of the outlet. In the blade tip of the axial flow fan, swept-back tip winglet was applied. The tip clearance was 3.2% of the fan's radius. The fan's hub was in cylindrical shape and its diameter was approximately 40.5% of the fan's diameter. The height of the hub is determined by how far the hub entrance and hub exit are apart from the blades' leading edge and trailing edge, respectively. In this study, the exit hub length (the distance between the hub exit and the trailing edge) was fixed as 5.4% of the fan's radius whereas the entrance hub length (the distance between the hub entrance and the leading edge) was tested within the range between 0% and 21.6% of the fan's radius. The edge of the hub entrance can be rounded in order to suppress the occurrence of the vortex in the hub; and the radius of the rounded edge of the hub entrance was tested within the range between 0% and 2.16% of the fan's radius.

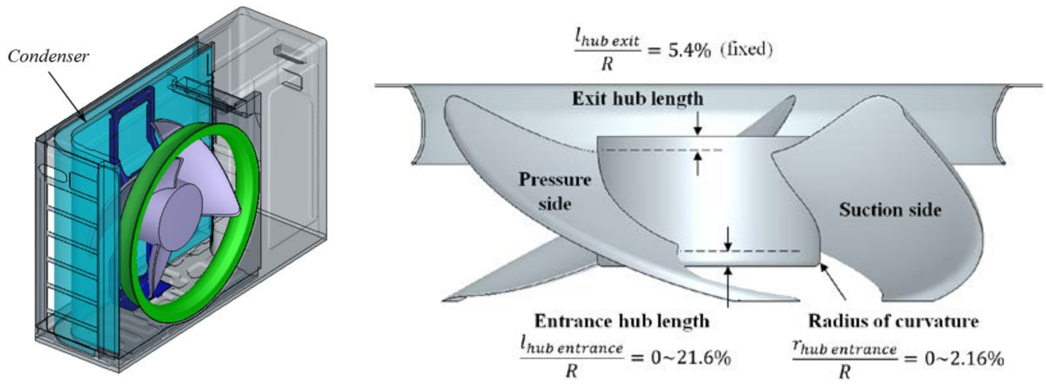


Figure 1 – Shape of tested outdoor unit including axial flow fan

2.2 Numerical method

The calculation domain is shown in Figure 2. The pressure drop through the heat exchange in an outdoor unit was obtained from experiments and simulated as the 'volume force' for numerical simulations as shown in Figure 2. The diameter of the axial flow fan was 370 mm and its rotation speed was 850 rpm. Commercial CFD solver SC/Tetra v12 was used for numerical analysis. It is beneficial to use ST/Tetra for simulation of massive mesh because the amount of calculation can be reduced compared to the cell-centered formulation as node-based scheme is applied with regard to tetrahedral mesh in ST/Tetra. In this study, the flow was assumed to be in steady state. SST k-ω model with more accurate calculation of the boundary layer flow on the wall surface was applied as a turbulence model. Further explanation on the numerical method and the turbulence model is described in detail in Jung et al., written by the authors(6).

