Technical Brief …

NATURAL (FREE) CONVECTION - NO FAN
WHAT IS IT AND HOW DOES IT WORK?

Background:

Numerous electronic devices rely on Natural Convection for their cooling. It is an attractive technique because it is simple, and with no fan – both reliable and silent. The drawback is that the cooling capacity is limited. All possible means to enhance Natural Convection performance are therefore very interesting for thermal designers. The issue for this article is the combination of Natural Convection and Chimneys. The impact of a Chimney is by no means radical but if properly designed it can sometimes contribute with the BOOST needed to make a solution attractive.

The Chimney Effect:

Figure 1 shows a volume that is fully and partially filled with parallel plates. The latter will here be referenced as the chimney case. Weather one case can dissipate more heat than the other is an interesting question. A correct answer requires extensive calculations but it is possible to use much simpler means to get a notion.

The driving force for the convection is the density difference between the air inside and outside of the volume. The lowest density is always found at the highest air temperature, which in both cases is at the outlets from the channels formed by the plates. For the Chimney Case however, this happens at a low section of the volume and the result is a higher average density difference than for the fully filled case, Figure 1. This advantage can either be used to increase the Air Velocity or to increase friction and the plate surface by placing the plates closer together. Both will result in an improved cooling. It is on the other hand not probable that the Chimney Case can house as much plate surface as the fully filled case, which is a disadvantage.

Optimum Condition Equations for the Chimney Case:

Optimum plate spacing

(1) $S_{opt} = k_s \cdot \left(\frac{H_s}{dT}\right)^{0.25} \cdot \left(\frac{H}{H_s}\right)^{0.5}$

Optimum heat transfer cofficient

(2) $h_{opt} = 0.862 \cdot k_l \cdot \left(\frac{dT}{H_s}\right)^{0.25} \cdot \left(\frac{H_s}{H}\right)^{0.5}$

Total heat dissiaption

(3) $\dot{Q} = k_v \cdot A_c \cdot H_s^{0.5} \cdot dT^{1.5}$

Applied equation

(4) $\dot{Q} = \eta_v \cdot k_v \cdot A_c \cdot H_s^{0.5} \cdot dT_{ref}^{1.5}$

h_0 = Heat transfer coefficient [W/m^{2}K]
dT = Temperature difference [K]
H = Height [m]
S = Plate spacing [m]
$A_c = W \cdot D$
η_v = Volumetric efficiency [--]
$\dot{Q}$ = Heat dissipation [W]

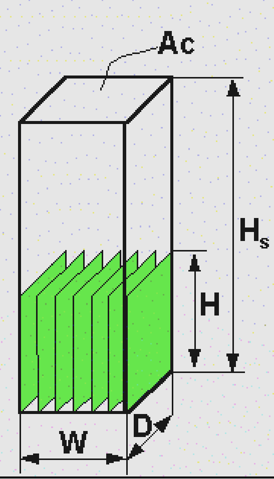


Figure 2

Figure 2 shows the most important optimum equations for isothermal parallel plates. The corresponding proportionality parameters are listed in table 1. The equations are valid for chimneys of modest height, or more precisely expressed; when the friction forces in the Chimney are small compared with those between the plates. This is also typically the case for Electronics Cooling Applications.

The equations are almost identical with the ones for pure parallel plates. The only difference is that there is a Chimney-to-Plate Height Ratio term in some of them. Another and very interesting phenomenon, is that the total heat dissipation is independent of Chimney-to-Plate Height Ratio! In more practical terms this means that each volume has a Heat Dissipation Ceiling that not can be exceeded - no matter how the plate surfaces in that volume are arranged.

