Evaluation of Airflow Prediction Methods in Compact Electronic Enclosures

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Abstract

During the design of forced convection cooled electronic enclosures, one of the most important parameter needed is the airflow through the enclosure. The airflow through the enclosure mainly depends upon the pressure drop in the enclosure and fan characteristics. Fan curves are often used in conjunction with the system pressure drop (impedance) characteristics to determine the airflow. The accuracy of the computational fluid dynamics tools depends mainly on accurately modeling pressure loss in the system (grilles, filters, etc.) and the accuracy of the fan curve data. These fan curves, which show the air delivery capacity of the fan at various pressure drops, are usually generated with no obstructions close to the fan. However, most modern electronic systems contain densely packaged components including airflow obstructions such as inlet and outlet grilles in close proximity to the fan. It is therefore possible that methods that use fan curves can often be inaccurate for predicting the airflow. Inaccuracies can also occur by using pressure loss data of grilles from engineering handbooks. The main objective of this study is to understand the accuracy of different methods of airflow prediction that rely on pressure loss and fan curve data when compared to experimental results obtained in a wind tunnel. The system used in this study is chosen to be representative of typical electronic systems, which include the major components such as fans, inlet and outlet grilles and an array of stacked printed circuit boards (PCB). Additional components such as capacitors, inductors, transformers and heat sinks are also included to increase the total pressure drop in the system. Variations to the base configuration are made by changing grille open area, fan size and using fans in series and parallel configurations. It is found that differences of up to 20% can occur depending on the method used to calculate the flow relative to the experimental method.

Key words: Fan curves, IcePak, wind tunnel, pressure loss coefficient, system pressure drop (Impedance) characteristics, CFD, series, parallel.

Introduction

With microelectronics technology steadily packing more chip power into small packages, cooling issues are becoming increasingly critical in the design process. Most modern electronic systems containing densely packaged components uses fans or blowers for forced air-cooling. Designing forced convection electronic systems requires one or more reliable ways to predict the airflow and pressure drop characteristics of the system. Although experiments continue to be important, especially when the flow is very complex, the trend is clearly towards greater reliance on computer based predictions in design because of cost effectiveness.

Most airflow prediction methods use one or both of the following: fan curves obtained from the manufacturer and pressure loss data of grilles and filters obtained from engineering handbooks. Before these methods can be used confidently some kind of validation needs to be done to verify whether the fan curves and the pressure loss coefficients of the grilles are estimated accurately. Biswas, Agarwal, Goswami and Mansingh (1998) [1] attempted this comparison with a simplified enclosure, which included only the major components as grilles, fan and stack of pcbs. The study showed that even in a simple model, differences of up to 20% can occur relative to the experimental results.

Mansingh and Misegades (1990) [2] used iterative technique with another CFD package called FIDAP to determine the operating flow and pressure drop through a computer system-processing unit. Numerically stimulated particle traces were recorded using video equipment. The particle traces showed some extremely interesting flow characteristics. Although they reported good agreement between measured and calculated pressures, the process required twelve iterations to reach a flow value within 10% of actual. This required ten hours of run time on a Cray Y-MP supercomputer. Additionally, 1.5 man-months were invested in creating the computer model. The authors indicated that qualitatively accurate flow paths were calculated A problem of similar complexity could be run on modern high-end workstations within a couple of hours.

Deiters and Hill (1991) [3] attempted comparison of test versus analysis using FloTHERM in a typical electronic enclosure used in industry. This particular study didn't yield a satisfactory agreement because measurements were made and related to analytic predictions close to the exhaust of the fan where airflow is very turbulent and simplified axial flow assumptions were not valid. However, Deiters and Hill did demonstrate the critical need to have an accurate measurement of the volume flow rate to be used as an input condition to the analytic method.

Physical Model

The system chosen in this study was similar to [1] that represented a typical simplified electronic system including all the major components such as fans, inlet and outlet grilles and an array of printed circuit boards. In this study, additional components on the PCBs such as capacitors, inductors and heat sinks were also included. No heat loads were applied in the system.

A test mockup (see figure 1) was built with the outer wall fabricated from plywood and internal components including boards and packages made of foamcore. The internal dimensions of the enclosure were 29cm lengths by 16 cm width by 14 cm height. An off-the-shelf 60x25mm axial flow, circular fan was selected to pressurize the system. The fan having a free airflow of 22 CFM was mounted on the center of the plate and was oriented to push air into the box. The distance of the fan from the front face of the enclosure was 3.25 cm. The width and height of the plate was 16 cm and 14 cm respectively.

