Study on the natural air cooling design of electronic equipment casings: Effects of the height and size of outlet vent on the flow resistances

M Ishizuka, T Hatakeyama, R Kibushi and M. Inoue

Department of Mechanical Systems Engineering, Toyama Prefectural University, 5180 Kurokawa, Imizu, Toyama 939-0398, Japan

E-mail: ishizuka@pu-toyama.ac.jp

Abstract. This paper describes the effects of the outlet vent size and the distance between the outlet vent location and the power heater position on the flow resistance in natural-air-cooled electronic equipment casings. An experiment was carried out using a simple model casing simulated for the practical natural-air-cooled casing which is composed of 4 side walls, a top plate and bottom plate which has an inlet opening. A power heater to served as a power dissipation unit was placed at its open bottom. An outlet opening was set on one of the side walls. The opening area, the height of the outlet and the heater location were varied. The experimental results were analyzed using the flow resistance coefficient K which was related to the distance between the outlet vent and the power heater position and the heat removal from the outlet vent, and K values were plotted against a pair of Reynolds numbers Re and the outlet vent porosity β which is defined as the ratio of outlet vent open area to the top surface area of the casing.

1. Introduction

The power dissipation density of electronic equipment has increased in recent years and, as a result, it is necessary to consider the cooling design of electronic equipment in order to develop suitable cooling techniques. The literature [1] includes a number of studies on air cooling, which is an important subject because almost all electronic equipment are cooled by air convection. Of the various cooling systems of electronic equipment, natural air cooling is often utilized for those applications, such as telecommunication, for which high reliability is essential. The main advantage of natural convection is that no fan or blower is required, because air movement is simply generated by density differences in the presence of a gravity field. The optimum thermal design of electronic devices, cooled by natural convection, depends on an accurate choice of geometrical configuration and on heat sources distribution to be able to promote the thermo-circulation flow rate that minimizes the temperature rise (mean and maximum values) inside the casings.

Guglielmini et al. [2] have reported on the natural air cooling of electronic cards in ventilated enclosures. Ishizuka et al. [3] presented a simplified set of equations which represents the cooling capability on the basis of data on natural air cooling of electronic equipment casing. However, there are insufficient data for detailed discussion regarding the design of practical electronic equipment. For example, the simplified set of equations was introduced using a ventilation model like a chimney.
which was formed by a heater at the duct bottom and an outlet vent on the top. However, in practical
electronic equipment, the outlet vent is located at the upper part of the side walls, not on the top, and a
casing is not composed of a circular duct. Therefore, in the present work, an attempt is made to study
the effects of the distance between the outlet vent location and the power heater position on the
cooling capability of natural-air-cooled electronic equipment casings.

2. Set of equations

Ishizuka et al.\cite{3} proposed the following set of equations for engineering applications in the thermal
design of electronic equipment

\begin{equation}
Q = 1.78S_{eq} \Delta T_m^{1.25} + 300A_o \left( \frac{h}{K} \right)^{0.5} \Delta T_o^{1.5}
\end{equation}

\begin{equation}
K = 2.5(1 - \beta) / \beta^2
\end{equation}

\begin{equation}
\Delta T_o = 1.3 \Delta T_m
\end{equation}

\begin{equation}
S_{eq} = S_{top} + S_{side} + 1/2S_{bottom}
\end{equation}

where, $Q$ denotes the total heat generated by the components, $S_{eq}$ is the equivalent total source of the
casing, $\Delta T_m$ is the average temperature rise in the casing, $A_o$ is the outlet vent area of the casing, $h$
is the distance from the heater position to the outlet, $K$ is the flow resistance coefficient arising from the
air path blockage at the outlet, $\Delta T_o$ is the air temperature rise at the outlet vent and $\beta$ is the porosity
coefficient of the outlet vent. The flow resistance coefficient $K$ was obtained approximately as a
function of porosity coefficient of the outlet vent in the casing. The following relations for wire nets
have been obtained in a low Reynolds number range by Ishizuka\cite{4}.

\begin{equation}
K = 40(Re(1 - \beta) / \beta^2)^{-0.95}
\end{equation}

where $Re$ is defined on the basis of the wire diameter used in the wire nets. However, since the effect
of the Reynolds number on $K$ is less prominent than that of the porosity coefficient $\beta$, $K$ can be
approximated as a function of the porosity coefficient only. Therefore, Eq.(2) is considered to be a
reasonable expression for practical applications. The $h$ value in Eq.(1) is called chimney height and
Eq.(1) was introduced assuming a ventilation model like a chimney which is composed of a heater at the
duct bottom and outlet vent on the top as shown in Fig.1. However, in practical electronic
equipment, the outlet is located at the upper part of the side walls, not on the top, and casings are not
circular ducts. Following experiments were carried out by taking those factors into account.

\begin{figure}
\centering
\includegraphics[width=0.3\textwidth]{figure1.png}
\caption{Ventilation model}
\end{figure}

