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COOLING OF HIGH-DENSITY AND POWER ELECTRONICS BY MEANS OF HEAT PIPES

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ABSTRACT

This report describes how heat pipes can be used for cooling modern electronic equipment, with numerous advantages over air-cooled systems. A brief review of heat-pipe properties is given, with a detailed description of a functioning prototype. This is a single-width CAMAC unit containing high-density electronic circuits cooled by three heat pipes, and allowing a dissipation of over 120 W instead of the normal maximum of 20 W.

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CONTENTS

1. INTRODUCTION

The component density of and the power absorbed by modern electronics have been steadily increasing over the last ten years. A NIM crate dissipated typically 150 W about ten years ago, while the present crates for fast data handling already surpass the 1 kW level. A few of the problems that are related to the increased power consumption are as follows:

i) The quantity and the speed of the air required for adequate cooling have attained values that become difficult to handle, especially since the component packing is generally dense. Moreover, fans and air ducts take up valuable space in the racks.

ii) The powerful fans required for producing sufficient airflow, and the turbulence of the moving air, produce an uncomfortably high noise level.

iii) The rejected heat puts very high demands on air-conditioning equipment in control rooms.

iv) The complexity of fast integrated circuits is limited because of the difficulty in getting rid of the dissipated heat.

v) The high-power supply currents (sometimes in excess of 100 A) at low voltage create stability problems due to the contact resistance of connectors, resistance of power supply leads, etc.

Unless a radically new way of attacking these problems can be found, we will soon have reached the point where the conventional methods do not offer an acceptable solution.

It is the author's opinion that for the new generation of data-handling equipment, the use of heat pipes offers a solution to the above problems.

It was with this idea in mind that the work was undertaken to construct a number of heat pipes and to equip a standard single-width CAMAC unit with them to see how much heat could be removed from so limited a space.

2. WORKING PRINCIPLES

The first publication written by scientists of the Los Alamos Scientific Laboratory on the principles of heat pipes dates from 1964. A heat pipe consists of an evacuated tube (usually of metal), lined with a porous material (a wick) which is soaked in a suitable liquid and hermetically sealed (Fig. 1).

If a temperature difference exists between the two ends, some fluid will evaporate at the warmer end, thereby absorbing the latent heat of vaporization and slightly increasing

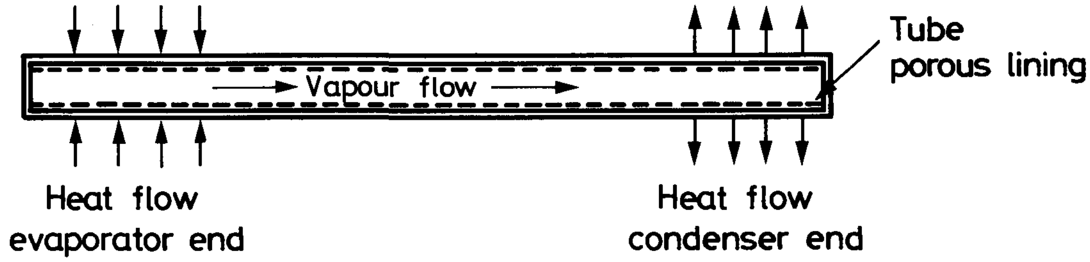


Fig. 1 Schematic drawing of a pipe

the vapour pressure in the tube, causing condensation at the colder end, where it will give off its latent heat of vaporization. The result is vapour flow from the evaporator to the condenser, and heat transfer from the warm end to the cold end.

The condensed liquid returns through the wick by capillary force (plus or minus gravity, depending on the pipe orientation).

According to these principles, heat pipes are capable of transporting large amounts of heat very efficiently and are essentially isothermal. Their thermal conductivity is several orders of magnitude better than that of metallic conductors. Furthermore, they are simple to construct, robust, and extremely reliable; and if made in large quantities, they are cheap.

3. CHOICE OF DIMENSIONS AND MATERIALS

Although the idea was to investigate and, if possible, prove the suitability of heat-pipe cooling for the coming generation of electronics equipment, the actual work was done on a standard single-width CAMAC unit. This was because the mechanical parts and crate were readily available, and thus only a minimum amount of time had to be spent on mechanical work.

The maximum diameter of a heat pipe that can be accommodated in a single-width unit is about 6 mm; the minimum length of the pipe equals the height of the unit (190 mm), plus the length required for heat transfer to a heat sink, which should preferably be less than 44 mm (= 1 U).

As pipe material, stainless steel with a wall thickness of 0.5 mm was chosen because of its compatibility with nearly all working fluids, its good mechanical properties, and its easy availability. For the wick material, a stainless-steel mesh with wire diameter $d = 0.05$ mm and pitch $D = 0.5$ mm and a phosphor-bronze mesh with $d = 0.05$ mm and $D = 0.085$ mm were available.

