OVERVIEW

Simplified thermal management is one of the benefits of using Vicor converters. High operating efficiency minimizes heat loss, and the low profile package features an easily accessible, electrically isolated thermal interface surface.

Proper thermal management pays dividends in terms of improved converter and system MTBFs, smaller size and lower product life-cycle costs. The following pages provide guidelines for achieving effective thermal management of Vicor converters.

EFFICIENCY AND DISSIPATED POWER

A DC-DC converter takes power from an input source and converts it into regulated output power for delivery to a load. Not all of the input power is converted to output power however; some is dissipated as heat within the converter. The ratio of delivered output power to converter input power is defined as the converter's efficiency. Efficiency is a basic figure of merit that can be used to relate power dissipation directly to converter output power, as illustrated in Figures 20–1a and 20–1b.

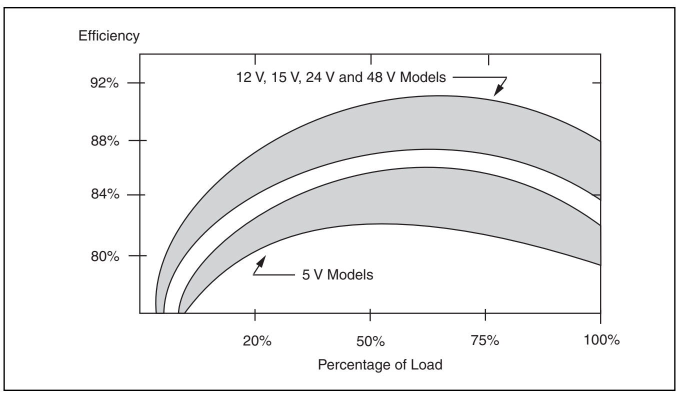

***Figure 20–1a** — Module efficiency*

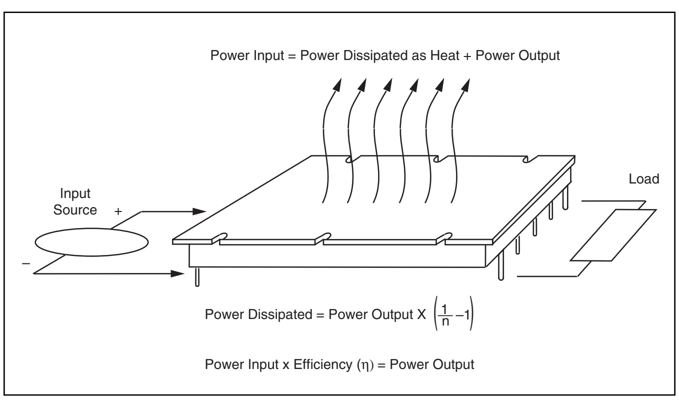

***Figure 20–1b** — Dissipated power*

The first step in evaluating cooling requirements is to calculate worst-case dissipation based on converter efficiency and worst-case anticipated load power. Clearly, higher efficiency will translate into lower power dissipation and simplify the cooling problem. Vicor converters are among the most efficient converters available, with full load efficiencies typically in excess of 80%.

REMOVING HEAT FROM VICOR CONVERTERS

Heat is removed from Vicor converters through the flat metal baseplate on top of the module. The baseplate is thermally coupled to, but electrically isolated from, all internal heat-generating components. The basic thermal design problem is to transfer heat from the baseplate into the surrounding environment as a means of maintaining baseplate temperature at or below rated maximum.

Heat energy is transferred from regions of high temperature to regions of low temperature via three basic mechanisms; radiation, conduction and convection.

Radiation. Electromagnetic transfer of heat between masses at different temperatures.

Conduction. Transfer of heat through a solid medium.

Convection. Transfer of heat through the medium of a fluid; typically air.

All three of these heat transfer mechanisms are active to some degree in every application. Convection will be the dominant heat transfer mechanism in most applications. Nondominant effects will provide an added contribution to cooling; in some cases, however, they may result in undesirable and unanticipated thermal interactions between components and subassemblies.

All three of these mechanisms should be given consideration when developing a successful cooling strategy.