3. Experiments

3.1 Experimental apparatus
The experimental apparatus is shown in Fig.2 and consists of a casing having the following external dimensions: length x width x height = 220 x 230 x 310 mm$^3$. The casing surfaces were made of plastic whose thickness and thermal conductivity were 10 mm and 0.01 W/(mK), respectively. A wire heater which is shown in Fig.3, served as heat dissipation unit and is placed at a position 25 mm above the bottom of the casing. The diameter of the heater was 1.6 mm. On one of the side walls of the box, one rectangular shaped opening is left and the size and location of which were varied. The opening dimensions have six patterns and are showed in Fig.4. The experiments were performed for three different distances between the wall bottom and the center of the opening of the outlet vent ($H_v = 275$, 200, and 150 mm) and for three different vertical locations of the heater from the bottom of the wall($H_h = 25, 125, and 225$ mm). However, the experiments were carried out in such a way that the heater was always placed below the outlet vent. The cooling air enters through the casing bottom and is exhausted through the upper outlet opening. There is an inlet vent whose area is 150 mm x 130 mm in the central region of the casing bottom. The air temperature distribution inside the casing, the room temperature and the wall temperature have been measured by means of 14 calibrated K-type thermocouples ($\pm 0.1K$ and $0.1$ mm in diameter). Inside the casing, they are arranged at the different horizontal locations (30, 115 and 115 mm) from the inside of the left side wall for each three different height values from the wall bottom (75, 175, and 275 mm) as shown in Fig.2.

3.2 Estimation of heat removed from casing surfaces

At first, the amount of heat removed from the casing surfaces was estimated from the experiment. For the purpose of this estimation, the casing in Fig.2 was considered to be closed unit casing with no vents except the open bottom. That is, it consisted of a top and four side walls. Using the natural convective heat transfer equations for individual surfaces presented by Ishizuka et al[4], the amount of heat removed from the casing surfaces, $Q_s$, was expressed as follows.

$$Q_s = D_h \Delta T_m^{1.25}$$  \hspace{1cm} (6)
The Eq.(6) was introduced in the same way as that of the first term on the right hand side of Eq.(1). In Eq.(5), constant $D_1$ includes a radiative heat transfer factor as well. The amount of heat generation was within the $Q_s = 6 \text{ W} - 40 \text{ W}$ range, and the room temperature $T_a$ was 298K. Results obtained from this experiment are presented in Fig.5. It is found from this figure that the results are well-expressed by the following Eq.(7), in which the arithmetic mean temperature rise $\Delta T_m$ was obtained from measuring the temperature of the 9 internal locations in the casing.

$$Q_s = 0.445 \Delta T_m^{1.25}$$ (7)

where, the constant $D_1$ was determined as 0.445 to meet the measured values. Temperature distribution in the casing was relatively uniform and the maximum ratio of the temperature difference value to the temperature in the casing was 5%. Hereafter, the amount of heat removed from the outlet vent $Q_v$ was calculated on the basis of the following equation.

$$Q_v = Q - Q_s$$ (8)

3.3 Influence of outlet vent size on the temperature rise in the casing

The experiment was carried out using a reference casing in Fig.2 by varying the outlet vent size. The reference casing was defined as that outlet vent position was at $H_v = 275 \text{ mm}$ and the heater position was at $H_h = 25 \text{ mm}$. Porosity coefficient $\beta_0$, presented in Fig.4, is defined as the ratio of the open area of each individual vent to the area of the reference vent (150 mm x 50 mm). The results are shown in Fig.6.
In Fig.6, mean temperature rise $\Delta T_m$ in the casing is logarithmically plotted against the porosity coefficient of the openings, $\beta_o$ in log scale by varying the input power, $Q$ value. As the $\beta_o$ decreases, $\Delta T_m$ increases linearly in the logarithm plot. $\Delta T_m$ also increases, as the input power increases.

3.4 Influence of outlet vent position on the mean temperature rise in the casing

The relationship between internal temperature rise $\Delta T_m$ and outlet vent position $H$ was investigated by varying opening size under the condition that the heater position was fixed at the bottom. Fig.7 shows the experimental results. The outlet vent height was varied by preparing 3 kinds of side walls with 3 different outlet vent height values. $\Delta T_m$ decreases as $H$ value increases. However, the slope which was a little larger in the case of small $H$ value, becomes smaller in the range of $H>200$ mm. The relation between $\Delta T_m$ and $H$ for two opening size values exhibited a similar pattern. 3.5 Influence of the distance between the outlet vent position and the heater position on the mean temperature rise in the casing

The relationship between internal temperature rise $\Delta T_m$ and the distance between the outlet vent position and the heater position $h$ was investigated by varying opening size and input power value under the condition that the outlet was fixed at $H_o=275$ mm. Fig.8 shows the experimental results. As $h$ value increases, $\Delta T_m$ decreases. $\Delta T_m$ was defined as the average of the temperature values measured at the locations above the heater position in the casing. That is, the temperature values at the temperature locations under the heater position were not considered. The slopes of three point groups are a little different from each other.
4. Correlation of the data using non-dimensional parameters