The working temperature required is somewhere between 30 °C and 40 °C.

It can be found in the literature[1)] that the most suitable working fluid at these temperatures is liquid ammonia (NH_3), with methanol as a not too bad second choice. In view of the rather primitive conditions under which the heat pipes were made in our laboratory, it was decided to use methanol as a working fluid because it is easier to handle.

4. LIMITING FACTORS

There are several ways in which a heat pipe is limited in the amount of heat it can transport:

i) By *Capillary Limitation*, where the capillary pumping force (which is a function of the pitch of the mesh and the surface tension of the working fluid) in combination with the gravitational force is not capable of forcing sufficient return liquid through the wick; this causes drying out of the wick and thus local overheating.

ii) By *Sonic Limitation*, where the velocity of the vapour approaches the velocity of sound. In fact, at vapour velocities over Mach 0.2, the vapour flow cannot be considered as incompressible, which causes thermal gradients along the pipe.

iii) By *Turbulence Limitation*, where the dimensions and shape of the vapour core are such that above a certain power level the vapour flow becomes turbulent, this again leads to thermal gradients along the pipe.

iv) By *Entrainment Limitation*, where the speed of the vapour flow is such that small droplets of liquid are torn from the wick surface; this increases the amount of liquid that circulates, and very soon causes drying out of the wick.

v) By *Boiling Limitation*, where the radial heat flux is such that the temperature drop across the wick thickness causes boiling inside the wick, preventing adequate circulation of the working fluid and causing drying out of the wick. Contrary to the first four limiting factors, which relate to the total amount of heat transported by the pipe (axial heat flow), this limitation concerns the maximum radial heat flux that can be applied per unit of pipe surface (or unit of pipe length).

Examples of the heat-pipe calculations are given in the Appendices; therefore only the results of the calculations will be given below:

Heat pipe No. 1 (see Appendix I):

Material	:	Stainless steel
Outer diameter	:	6 mm
Inner diameter	:	5 mm
Length	:	190 mm
Wick	:	Five layers of wrapped stainless-steel mesh, d = 0.05 mm, D = 0.5 mm
Wick thickness	:	0.5 mm
Working fluid	:	Methanol
Working temperature	:	∿ 40 °C
Limitations:		
Capillary, vertical (gravity assisted)	:	∿ 650 W
Capillary, horizontal	:	∿ 46.5 W
Sonic	:	∿ 1400 W
Turbulence	:	∿ 87 W
Entrainment	:	∿ 122 W
Boiling	:	∿ 2 W/cm pipe length

These values are already quite useful for a heat pipe of the simplest possible construction, especially since it can be used in the vertical position. The severest limitation in this case is the boiling limitation. For this reason another heat pipe with a different wick was calculated; here a single layer of much finer mesh was used. In order to avoid severe capillary limitation, an artery wick was preferred (see Fig. 2). The artery serves as a low-resistance return for the liquid flow and is made of the same mesh as the wick.

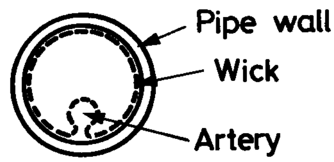


Fig. 2 Heat pipe with artery wick

The following data apply:

Heat pipe No. 2 (see Appendix II):

Material	:	Stainless steel
Outer diameter	:	6 mm
Inner diameter	:	5 mm
Length	:	190 mm
Wick	:	Artery wick of one layer of phosphor-bronze mesh $d = 0.05$ mm, $D = 0.085$ mm
Wick thickness	:	0.1 mm
Inner diameter of artery	:	1.5 mm
Working fluid	:	Methanol
Working temperature	:	∿ 40 °C
Limitations:		
Capillary, vertical (gravity assisted)	:	∿ 311 W
Capillary, horizontal	:	∿ 113 W
Sonic	:	∿ 1800 W
Turbulence	:	∿ 140 W
Entrainment	:	∿ 370 W
Boiling	:	∿ 26 W/cm pipe length

As these values are more than adequate for our purpose, a number of prototype heat pipes of this latter design were made.

