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Local improvement of flow and noise performances of axial-flow fans in a household refrigerator

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ABSTRACT

In this paper, numerical and experimental investigations on an axial-flow fan with grooved/double-sectional blades are performed to improve its performance in terms of volume flow rate and noise at a local operating point. The target fan is used to cool a mechanical room of a household refrigerator. However, it is found that the target system has more resistance than that for which the fan is originally designed and the fan's operating point is not optimal. First, the performance of the target fan unit is measured from the fan performance tester. Then, the P-Q curve is predicted using the virtual fan performance tester based on the CFD techniques to solve the incompressible RANS equations. The surface acoustic power of the fan unit is also predicted using the broadband noise sources modeled from the CFD results. The predicted P-Q curve is compared to the measured data, through which the validity of the numerical techniques is confirmed. On the basis of the prediction results, new fan units with modified fan's housing structures are devised. Further predictions using these new fan units show that these fan units generate higher volume flow rate while keeping surface acoustic power levels lower than the existing fan system on the local operating point

Keywords: fan noise, flow rate, refrigerator machine room, axial flow fan, I-INCE Classification of Subjects Number(s): 11.4.1, 12.7.2

1. INTRODUCTION

With increasing concern about energy efficiency, recently, more and more customers consider the energy consumption rate of home appliances the most important index for their selecting products. Among many household appliances, considering that a refrigerator should run 24 hours a day, its efficiency is recognized as an extremely important performance.

Given that a refrigerator is based on a refrigerating cycle where heat is absorbed inside the refrigerator while being emitted outside of it to keep food fresh, the energy consumption efficiency of a refrigerator is significantly influenced by the refrigerating cycle. The refrigerating cycle entails four processes: compression, condensation, expansion, and evaporation. The evaporation process means a liquid-vapor mixture is vaporized by absorbing a heat inside of the refrigerator. An evaporator is used in the refrigerator and is generally installed in freezer compartment. The condensation process indicates a vapor is condensed into a liquid by emitting a heat into the refrigerator. A condenser used for the condensation process is located in a machine room of the refrigerator. The cooling fan system installed in the machine room of the refrigerator for efficient heat release is known to make a significant contribution to improving the total energy consumption efficiency of the refrigerator [1]. Generally, an axial-flow fan system is used as the cooling fan system in the machine room of the household refrigerator because it has a high volume flow rate performance at low pressure. The high volume flow rate performance induces the effective heat exchange procedure. Accordingly, the axial-flow fan system in the machine room of the refrigerator has an important role in the overall energy efficiency of the refrigerator while also acting as a load.

Many experimental/numerical studies have been conducted in order to develop a high-performance low-noise axial fan system. From an experimental study, Venter et al. [2] found as the tip clearance became smaller, the volume flow rate and static pressure grew larger. The performance of the axial-flow fan system is affected not only by elements constituting the fan unit but also by the environmental conditions where the axial-flow fan system is installed. Thiart et al. [3] conducted a numerical/experimental investigation on the influences on distortion of the inflow on the axial-flow fan system. They identified the disadvantage that inflow with the distortion significantly increased power consumption of the axial-flow fan system, consequently reducing efficiency. Lin et al[4] carried out an experimental study on the effects of blockage attached to the axial-flow fan system on performance and noise and discovered that the blockage caused considerable loss of static pressure, although it did not contribute greatly to volume flow rate reduction. Additionally, the blockage of the axial-flow fan system substantially increased broadband noise. Gue et al. [5] and Ren et al. [6] proposed new designs of fan units used in the machine room in the household refrigerator for higher performance and lower noise. Heo et al. [7] reported that low noise centrifugal fans were developed by changing the linear trailing edge line into the inclined S-shaped trailing edge of the fan blades.

This paper proposes a new axial-flow fan system designs that are suitable to the machine room in refrigerator for high performance and low noise. To develop the new designs, the virtual fan performance tester which is based on the CFD technique is considered. From analysis of the flow field around the target axial-flow fan system, the new axial-fan systems are proposed through changing the length of the axial-flow fan's housing. The proposed design adjusts the operating point by redesigning structures around the fan and consequently improves the static pressure or volume flow rate performance of the axial fan system in line with the purpose of the modified fan system.