RADIATION

Radiant heat transfer occurs continuously between objects at different temperatures that are exposed to each other. The net effect on the temperature of an individual part is dependent on a great many factors, including its temperature relative to other parts, relative part orientations, surface finishes and spacing. The difficulty in quantifying many of these factors, combined with the universal presence of radiant energy exchange, makes calculation of radiational temperature effects both a complex and generally imprecise task.

Temperature differentials encountered in practical applications of Vicor converters are never large enough to cause radiational cooling to be the dominant heat transfer mechanism. Radiation will account for less than 10% of total heat transfer in the majority of cases. For these reasons, the presence of radiant cooling is often assumed to provide safety margins over and above the dominant cooling mechanism, and detailed consideration of its effects are neglected. A valid assumption, in most cases, is that the converter will be warmer than its surroundings and radiant energy transfer will aid cooling. In some cases, however, nearby objects (PC boards, power resistors, etc.) may be much hotter than the converter and net radiant energy transfer may actually increase the converter's temperature.

Surveying the relative positions and estimated temperatures of converters and surrounding parts is advisable as a means of anticipating the potential effects of radiant transfer. In cases where hot components are in close proximity to the converter, the use of interposing barriers can generally moderate undesirable radiational heating effects.

CONDUCTION

In most applications, heat will be conducted from the baseplate into an attached heat sink or heat conducting member. Heat conducted across the interface between the baseplate and mating member will result in a temperature drop which must be controlled. As shown in Figure 20–2, the interface can be modeled as a "thermal resistance" in series with the dissipated power flow. The baseplate temperature will be the sum of the temperature rise in the interface and the temperature of the member to which the baseplate is attached.

Temperature rise across a surface interface can be significant if not controlled. The area of the interface should be as large as possible, and the surface flatness of the attached member should be within 5 mils. Thermal compound or a thermal pad should be used to fill surface irregularities. Thermal resistance across surface interfaces can be held to under 0.1˚C/Watt with proper measures.

Many applications require that heat be conducted from the baseplate of the converter to a "remote" dissipative surface via a thermally conductive member. The resulting baseplate temperature will be the sum of the temperature of the dissipative surface, the temperature rise in the heat conducting member, and the rises across the two surface interfaces. The thermal resistance of the conductive member is proportional to its length, and inversely proportional to both its cross-sectional area and thermal conductivity (Figure 20–3). Minimizing total temperature rise is dependent on controlling interface resistance, as described above, and controlling the thermal resistance of the transfer member through appropriate material selection and dimensioning.

