Simple Analytical Calculations for

Free Convection Cooled Electronic Enclosures

An electronic system is not complete without a rugged enclosure (a case or a cabinet) that will house the circuit boards and the necessary peripheral equipment and connectors, protect them from the detrimental effects of the environment, and provide a cooling mechanism. In a small electronic system such as a personal computer, the enclosure can simply be an inexpensive box made of sheet metal with proper connectors and a small fan. But for a large system with several tens of PCBs, the design and construction of the enclosure are challenges for both electronic and thermal designers.

Natural convection cooling is the preferred method of cooling for such enclosures, since, it is silent, reliable, and it is environmentally sound because no additional energy is used to remove the excess heat. On the other hand, natural convection cooling is more complicated to design because it may be challenging to identify if a certain application is suitable for fan-less cooling at the very early stage of the product development process. Computational Fluid Dynamics is one way of analyzing this problem this but it is time consuming and not a feasible way of determination at the very early stage. Instead, a quick estimation is required which will help this assessment. The "Better Box" analytical model developed by Luiten [1] is instrumental in doing quick calculations for convection cooled enclosures.

The "Better Box" model is actually an outcome of 2 different analytical models:

1. Simple Box Model: Box shaped enclosure that estimates air temperature rise inside the enclosure and the enclosure temperature with the available airflow.

2. Second Analytical Model: Estimates the airflow through the box as a function of temperature rise of the air.

The analytical model is placed in an excel spreadsheet and results are compared with those obtained from CFD simulations on 3 different LCD TV's.

The following text begins with the development of Simple Box Model followed by flow estimation and concluding with the comparison with CFD results.

Nomenclature

T = Temperature (^{o}C)
q = Heat flow, power dissipation (W)
A = Area (m^2)
h = Heat transfer coefficient ($Wm^{-2}\ K^{-1}$)
t = Thickness of the box wall (m)
ρ = Density of air(Kgm^{-3})
C_p = Specific heat of air ($Jkg^{-1}\ K^{-1}$)
Φ = Volume flow of air ($m^3\ s^{-1}$)
R = Thermal resistance (KW^{-1})

L = Length of the heated air column (m)
X = Vertical coordinate in the box (m)
F = Force (N)
G = Gravity (ms^{-2})
P = Pressure (Pa)
μ = Dynamic viscosity (Pas)
w = Flow velocity (ms^{-1})
f = Resistance factor (-)
CFD = Computational Fluid Dynamics
H, h = Height coordinate in the box (m)
FTV = Flat TV

The Simple Box Model

A box like product enclosure is considered as shown in figure 1 with uniform power dissipation inside the enclosure.

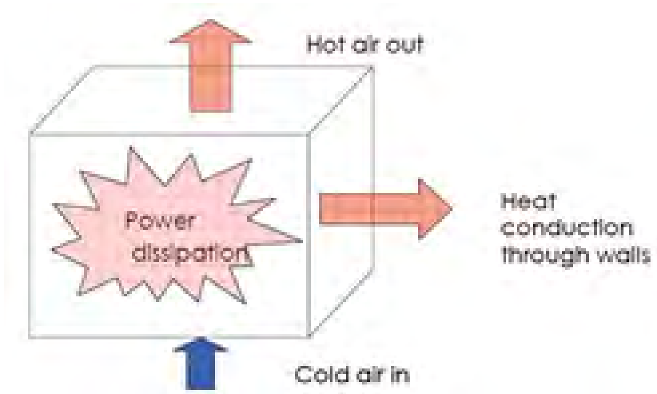

Figure 1. Simplified Box-Like Product [1]

For Energy balance, all the power dissipated inside the box must leave the box. There can be two ways the power can be dissipated; (a) one in which, the box is considered as closed box and all the power exits through the walls through material connected? to the walls. (b) The other can be, the box is considered as open box and all the power is put into the air through convection as a result of flow induced inside the box. It is assumed that, all the walls are at the same temperature and also that the power source is centered in the box, with no direct contact with the walls of the box. In this scenario, all the power is required to be transferred through convection and radiation only.