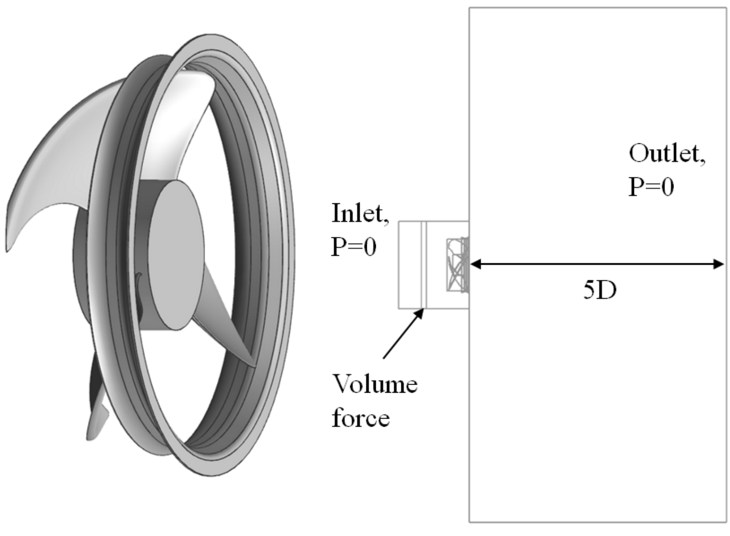


Figure 2 – The calculation domain and boundary conditions

2.3 Experiments

To measure the flow noise occurring by the axial flow fan of the outdoor unit experimentally, the outdoor unit and microphones are installed as shown in Figure 3. In the experiment, the noise was measured in an anechoic room by using microphones. The anechoic room was the size of 4.8 m in width and length and 4 m in the height and the 6 walls including the floor and the ceiling were surrounded by wedge. The microphones were located in two different spots in order to measure the anterior and the posterior noise, respectively. In order to minimize the influence of the noise reflected from the floor on where the outdoor unit was installed, the microphones were installed 1 m apart from the floor and 1 m apart from the surface of the outdoor unit. The microphones used in this experiment were Type 2669 (B&K) and noise data acquisition equipment was 3560B (B&K). The ambient condition during the experiment was the temperature of 23.8°C, the relative humidity of 33.0%, the atmospheric pressure of 760 mmHg, and the consequent air density was $\rho = 1.184\ \mathrm{kg/m^3}$. The background noise was 15.5dB(A).

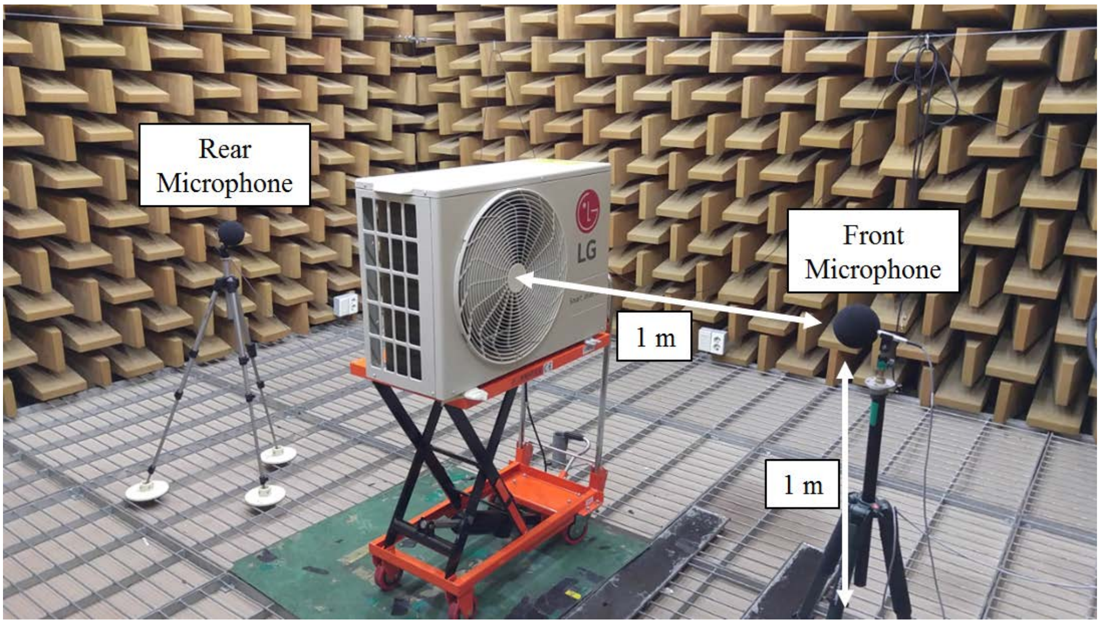


Figure 3 – The setup of the experiment measuring fan noise by using microphones in the anechoic room

3. RESULTS

3.1 The influence of the hub vortex on the fan performance

An axial flow fan used in an air conditioner usually has a blunt-ended hub. As shown in Figure 4(b), flow is separated at the hub corner edge and reattached to create a separation bubble. Then, separation bubble is shed from the hub edge to form vortex due to the interaction with the potential field upstream of the blade. In this study, this vortex is called as 'hub vortex', and the effects of the hub vortex were investigated with the entrance hub length and shape of the hub on the characteristics, pressure loss distribution, and fan efficiency. Stream-wise absolute vorticity defined in Equation (1) was used to visualize the movement and direction of the hub vortex as shown in Figure 4(c).

