[Technical Paper]

**MRF Modeling of Axial Fan for Thermal Simulation of Electronic Equipment**

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(Received July 29, 2014; accepted October 16, 2014)

Abstract

Nowadays, thermal flow simulation based on computational fluid dynamics (CFD) is applied to the thermal design of electronic equipment. In this paper, we discuss the applicability of the multiple reference frame (MRF) approach, which is a new method for modeling an axial cooling fan, to the thermal flow simulation of electronic equipment using the CFD code. In this study, as the first step, flow visualization of the exhaust air flow pattern of the experimental axial fan was conducted. Then the flow visualization results were compared with the MRF fan model simulation results. Finally, the P-Q characteristic obtained by MRF fan model simulation was compared with the measured P-Q characteristic to validate the applicability of the MRF approach to the thermal flow simulation of electronic equipment.

**Keywords:** Thermal Modeling, Thermal Measurements, Multiple Reference Frame, Axial Fan, P-Q Characteristic

1. Introduction

In recent years, thermal flow simulation based on the computational fluid dynamics (CFD) code has been applied to the thermal design of electronic equipment.[1] Figure 1 shows a switch mode power supply (SMPS), which is a type of electronic equipment, and Fig. 2 shows the thermal flow simulation result of the component temperature. In order to conduct thermal flow simulation using the CFD code, it is necessary to develop a compact and accurate thermal model of a component to predict the temperature at a practical calculation cost. Figure 3 shows a forced-air-cooling-type SMPS. In forced-air-cooled equipment, the axial fan is one of the most commonly used air-moving devices. In previous CFD simulations, the most popular modeling method of the axial fan was to use the lumped fan model[2] shown in Fig. 4. The lumped fan model is the simplest method in terms of both model geometry and boundary condition to set the operating conditions.

![Fig. 2 Simulation result of component temperature.](image-url)

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![Fig. 3 Forced air cooled switch mode power supply.](image-url)

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point of the fan. In this modeling method, the geometry of the fan is modeled by a circular planar face and the P-Q characteristic is simplified as the pressure boundary condition. However, this simplified fan model has the following issues: the flow distribution is unrealistically uniform and the operating point on the P-Q characteristic is affected by the pressure definition of the CFD code. The governing equations of the lumped fan model which determines flow volume through the bounded planar face is only the continuity equation, and the full Navier-Stokes equations need not be solved. In this modeling method, the flow volume is determined based on the P-Q characteristic that is set as the boundary condition of the lumped fan model and the pressure difference between the upstream and downstream regions of the bounded planar face.

Recently, an MRF approach that involves a new modeling method for the axial fan was examined for possible application to the CFD simulation of electronic equipment. [2, 3] The MRF fan modeling method has found a use in the automotive industry.[4, 5] However, there are few practical studies of the thermal flow simulation of electronic equipment. The MRF approach is a steady-state approximation in which the model comprises the CAD geometry of the fan blade and the rotating frame MRF zone that circumscribes the fan blade. The MRF approach has low calculation cost compared with that of the sliding mesh model, which is inherently unsteady. The MRF fan modeling method is expected to provide a practical and compact solution for the thermal simulation of electronic equipment cooling. Figure 5 shows a schematic of the MRF fan model. The fan blade is modeled to be stationary and the MRF zone is positioned to circumscribe the fan blades. As the MRF zone is in a rotating frame, the presence of the stationary blade provides flow components from the fan.

The objective of this study is to examine the possible application of the MRF fan modeling method to the thermal flow simulation of electronic equipment. In this report, we first conducted flow visualization of a 1U-size enclosure equipped with a 40 mm-size axial fan. Then the air-flow simulation using MRF modeling was conducted, and the flow distribution of fan exhaust flow obtained by MRF fan modeling simulation and the visualized flow distribution were compared.

2. Flow Visualization of 1U-Size Simplified Experimental Enclosure

Figure 6 shows the experimental fan used in this study. A 40 mm-size axial cooling fan with a stator blade was used in this experiment. Figure 7 shows the dimensions of the 1U-size simplified experimental enclosure for the visualization of fan exhaust flow. The experimental enclosure is made of transparent acrylic and the fan is positioned on one side of this enclosure to realize the push direction of air flow. Figure 8 shows the experimental setup for flow visualization. Tracer particles are introduced into the enclosure from the side of the fan intake and the exhaust side near the fan is illuminated by a laser light sheet. The air flow pattern is captured by a high-speed camera with its optical axis perpendicular to the laser plane.