2. Numerical method

2.1 Governing Equations and Numerical Methods

To predict target fan performance, the flow field is calculated through Computational Fluid Dynamics (CFD) to solve incompressible RANS equations in the form,

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l})\right) + \frac{\partial}{\partial x_j}(-\rho\overline{u_i^{'} u_j^{'}}), \frac{\partial}{\partial x_i}(\rho u_i) = 0 \qquad (1)$$

Equation (1) has the same form as the instantaneous Navier-Stokes equation, with the velocities and other solution variables now representing ensemble-averaged (or time averaged) value. The governing equation is spatially discretized with tetrahedral meshes, and is solved using the finite-volume method. The Reynolds stress, $-\rho\overline{u_i^{'} u_j^{'}}$, representing the effects of turbulence must be modeled in order to close Equation (1)

Reynolds stress term in the RANS equations is modeled using the RNG k-ε model. These are numerically realized by using the commercial CFD program, FluentTM.

2.2 Noise Source Models

The sound energy in many practical applications involving turbulent flows, such as the axial-flow fan system in the present study, is continuously distributed over a broad range of frequencies. It is not easy to predict the sound spectrum including broadband components due to the intrinsic properties of turbulences, of which the accurate predictions require the LES(Large Eddy Simulation) or DNS(Direct Numerical Simulation) techniques. However, comparison of broadband noise source strength between different fans can be made by using the acoustic analogy. For fans, far field sound generated by turbulent boundary layer flow over a solid body at low Mach numbers is of practical interest. On a basis of Curle's acoustic analogy, the sound pressure in far field can be computed using the time-derivative of static pressure on the surface of the fans in the form:

$$p^{'}(\vec{x},t) = \frac{1}{4\pi a_0}\int_S \frac{(x_i - y_i)n_i}{r^2}\frac{\partial p}{\partial t}(\vec{y},\tau)dS(\vec{y}) \qquad (2)$$

The sound intensity can be, therefore, approximated by the following equations,

$$\overline{p'^2} \approx \frac{1}{16\pi^2 a_0^2} \int_S \frac{\cos^2\theta}{r^2} \overline{\left[\frac{\partial p}{\partial t}(\vec{y},\tau)\right]^2} A_c(\vec{y}) dS(\vec{y}) \tag{3}$$

where A_c is the correlation area. The total acoustic power emitting from the entire body surface can be expressed in the form,

$$P_A = \frac{1}{\rho_0 a_0} \int_0^{2\pi} \int_0^{\pi} \overline{p'^2} r^2 \sin\theta d\varsigma d\psi = \int_S I(\vec{y}) dS(\vec{y}) \tag{4}$$

Here, $I(y)$ can be interpreted as the local contribution per unit surface area of the body surface to the total acoustic power.

3. Experimental validation

Figure 1 shows the target fan system consisting of blades with grooved surface and double sections, inlet and outlet ducts.. As above-mentioned, the performance of the target fan system is analyzed to find design factors for higher performance and lower noise. First, the performance of the target fan is experimentally assessed using a fan performance tester shown in Figure 2. The fan performance tester consists of a couple of nozzles and sensors that measure data used to determine static-pressure and volume flow rate of fans. The measured P-Q curve of the target axial-flow fan system is shown in Figure 2. To find the operating point of the fan which is installed in a mechanical room for cooling two compressors and two evaporators, the resistance curve of the machine room is also measured and shown in Figure 2.

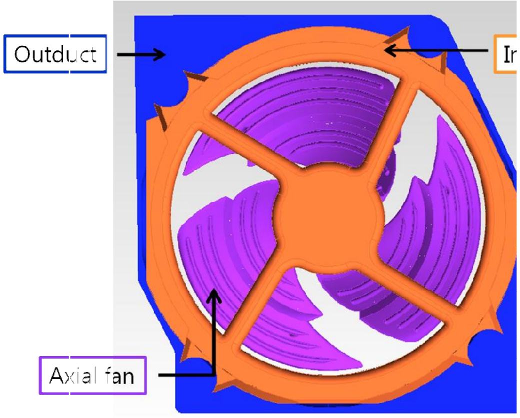


Figure 1 - Target axial-flow fan system

Figure 2 - Fan performance tester

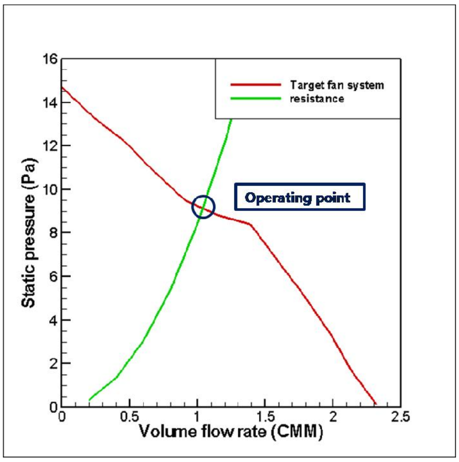


Figure 3 – target fan P-Q curve [8]