$$\omega_S = \frac{\vec{\omega}\cdot\overrightarrow{u_{rel}}}{2\Omega|\overrightarrow{u_{rel}}|} = \frac{\omega_i\cdot u_i+\omega_j\cdot u_j+\omega_k\cdot u_k}{2\Omega\sqrt{u_i^2+u_j^2+u_k^2}} \tag{1}$$

where $\vec{\omega}$ is the vorticity, $\overrightarrow{u_{rel}}$ is the relative velocity, and Ω is the angular velocity. The positive (red color) and negative (blue color) signs of stream-wise absolute vorticity represent the clock-wise and the counter-clock-wise directions of the rotation on the direction of the main stream. As shown in Figure 4(c) as well, occurrence of a vortex strongly rotating clockwise was confirmed at the front edge of the hub due to the velocity component in radial direction caused by the blunt-ended hub. In Figure 4(d), vorticity is presented as the iso-surface to observe the 3-D configuration of the hub vortex. The separation bubble that occurs from the edge of the hub's front end creates the hub vortex as shed from the hub edge proximal to the suction side of the blade. The vortex shed from the edge flows toward the blade passage. Figure 4(c) and (d) indicates that the distance from the front end of the hub to the blade may affect the blade performance.

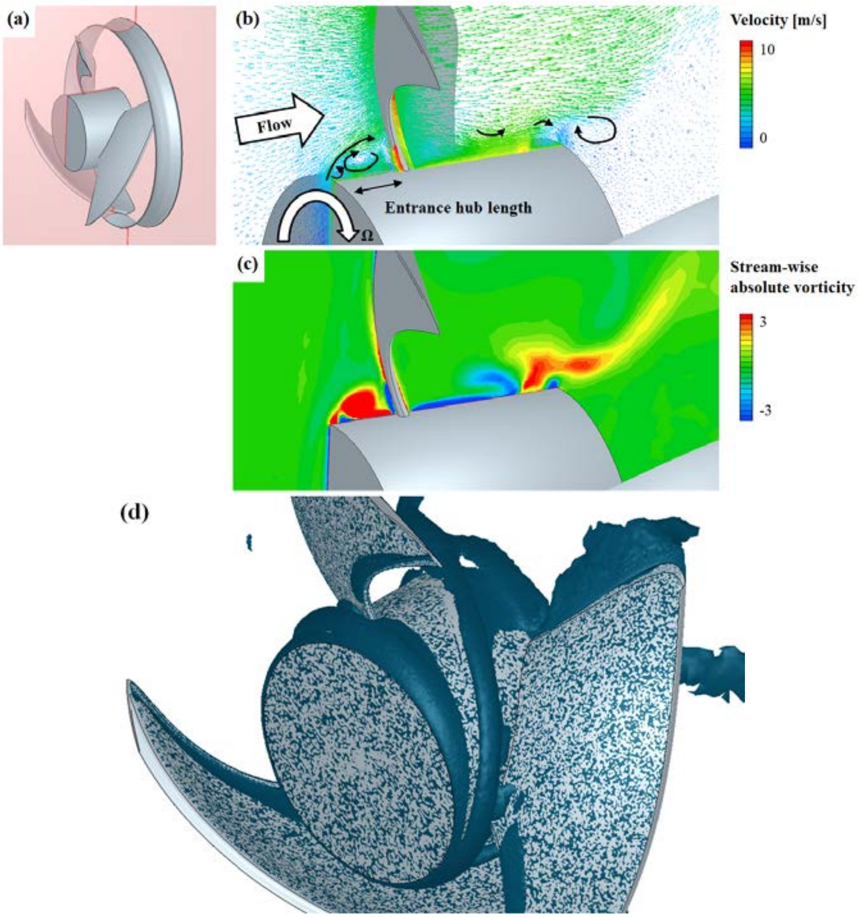
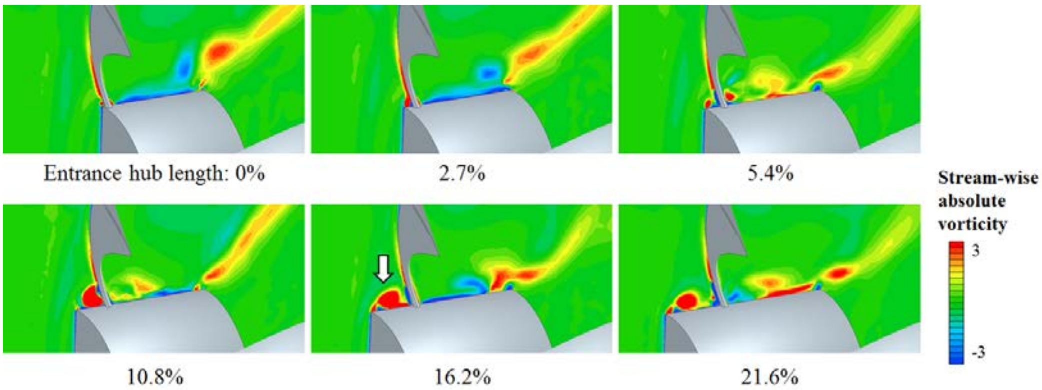