4.1 Flow Resistance Coefficient $K$

The flow resistance coefficient $K$ was used to correlate the data obtained in the experiments, since the $K$ value was related to $Q_v$ and the distance $h$ as follows. A ventilation model, as shown in Fig.1, is considered. Assuming a uniform temperature distribution and a one-dimensional steady state flow, the two expressions below can be written, one for the overall energy balance and the other for the balance between flow resistance and buoyancy force.

$$Q_v = \rho c_p A u \Delta T$$  \hspace{1cm} (9)

$$(\rho_s - \rho) g h = K \rho u^2 / 2$$  \hspace{1cm} (10)

where, $Q$ is dissipated power, $\rho$ is air density, $c_p$ is specific heat of the air at constant pressure, $u$ is airflow velocity, $A$ is the cross-sectional area of the duct, $\Delta T$ is temperature rise, $g$ is acceleration due to gravity, $h$ is the distance between the outlet and the heater and $K$ is the flow resistance coefficient for the system. The subscript, $a$, is used to represent the atmosphere condition. Since the pressure change in the system is small, the expression is obtained assuming a perfect gas.

$$(\rho_s - \rho) / \rho = (T - T_a) / T_a$$  \hspace{1cm} (11)

and the equation for $K$ is obtained as follows,

$$K = 2gh\Delta T^3 / (T_a(\rho c_p A / Q_v)^2)$$  \hspace{1cm} (12)

Eq.(12) indicates the relationship among $K, h$ instead of $H$ in Fig.7 and $Q_v$.

4.2 Porosity coefficient $\beta$

In Fig.3, although the porosity coefficient of opening $\beta_o$ was defined as the ratio of the area of the opening to the reference opening, its value was generally unknown, since the data on reference opening was generally unavailable. Thus, the porosity coefficient $\beta$ is defined as follows in this study.

$$\beta = \text{open area in the outlet vent} / \text{inner casing top surface area}.$$  \hspace{1cm} (13)
4.3 Reynolds number $Re$

The Reynolds number was also used in order to correlate the data. The velocity $u$ was obtained from Eq.(9) and the hydrodynamic equivalent diameter of the openings was used as a reference length as follows with the height $A$ and the width $B$ of the opening.

$$L = \left(4 \times A \times B \right)/2 (A + B)$$  \hspace{1cm} (14)

Thus, the Reynolds number $Re$ is defined as follows.

$$Re = u L / \nu$$  \hspace{1cm} (15)

4.4 Relationship among $K$, $Re$ and $\beta$

In Figure 9, the variation of $K$ with $X$ is plotted. The parameter $X$ was defined on the basis of the method of correlation of the $K$ values for wire nets and perforated plates reported the author et al[3,-4].

$$X = Re \cdot \beta^2 / \left(1 - \beta \right)$$  \hspace{1cm} (16)

Almost all the $K$ values lie on the line obtained by the empirical correlation. The $K$ values obtained in the case of the reference condition that the outlet vent was at the upper position and the heater was kept at the bottom lie on the line plotted by the empirical correlation. However, the $K$ values in the non-reference case indicate slightly smaller values. This reason behind this discrepancy is expected to be due to measurement uncertainty, other than the change of basic phenomena. And the error due to measurement uncertainty composed of mean temperature rise values. The line was obtained from the best fit to the measure data as follows.

$$K = 0.4X^{1.5}$$  \hspace{1cm} (17)

where, the coefficient having a value of 0.4 is inherent in this apparatus and not a general value. This indicates that the distance between the outlet vent and the heat dissipation unit position $h$ can be considered as a chimney height in practical equipment as well and the definition of $\beta$ is reasonable. A more detailed discussion requires 3-dimensional thermo-fluid analysis and more precise measurement. However, it is considered that a useful relationship among the flow resistance coefficient, Reynolds number and porosity coefficient can be clarified from a practical point of view.
5. Conclusion

The effects of the size of outlet vent opening and the distance between the outlet vent and the power heater location on the flow resistance in natural air cooled electronic equipment casings were studied experimentally. From the results, the relationship among flow resistance coefficient $K$, Reynolds number $Re$ and the outlet vent porosity coefficient $\beta$ defined on the basis of the top surface are obtained as follows.

$$K = B \cdot X^{-1.5}$$

$$X = Re \cdot \beta^2 / (1 - \beta)$$

$B$ is the inherent coefficient of the casing in question.

6. References

[1] Bergles AE, 1990, Heat Transfer in Electronic and Microelectronic Equipment, Hemisphere. New York.
[2] Guglielmini D., Milano G., and Misale M., 1988, Second UK National Conference on Heat Transfer, 1 199.
[3] Ishizuka M., 1995, ASME-HTD-Vol.303, National Heat Transfer Conference-1 65.
[4] Ishizuka M., Miyazaki Y. and Sasaki T., 1987, ASME J.of Heat Transfer, 109 540