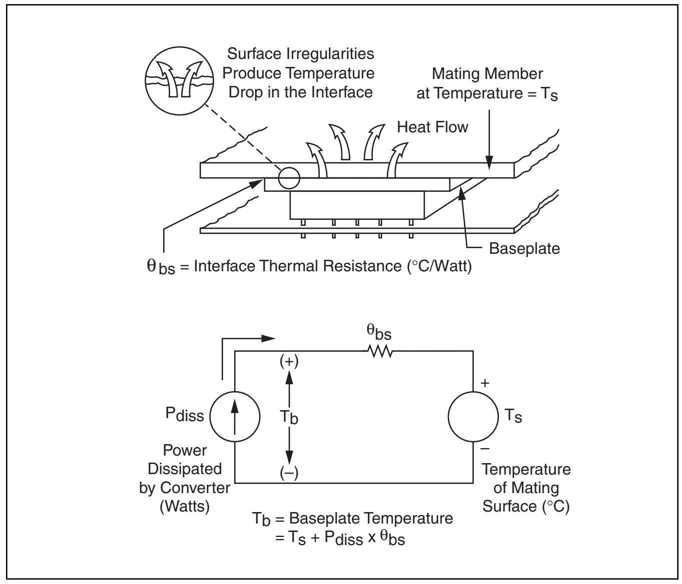

Figure 20–2 *— Baseplate thermal considerations*

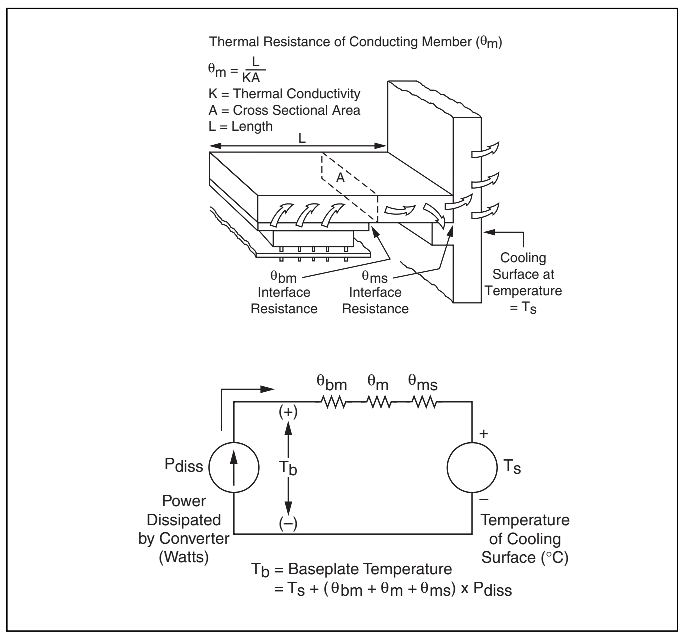

Figure 20–3 *— Interface thermal considerations*

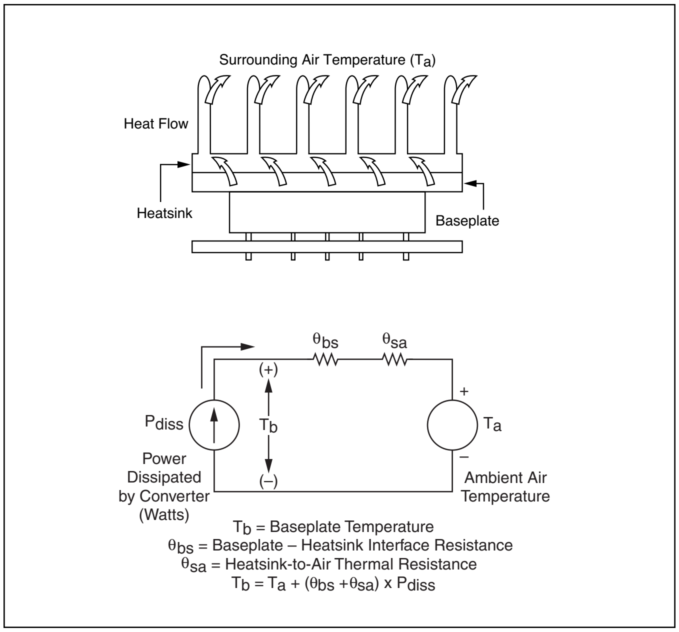

Figure 20–4 *— Heat sink thermal considerations*

CONVECTION

Convective heat transfer into air is a common method for cooling Vicor converters. "Free" or "natural" convection refers to heat transfer from a dissipative surface into a cooler surrounding mass of otherwise still air; forced convection refers to heat transfer into a moving air stream.

The convection cooling model is shown in Figure 20-4. Baseplate temperature depends on the temperature of the air, total dissipated power and the values of two thermal resistances; the thermal resistance of the surface interface between the baseplate and the heat sink, and the heat sink-to-air thermal resistance. Surface interface resistance can be minimized as discussed under Conduction. The heat sink-to-air resistance is dependent on a variety of factors including heat sink material and geometry, air temperature, air density and air flow rate. Fortunately, thermal resistance data is available for a very wide range of standard heat sinks for use in both free and forced convection applications. The following sections will provide guidelines for both free and forced convection cooling of Vicor converters and configurables.

FREE CONVECTION

The benefits of free convection include low cost of implementation, no need for fans, and the inherent reliability of the cooling process. Compared to forced air cooling, however, free convection will require more heat sink volume to achieve an equivalent baseplate temperature.

To select a suitable heat sink for free convection cooling, follow these steps:

1. Determine the power to be dissipated by the heat sink. This should be based upon converter efficiency and worst-case converter power output using the formula given in the section on Module Efficiency and Dissipated Power. (Figures 20–1a and 20–1b)

$$\text{Power Dissipated} = \text{Power Output } \times \left(\frac{1}{n} - 1\right)$$

2. Estimate or experimentally determine the surface interface thermal resistance. Use of thermal compound or a thermal pad is recommended to minimize this resistance. An estimate of 0.2°C/Watt should provide an adequate safety margin.
3. Referencing Figure 20–4, we can derive the following formula for heat sink-to-air thermal resistance:

$$\theta_{sa} = \left(\frac{T_b - T_a}{P_{diss}}\right) - \theta_{bs}$$

T_a = Worst case anticipated operating ambient air temperature.