A group of five printed circuit boards made with foam core having thickness of 5 mm was placed at a distance of 8.5 cm from the front face. The dimensions of printed circuit boards were 16cm length by 13.2cm width. The boards fit into the slots made with the plywood. The spacing between the boards was 2.6 cm. The components were placed on the printed circuit boards in a similar fashion (as shown in the figure) for simplicity. Both ends of the enclosure were covered with 33% open area grille. The grilles are made up with low carbon steel. The perforated metal has a hole diameter of 0.094 inch and center to center spacing of 5/32 inch staggered at 60 degree. Grille area is critical in an enclosure.

In addition to using the basic model described above, three variations to the model were studied. In the first case, the percent open area of the inlet and outlet grilles was varied from 23% to 40%. The grilles were 60° staggered and had round holes. The hole diameter of 23% open grille was 1/16 inch and center to center spacing was 1/8 inch. The hole diameter of 40% open grille was 1/8 inch and center to center spacing was 3/16 inch. In the second case the fan size was varied from 60mm to 92mm. In the third case, two fans were used in series and parallel. Fans used in series mode were 60×25mm fans having a free airflow of 22 cfm. The two fans were mounted centrally on two plates, placed at a distance of 3.25 cm from the front end and back end of the enclosure respectively. In the parallel mode both fans were mounted on a plate side by side located at a distance of 3.25 cm from the front end of the enclosure.

CFD Model

A computational fluid dynamics (CFD) model of the system was created using a software package called IcePak. It included the major air flow blockages such as printed circuit boards, inlet and outlet grilles, fans, etc. The fan was modeled using the fan curve data obtained from the manufacturer. The inlet and outlet grilles were modeled by specifying the pressure loss coefficient. The components such as capacitors, inductors and heat sinks on the PCBs were modeled by using blocks. The pressure loss coefficient was first found from an engineering handbook [4] and then measured experimentally using a wind tunnel. In IcePak the basic equation for mass, momentum and energy (Navier-Stokes) are solved numerically using the finite volume method. The total airflow was calculated from the velocity field obtained as part of the CFD solution.

Air flow prediction methods

1. Experimental method

In this method, the airflow through the system was measured directly using a wind tunnel. Since this data is independent of the fan curve and pressure loss data, this was chosen as the reference for comparing the other methods.

After turning on the power supply to the fan, the airflow in the system was allowed to stabilize for approximately 20 to 30 seconds. The differential pressure across the nozzle was measured and the corresponding airflow was determined using calibrated charts. In this study, the air flow ranged from 14.25cfm to 16.25cfm based on the variation of the grille open area.

2. Using performance curves

In this approach, the airflow was determined using fan performance curves provided by the manufacturer. It also involved generating a system impedance curve, showing the pressure drop variation with mass flow, using wind tunnel tests.

The system impedance curve was superimposed on the fan curve obtained from the fan manufacturer. The system air flow was found from the intersection of these curves.

3. Computational Fluid Dynamics (CFD):

The CFD software package used in this study is called IcePak. It solves the governing mass, momentum and energy (Navier-Stokes) equations numerically using the finite volume method. Since no heat loads were applied in the system and the energy equation is uncoupled from the other equations in forced convection, only the mass and momentum equations were solved.

The fan modeled was a circular, internal fan with a specified fan curve. The fan curve, which shows the relationship between the pressure drop and the airflow, was obtained from the fan manufacturer.

The pressure drop through grilles is usually provided as an input to the CFD tool as a pressure loss coefficient. In terms of the total number of mesh elements, this is a far cheaper method than modeling the actual hole pattern of the grille. These coefficients are generally found from handbooks using the percentage open area of the grille. Here, two different approaches were used for the pressure loss coefficients of the inlet and outlet grilles. In the first approach (called CFD-1), the pressure loss coefficients of the inlet and outlet grilles were obtained from an engineering data handbook by Idelchik [4]. The values used were from diagram 3-12. The pressure loss coefficient in this case was obtained using holes with sharp edges. In the second approach (called CFD-2), the pressure loss coefficients of the grilles were measured experimentally using wind tunnel tests.