Table 1
Parameters for the Equations in Figure 2

Air temp [C]	Kl	Ks	Kv
-20	1.51	0.0226	116
-10	1.50	0.0236	109
0	1.48	0.0247	104
10	1.47	0.0257	98.5
20	1.46	0.0268	93.9
30	1.44	0.0279	89.3
40	1.43	0.0290	85.2
50	1.42	0.0301	81.5
60	1.41	0.0311	78.0

Equation 4 is a more applied version of equation 3. It has an additional efficiency term that handles all deviations from the ideal case. The temperature difference has also been indexed in order to clarify that it can be used for non-isothermal cases. The Volumetric Efficiency for actual designs can vary greatly. For heat sinks it is usually in the 70% - 90% range. For natural convection enclosures filled with PCBs its much lower, typically 20% - 30%.

Filling a volume with heat transfer surfaces:

Figure 3
The Maximum Heat Dissipation is Independent of the Chimney-to-Plate Height Ratio.

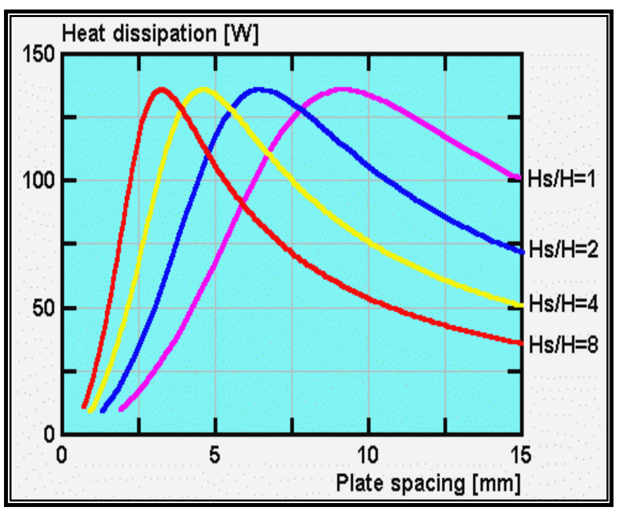


Figure 3 shows a typical result of a calculation that involves several Chimney-to-Plate Height Ratios, (Hs/H). The main impact of this parameter is that it tends to decrease the optimum plate spacing. The total heat dissipation is, as expected, the same for all cases.

There are several issues to consider when going from the theoretically perfect world into the real world. A discussion on this level can never be quite scientifically stringent. The problems brought up are however typical for applied thermal design and must therefore be managed somehow.

The first problem is very rudimentary and has to do with the Plate, (Fin) Spacing. It is always confined to discrete values. It is therefore rarely possible to create a design that operates exactly on the optimum point. The selected spacing should however rather be on the right side of the optimum point than on the left side. The simple reason is that the curves on the right side have a lower slope.

A second problem is that the orientation of the heat transfer surfaces sometimes not even is near a parallel plate arrangement. A reasonable assumption here is that the general tendencies in Figure 3 are valid for all kinds of vertical surfaces, regardless of their orientation. A volume would - in this line if thinking- rather be characterized by an optimum total heat transfer surface than by an optimum plate spacing. An issue that can be approached in this way is heat sinks mounted on Natural Convection cooled PCBs. It is apparent that it is problematic to introduce a heat sink between two PCBs that already are on an optimum distance. The result will in most cases be an increased temperature! Natural convection is however very sensitive to Delta T, (dT) and if that heat sink could be given a temperature considerably above that of the PCB, there could possibly possible be some gains. This does not mean that heat sinks should not be used for Natural Convection Cooled PCBs. It simply means that the PCBs in that case must be placed on a larger than optimum distance. This distance is quite difficult to calculate but simple considerations based on the parallel plate case can at least give an indication in which range it can be found.

A third problem is that plates always have a thickness. For flat plates with a thickness of a millimeter or two, (0.040” to 0.080”) it is seems reasonable to interpret the spacing as a wall-to-wall distance rather than as a centre-to-centre distance. For PCBs it is more complicated. The components must in that case be simulated as an extra surface layer with a thickness that by no means is simple to calculate. Typical PCB pitches are however above 15 mm and if the height of the components are only a couple of millimeters, it is apparent that an exact thickness compensation is not crucial. It can in this context also be added that the boundary layer thickness for Natural Convection typically is in the 4 to 10 mm, (0.160” to 0.397”) range. Most components will be embedded in that layer. Their physical surface enhancements, (i.e. surface roughness), will therefore have a negligible effect on the heat transfer at this level.