θ_{bs} = Surface interface thermal resistance, from Step 2.

P_{diss} = Worst-case power dissipation, from Step 1.

T_b = Baseplate temperature.

Start with a value of Tb = 85°C (or 100°C, VI-J00) to determine the maximum acceptable heat sink-to-air thermal resistance.

4. Select several heat sinks that appear physically acceptable for the application. Using data provided, obtain values for their free convection thermal resistance, preferably at worst-case ambient temperature, Ta. If values obtained are less than the value calculated in Step 3, go on to Step 5. If the values are greater, then either a physically larger heat sink will be required or a different cooling method will need to be used (i.e., forced air, etc.).
5. Select the heat sink with the lowest available thermal resistance consistent with space and cost limits. Keep in mind that small reductions in baseplate temperature produce dramatic improvements in MTBF.
6. Baseplate temperature can be estimated by using the following formula:

$$T_b = T_a + P_{diss} \times (\theta_{bs} + \theta_{sa})$$

7. Test to verify that performance is in line with expectations.

Heat sink data is almost always given for vertical fin orientation. Orienting the fins horizontally will reduce cooling effectiveness. If horizontal mounting is mandatory, obtain relevant heat sink performance data or use forced convection cooling.

Free convection depends on air movement caused by heat-induced density changes. Thermal resistance data is dependent on the heat sink fins being completely exposed to the ambient air without any significant interference to air flow at the ends of or along the length of the fins. If packaging will tend to block or baffle air movement over the fins, a larger heat sink might be required. In the worst case, free convection may be ineffective. Make sure that the fins are well exposed to ambient air.

It is not necessary to limit the size of the heat sink to the size of the baseplate. Heat sinks with footprints larger than the baseplate area can often be used to advantage. In the latter case, heat must be conducted along the base surface of the heat sink to get to the outer fins, so don't count on achieving full cooling capability. Also, several modules can be mounted to a common heat sink, but cooling calculations must now take into account total power dissipation with consideration given to possible localized overheating if worst-case converter power dissipations are greatly imbalanced. When securing a PC board containing two or more converters to a heat sink, it is good practice to use sockets on the converter pins to allow for mechanical alignment. If sockets are not used, be sure to mount the converters first mechanically, then solder the units in place. A fixture should be used to maintain alignment if soldering must be performed before attachment.

When mounting heat sinks to Vicor modules, use #6 or M3.5 screws torqued uniformly through the mounting slots provided. The following tightening sequence should be used:

- Lightly finger-tighten all screws
- Torque screws to 6 in-lbs (0.7 N – m) per Figure 20–5.

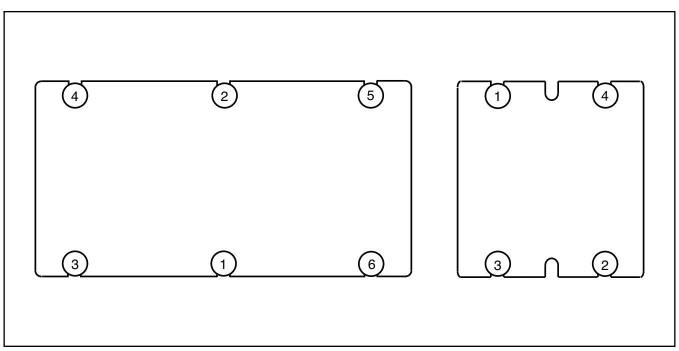

Figure 20–5 *— Heat sink torquing sequence VI-200 / VI-J00*

Multiple Modules Using Common Fasteners. The following mounting scheme should be used to attach modules to a heat sink for two or more modules. A large, heavy washer should be used on the common fasteners to distribute the mounting force equally between modules. The torquing sequence shown in Figure 20–6 can easily be expanded from two to any number of modules. An array of three is shown.

