THERMAL DESIGN

Prepared by Dr K A Walshe

1 Preliminary Information:

References for this chapter are

Mohan ; chapter 29 Component Temperature Control and Heat Sinks

Copper Development Association Handbook "Copper for Busbars" publication no 22

German Standard VDE 0660 part 500

various manufacturers literature on heatpipes , heat sinks and the like (supplied as pdf files on course web site)

Note :
Anybody intending to have any business with heavy electrical currents ought to obtain a copy of the Copper for Busbars handbook - contact the CDA at http://www.copper.org/pub_list/energy-efficiency.html

2 Introduction

Electrical plant can be killed in three main ways :

- Over / under voltage
- Over current, and
- Over temperature

When we look at the way most plant is designed overvoltage protection is almost an in-built feature due to the fact that system voltages are associated with steady state and transient maximum values specified in a raft of Standards. The users responsibility is to ensure that his installation ensure that voltages fall within these limits.

Current tends to be more of a dependant variable thus in-service currents are largely determined by the user. Many items of plant may have their own internal over current limiters but it is generally left to the users to ensure that the associated circuit breakers have adequate over current protection fitted.

Temperature is however the least regulated and possibly most poorly protected killer disease. In general equipment manufactures can fit a range of protective devices such as

- Winding temperature sensors (thermocouples, temperature dependant switches, temperature dependant resistors etc),
- Winding temperature estimators based on history of current drawn,
- Oil temperature monitors,
- Air temperature monitors,
- Heat sink water or air outlet temperature

In most cases the protection given by these methods is either or both of

- indirect - that is measure one thing and estimate another, the one you want, from it or

- highly approximate - measure an average or single spot and assume a fixed relationship between that and the highest temperature

Some examples of this problem are motor winding temperature, transformer winding temperature and semi-conductor temperature.

In the case of motors a thermocouple is sometimes placed in between the two windings in 4 or 6 volts (note that this has a distorting effect on the winding package and thus can lead to winding failure if excessive mechanical force is applied to holding the windings in place). It is then assumed that these thermocouples or RTDs indicate the average winding temperature - this takes no account of possible disruption to cooling air flow and thus it is possible, if unlikely, for the protection to say that all is well when some insulation is burning.

In the case of a transformer and average winding temperature is measured and then increased by a fixed amount to give the "hottest spot temperature".

Pretty much the same assumption is made with heatsinks - junction temperature is a fixed increment over heat sink temperature. When water cooled heat sinks are used the temptation to use an easy water temperature measurement leads to an even greater leap of faith water temperature + fixed rise on heat sink + fixed rise in device = junction temperature.

3 Units Conversion

Despite the Metric system being the preferred system of measurements, the USA still retains a variant of the old Imperial system of Feet, Stones, Slugs and the like. It is therefor necessary to be able to convert between MKS and Imperial. The USA uses CFM (Cubic Feet per Minute) for airflow whilst m3/min or m3/h is used, in the MKS system.

air flow unit conversion table

cubic meter / minute 35.28 1 60 16.67 1000

cubic meter / hour 0.588 0.017 1 0.28 16.67

litre / second **1** 2.12 0.06 3.6 1 60

litre / min **1 =** 0.035 0.001 0.06 0.017 1

	CFM	M3/min	M3/hour	L/s	L/min
1 CFM =	1	0.028	1.7	0.47	28.3
1 m3/min =	35.28	1	60	16.67	1000.
1 m3/h =	0.588	0.017	1	0.28	16.67
l/s =	2.12	0.06	3.6	1	60
l/min	0.035	0.001	0.06	0.017	1

Fahrenheit / Celsius / Kelvin conversion

Fahrenheit to Celsius	Celsius to Fahrenheit	Celsius to Kelvin
°C = (5/9) * (°F-32)	°F = (°C * (9/5)) + 32	K = °C + 273.15

4 Heat Transfer

4.1 Thermal resistance

4.1.1 Steady State

Two concepts

- Adiabatic heating for short time overload conditions - no heat loss hence calculated temperature rise is pessimistic & hence conservative.
- Linear heat flow - uniform heat density allows mass and specific heat to be used to calculate temperature ***rise*** . Starting with the formula for Power (watts heat) transmitted

P = c * A *dT / l Watts where :
c = thermal conductivity W/m/degC
l = length m
dT = temperature difference across sectional length l degC

The thermal resistance of a body is then
R_{th} = dT / P degC / watt

Mohan quotes c= 220 W/m/degC for 90% pure aluminium such as typically used in heat sinks. The Copper Development Association quotes 397 for C101 grade electrical copper and 230 for 1350 grade Al. Other useful formulae and constants are :

Convection heat loss from vertical surface : Wv= (7.66*dT^1.25)/ L^0.25
" " " " horizontal " : Wh= (5.92*dT^1.25)/ L^0.25
" " " " round " : Wc= (7.66*dT^1.25)/ d^0.25
all in Watts / m^2

were L is height or width of surface (m), dT is temperature rise (Cdeg) and d is diameter (m)

The above hold for airflows of <0.5 m/sec. Above that value use
Wa= 120 * sqrt(v) * (A/Length) * dT Watts /m

Radiation loss is calculated by

Wr= 5.7 * 10^-8 *E * (T1^4 - T2^4)

were

E = relative emmissivity (see below)

T1 = absolute temperature of body 1 in degKelvin, and

T2 = " " " " 2 " "

Relative emmissivity	
Bright metal	0.1
Partially oxidised	0.3
Heavily oxidised	0.7
Dull non-metallic paint	0.9

When heat flows through several different materials the thermal resistances are added just as for resistances in series. Provided that the ambient temperature is known (or a maximum value specified) we can say

$T_{actual} = P * (R_{th1} + R_{th2} + ...) + T_{ambient}$

The most frequent application of this formula is in the calculation of junction temperature rise. R_{th1} etc represent the thermal resistances of each discrete block of material from the junction to the true ambient. In practice a single figure is usually quoted for the device and a single figure for the heat sink. For example

$R_{thDevice}$ = 0.14 degC/W

P loss = 400 W

$R_{thHeatSink}$ = 0.058 degC/W (with forced air cooling at 180 m^3 per hour)

$T_{ambient}$ = 45 degC

T_{actual} = 400 *(0.18 +0.058) + 45 = 95.2 + 45 = 140.2 deg C (ie only OK if diodes are being used)

These calculations are based on an assumed P(t) waveform. Consider the basic model of an active device

$V(I) = V_j + I * R_d$

where :

V_j = intrinsic junction volt drop = about 0.7 volts for silicon devices

R_d = bulk junction resistance in the On state - usually milli ohms for power devices.

The power dissipation in the junction ignoring switching losses approximates to P_d (t) = I(t) * (V_j + I * R_d)

In bridge rectifiers the current is a near rectangular pulse of 60 degrees width symmetrical about the peak of the voltage. Hence the power pulse is also rectangular and of 120 degrees width per cycle. Other types of circuits will impose a range of current waveshapes on the active device.