5. HEAT TRANSFER

i) The problems encountered in transferring the heat from the components to the ultimate cooling agent -- which in our case is water at about 17 °C -- are nearly all interface problems. Since the units to be cooled are of the plug-in type, a water connection to the plug-in unit would be impractical. For this reason it was decided to equip the top of the crate with water-cooled strips (see Fig. 3), and to have the heat pipes make sliding contact with these strips via spring-loaded copper blocks (see Fig. 4). The copper blocks are necessary in order to have sufficient contact area and thus low thermal resistance.

ii) The first prototypes used a copper block of 60 × 40 × 8 mm that was shrunk onto the heat pipe. Measurements showed that the heat resistance between pipe and copper block was slightly higher than the heat resistance between copper block and cooling strip.

iii) Considerable improvement could be achieved by making the copper block an integral part of the heat pipe. This was done by drilling (and afterwards plugging) a number of channels in the block, as illustrated in Fig. 5. Condensation of the working fluid

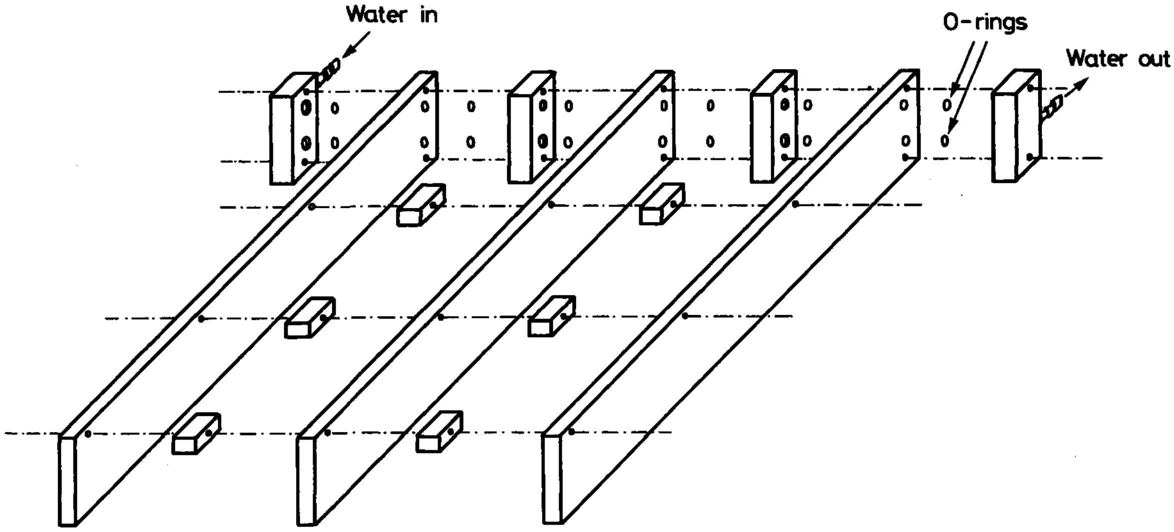


Fig. 3 Modular assembly cooling strips

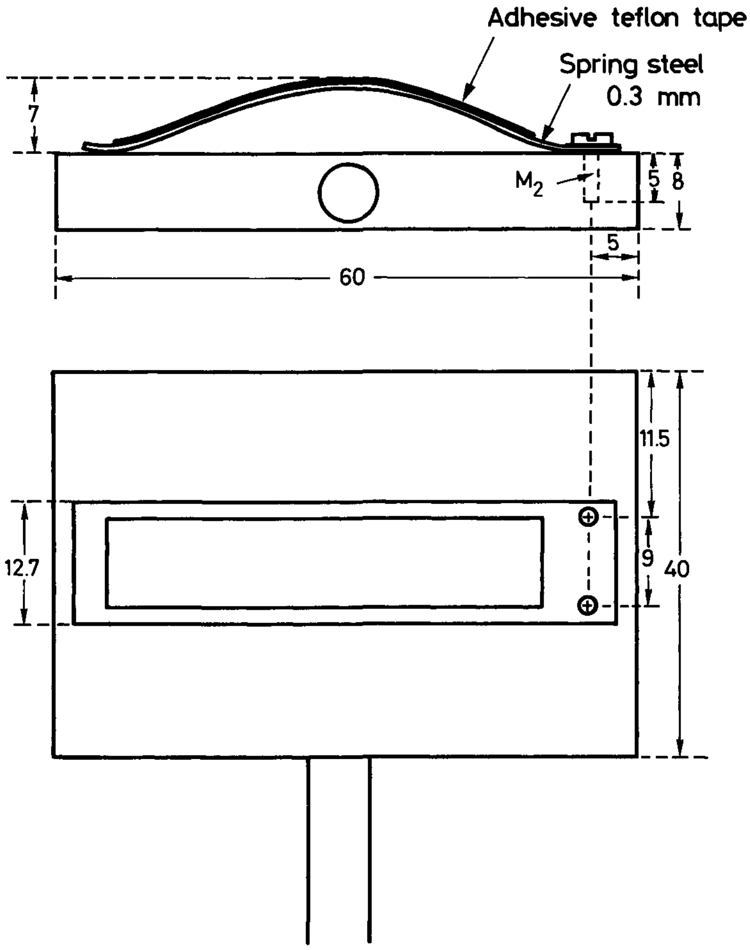


Fig. 4 Spring-loaded copper block

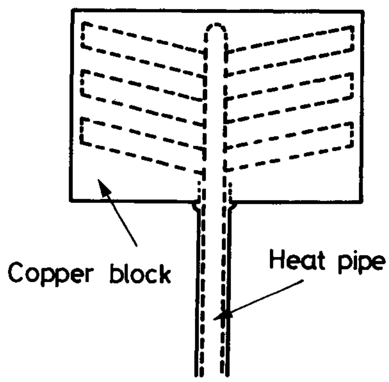


Fig. 5 Integral condenser block

is on the walls of the channels, so the heat is deposited inside the block. The advantages are that the condenser area is much greater, which means that the radial heat flux per area is smaller and that the average distance to the cooling surface is smaller. Since it was not practical to line the channels with a wick, the return of the condensed liquid is exclusively by gravity. This implies that the heat pipe will only work properly in a vertical or near-vertical position.