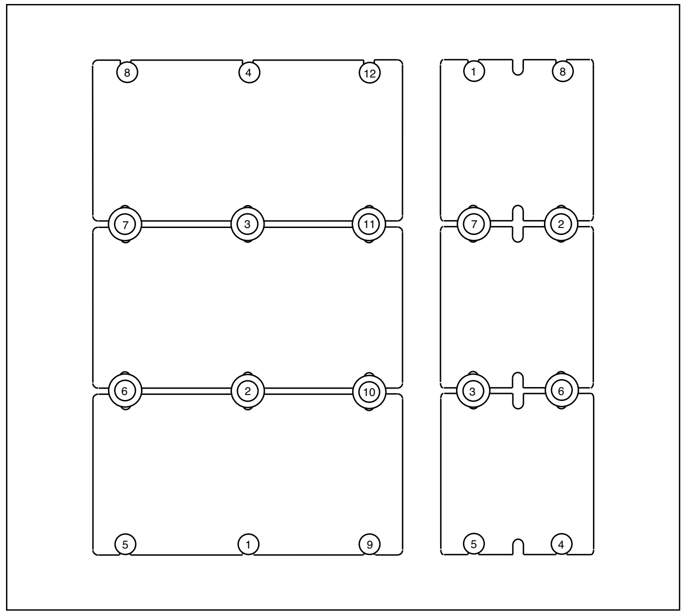

Figure 20–6 *— Torquing sequence, multiple VI-200 / VI-J00 converters*

FORCED CONVECTION

Forced air can make a great difference in cooling effectiveness. Heat sink-to-air thermal resistance can be improved by as much as an order of magnitude when compared to free convection performance, by using suitable heat sinks. Consider the following data for baseplate-to-air thermal resistance (no heat sink) of a VI-200 or VI-J00 module at various airflow rates:

Airflow	VI-200 Baseplate to Air Thermal Resistance	VI-J00 Baseplate to Air Thermal Resistance
Free Air	5.1˚C/W	8.1˚C/W
200 LFM	2.8˚C/W	5.1˚C/W
400 LFM	1.8˚C/W	2.7˚C/W
600 LFM	1.4˚C/W	2.3˚C/W
800 LFM	1.2˚C/W	1.7˚C/W
1,000 LFM	1.0˚C/W	1.4˚C/W

Table 20–1 *— Baseplate-to-airflow thermal resistance (no heat sink)*

Forced air implies the use of fans. Many applications require that fans be used to achieve some desired combination of overall system reliability and packaging density. Industrial environments will require filters that must be changed regularly to maintain cooling efficiency, and neglecting to change a filter or the failure of the fan could cause the system to shut down or malfunction.

The steps involved in selecting a heat sink / fan combination for forced convection are essentially the same as those followed for free convection, with the additional requirement that the heat sink and fan be matched to achieve desired heat sink-to-air thermal resistance. Attention must also be paid to proper channeling of fan airflow so that maximum utilization of its cooling capability is realized. Selection of a heat sink / fan combination involves the following three steps:

1. Determine maximum acceptable heat sink-to-air thermal resistance by following the first three steps of the heat sink selection procedure given in the Free Convection section.

2. Selection of a heat sink / fan combination requires that forced convection data for both the heat sink and fan be available. Forced convection characteristics for heat sinks define both heat sink-to-air thermal resistance and pressure drop through the heat sink as a function of airflow. Fan characteristics define airflow as a function of pressure drop. The intersection point of the airflow versus pressure curves for the fan and heat sink will define the operating airflow through the heat sink. (Figure 20–7) The heat sink-to-air thermal resistance for this airflow may be read directly off the airflow versus resistance curve for the heat sink.