A fourth problem has to do with Natural Convection Cooled Enclosures. They can be designed in a many different ways, with and without Chimneys. From an overview perspective and just considering the total heat dissipation, it can be concluded that none of these designs has any advantage over the other. The difference is that Chimney arrangements usually result in a higher allowed heat density on each single PCB.

Making a Chimney work:

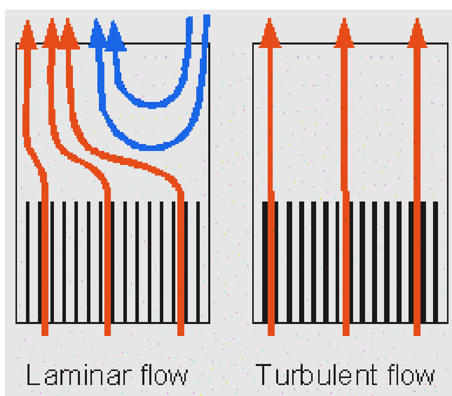


Figure 4

Laminar Flow in the Chimney creates a Re-Circulation Region at the Air Outlet, Completely eliminating the Benefits of the Chimney.

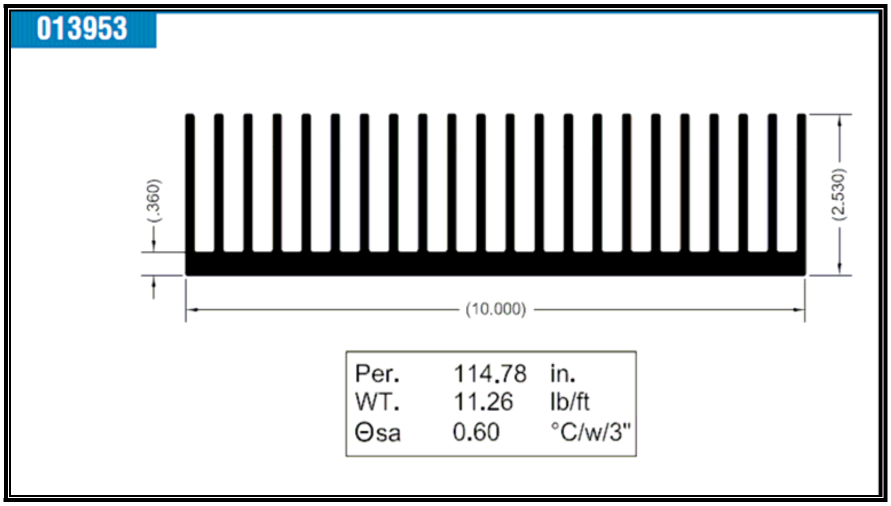
There are also some difficulties associated with the Chimney Effect. One of them is the Flow Characteristic in the Chimney. If it is laminar it will also be non-uniform and tend to create a region of re-circulation at the outlet, Figure 4. The phenomenon is closely related to the Cohanda Effect, (adherence of a jet to a wall). This impact totally neutralises the benefits of a Chimney. A coarse estimate, based on 0.3 m/s Air Velocity, indicates that the critical size limit is somewhere around a Hydraulic Diameter of 140 mm.

A similar phenomenon also appears for very large cross sections. An example of this is convectors in cold stores. If there is a Chimney it is in this case rather a skirt but the physics is the same. That skirt will nevertheless only function well provided that it is subdivided into a number of smaller sections.