Since the rate of rise of temperature depends on the thermal mass and the fall of temperature on the dissipation constants, it follows that the junction temperature will vary as a low pass filtered version of the junction power dissipation. If the thermal inertia of the junction is small, the associated thermal time constant will also be small (short) and hence the junction temperature will exhibit time variations. In power semiconductors this typically gives a lag in the range 0.5 ~ 2.5 milliseconds.

Since the aim of the steady state junction calculation is to arrive at a heat sink design that limits the crest junction temperature (device failure is based on absolute temperatures reached) devices are usually quoted with R_j values for 120 degree rectangular, 180 degree half sinewave and continuous conduction cases. As an example the Semikron SKKT 250 (a thyristor / diode module nominally rated for 445A rms in a three phase Greatz bridge configuration) has values of

Rthjc = 0.14 / 0.15 / 0.165 degC/W for cont / sin180 / rect 120

Rd = 0.45 milliOhms
Vj = 1.0625 V (NB this has 2 devices in series)

4.1.2 Transient Loads

If a load imposes a short duration over current on the devices, ie a motor starting application, then we must ensure that the peak junction temperature does not exceed that specified (usually 125 for a thyristor etc and 150 for a diode). When the short duration over current flows the junction will heat up far quicker than the body of the device which in turn is quicker than the case of the device etc. If a large air naturel heat sink is being used, its thermal heating time constant might be measured in minutes rather than milliseconds for the junction.

In view of the above devices also have a ***Transient thermal resistance*** value quoted. Just as the steady state values quoted allow for various waveshapes so does the transient thermal resistance (impedance) account for these various lags; it is usually quoted as 3 or 4 series first order low pass time constants :

$$T_{j/D2}(t) = T_C + T_{coup/T2} + P_{D2}(t) \cdot \sum_{v=1}^{n} R_{thv/D2}\left[1 - \exp\left(-t/\tau_{thv/D2}\right)\right]$$

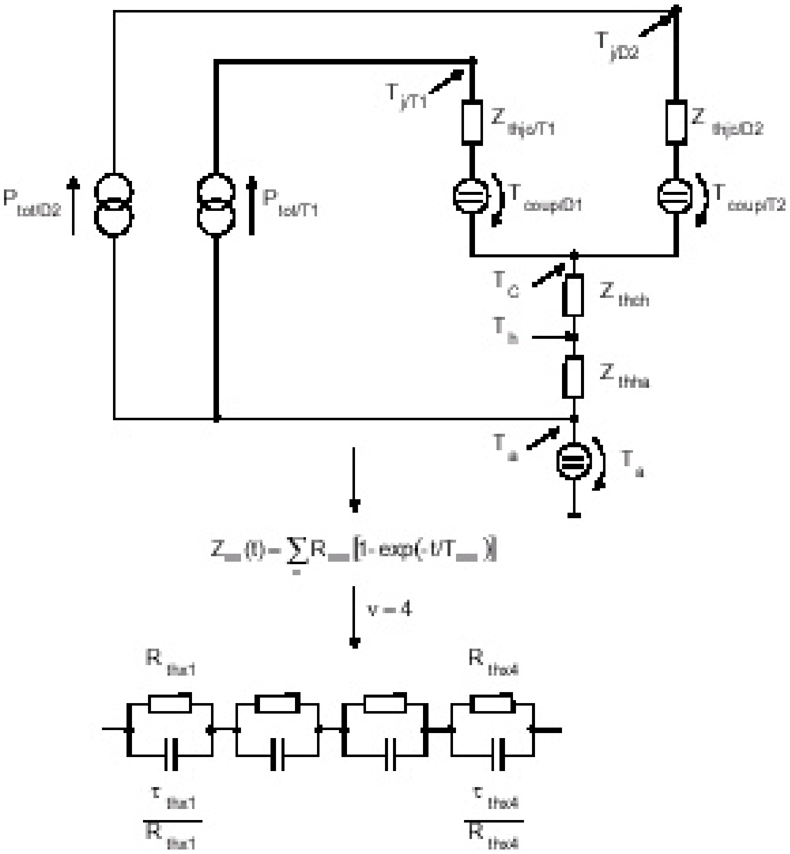


Figure 3.8 Simplified thermal equivalent block diagram of IGBT and free-wheeling diode in a power module

Explanation of designations:

P_{tot}	Total power dissipation within transistor and free-wheeling diode
T_j	Junction temperatures
Z_{thjc}	Thermal impedance from junction to module case
T_c	Case temperature
Z_{thch}	Thermal impedance from module case to heatsink
T_h	Heatsink temperature
Z_{thha}	Thermal impedance from heatsink to ambient temperature (see chapter 3.3)
T_a	Ambient temperature

from Semikron "Applications Manual"

When this is used a piece wise solution must be adopted unless a digital time domain solution can be formed.

NB Mohan points out that this representation is an approximation of the thermal diffusion process; however the values are usually conservative and hence errors introduced by such modelling are not important.

4.2 Heat Sinks & busbars

Ref : Mohan 29-4 & CDA handbook "Copper for Busbars)

4.2.1 General Concepts

The heat transfer processes are radiation and convection. The first general rule is matt black vertical surfaces are best, bright shiny horizontal surfaces are worst for heat transfer. The steady state heat transfer by both radiation and conduction can be reduced to a form

R = constant / Area degC / W

And the only problem is then to define the constant (see Mohan equations 29-15 to 29-18).

In the design of heat sinks a number of important factors arise

- Ensuring a very small temperature gradient in the bulk material in contact with the device,
- Preventing thermal clogging of the space between the fins, and
- Ensuring a very small temperature gradient from the root to the tip of a heat sink fin.

4.2.2 Spreading Resistance

Before we proceed with the analysis, let us attend to what the temperature distributions shown in Fig. 1 are telling us. The first obvious one, as noted earlier, is that the maximum temperature at the center increases as the heat source becomes smaller. Another important observation is that, as the temperature rises in the center, the temperatures along the edges of the heat sink decrease simultaneously. It can be shown that this happens in such a way that the area-averaged surface temperature of the heat sink base-plate has remained the same. In other words, the average heat sink thermal performance is independent of the size of a heat source. In fact, as will be seen later, it is also independent of the location of the heat source.