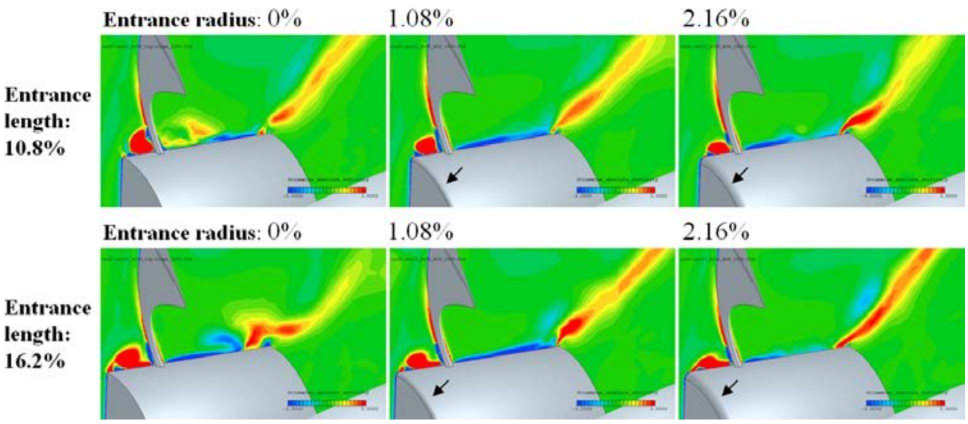
Figure 4 – The visualization of the hub vortex using stream-wise absolute vorticity. (a) A location of plotting section, (b) vector plot, (c) stream-wise absolute vorticity plot and (d) 3-D iso-surface of stream-wise absolute vorticity $\omega_s = 2$.[7]

The entrance hub length of the fan can influence the behavior of vortex occurring around the hub and hence influence fan efficiency. The entrance hub length was the distance in axial direction from the front end of the fan hub to the leading edge of the blade, as defined in Figure 1, and it was nondimensionalized by the radius of the fan. The changes in vortical flow for the entrance hub length of the axial flow fan are predicted in Figure 5(a). In the figure, the contour is the stream-wise absolute vorticity; hence, greater value means stronger component rotating in positive direction (clockwise in the figure) on the flow direction. The separation bubble occurring in the entrance hub corner produced strong vortex (indicated with arrows) rotating clockwise in Figure 5(a) as it meets the main flow component. Longer entrance hub length means farther distance between the hub entrance and the leading edge of the blade. Hence, the influence of the vortex occurring in the anterior region of the hub on the blade is reduced when the entrance hub length gets longer. In addition, no change in the pattern of the hub vortex was observed when the entrance hub length was lengthened over 16.2%, indicating that hub vortex no longer had influence on the blade. It was interesting that hub vortex in cleaved shape was observed at the specific entrance hub length (5.4%) as the hub vortex encounters the leading edge of the blade.

In Figure 5(b), the shapes of the hub vortex according to the radius of curvature of the hub entrance were compared. As the radius of curvature gets larger, the size of the hub vortex was found to be decreased.



(a) Hub entrance vortex depending on the entrance length of the axial flow fan's hub



(b) Hub entrance vortex depending on the radius of the axial fan's hub

Figure 5 – The effect of the entrance hub geometry on the hub vortex[7]

Figure 6 is showing the fan efficiency depending on the length and the radius of curvature of the hub entrance . Depending on the entrance length of the hub, the degree of the efficiency of the hub vortex on the fan blade is shown to be changed and the fan efficiency is also shown to be rapidly changed accordingly. Except for the specific entrance length (5.4%) presenting complicated flow characteristics, the fan efficiency was decreased by over 3% when the entrance length was short enough so that the hub vortex had great influence on the blade. In case of having the round-shaped edge of the hub entratnce, which suppresses the hub vortex, the influence on the fan blade was decreased and efficiency was not found to be reduced even with the short entrance length.