There is also a problem associated with heat sinks. They can be looked at as parallel plates if the fins are high enough. The difficulty is that the fins must be made sufficiently thick to keep the fin temperature on a reasonable level. Placing a heat sink in a Chimney always results in a reduction of the total heat transfer surface, which increases the heat density and thus require thicker fins. Very high Chimney-to-Plate Height Ratios are therefore excluded simply because the fins would become too thick and choke the airflow. It is difficult to give any general guidelines for this effect but Chimney-to-Plate Height Ratios as high as 4 have been successfully tested.

Example

A side of a box must be cooled by Natural Convection, The available volume is (width, height, depth), 10.0",(0.254m) x 2.53",(0.0643m) x 8.0",(0.203m),. The allowed temperature difference is 10 C and the maximum room temperature is 50 C.



Solution(s)

A. Estimate how much heat that can be dissipated by Natural Convection from this volume with a temperature rise of 10C?

- n_v = 85% = 0.85 (Assumed Volumetric Efficiency)
- K_v = 81.5 (From Table 1 – Parameters - at 50C Ambient)
- A_c = cross section - Width X Depth = 10.0",(0.254m) X 8.0",(0.203m) = 0.0516m^2
- H_s = Heat Sink Height = 2.53",(0.0643m)
- dT_{ref} = Delta T between Heat Sink and Ambient – 10C

Applied equation

$$(4)\quad \dot{Q} = \eta_v \cdot k_v \cdot A_c \cdot H_s^{0.5} \cdot dT_{ref}^{1.5}$$

PWR = 0.85 X 81.5 X 0.0516 X $(0.0643)^{0.5}$ X $(10)^{1.5}$

25.0W = 0.85 X 81.5 X 0.0516 X 0.254 X 31.62

B. Estimate how much heat that can be dissipated by Natural Convection if we increase the height of the heat sink – 2X - from 2.53" to 5.060" with a temperature rise of 10C?

- n_v = 85% = 0.85 (Assumed Volumetric Efficiency)
- K_v = 81.5 (From Table 1 – Parameters - at 50C Ambient)
- A_c = cross section - Width X Depth = 10.0",(0.254m) X 8.0",(0.203m) = 0.0516m^2
- H_s = Heat Sink Height = 5.060", (0.128m)
- dT_{ref} = Delta T between Heat Sink and Ambient – 10C

Applied equation

$$(4)\quad \dot{Q} = \eta_v \cdot K_v \cdot A_c \cdot H_s^{0.5} \cdot dT_{ref}^{1.5}$$

PWR = 0.85 X 81.5 X 0.0516 X $(0.128)^{0.5}$ X $(10)^{1.5}$

40.6W = 0.85 X 81.5 X 0.0516 X 0.359 X 31.62

C. Estimate how much heat that can be dissipated if the heat sink at 2.53" height is embedded in an 10.0", (254mm) high chimney with a temperature rise of 10C?

Calculate the Hydraulic Diameter of the chimney as:

(4) X (cross section) / (circumference) = (4) X (0.254 X 0.203) / (2 X (0.203 + 0.254)) = 0.225 m.

Assume Kinematic Viscosity = 16.9E-6 m^2/s.

Calculate Reynolds Number as:

Assume an Air Velocity of 0.25 m/s, (50 LFM).

(Velocity) X (Hydraulic Diameter) / (Kinematic Viscosity) = 0.25 X 0.225 / 16.9E-6 = 3328

Compare this number with the laminar limit, which is 2300.

- It is clearly on the turbulent side.
- The velocity assumption was therefore valid.

With the flow in the Chimney as turbulent and based on the same data as above except the height, which now is ~ 10.0", (0.254m), the answer is:

Applied equation

$$(4)\quad \dot{Q} = \eta_v \cdot K_v \cdot A_c \cdot H_s^{0.5} \cdot dT_{ref}^{1.5}$$

PWR = 0.85 X 81.5 X 0.0516 X $(0.254)^{0.5}$ X $(10)^{1.5}$

57.0W = 0.85 X 81.5 X 0.0516 X 0.504 X 31.62