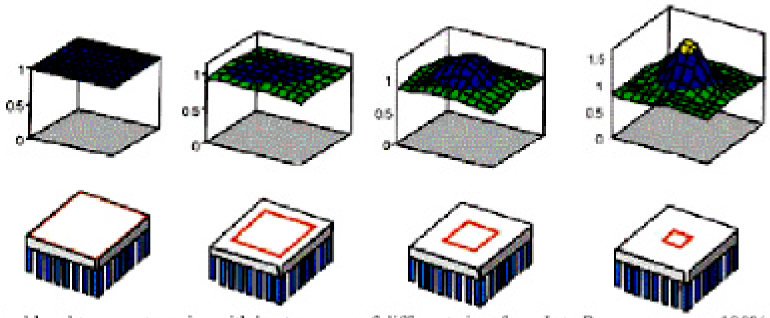
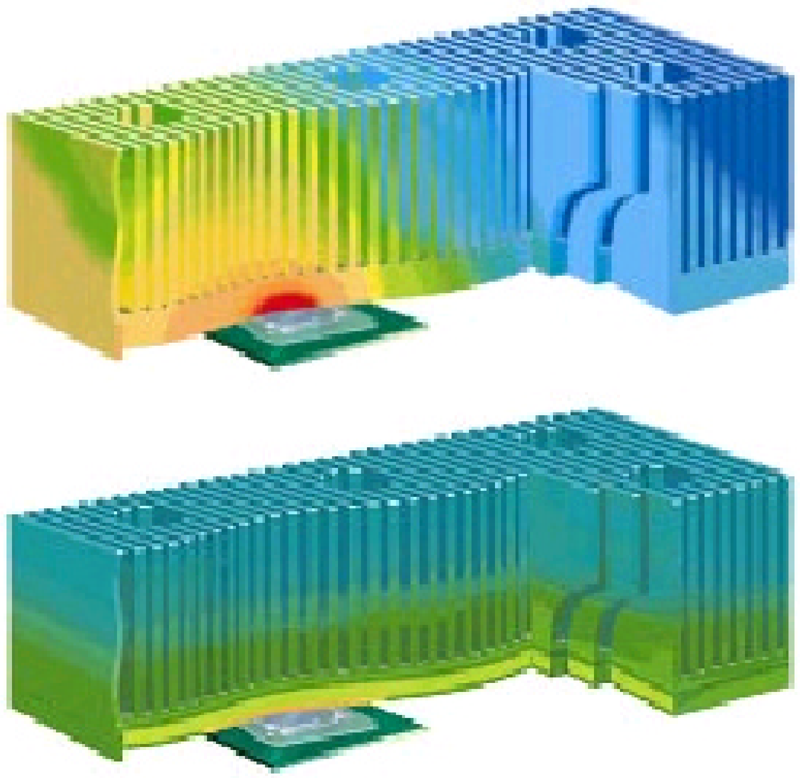


Figure 1 - Normalized local temperature rise with heat sources of different size; from L to R, source area = 100%, 56%, 25%, 6%, of heat-sink area

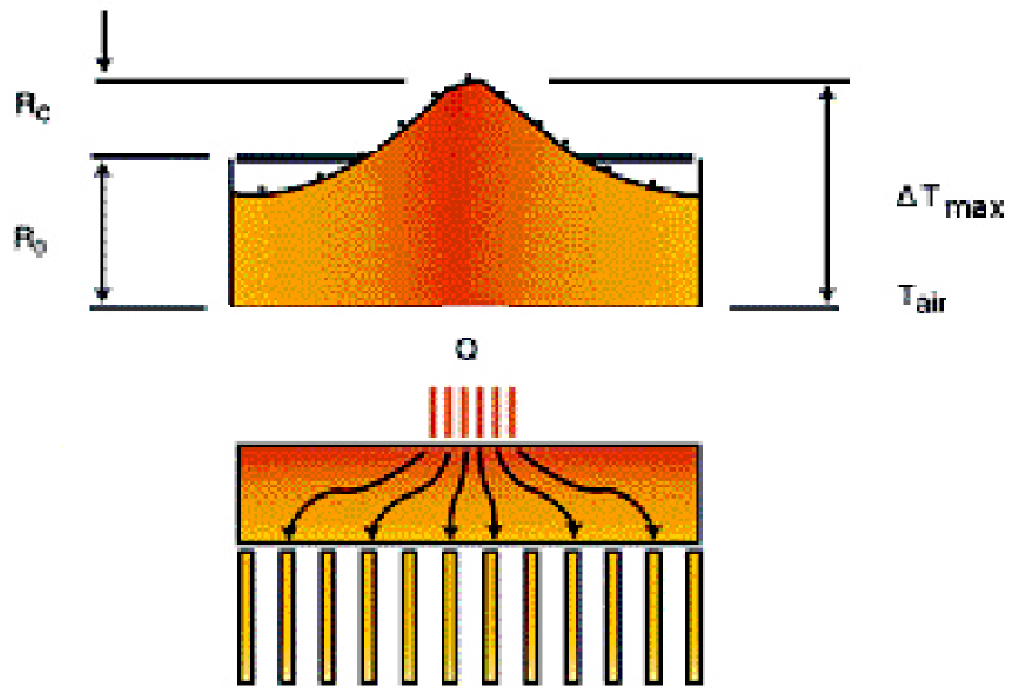


Normalized local temperatures for different heat source sizes and locations

The spreading resistance can be determined from the following set of parameters:

- footprint or contact area of the heat source, A_s
- footprint area of the heat sink base-plate, A_p
- thickness of the heat sink base-plate, t
- thermal conductivity of the heat sink base-plate, k
- average heat sink thermal resistance, R_0

We will assume, for the time being, that the heat source is centrally mounted on the base-plate, and the heat sink is cooled uniformly over the exposed finned surface. These two assumptions will be examined in further detail. Figure 2 shows a two-



Heat Flow in Typical Heat Sink

Figure 2 - Two dimensional schematic view of local resistance or temperature variation of a heat sink shown with heat flow Lines Note that the correlation addresses neither the shape of the heat source nor that of the heat sink base-plate. It was found in the earlier study that this correlation typically results in an accuracy of approximately 5% over a wide range of applications with many combinations of different source/sink shapes, provided that the aspect ratio of the shapes involved does not exceed 2.5. See references 1 and 2 for further discussions.

dimensional side view of the heat sink with heat-flow lines schematically drawn in the base-plate whose thickness is greatly exaggerated. At the top, the corresponding surface temperature variation across the center line of the base-plate is shown by the solid line. The dotted line represents the average temperature of the surface which is, again, independent of the heat source size and can be easily determined by multiplying R_0 with the total amount of heat dissipation, denoted as Q.

As indicated in Fig. 2, the maximum constriction resistance R_c, which accounts for the local temperature rise over the average surface temperature, is the only additional quantity that is needed for determining the maximum heat sink temperature. It can be accurately determined from the following correlation.

4.2.3 References

1. S. Lee, S. Song, V. Au, and K.P. Moran, Constriction/Spreading Resistance Model for Electronic Packaging, Proceedings of the 4th ASME/JSME Thermal Engineering Joint Conference, Vol. 4, 1995, pp. 199-206.

2. S. Song, S. Lee, and V. Au, Closed Form Equation for Thermal Constriction/Spreading Resistances with Variable Resistance Boundary Condition, Proceedings of the 1994 IEPS Conference, 1994, pp. 111-121.

3. D. P. Kennedy, Spreading Resistance in Cylindrical Semiconductor Devices, Journal of Applied Physics, Vol. 31, 1960, pp. 1490-1497.