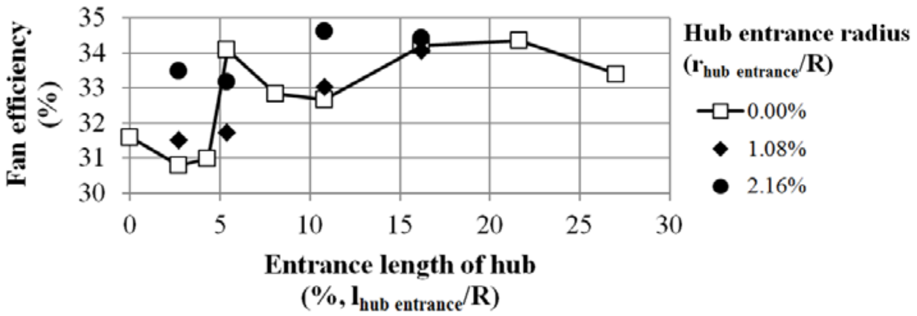


Figure 6 – Variations of efficiency depending on the entrance length and radius of the axial fan's hub[7]

3.2 Development of the Hubless fan to avoid the influence of the hub vortex

In order to reduce the influence of the hub vortex, which reduces the efficiency of the axial flow fan, the axial flow fan with a hub in extremely reduced size was designed and named as "Hubless fan". Figure 7 is showing the illustration of the Hubless fan. While maintaining the same specification of the connecting region to the motor shaft to allow the installation of the fan replacing the existing fan in the outdoor unit, the hub size was mimimized and the linkage to the wings were newly designed with curvature. The volume of the hub was decreased by 86% of the existing fan's hub as the diameter and height of the hub was decreased by 36% and by 33%, respectively. The new design of the linkage to the wings enabled securing large pitch angle on the hub side of the wings while using the hub in minimal size.

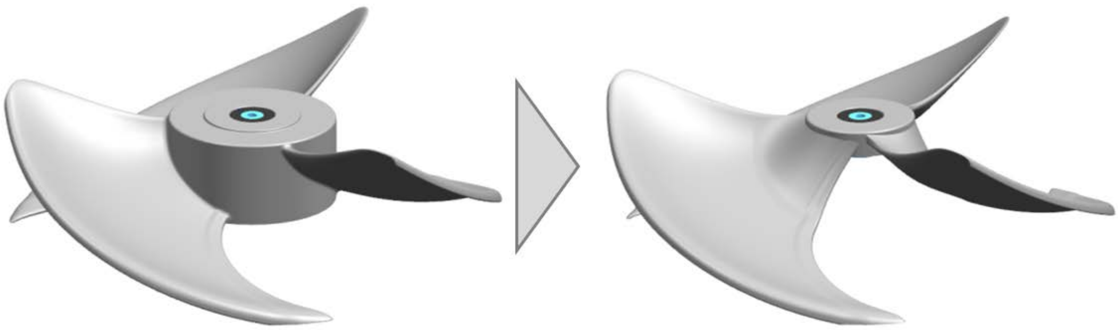
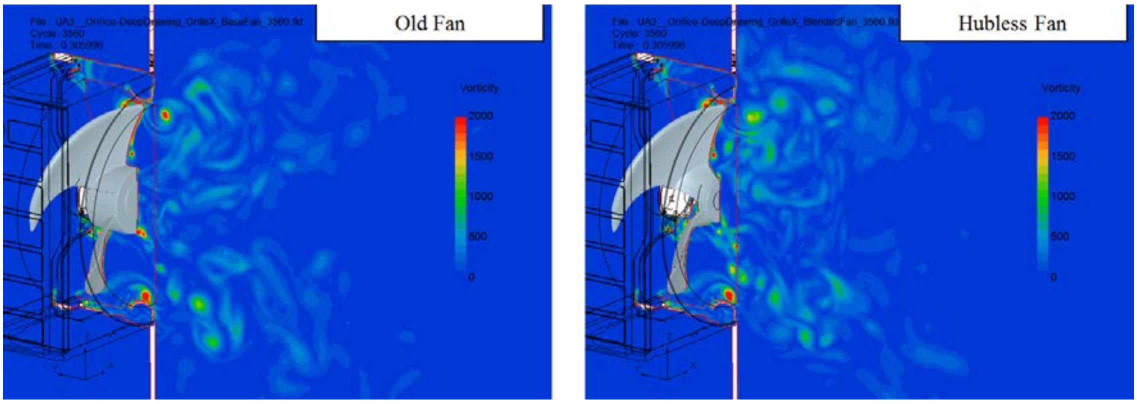