The closed box situation can be described as:

$$T_{in} - T_{out} = q_{wall} \cdot \left(\frac{1}{Ah_{in}} + \frac{t}{Ak} + \frac{1}{Ah_{out}} \right) \qquad (1)$$

Where:
T_{in} = temperature inside the box
T_{out} = temperature outside the box

For the open box situation the heat flow convected as a result of induced flow is:

$$q_{air} = \rho C_p \Phi \cdot (T_{heated} - T_{cold}) \qquad (2)$$

Air temperature rise inside the box:

$$\Delta T = T_{heated} - T_{cold} = T_{in} - T_{out} \qquad (3)$$

From Conservation of Energy: Total Power dissipated inside the box = Total power leaving the box through wall and air

$$q = q_{wall} + q_{air} \qquad (4)$$

Thermal network representation of Equation (1) to (4) is shown below:

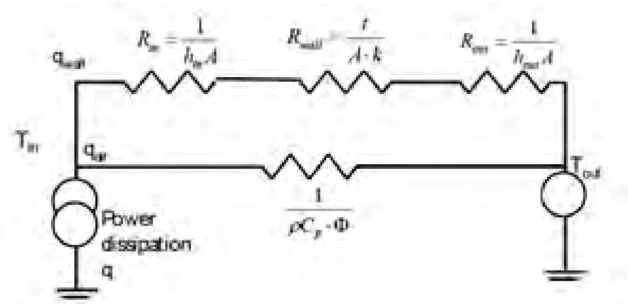

Figure 2. Thermal Network for the Simple Box [1]

Considering a linear temperature gradient of the air inside the box, from temperature at the bottom to the top of the box, equation (1) for the wall losses can be modified as:

$$(T_{top} + T_{bottom})/2 - T_{in} - T_{out} = q_{wall} \cdot \left(\frac{1}{Ah_{in}} + \frac{t}{Ak} + \frac{1}{Ah_{out}} \right) \qquad (5)$$

Assuming, ambient air enters at the bottom, such that, $T_{bottom} = T_{cold} = T_{out} = T_0$ and the hot air leaves the top of the box at $T_{heated} = T_{top} = T_1$ and rearranging equation (5) and (2) along with equation (4), results in:

$$(T_1 - T_0)/2 = q_{wall} \cdot \left(\frac{1}{Ah_{in}} + \frac{t}{Ak} + \frac{1}{Ah_{out}}\right) \quad (6)$$

$$q_{air} = \rho C_p \Phi \cdot (T_1 - T_0) \quad (7)$$

$$q = q_{wall} + q_{air} \quad (8)$$

Thus, using Simple Box model, we can determine the air temperature inside the box and the average temperature of the enclosure, provided that we know the air flow through the box, total power dissipation inside the box and the surface area of the box.

Flow Estimation

For a Natural convection cooled system, the air flow through the system and the temperature rise of the airflow depend on each other. This airflow is not known a priori and is determined by the balance between the driving pressure and the pressure drop from viscous and dynamic flow resistances.

The driving pressure is caused by buoyancy effect which needs to be determined. Let us assume that we have a box and we consider a slice of air from the box as shown in figure 3. A linear temperature distribution is considered over the height of the box such that, $T_0 = T_{ambient}$ at the bottom of the box at X=0, and T_1 at the top of the box at X = L. This leads to the following temperature relation:

$$T(x) = T_0 + \frac{(T_1 - T_0)}{L} x \quad (9)$$

The slice of air at location X and having thickness dx is subjected to the upward and downward force.