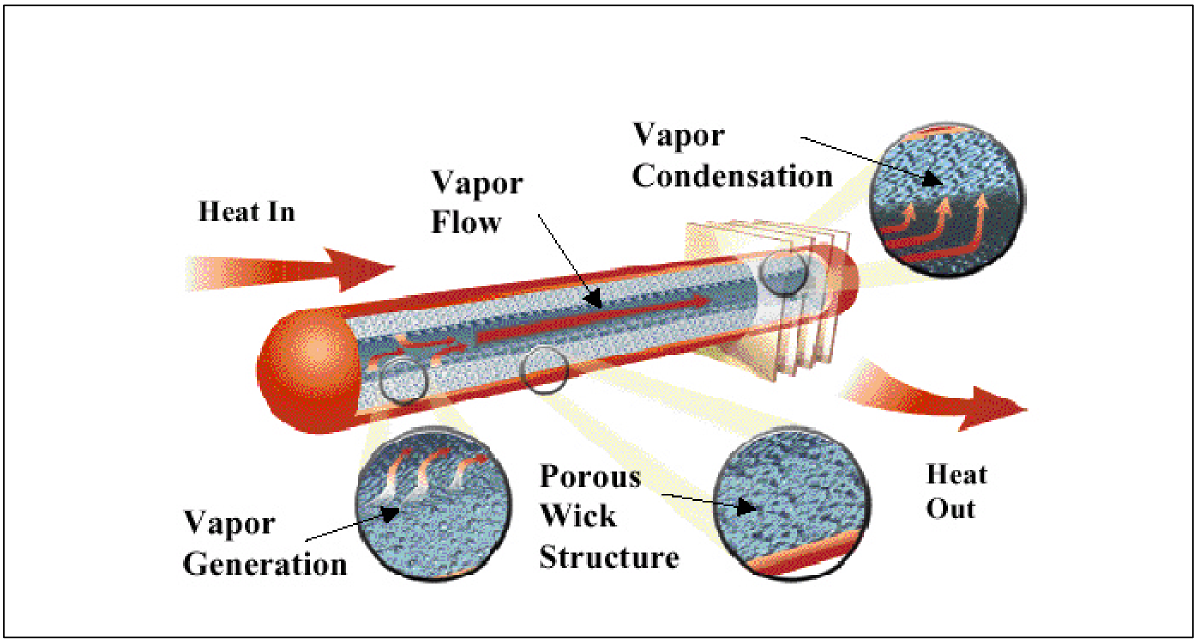
4.3 Heat Pipes

4.3.1 Introduction

Today, many engineers typically use discrete heat sinks that attach to the largest heat producing components within an enclosure. These heat sinks have typically been constructed from extruded aluminum. However, as the performance of power semiconductors increase, the usual method of their thermal control becomes the gating factor. Increasing the performance of extruded heat sinks to keep up with the exponential performance of power semiconductors is inadequate due to the inherent limitations of the extruding process. Consequently, meeting future thermal demands with extruded heat sinks is highly unlikely. The alternative solution is to take advantage of the additive thermal performance of heat pipes.

A heat pipe, in the simplest sense, is a heat mover or spreader; it acquires heat from a source, such as the power semiconductors, and moves or spreads it to a region where it can be more readily dissipated. The heat pipe moves this heat with very a minimal drop in temperature.

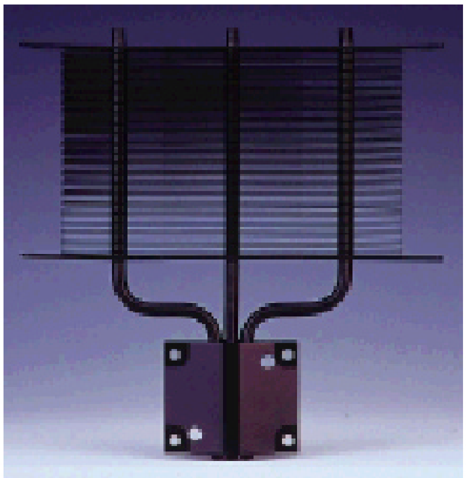
A typical heat pipe is a sealed and evacuated tube with a porous wick structure and a very small amount of working fluid on the inside. Figure 1 is a heat pipe illustration that depicts its internal design and operation. A porous wick structure, such as sintered powder metal, lines the internal diameter of the tube. The center



core of the tube is left open to permit vapor flow. The heat pipe has three sections: evaporator, adiabatic and condenser. As heat enters the evaporator section, it is absorbed by the vaporization of the working fluid. The generated vapor travels down the center of the tube through the adiabatic section to the condenser section where the vapor condenses, giving up its latent heat of vaporization. The condensed fluid is returned to the evaporator section by gravity or by capillary pumping in the porous wick structure. Heat pipe operation is completely passive and continuous. Since there are no moving parts to fail, a heat pipe is very reliable.

4.3.2 Heat Pipe Heat Sinks:

Heat pipes are widely used to cool electronics in computers, telecommunication and industrial equipment. The most versatile feature of using heat pipes is the wide variety of geometry's that can be constructed to take advantage of the available space around the electronics to be cooled. For example, Figure 2 is a photograph of several heat pipes geometry's that are in high volume production. The tall "Tower" assemblies are for those applications where there is vertical space available above the electronics. In many applications, the available heat sink volume above electronics is limited by the board-to-board spacing. In this situation, heat pipes are used in a low profile design that transports the heat to a large fin stack. In general, the smaller component cooling approaches transport 10W to 100W each.



The heat pipe heat sink converts a small heat radiating area into a large one and in a suitable location for cooling

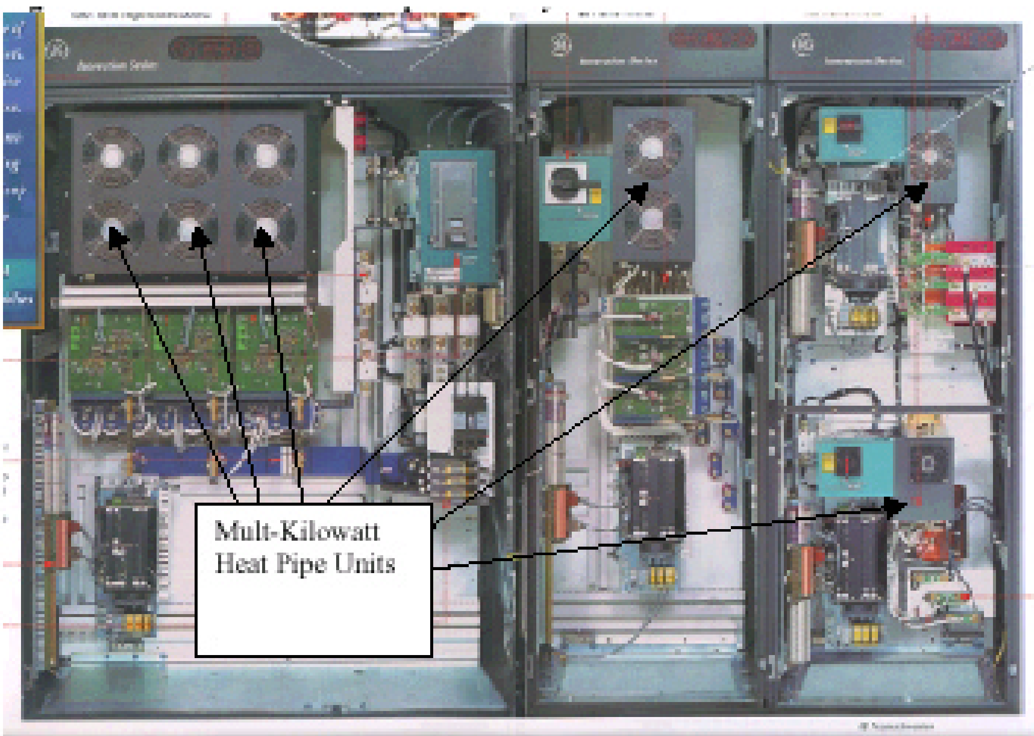
In those situations where there are several hundred watts to multi-kilowatts to be rejected, the electronics are mounted on large heat pipe units inside cabinets. Figure 3 shows a large heat pipe unit that has several IGBTs mounted on it. The IGBTs are attached to a mounting plate and heat pipes embedded in the plate, transports the heat to an air-cooled fin section. There are several different sized units like this being used in the field. Heat rejection from units like these is from 500W to 8.3 kW with thermal resistance values from 0.004 ^{o}C/W to 0.062 ^{o}C/W. Figure 4 is an example of some multi-kilowatt heat pipe units installed in a motor drive cabinet.