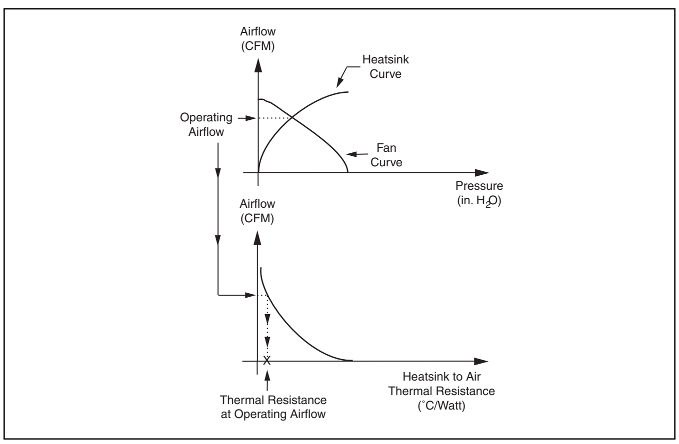

Figure 20–7 — *Airflow vs. resistance*

Finding and interpreting the operating point requires consideration of the following:

Units of pressure drop are generally given in inches of water. Units of fan airflow are in cubic feet per minute (CFM). Occasionally metric units are used, but conversion is straightforward.

Heat sink airflows may be given either in CFM or LFM (linear feet per minute). The conversion between LFM and CFM is dependent on the cross-sectional area through which air is flowing: CFM = LFM X Area

The cross-sectional area between the fins is the area through which the total airflow must pass. (Figure 20–8) Correct interpretation of heat sink data requires that only the airflow through this area be considered. Simply pointing a fan at a heat sink will clearly not result in all of the flow going through the cooling cross-section of the sink; some channeling of air is usually required to get the full benefit of fan output.

The fan curves give output in CFM versus pressure drop. Fan pressure drop is the total of all drops encountered by the fan airflow. The heat sink, any ducting that is used, and air entry and exit channels all contribute to pressure drop. Pressure drop represents the work done by the fan in moving air through a region, so care should be taken to minimize unproductive pressure losses. Ensure that air entry and exit locations and internal air channels are not unduly constricted, and avoid sharp turns in airflow paths.

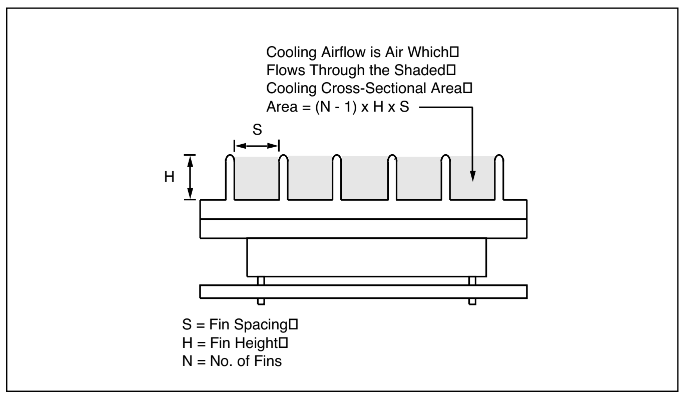

Figure 20–8 — *Heat sink cross section*

The thermal resistance that was determined by overlapping the fan and heat sink curves will represent an optimistic estimate since it assumes that all the fan output flows through the heat sink cooling cross section, and that all the pressure drop occurs along the heat sink. If the estimated thermal resistance is close to the minimum value determined in Step 1, then it is likely that a larger fan or different heat sink is required. This will not be a problem in most cases; relatively modest heat sinks and fans usually provide ample cooling.

Careful channeling and ducting of airflow as a means of both maximizing flow through the cooling cross-section of the heat sink and minimizing extraneous flow of air around the sink is well worth the small extra design effort required. Every degree of improvement in baseplate temperature results in significant improvement in MTBF. If you are paying for a fan, you may as well leverage it for all that it is worth.

3. Steps 5 through 7 in the Free Convection section will complete the heat sink selection process. Select the fan / heat sink combination with the lowest thermal resistance consistent with cost and space constraints, calculate the estimated baseplate temperature and test to verify.

NOTE: The values of θsa incorporating add-on or integral heat sinks include the baseplate-to-heat sink thermal resistance θba. When using heat sinks from other sources, the thermal impedance baseplate-to-air will be the sum of the thermal impedance heat sink-to-air specified by the heat sink manufacturer and the baseplate-to-heat sink impedance from the following Thermal Impedance Charts that follow.

Thermal Impedance Table (°C/W)

TABLE USAGE: The forced convection thermal impedance data shown in the tables below assumes airflow through the heat sink fins. Actual airflow through the fins should be verified. For purposes of heat sink calculation, assume efficiencies of 81% for 5 V outputs and 85% for 12 V and above.