Figure 9 shows the flow visualization results for the exhaust side of the axial fan center plane. The air flow pattern exhibits radially spreading flow and backflow to the hub center, as schematically shown in Fig. 9.
3. MRF Fan Model Simulation

To model the actual experimental fan faithfully on the CFD code, computed tomography (CT) of the fan blade was conducted to capture the actual blade geometry. Figure 10 shows the fan blade CAD geometry obtained by CT of the actual experimental fan blade. Figure 11 summarizes the modeling procedure of the fan 3D geometry. The MRF fan model simulation was conducted by setting the MRF zone to circumscribe this captured fan blade object as shown in Fig. 12.

As mentioned above, by employing the MRF approach, an unsteady problem in a stationary reference frame can be modeled as a steady-state problem with respect to a moving reference frame. In order to solve fluid flow in a moving reference frame of the MRF fan modeling method, we use the Navier-Stokes equations (momentum conservation law) which are the governing equations of fluid flow and include two additional acceleration terms: the Coriolis acceleration and the centripetal acceleration.

3.1 Flow distribution of MRF fan model simulation

In order to confirm the applicability of the MRF approach to the modeling of the axial cooling fan, we verified the simulation results focusing on the following points.

– Influence of air flow direction
– Influence of MRF zone size
– Influence of stator blade

The first point is the influence of air flow direction. In practical applications, forced cooling equipment employs both exhaust and intake air flow directions depending on the system application. Therefore, confirming the influence of air flow direction is essential to demonstrate the applicability of the MRF approach to the thermal flow simulation of electronic equipment. As the second point, to provide specific guidance to modeling using the MRF fan model, the influence of MRF zone size is examined by changing the MRF zone length in the axial direction. In this study, we confirmed two MRF zone sizes: normal-size MRF zone and extended MRF zone. The MRF zone length in the axial direction of the normal-size MRF zone is the size that exactly encloses the moving blade, as shown in Fig. 12. In the extended MRF zone model, the MRF zone length in the axial direction is extended to half the length of the fan casing thickness from the upstream and downstream side surfaces of the fan casing. The third point is the influence of stator blade. The cooling fan used in electronic equipment may sometimes have a stator blade, par-
particularly in a small-sized or high-speed axial fan. Today’s electronic equipment is extremely compact and has high density due to complex functions. Therefore, it is important to confirm the influence of the stator blade to extend the applicability of the MRF fan modeling method to other equipment.

At first, the results of simulation that examined the influence of air-flow direction are shown in Fig. 13. In this simulation, MRF zone size is normal and no stator blade is equipped. Figure 13 shows the simulated flow speed contours for both exhaust and intake air flow directions. As both simulation results demonstrated almost the same flow volume and flow pattern, the influence of air flow direction could not be confirmed from the simulation result. Next, the results of simulation that determined the influence of MRF zone size are shown in Fig. 14. Figure 14 shows the simulated flow speed contours of the extended MRF zone model. The extended length of the MRF zone for both upstream and downstream sides is half the fan case thickness. The simulation results of flow volume and flow pattern are almost the same as those of the normal-size MRF model shown in Fig. 13. Through a comparison of Fig. 13 with Fig. 14, it was confirmed that the MRF zone size has little influence if it exactly circumscribes the moving blade.

The above mentioned results are considered below. In Fig. 13, the difference between exhaust setup and intake setup is the position of the fan enclosure wall. In the exhaust setup, the enclosure wall is set at the downstream side of the fan, as shown in Fig. 13(a). In the intake setup, the enclosure wall is set at the upstream side of the fan, as shown in Fig. 13(b). In general, the cooling fan in the electronic equipment is set on the enclosure side surface. In the intake setup, the enclosure wall is adjacent to the MRF zone of the fan model. To consider the positional relationship between the MRF zone and the enclosure wall, streamline flow into the fans of the exhaust setup and the intake setup is shown in Fig. 15 and Fig. 16, respectively. Those figures demonstrate that the streamlines flow into
the fans is uniform and almost inflexible. From these results, it is considered that the uniformity of streamline flow into the MRF zone in the case of both exhaust setup and intake setup yields almost the same flow pattern in both exhaust and intake setups, as shown in Fig. 13. Regarding the influence of the MRF zone size shown in Fig. 14, the result that the MRF zone size has little influence on the flow pattern is also explained by the uniformity of flow into the boundary of the MRF zone regardless of MRF zone size.