4.3.3 Sealed Cabinet Heat Exchanger Cooling

In many cabinet cooling situations, there is always additional heat to be rejected form those electronics that are not cooled with heat sinks directly. In those situations, it is best to use an air-to-air heat exchanger designed for cooling indoor/outdoor electronic cabinets.

For example, Figure 5 shows a double-sided impingement heat exchanger designed to achieve significantly improved heat transfer while reducing heat exchanger size. Figure 6 is an illustration on how it works. Clean air on the inside impinges on one side of the heat exchanger; dirty air on the outside impinges only on the outside. Dirt or moisture never gets inside the cabinet. Heat load dissipation from 500W to 2000W has been demonstrated.

In summary, this article shows a progression of cooling approaches that thermal engineer's can use to address the demands of the growing heat dissipation levels of power semiconductors inside of cabinets. Small heat pipe assemblies (10W to 100W) allow increased heat sink performance within the volume available with potentially little impact on the existing system design. In those situations where the heat to be rejected is several kilowatts, large multi-kilowatt heat pipe assemblies



Multi-kW coolers mounted inside panel

can be used. Ultimately, these multi-kilowatt cooling units might be the best solution for keeping pace with the increasing heat loads of power semiconductors. After cooling those electronics which generate the most heat, there is always heat from other electronics in the cabinet. A sealed air-to-air heat exchanger is best for dealing with the residual heat inside the cabinet.

4.4 Thermoelectric coolers

The following is an article in the public domain by :

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4.4.1 Introduction

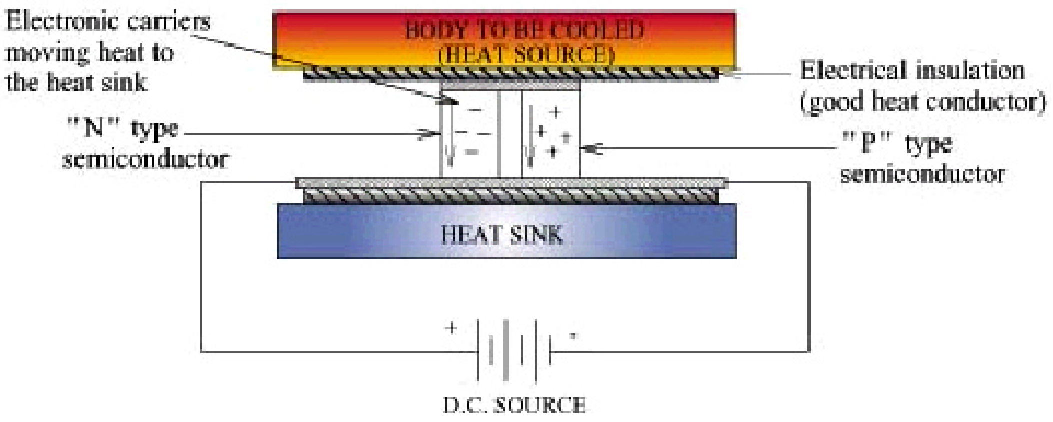


Figure 1: Cross Section of a Typical TE Couple

Thermoelectric coolers are solid state heat pumps used in applications where temperature stabilization, temperature cycling, or cooling below ambient are required. There are many products using thermoelectric coolers, including CCD cameras (charge coupled device), laser diodes, microprocessors, blood analyzers and portable picnic coolers. This article discusses the theory behind the thermoelectric cooler, along with the thermal and electrical parameters involved.

4.4.2 How the Thermoelectric Works . . .

Thermoelectrics are based on the Peltier Effect, discovered in 1834, by which DC current applied across two dissimilar materials causes a temperature differential. The Peltier Effect is one of the three thermoelectric effects, the other two are known as the Seebeck Effect and Thomson Effect. Whereas the last two effects act on a single conductor, the Peltier Effect is a typical junction phenomenon. The three effects are connected to each other by a simple relationship.

The typical thermoelectric module is manufactured using two thin ceramic wafers with a series of P and N doped bismuth-telluride semiconductor material sandwiched between them. The ceramic material on both sides of the thermoelectric adds rigidity and the necessary electrical insulation. The N type material has an excess of electrons, while the P type material has a deficit of electrons. One P and one N make up a couple, as shown in Figure 1. The thermoelectric couples are electrically in series and thermally in parallel. A thermoelectric module can contain one to several hundred couples.

As the electrons move from the P type material to the N type material through an electrical connector, the electrons jump to a higher energy state absorbing thermal energy (cold side). Continuing through the lattice of material, the electrons flow from the N type material to the P type material through an electrical connector, dropping to a lower energy state and releasing energy as heat to the heat sink (hot side).

Thermoelectrics can be used to heat and to cool, depending on the direction of the current. In an application requiring both heating and cooling, the design should focus on the cooling mode. Using a thermoelectric in the heating mode is very efficient because all the internal heating (Joulian heat) and the load from the cold side is pumped to the hot side. This reduces the power needed to achieve the desired heating.

4.4.3 Thermal Parameters Needed

The appropriate thermoelectric for an application, depends on at least three parameters. These parameters are the hot surface temperature (T_h), the cold surface temperature (T_c), and the heat load to be absorbed at the cold surface (Q_c).

The hot side of the thermoelectric is the side where heat is released when DC power is applied. This side is attached to the heat sink. When using an air cooled heat sink (natural or forced convection), the hot side temperature can be found by using Equations 1 and 2.

$$T_h = T_{amb} + (O)(Q_h) \quad (1)$$

Where

T_h = The hot side temperature (°C).