VI-200 MI-200 θbs = 0.2	Baseplate	Part #30089 0.9"L Fins[a] (22,86 mm)	Part #30775 0.7"L Fins (17,78 mm)	Part #30090 0.9"T Fins[b] (22,86 mm)	Part #30780 1.45"L Fins (36,83 mm)	Part #30193 0.7"T Fins (17,78 mm)	Part #30194 0.4"T Fins (10,16 mm)	SlimMod	FinMod –F1 / –F3	FinMod –F2 / –F4
	θsa	θsa	θsa	θsa	θsa	θsa	θsa	θsa	θsa	θsa
Free Air	5.10	3.40	4.08	2.70	2.60	3.15	3.80	5.40	5.00	3.70
200 LFM	2.80	1.50	1.80	1.10	1.00	1.28	1.55	3.20	2.40	1.80
400 LFM	1.80	1.00	1.20	0.80	0.60	0.93	1.13	2.20	1.50	1.20
600 LFM	1.40	0.80	0.96	0.60	0.50	0.70	0.84	1.60	1.10	0.90
800 LFM	1.20	0.60	0.72	0.50	0.40	0.58	0.70	1.30	0.90	0.70
1,000 LFM	1.00	0.50	0.60	0.40	0.30	0.47	0.56	1.20	0.80	0.60

***Table 20–2a** — Thermal impedance for VI-200/MI-200*

VI-J00 MI-J00 θbs = 0.4	Baseplate	Part #30191 0.9" L Fins (22,86 mm)	Part #30771 0.9" T Fins (22,86 mm)	Part #30140 0.4" T Fins (10,16 mm)	SlimMod	FinMod –F1 / –F3	FinMod –F2 / –F4
	θsa	θsa	θsa	θsa	θsa	θsa	θsa
Free Air (H)	8.10	4.20	4.00	5.63	8.50	8.00	7.00
Free Air (V)	7.60	4.00	3.90	5.49	8.40	7.30	6.70
200 LFM	5.10	1.60	1.60	2.25	5.50	5.00	2.70
400 LFM	2.70	1.30	1.30	1.83	3.60	2.50	1.50
600 LFM	2.30	0.90	0.90	1.27	2.90	2.10	1.20
800 LFM	1.70	0.70	0.70	0.99	2.30	1.30	0.80
1,000 LFM	1.40	0.60	0.60	0.84	2.00	1.10	0.70

***Table 20–2b** — Thermal impedance for VI-200/MI-J00*

Configurables (also applies to MI-ComPAC and MI-MegaMod)	FlatPAC[c]			ComPAC[c]			MegaMod[c]		
	1-Up	2-Up	3-Up	1-Up	2-Up	3-Up	1-Up	2-Up	3-Up
	θbm	θbm	θbm	θbm	θbm	θbm	θbm	θbm	θbm
	0.1	0.05	0.03	0.1	0.05	0.03	0.1	0.05	0.03
	θsa	θsa	θsa	θsa	θsa	θsa	θsa	θsa	θsa
Free Air	2.1	1.3	1.0	3.6	1.7	1.4	4.4	2.1	1.7
50 LFM	1.5	1.1	0.9	2.7	1.4	1.3	3.3	1.7	1.6
100 LFM	1.2	0.9	0.7	2.3	1.3	1.1	2.8	1.6	1.3
250 LFM	0.7	0.5	0.4	1.6	1.0	0.8	2.0	1.2	1.0
500 LFM	0.4	0.3	0.3	1.2	0.7	0.6	1.5	0.9	0.7
750 LFM	0.3	0.2	0.2	0.9	0.5	0.5	1.1	0.6	0.6
1,000 LFM	0.2	0.2	0.2	0.8	0.4	0.4	1.0	0.5	0.5

***Table 20–2c** — Thermal impedance for FlatPAC, ComPAC/MI-ComPAC and MegaMod/MI-MegaMod Families*

[a] Longitudinal fins
[b] Transverse fins
[c] Assumes uniform loading of two and three output units.

DEFINITIONS

T_{max} = maximum baseplate temperature
(Available from converters data sheet, which can be found on vicorpower.com.)

T_a = ambient temperature

η = efficiency = $\frac{P_{out}}{P_{in}}$ (Assume efficiencies of 81% for 5 V outputs and 85% for 12 V out and above.)