The meshing was done using an unstructured mesh. This allowed the mesh density to vary from fine mesh to coarse mesh within the same model. The solutions obtained were also checked for mesh independence.

Results:

The air flow results obtained using the various prediction methods were compared with that obtained from experiment. The comparison metric is the percentage error, which is defined as

$$\text{Percentage Error} = \frac{\text{Method - Experiment}}{\text{Experiment}} \times 100\%$$

Method refers to CFD or using performance curves.

Figure 2 shows the comparison between the percentage error obtained using the different methods for different open area of the grilles. From this figure, it is found that as the percent open area increases, the relative error decreases. Figure 3 shows the comparison of the relative error for different fan sizes. The absolute value of percentage error increases with an increase in the fan size. As the fan size became closer to the enclosure dimensions, all the methods under-predicted the airflow compared to the experimental result.

Figure 4 shows the variation of percentage error with fans in series-parallel combination. No trends were observed in these cases. Nevertheless, the percentage error lies within 12%.

The maximum percentage error is the cumulative effect of the errors due to the fan curve, the pressure loss coefficient of the grilles and the numerical error. It was found that the maximum percentage error using the different airflow prediction methods was in the range of 5 to 20%. It reduced to 13% when the pressure loss coefficients were experimentally measured. This error could be further reduced to the range of 5 to 10% if the fan curve can be properly measured to account for flow obstructions close to the fan.

When the results were compared to the previous work [1] it was found that the total airflow and trends do not change significantly after adding components to the PCBs. This means that the pressure drop across the enclosure is mainly due to the presence of inlet and outlet grilles.

Conclusions:

In compact electronic enclosures the pressure loss due to the presence of the inlet and outlet grilles accounts for a substantial part of the total system pressure loss. For higher accuracy, the pressure loss coefficient should preferably be measured experimentally instead of using the values from data handbooks. The fan curve obtained from the manufacturer should be used with caution when predicting the airflow in a compact, densely-packed system, especially if the airflow is closely ducted around the fan. This study also confirms that CFD is a very useful and accurate tool in the cooling design of electronic systems.

References

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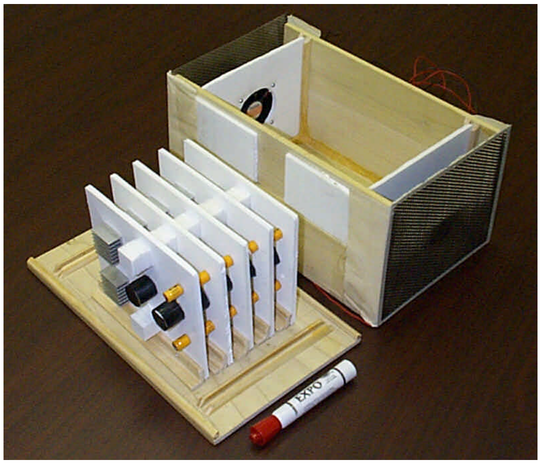