T_{amb} = The ambient temperature (°C).

O = Thermal resistance of heat exchanger (°C/watt).

And

$$Q_h = Q_c + P_{in} \quad (2)$$

Where

Q_h = the heat released to the hot side of the thermoelectric (watts).

Q_c = the heat absorbed from the cold side (watts).

P_{in} = the electrical input power to the thermoelectric (watts).

The thermal resistance of the heat sink causes a temperature rise above ambient. If the thermal resistance of the heat sink is unknown, then estimates of acceptable temperature rise above ambient are:

Natural Convection 20°C to 40°C

Forced Convection 10°C to 15°C

Liquid Cooling 2°C to 5°C (rise above the liquid coolant temperature)

The heat sink is a key component in the assembly. A heat sink that is too small means that the desired cold side temperature may not be obtained.

The cold side of the thermoelectric is the side that gets cold when DC power is applied. This side may need to be colder than the desired temperature of the cooled object. This is especially true when the cold side is not in direct contact with the object, such as when cooling an enclosure.

The temperature difference across the thermoelectric (dT) relates to T_h and T_c according to Equation 3.

$dT = T_h - T_c$ (3)

The thermoelectric performance curves in Figures 2 and 3 show the relationship between dT and the other parameters.

Estimating Q_c, the heat load in watts absorbed from the cold side is difficult, because all thermal loads in the design must be considered. Among these thermal loads are:

- Active: I^2R heat load from the electronic devices Any load generated by a chemical reaction
- Passive: Radiation (heat loss between two close objects with different temperatures) Convection (heat loss through the air, where the air has a different temperature than the object)
  Insulation Losses
  Conduction Losses (heat loss through leads, screws, etc.)
  Transient Load (time required to change the temperature of an object)

4.4.4 Powering the Thermoelectric

All thermoelectrics are rated for I_{max}, V_{max}, Q_{max}, and T_{max}, at a specific value of T_h. Operating at or near the maximum power is relatively inefficient due to internal heating (Joulian heat) at high power. Therefore, thermoelectrics generally operate within 25% to 80% of the maximum current. The input power to the thermoelectric determines the hot side temperature and cooling capability at a given load.

As the thermoelectric operates, the current flowing through it has two effects: (1) the Peltier Effect (cooling) and (2) the Joulian Effect (heating). The Joulian Effect is proportional to the square of the current. Therefore, as the current increases, the Joule heating dominates the Peltier cooling and causes a loss in net cooling. This cut-off defines I_{max} for the thermoelectric.

For each device, Q_{max} is the maximum heat load that can be absorbed by the cold side of the thermoelectric. This maximum occurs at I_{max}, V_{max}, and with T = 0°C. The T_{max} value is the maximum temperature difference across the thermoelectric. This maximum occurs at I_{max}, V_{max} and with no load (Q_c = 0 watts). These values of Q $_{max}$ and T $_{max}$ are shown on the performance curve (Figure 3) as the end points of the I $_{max}$ line.

4.4.5 An Example

Suppose a designer has an application with an estimated heat load of 22 watts, a forced convection type heat sink with a thermal resistance of 0.15°C/watt, an ambient temperature of 25°C, and an object that needs to be cooled to 5°C. The cold side of the thermoelectric will be in direct contact with the object.

The designer has a Melcor CP1.4-127-06L thermoelectric in the lab and needs to know if it is suitable for this application. The specifications for the CP1.4-127-06L are as follows (these specifications are at T $_h$ = 25°C):

To determine if this thermoelectric is appropriate for this application, it must be shown that the parameters T and Q $_c$ are within the boundaries of the performance curves. The parameter T follows directly from T $_h$ and T $_c$. Since the cold side of the thermoelectric is in direct contact with the object being cooled, T $_c$ is estimated to

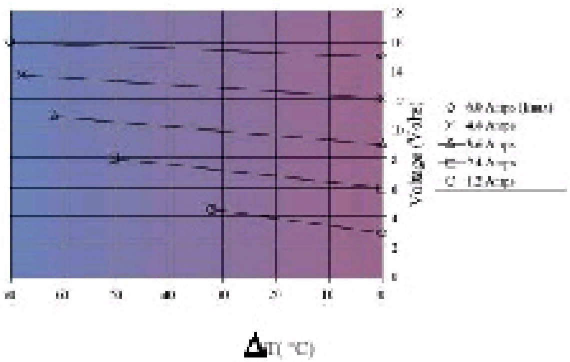


Figure 2: Melcor CP1.4-127-06L, ^T vs. Voltage

Performance Curve (ΔT vs. Qc)

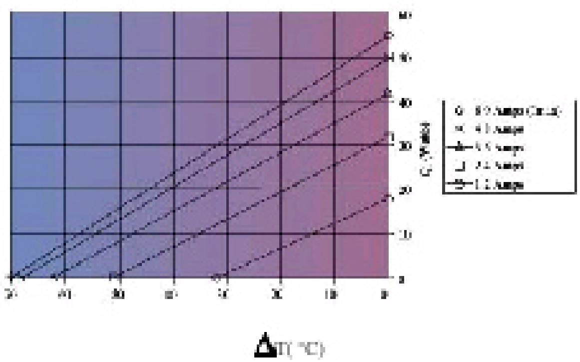


Figure 3: Melcor CP1.4-127-06L, ^T vs. Qc

be 5°C. Assuming a 10°C rise above ambient for the forced convection type heat sink, T_h is estimated to be 35°C. Without knowing the power into the thermoelectric, an exact value of T_h cannot be found. Equation 3 gives the temperature difference across the thermoelectric:

$T = T_h - T_c = 35°C - 5°C = 30°C$

Figures 2 and 3 show performance curves for the CP1.4-127-06L at a hot side temperature of 35°C. Referring to Figure 3, the intersection of Q_c and T show that this thermoelectric can pump 22 watts of heat at a T of 30°C with an input current of 3.6 amps.

4.4.6 Performance Curve (T vs. Voltage)

I_{max} = 6.0 amps
Q_{max} = 51.4 watts
V_{max} = 15.4 volts
T_{max} = 67°C

These values are based on the estimate T_h = 35°C. Once the power into the thermoelectric is determined, Equations 1 and 2 can be used to solve for T_h and to determine whether the original estimate of T_h was appropriate. The input power to the thermoelectric, Pin, is the product of the current and the voltage. Using the 3.6 amp line in Figure 2 for the current, the input voltage corresponding to T = 30°C is approximately 10 volts.

Using Equations 1 and 2, T_h can now be calculated.

The calculated T_h is close enough to the original estimate of T_h, to conclude that the CP1.4-127-06L thermoelectric will work in the given application. If an exact solution needs to be known, the process of solving for T_h mathematically can be repeated until the value of T_h does not change.