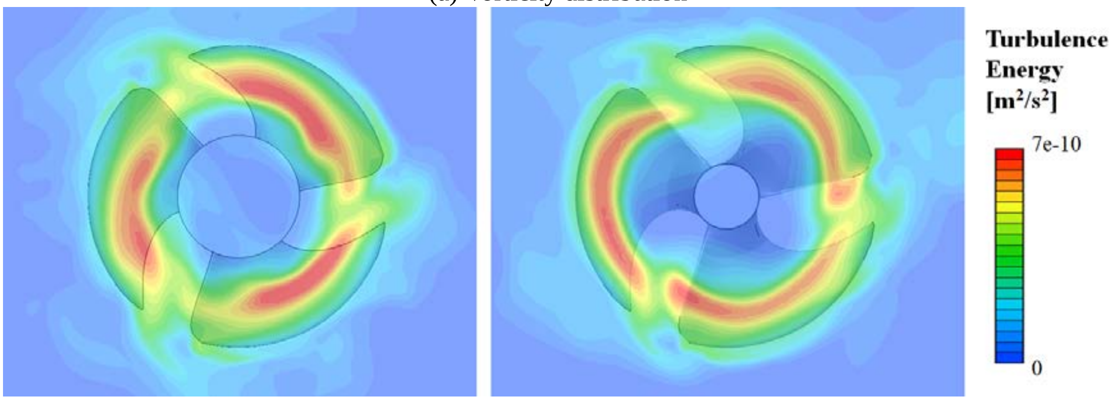
Figure 7 – The Hubless fan with a hub in extremely reduced size

Figure 8(a) is showing the comparison of the vorticity in the wake region between the existing fan and the Hubless fan. With the changes in the hub shape, the reduction of the vortex shed from the hub exit was found. In Figure 8(b), the turbulence energy on the cross-sections perpendicular to the rotational axia in the wake region were compared. According to the comparison, the maximum turbulent energy in the wake region of the Hubless fan was found to be reduced by approximately 5.6%.

To compare the performance of the existing fan and the new Hubless fan, the motor input power and the fan noise were compared through the experimental measurement and the results are shown in Figure 9. Since the noise measured in the front side and the rear side were in the same trend, the values from the front microphone were used as the representative values. In order to examine the effect of the hub shape, other factors of the wings were identically designed. The comparison in the same flow rate condition revealed the reduction in the motor input power by 5% compared to the existing fan. The comparison of the fan noise at the same flow rate revealed to be reduced by approximately 1.3 dB in all flow rate, resulting approximately 5% increase in the flow rate at the same noise.



(a) Vorticity distribution



(b) Turbulence energy distribution

Figure 8 – Comparison of the flow field in the wake region between the existing fan and the Hubless fan

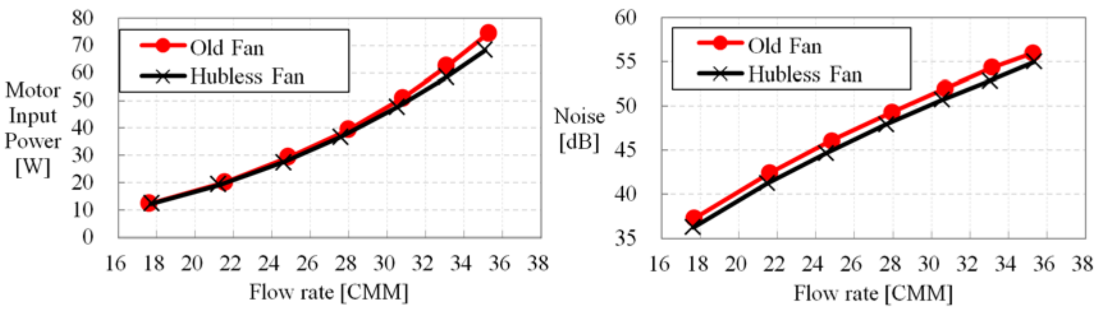


Figure 9 – Experimental results on performance comparison between old and new fans for fan's power consumption and noise

Figure 10 is showing the sound spectrum of the noise measured from the two fans. There was no big difference was found in the trend of the BPF peak noise, which is commonly found in the low frequency, between the two fans. However, obviously reduced broad band noise in the region of 1,000-2,000 Hz and 3,000-5,000 Hz was found. This was thought to be due to the reduction in noise owing to the reduced turbulence energy in the fan’s wake region.