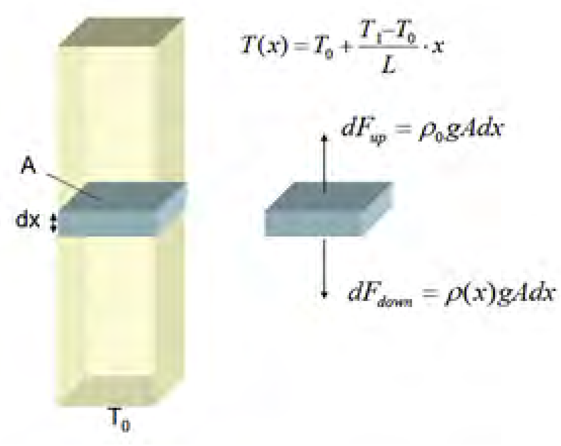

Figure 2. Thermal Network for the Simple Box [1]

The downward force is due to its weight and an upward force results from the Archimedes buoyancy law. Integrating the net force over the length L, we can obtain the buoyancy pressure over the box, using the ideal gas law [1] [2].

$$\frac{\Delta\rho}{\rho} = \frac{\Delta T}{T} \quad (10)$$

$$F\uparrow = \int_0^L (\rho_0 - \rho_x) \cdot g \cdot Adx = \int_0^L \frac{\Delta T(x)}{T} \cdot \rho \cdot g \cdot Adx = \frac{\rho \cdot g \cdot A}{T} \int_0^L \frac{T_1 - T_0}{L} \cdot xdx \quad (11)$$

$$F\uparrow = \frac{\rho \cdot g \cdot A}{T} \cdot \frac{(T_1 - T_0)}{L} \cdot \frac{1}{2} L^2 = \frac{\rho \cdot g \cdot A}{T} \cdot (T_1 - T_0) \cdot \frac{1}{2} L \quad (12)$$

$$\Delta\rho_{buoyancy} = \frac{\rho \cdot g}{T} \cdot (T_1 - T_0) \cdot \frac{1}{2} L \quad (13)$$

The airflow through the box is subjected to viscous and dynamic flow resistances which causes a pressure drop along the flow path. The viscous part of the pressure drop is related to viscous behavior of the flow and is proportional to the flow velocity w. The dynamic pressure drop is related to changes in either flow speed or flow direction, and it is proportional to $1/2\ \rho w^2$ [3] [4].

The total pressure drop due to viscous and dynamic flow resistance in the box is given as

$$\Delta\rho = f_{viscous} \cdot W + f_{dynamic} \cdot W^2 \qquad (14)$$

Where $f_{viscous}$ and $f_{dynamic}$ is the summation of the viscous flow resistance factors and dynamic flow resistance factors along the flow path, respectively. Both of these factors are with respect to the same velocity w. For flow through a cross sectional A, volume flow rate Φ, is given by Φ=A·w

Balancing the total pressure drop we get the following equation as:

$$\Delta\rho_{buoyancy} = \Delta\rho_{viscous} \cdot \Delta\rho_{dynamic} \qquad (15)$$

$$\frac{\rho \cdot g}{T} \cdot (T_1 - T_0) \cdot \frac{1}{2} L = f_{viscous} \cdot W + f_{dynamic} \cdot W^2$$

$$\frac{\rho \cdot g}{T} \cdot (T_1 - T_0) \cdot \frac{1}{2} L = \frac{f_{viscous}}{A} \cdot \Phi + \frac{f_{dynamic}}{A^2} \cdot \Phi^2$$

The resistance factors can be calculated from standard equations for blockages, filters, contractions, etc.

The Better Box Model

The combined set of Equations (6), (7), (8) and Equation (15) forms a "Better Box" model. So there are basically 4 set of equations for 4 unknowns T_1 and Φq_{air} and q_{wall}. The solution of the equation gives us the volume flow Φ through the box and $(T_1 - T_0)$, the temperature rise over the height of the box.

Comparison of Better Box Model with CFD Simulations

LCD TV's of 3 different configurations were selected for this study [1] [3]. The screen size and grid hole pattern were kept the same for all 3 TV designs, and power dissipation, vent area and product depth were varied. The better box model is implemented in commercial spreadsheet software which calculates different constants in the equations based on the mechanical design dimensions and total power dissipation, and the resulting equations are solved. The spreadsheet is additionally adapted to accommodate the presence of the display. In the adapted model, the heat transfer from the front surface (i.e. display surface) and the back surface is treated separately.