It is seen that the target axial-flow fan system draws and exhausts air at the volume flow rate, 1CMM with static pressure 8.5Pa. However, the current operating point of the fan seems to be located outside the design range, which means that the operating point of the target axial-flow fan system is not optimal and thus new designs are needed to improve the performance of the target fan system.

4. Virtual fan tester

Virtual fan tester (VFT) is used as an analysis tool. Figure 4 shows the entire computation domain of the VFT with applied boundary conditions. The volume flow rate of the target fan in the VFT is predicted by varying the pressure difference between the inlet and outlet boundaries in the VFT.

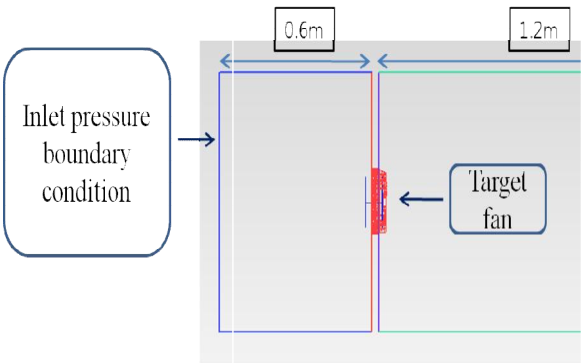


Figure 4 - Virtual fan performance tester with boundary condition

Numerical methods using the VFT are validated by comparing its prediction results with the measured data show in Section 3. The P-Q curve predicted using the VFT is compared with that measured through the fan performance tester shown in Figure 3. It is seen that there are good agreements between two curves in terms of increasing rate of the volume flow rates according to the variation of static pressure. Slight difference between two curves seems to be due to simplified resistance of the VFT. However, since the VFT provides the reliable prediction results in terms of the variation rate of the volume flow rate versus the static pressure, the VFT is used to assess the relative performances of the newly proposed fan systems compared to the original system.

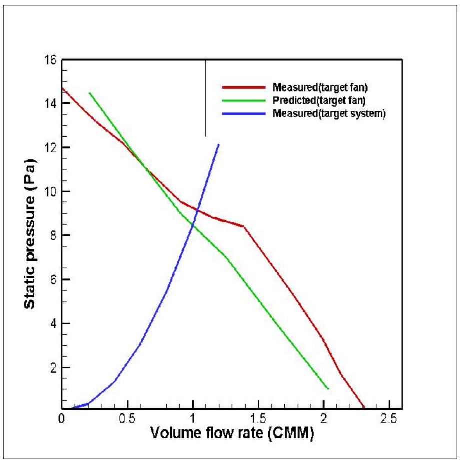


Figure 5 – Comparison between the values predicted from VFT and the measured data

5. CFD analysis

5.1 Design factors

First, to fine design factors to improve the performance of the target fan system, the flow field around the target fan system predicted using the VFT is closely examined. Figure 6 shows the flow velocity fields around the target fan system. As it is found that reverse flow in secondary flow region and strong tip vortices occur in the gap between the fan blade and its housing. These two flow patterns have adverse effects on the fan performance. On a basis of these findings, new designs are proposed to reduce the observed secondary flows and tip vortex strength.