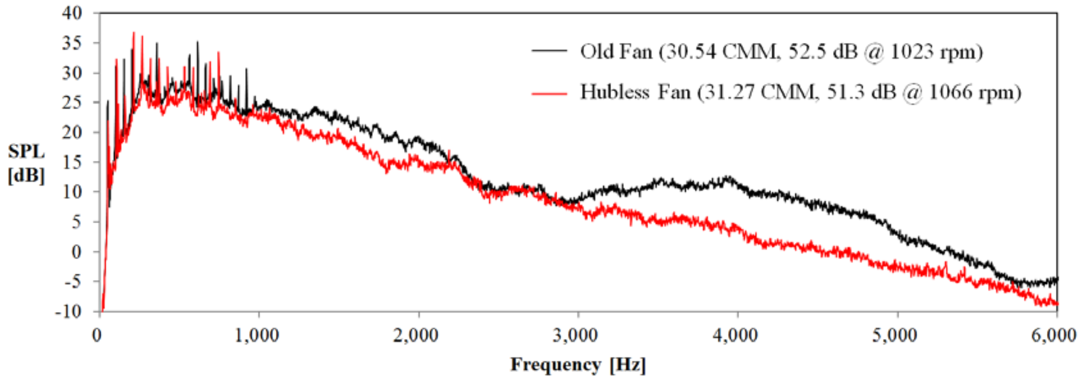


Figure 10 – Comparison of noise spectra obtained by and experiment between old fan and Hubless fan

4. CONCLUSIONS

In this study, the influence of the vortex occuring near the hub of an axial flow fan on the fan blade was numerically analysed and the size and shape of the hub vortex was found to affect the fan efficiency. In order to reduce the loss occuring by the hub vortex, a Hubless fan with exteremely reduced size of the hub was designed. Through the experiment examining the improvement by the design change, not only the improvement in the fan efficiency but also the noise reduction were confirmed. Due to the change in the hub shape, reduction of the input power required for the fan motor by 5% and reduction of the fan noise by 1.3 dB in the same condition of the flow rate were confirmed. This was thought to be due to the reducion of the vorticity and turbulence energy in the fan's wake region in consequence of the greatly reduced influence of the hub vortex owing to the minimized hub size. Consequently, the broad noise in the noise spectrum of the new Hubless fan was reduced.

5. ACKNOWLEDGEMENTS

The first author is enrolled in Ph.D. course as dispatched from LG electronics with tuition provided by LG electronics.

6. REFERENCES

1. Jang, C. -M., Furukawa, M. and Inoue, M., *"Analysis of vortical flow field in a propeller fan by LDV measurements and LES - Part I: three-dimensional vortical flow structures"*, ASME Journal of Turbomachinery, Vol. 123, pp. 748-754 (2001).

2. Jang, C. -M., Furukawa, M. and Inoue, M., *"Analysis of vortical flow field in a propeller fan by LDV measurements and LES - Part II: unsteady nature of vortical flow structures due to tip vortex breakdown"*, ASME Journal of Turbomachinery. Vol. 123, pp. 755-761 (2001).

3. Wang, H., Tian, J., Ouyang, H., Wu, Y. and Du, Z., *"Aerodynamic performance improvement of up-flow outdoor unit of air conditioner by redesigning the bell-mouth profile"*, Int. J. Refrigeration, Vol. 46, pp. 173-184 (2014).

4. Tian, J., Ouyang, H. and Wu, Y., *"Experimental and numerical study on aerodynamic noise of outdoor unit of room air conditioner with different grilles"*, Int. J. Refrigeration, Vol. 32, pp. 1112-1122 (2009).

5. Zhao, X., Sun, J. and Zhang, Z., "*Prediction and measurement of axial flow fan aerodynamic and aeroacoustic performance in a split-type air-conditioner outdoor unit*", Int. J. Refrigeration, Vol. 36, pp. 1098-1108 (2013).

6. Jung. J. H. and Joo, W.-G., "*Effect of tip clearance, winglets, and shroud height on the tip leakage in axial flow fans*", Int. J. Refrigeration, Vol. 93, pp. 195-204 (2018).

7. Jung. J. H. and Joo, W.-G., "*The Effect of the Entrance Hub Geometry on the Efficiency in an Axial Flow Fan*", Int. J. Refrigeration, Accepted (2019).