$T_h = T_{amb} + (O)(Q_h)$
where T_{amb} = 25°C O = 0.15°C/watt
$Q_h = Q_c + P_{in}$ = 22 watts + ((3.6 amps) * (10 volts)) = 22 watts + 36 watts = 58 watts

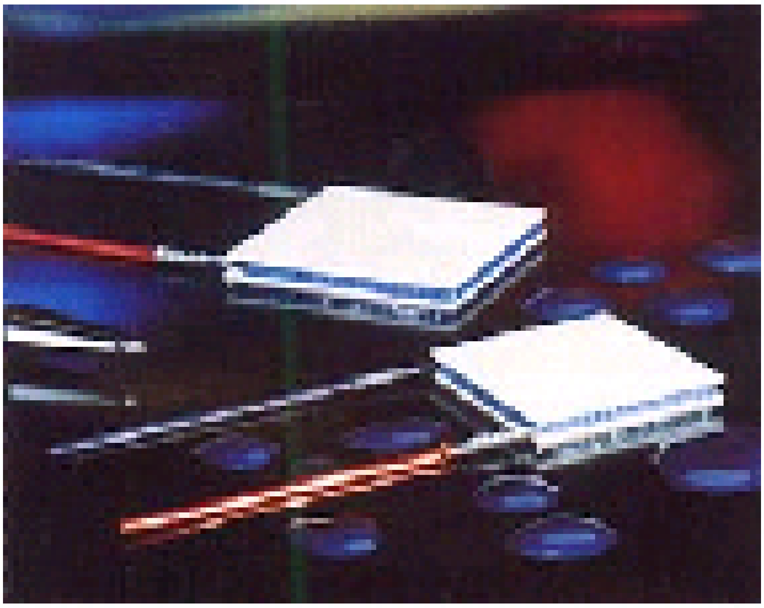


Figure 4: Typical thermoelectric from Melcor with a perimeter seal

therefore T_h = 25°C + (0.15°C / watt) (58 watts)

= 25°C + 8.7°C = 33.7°C

4.4.7 Other Parameters to Consider

The material used for the assembly components deserves careful thought. The heat sink and cold side mounting surface should be made out of materials that have a high thermal conductivity (i.e., copper or aluminum) to promote heat transfer. However, insulation and assembly hardware should be made of materials that have low thermal conductivity (i.e., polyurethane foam and stainless steel) to reduce heat loss.

Environmental concerns such as humidity and condensation on the cold side can be alleviated by using proper sealing methods. A perimeter seal (Figure 4) protects the couples from contact with water or gases, eliminating corrosion and thermal and electrical shorts that can damage the thermoelectric module.

The importance of other factors, such as the thermoelectrics footprint, its height, its cost, the available power supply and type of heat sink, vary according to the application.

4.4.8 Single Stage vs. Multistage

Given the hot side temperature, the cold side temperature and the heat load, a suitable thermoelectric can be chosen. If T across the thermoelectric is less than 55°C, then a single stage thermoelectric is sufficient. The theoretical maximum temperature difference for a single stage thermoelectric is between 65°C and 70°C.

If T is greater than 55°C, then a multistage thermoelectric should be considered. A multistage thermoelectric achieves a high T by stacking as many as six or seven single stage thermoelectrics on top of each other.

4.4.9 Summary

Although there is a variety of applications that use thermoelectric devices, all of them are based on the same principle.

When designing a thermoelectric application, it is important that all of the relevant electrical and thermal parameters be incorporated into the design process. Once these factors are considered, a suitable thermoelectric device can be selected based on the guidelines presented in this article.

4.4.10 References on Thermoelectrics

- Levine, M.A., Solid State Cooling with Thermoelectrics, Electronic Packaging & Production, Nov. 1989.
- Melcor Corporation, Thermoelectric Handbook, Sept., 1995.
- Rowe, D. M., CRC Handbook of Thermoelectrics, CRC Press, Inc., 1995.
- Smythe, Robert, Thermoelectric coolers take the heat out of today's hot chips, Electronic Products, Aug. 1995.

5 Cooling of panels

5.1 IP Ratings for panels

NEMA	IEC/ IP	Abbreviated Protection Description
1	IP23	Indoor protection from contact with contents.
2	IP30	Indoor with limited protection from dirt & water.
3	IP64	Outdoor with some protection from rain, sleet, and ice damage.
3R	IP32	Outdoor with some protection from rain, sleet, and ice damage.
4	IP66	Indoor and outdoor with some protection from windblown dust, rain, splashing water, hose- directed water, and ice damage
4x	IP66	Indoor and outdoor with some protection from corrosion, windblown dust, rain, splashing water, hose-directed water, and ice damage
6	IP67	Indoor and outdoor with some protection from hose directed water, entry of water during submersion at limited depth, ice damage.
12	IP55	Indoor with protection from dust, falling dirt, and dripping non- corrosive liquids
13	IP65	Indoor with protection against dust, spraying water, oil, and non- corrosive liquids

5.2 Cooling of Panels

Very often the power electronic device will be built into a control panel / switchboard of some form. The problem is then to ensure that the temperature rise

Enclosure Cooling Units

Hints for Practical Application

Wherever optimum operating conditions inside an enclosure are required, even with high external temperatures, a Rittal enclosure cooling unit offers just the answer. It is possible to cool the enclosure's internal temperature down to well below the ambient temperature.

The arrangement of air inlet and outlet apertures for the internal and external air circulation, which is favourable in terms of airflow technology, ensures optimum air circulation in the enclosure.

We would like to show you, by means of example, how you can quickly carry out the calculation of a cooling unit.

You should consider the following points:

- Where will the enclosure be sited (dust or oil laden air)?
- What type of installation in accordance with VDE 0660 part 500 does your calculation example allow for (see table)?
- What ambient temperatures must be anticipated (e.g. ambient temperature and humidity)?
- What is the heat loss $\dot{Q}_V$ in the enclosure?
- What ist the max. required interior temperature T_i of the enclosure?
- Is there a requirement with regard to the protection category in accordance with DIN 40 050?
- In enclosure suites, it will be necessary to take into account the heat which may have been penetrating from the adjacent enclosures.
- The irradiating heat, e.g. from the environment into the enclosure, must be taken into account.
- The cooling units should always be connected via door limit switches, in order to avoid any excessive condensation arising.

We are considering a wall mounted cooling unit. In this calculation example, the values entered in blue have been adopted.

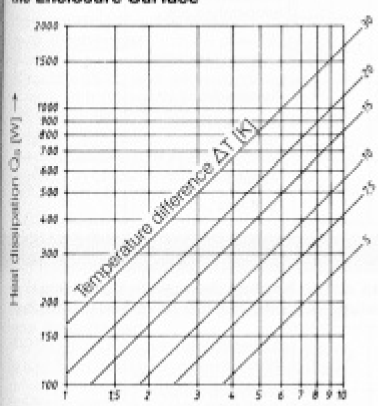
Enclosure dimensions W = 600 H = 2000 D = 500	A	4.4	Calculate the effective enclosure surface (from table 1)	A =	4.4
Temperature outside the enclosure	T_u =	+50	Calculate the temperature diff. $\lvert\Delta T\rvert = \lvert T_i - T_u\rvert$	ΔT =	15
Required temperature inside the enclosure	T_i =	+35			
Heat transition coeff. sheet metal ~ 5.5 W/m² K Plastic ~ 3.5 W/m² K	k =	5.5	Transfer the heat transition coefficient	k =	5.5
Heat loss	$\dot{Q}_V$ =	700	Transfer the heat loss	$\dot{Q}_V$ =	700

The value given between the two vertical lines | | must only be expressed as a positive value whether calculated as minus or positive.