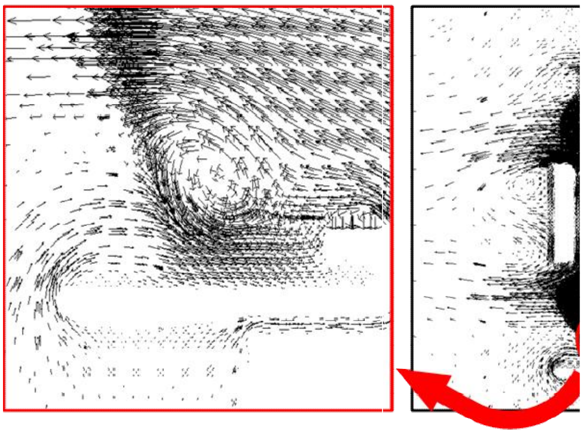
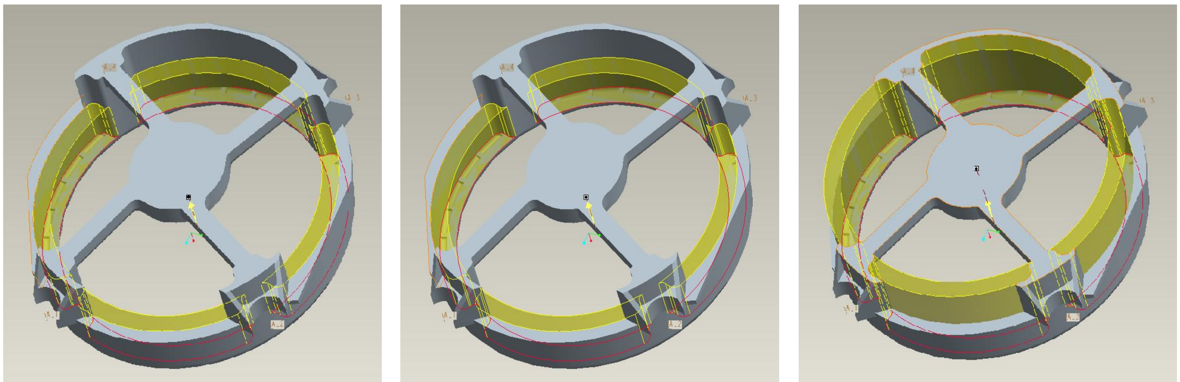


Figure 6 – Flow velocity fields in the target fan system predicted using the VFT



(1) 18mm **(2) 30mm** **(3) 41.1mm**

Figure 7 – Housing length as selected design factor

Housing length is selected as a design factor among several factors for reducing the flow losses. The housing length of the target fan system is limited by other structures in the machine room such as compressors and condensers. The maximum housing length allowed is 41.1mm. Three designs of the housing duct length, 18mm, 30mm and 41.1mm are tested as shown in Figure 7. Figure 8 shows the predicted P-Q curves and the surface acoustic power levels according to the volume flow rates of the proposed three designs.

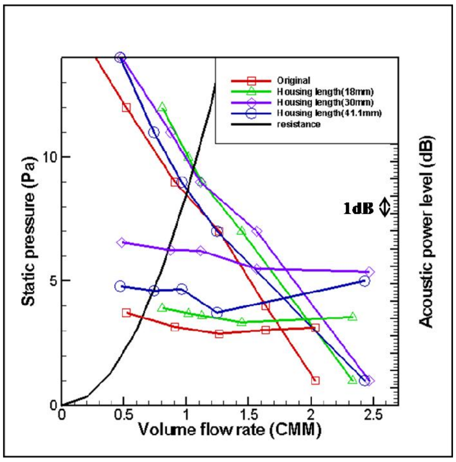


Figure 8 - P-Q and noise curve of length value at 18mm, 30mm and 41.1mm

It can be found that the new fan systems with the housing duct of length 30mm and 18mm have higher performance characteristics than that of 41.1mm at the operating point. The surface acoustic power of the fan system of the duct length 18mm is smallest among the new fans, though it is lightly higher than that of the original fan system while the value of 30mm design is high than that of the target fan system. Therefore, the fan system of the housing length 18mm is is selected as a final design, which increases the volume flow rate by 5.9% but generates noise of power level comparable to the original one.

6. Conclusions

Virtual fan tester (VFT) is developed to predict performance of fan system and to analyze the related flow characteristics. The validity of the VFT is confirmed by comparing its predictions with the measured data. On a basis of analysis result of flow fields around the target fan system from the VFT, the housing length is selected for a parametric study. Three lengths of 18mm, 30mm, and 41.1mm are selected and the performance and noise characteristics of the proposed designs are predicted using the VFT. In view of performance, the system with the housing of length 30mm and 18mm show higher volume flow rates than the target fan system. In aspect of noise, the system with the duct length of 18mm is predicted to generate lowest acoustic power among new systems, though it is slightly higher than the original. Experimental study to confirm the current results will be made in near future.

ACKNOWLEDGEMENTS

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