CFD simulation results for these 3 different TV sets matched well with the Better Box model [1] and the comparison is shown in Table 1 below.

Case	A		B		C	
Power Dissipation	100%		60%		60%	
Vent Area	100%		35%		100%	
Depth	100%		100%		70%	
	Spread-sheet	CFD	Spread-sheet	CFD	Spread-sheet	CFD
Air ΔT (°C)	28	28	30	26	28	24
Front Surface ΔT (°C)	19	18	10	10	10	10
Back Surface ΔT (°C)	7	10	8	8	7	6
Air Flow (liter/sec)	5.3	5.5	2.6	3.4	3.1	3.8

Table 1. Spreadsheet vs. CFD Results [1]

Comparison of Better Box Model with Thermal Measurements

Thermal measurements on 4 different FTV configurations were taken and compared with better box model. These FTV varied in screen size, set depth, ventilation grids and power dissipation. Thermocouples were mounted at different locations to measure the temperature of the outlet airflow. The temperature of the display surface and back surface were measured using Infra-Red camera [3]. The images are shown in Figure 4.

A good agreement is seen in the results obtained through Better Box model and measurement values. Snapshot of the same is shown in table 2.

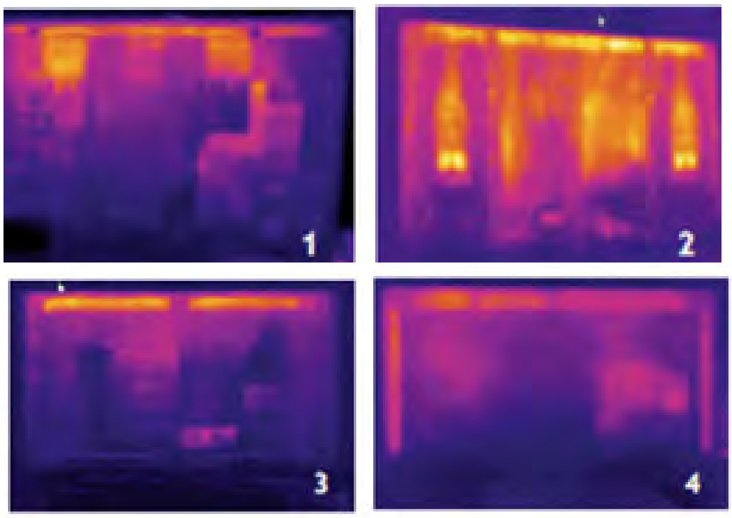

Figure 4. Thermal Images Using IR Photography [1]

Case	D		E		F		G	
Power Dissipation	85%		75%		100%		30%	
Vent Area	80%		30%		100%		70%	
Depth	100%		30%		100%		90%	
Screen Size	75%		100%		100%		80%	
Grid Hole Size	90%		100%		90%		50%	
	meas	calc	meas	calc	meas	calc	meas	calc
Air T-rise (°C)	22	19	29	30	20	17	16	16
Front Surface ΔT (°C)	18	23	20	21	11	17	9	8
Back Surface ΔT (°C)	6	6	13	10	5	6	6	5

Table 2. Spreadsheet vs. Thermal Measurements [1]

The comparison shows that the better box model can be used to determine if a certain product concept is suitable for natural convection or not.

The better box model requires very limited information available in the pre-concept product design phase. The evaluation is fast and it identifies opportunities for improved cooling, thus offering more promising cooling options in a pre-CFD stage and in time efficient manner. This facilitates effective and focused use of CFD time and CFD resources in the total product development cycle.

References:

1. **Luiten, W., Kadijk, S., Philips Consumer Lifestyle Innovation lab, Philips Applied Technologies, the Netherlands, "The Better Box Model: An Analytical Estimation of Temperature and Flow in a Free Convection Air Cooled Electronics Enclosure"**

2. **"Fundamentals of Heat and Mass Transfer", by Frank P. Incropera, David P. DeWitt**

3. **www.electronics-cooling.com**

4. **"Heat and Mass Transfer – A Practical approach", by Yunus Cengel**