Now calculate your required cooling output $\dot{Q}_E = (A \times \lvert\Delta T\rvert \times k) + \dot{Q}_V$

Required cooling output $\dot{Q}_E$ = Watt 1063

Heat Dissipation through the Enclosure Surface



Enclosure surface A [m²] in accordance with VDE 0660 part 500

Starting values
in interior installation in static air
Heat transition coefficient k = 5.5 W/m² K

$\dot{Q}_S = k \cdot A \cdot (T_i - T_u)$
$\dot{Q}_S > 0$: radiation $(T_i > T_u)$
$\dot{Q}_S < 0$: absorption $(T_i < T_u)$

Additionally:
$\dot{Q}_S = \dot{Q}_V - \dot{Q}_K$ and
$\dot{Q}_S = \dot{Q}_V + \dot{Q}_H$

If $\dot{Q}_K = \dot{Q}_H = 0$, it follows that
$\dot{Q}_S = \dot{Q}_V = k \cdot A \cdot (T_i - T_u)$

Calculation of the heat dissipation from exposed enclosure surfaces A [m²]

The calculating of A is in accordance with DIN 57 660 part 500 or VDE 0660 part 500, taking into account the type of configuration.

Type of enclosure configuration in accordance with VDE 0660 part 500

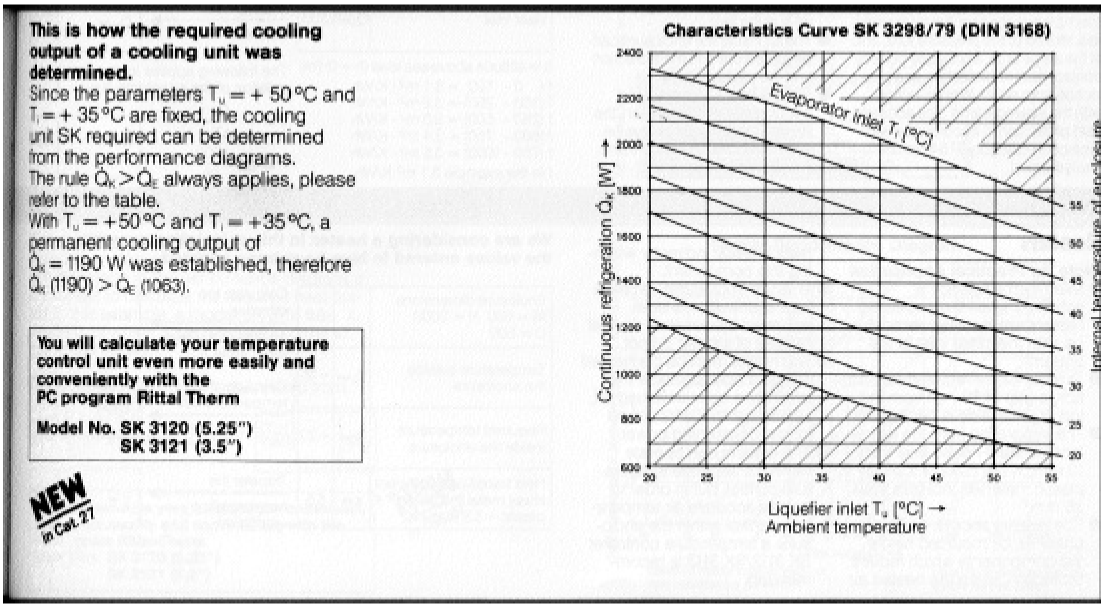
- ☐ Single cabinet, free standing
- ☐ Single cabinet against a wall
- ☐ First/last cabinet of a suite, free standing
- ☐ First/last cabinet of a suite, against a wall
- ☐ Cab. within a suite, free stand.
- ☐ Cab. within a suite against a wal
- ■ Cabinet within a suite against a wall with covered roof areas

Positioning in acc. with VDE 0660/500	Formula for calculating A [m²]
Single cabinet, free standing	A = 1.8 x H x (W + D) + 1.4 x W x D
Single cabinet against a wall	A = 1.4 x W x (H + D) + 1.8 x D x H
First/last cabinet, free standing	A = 1.4 x D x (H + W) + 1.8 x W x H
First/last cabinet, against a wall	A = 1.4 x H x (W + D) + 1.4 x W x D
Cab. within a suite, free stand.	A = 1.8 x W x H + 1.4 x W x D + D x H
Cab. within a suite against a wall	A = 1.4 x W x (H + D) + D x H
Cabinet within a suite against a wall with covered roof areas	A = 1.4 x W x H + 0.7 x W x D + D x H

W = cabinet width [m] H = cabinet height [m] D = cabinet depth [m]

Examples: Exposed cabinet surface for specified dimensions [m²]

W x H x D [mm]	☐	☐	☐	☐	☐	☐	■
600 x 1600 x 600	4.0	3.6	3.6	3.2	3.2	2.8	2.6
800 x 1800 x 500	4.8	4.2	4.4	3.8	4.1	3.5	3.2
600 x 2000 x 500	4.4	3.9	4.0	3.5	3.6	3.1	2.9
800 x 2200 x 600	6.2	5.5	5.7	5.0	5.2	4.5	4.1



Example :

A panel containing 720 Watts of dissipation is placed in the corner of a room where the maximum ambient (air) temperature can reach 35 degC. The panel dimensions are 1000(w) * 1800(h) * 600(d) and is made from painted mild steel.

The calculation shows an internal temperature of nearly 62 degC. If the maximum temperature that the inside of the panel could be allowed to rise to was 45 degC (why should there be any limit ?) then additional cooling would be required. This can be obtained in several ways :

- Larger cabinet
- Free standing not in corner (A increases by 1.25 times
- Add heat extraction

The problem here is that the ambient in the switchroom is quite high and so the panel might have to be very large to meet the internal temperature limit. Air vents could and often are) added to a panel but this has a few drawbacks including unpredictable cooling, dirt and grim entering the panel and the risk of the vents getting covered up at some future date.

Panel dimensions (m)

$W := 1.0$ $H := 1.8$ $D := 0.6$ all metres

Formula for positioning is

$A := 1.4 \cdot H \cdot (W + D) + 1.4 \cdot W \cdot D$ $A = 4.872$ sq m

Power to be dissipated $Q := 720$

heat transmission coeff for steel $k := 5.5$ Watts /(sq m)/degC

ambinet temperature $Tamb := 35$

temperature rise $dT := \frac{Q}{k \cdot A}$ $dT = 26.87$

internal ambient $Tinternal := Tamb + dT$ $Tinternal = 61.87$