Figure 1: Test mockup of an electronic box

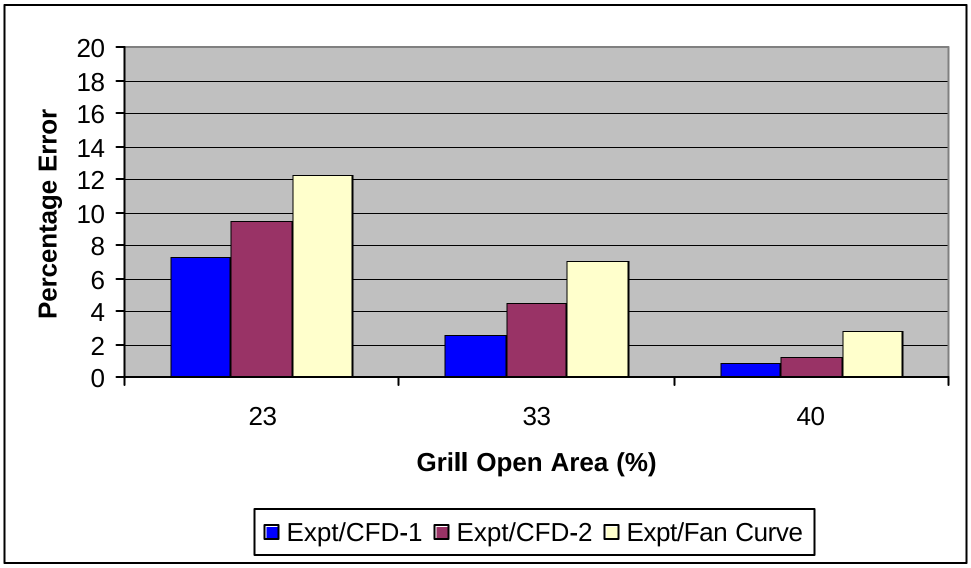

Figure 2: Percentage error in airflow versus percentage open area of grill

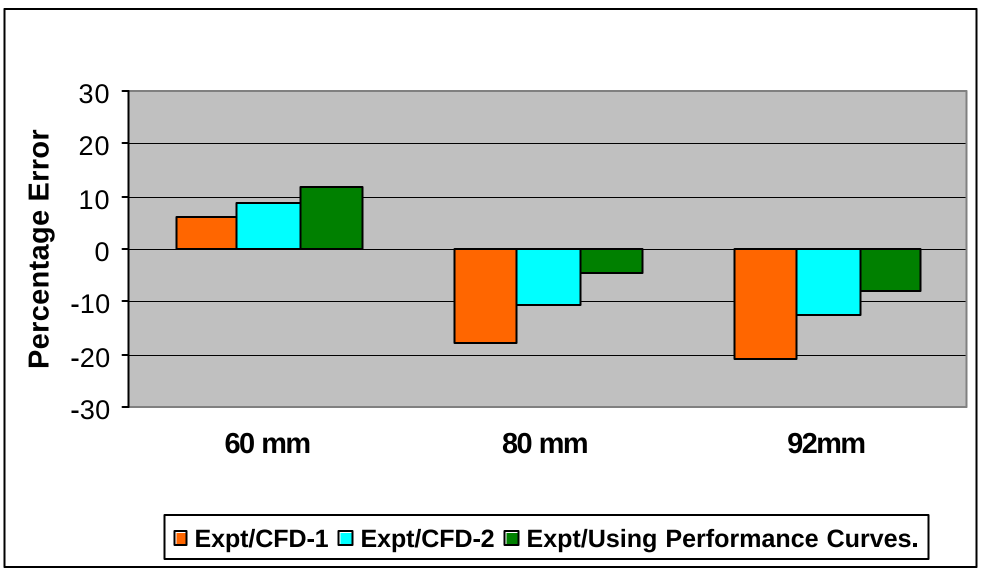

Figure 3: Percentage error in airflow versus fan size

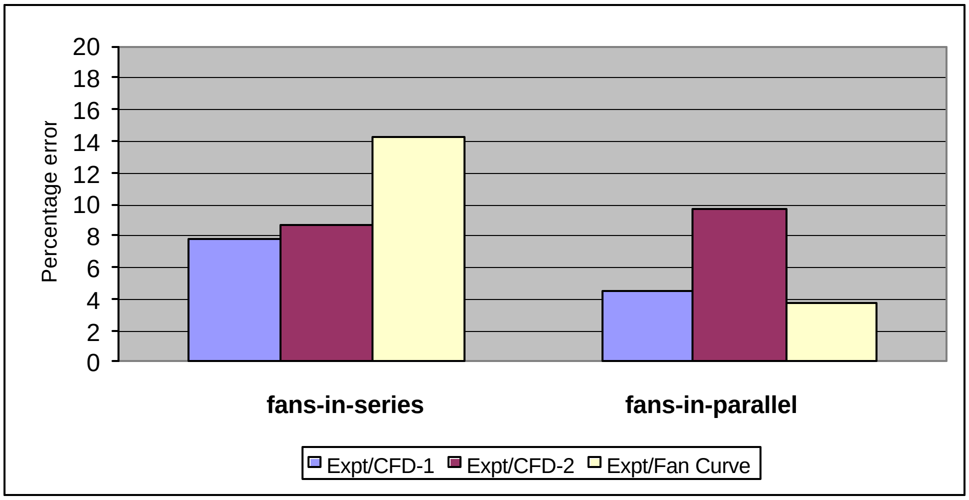

Figure 4. Percentage error in airflow with fans in series and parallel