Next, the influence of the stator blade on the simulation results was confirmed. The experimental fan in this study has a stator blade that is integrally molded together with the case as shown in Fig. 17. Therefore, to confirm the applicability of the MRF model to a fan equipped with a stator blade, we conducted the simulation by modeling the stator blade. Figure 18 shows the fan model geometry in which the stator blade was modeled. Figure 19 shows the simulated flow speed contours at the center plane of the fan with a stator blade. There was no notable difference from the simulation results of only the moving blade shown in Fig. 13(a). However, the fan operating point boundary condition of the simulation results shown in Fig. 19 was the maximum flow volume. Therefore, next, the simulation in which the boundary condition of the operating point was near the surging point was conducted. Figure 20(a) shows this simulation result. For comparison, Fig. 20(b) shows the simulation result obtained using the fan model with only a moving blade in which the boundary condition of the operating point was approximately the same flow volume as that in the case of Fig. 20(a). Without the stator blade, the flow pattern at the downstream side is diverted at a right angle immediately near the fan outlet. On the other hand, with the stator blade, airflow at the downstream side is comparatively straight. These results confirmed that MRF fan modeling could correctly describe the rectifying effect of the stator blade on the exhaust flow of the axial fan at the operating point near the surging point.

As a comparison, the simulated flow speed contours obtained by a lumped fan model are shown in Fig. 21. Here, the airflow pattern is unrealistically straight and uniform. In this result, the actual flow distribution pattern around the fan shown in Fig. 9 as obtained by the MRF fan...
model is not simulated. Table 1 summarizes the simulation conditions for a series of simulations of air flow around the fan shown above.

3.2 P-Q characteristics of MRF fan model simulation

In this study, actual measurement of the P-Q characteristic of the experimental fan was conducted to verify the simulated fan characteristic obtained using the MRF fan model. The experimental apparatus for P-Q characteristic measurement was constructed according to the JIS B 8330 standard.[6] Figure 22 shows the measured P-Q characteristic of the axial fan used in this study. The P-Q characteristic was measured for both “with stator blade” and “without stator blade.”

Figure 22 shows that the fan with a stator blade and that without one have different P-Q characteristics. At the operating point, which is around the air flow of 0.3 m³/min, the pressure characteristic of the fan with a stator blade is higher than that of the fan without a stator blade. On the other hand, in the zero static pressure condition, the maximum air flow delivered by the fan with a stator blade is lower than that delivered by the fan without a stator blade. The reason for the difference in the P-Q characteristics is considered. At the operating point of approximately 0.3 m³/min, the rectifying effect of the stator blade as shown in Fig. 20(a) induces a pressure increase compared to the fan without a stator blade. On the other hand, in the zero static pressure condition, air flow pattern of the fan downstream side is not diverted even in the absence of the stator blade, as shown in Fig. 19. Therefore, in the zero static pressure condition, it is considered that the stator blade acts to obstruct flow through the fan tube, thereby causing a decrease of the maximum air flow.

Figure 23 (with stator blade) and Fig. 24 (without stator blade) show the simulation results obtained using the MRF fan model, along with the measured P-Q characteristic. In both results, there is a difference in the high-static-pressure side above the surging region. However, in the region of practical use, the simulated P-Q characteristic is

| Air flow direction | MRF zone size | Stator blade | Simulation result | Note         |
|--------------------|---------------|--------------|-------------------|--------------|
| Exhaust            | Normal        | Without      | Fig. 13 (a)       |              |
| Intake             | Normal        | Without      | Fig. 13 (b)       |              |
| Exhaust            | Extended      | Without      | Fig. 14 (a)       |              |
| Intake             | Extended      | Without      | Fig. 14 (b)       |              |
| Exhaust            | Normal        | Without      | Fig. 19           | Max flow point |
| Exhaust            | Normal        | With         | Fig. 20 (a)       | Actual use point |
| Exhaust            | N/A (Lumped fan model) | | Fig. 21 | |
in good agreement with the measured P-Q characteristic. Although further research is needed to elucidate such phenomena as the separation of blade flow and prerotation flow, which are unique phenomena in the high-pressure region of the axial fan, the simulation results of this study demonstrated that the MRF fan model simulates well the practical operating region, which is the higher air flow region above the surging point.

4. Conclusion

In this study, we examined the applicability of the MRF fan modeling method to CFD simulation for thermal design of electronic equipment. The following results were reported.

– It was confirmed that the MRF fan model is a practical solution for predicting the spreading and swirling exhaust flow of an axial fan by comparing the visualization result of an actual flow and the simulation result.
– The MRF model is able to simulate the rectifying effect of the stator blade on the exhaust flow of the axial fan.
– The simulation result of the P-Q characteristic was in good agreement with the measured P-Q characteristic in the region of flow volume higher than the surging region.

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