θ_{bs} = baseplate-to-heat sink thermal resistance
(From thermal impedance tables)

θ_{sa} = heatsink-to-air sink thermal resistance
(From thermal impedance tables)

THERMAL EQUATIONS

$$P_{diss} = \text{dissipated power} = P_{out}\left(\frac{1}{\eta} - 1\right)$$

$$\text{Airflow (LFM)} = \left(\frac{\text{CFM}}{\text{Area}}\right)$$

$$\text{Maximum output power} = \frac{T_{max} - T_a}{\theta_{sa}\left(\frac{1}{\eta} - 1\right)}$$

$$\text{Maximum thermal impedance} = \frac{T_{max} - T_a}{P_{out}\left(\frac{1}{\eta} - 1\right)}$$

$$\text{Maximum ambient temperature} = T_{max} - \theta_{sa} \text{ x } P_{out}\left(\frac{1}{\eta} - 1\right)$$

$$\text{Temperature rise} = \theta_{sa} \text{ x } P_{out}\left(\frac{1}{\eta} - 1\right)$$

$$\text{Thermal drop} = \theta_{bm} \text{ x } P_{out}\left(\frac{1}{\eta} - 1\right)$$

TYPICAL EXAMPLES

Example 1. Determine the maximum output power for a 100 W, VI-200 converter, no heat sink, delivering 5 V in 400 LFM at a maximum ambient temperature of 45°C.

$$\text{Maximum output power} = \frac{T_{max} - T_a}{\theta_{sa}\left(\frac{1}{\eta} - 1\right)}$$

T_{max} = 85°C

T_a = 45°C

θ_{sa} = 1.8°C/W

η = 81% = (0.81)

$$\text{Maximum output power} = \frac{85 - 45}{1.8\left(\frac{1}{0.81} - 1\right)}$$

= 95 W max.

Example 2. Determine the maximum thermal impedance of a 50 W, VI-J00 converter, no heat sink, delivering 24 V at 45 W in free air convection at 55°C ambient.

$$\text{Maximum thermal impedance} = \frac{T_{max} - T_a}{P_{out}\left(\frac{1}{\eta} - 1\right)}$$

T_{max} = 100°C

T_a = 55°C

P_{out} = 45 W

η = 85% = (0.85)

$$\text{Maximum thermal impedance} = \frac{100 - 55}{45\left(\frac{1}{0.85} - 1\right)}$$

= 5.7˚C/W

Example 3. Determine the maximum ambient temperature of a 3-up FlatPAC delivering 12 V at 600 W in 500 LFM with no additional conduction cooling to the chassis.

$$\text{Maximum ambient temp.} = T_{max} - \theta_{sa} \times P_{out}\left(\frac{1}{\eta} - 1\right)$$

$T_{max} = 85°C$

$\theta_{sa} = 0.3°C/W$

$P_{out} = 600\ W$

$\eta = 85\% = (0.85)$

$$\text{Maximum ambient temp.} = 85 - 0.3 \times 600\left(\frac{1}{0.85} - 1\right)$$

$= 53°C$

Example 4. Determine the temperature rise of a 150 W, VI-200 converter delivering 5 V at 132 W with a Part #30090 heat sink in 200 LFM.

$$\text{Temperature rise} = \theta_{sa} \times P_{out}\left(\frac{1}{\eta} - 1\right)$$

$\theta_{sa} = 1.1°C/W$

$P_{out} = 132\ W$

$\eta = 81\% = (0.81)$

$$\text{Temperature rise} = 1.1 \times 132\left(\frac{1}{0.81} - 1\right)$$

$= 34°C$ Over ambient temperature

Example 5. Determine the baseplate to coldplate thermal drop for an MI-200 converter delivering 5 V at 50 W with a thermal pad.

$$\text{Thermal drop} = \theta_{bs} \times P_{out}\left(\frac{1}{\eta} - 1\right)$$

$\theta_{bs} = 0.2°C/W$

$P_{out} = 50\ W$

$\eta = 81\% = 0.81$

$$\text{Temperature rise} = 0.2 \times 50\left(\frac{1}{0.81} - 1\right)$$

$= 2.34°C$