iv) Heat transfer from the copper block to the water-cooled strip is one of the most critical points in the system. Both the copper block and the cooling strip were machined and polished to provide a flat, smooth surface. A small amount of silicon grease was applied to the surface in order to avoid having a layer of insulating air between the contact surfaces, and also to ensure minimum friction when sliding. In this way adequate heat transfer could be achieved. Numerical data will be given in Section 7.

v) The design of the cooling strips is given in Fig. 6. Since the power per plug-in unit (and thus per cooling strip) is not very high (in our test set-up ∿ 120 W), the required flow of cooling water is also fairly low, which results in laminar flow inside the strips. As a consequence, heat transfer to the coolant is rather poor. Apart from increasing the contact area between strip material and coolant there is not much that can be done about this.

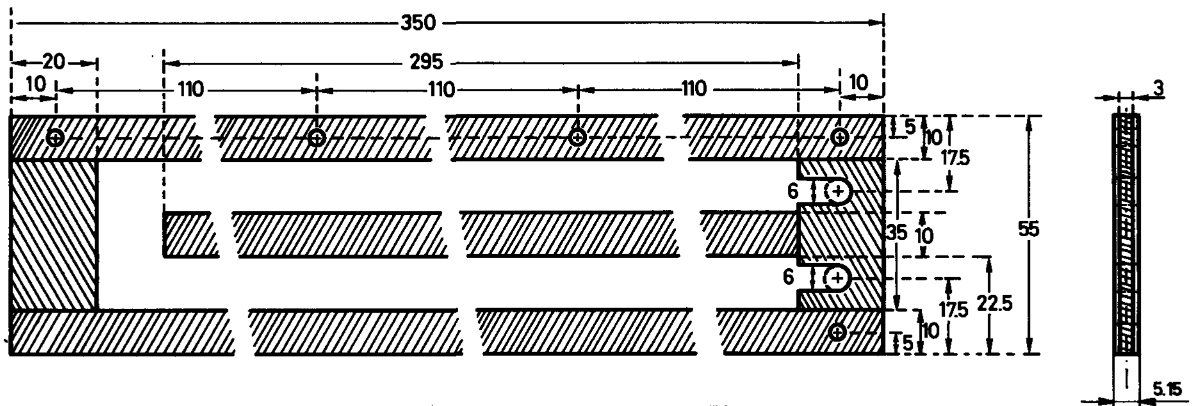


Fig. 6a Cooling strip

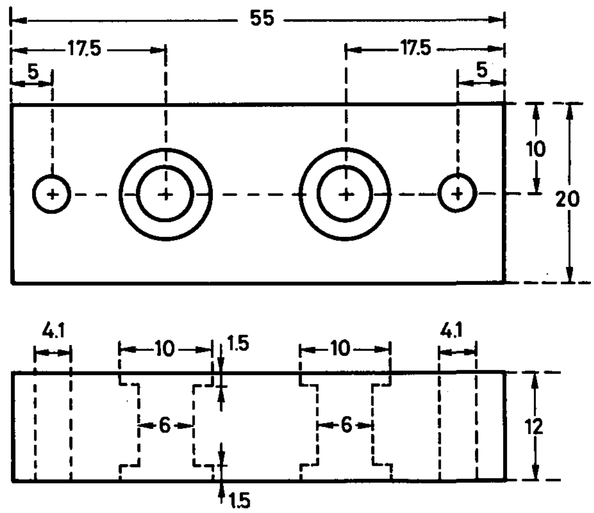


Fig. 6b Distance piece for cooling strips

vi) For the heat transfer from the components to the heat pipe, various adapters (heat links) have to be made. Since the capacity of the heat pipes is large compared to the heat developed by the individual components, with the exception perhaps of power transistors and integrated circuits for stabilization of voltages and currents, the links can generally be made to accommodate several components (see Figs. 7 and 8). The links have a semicircular hollow to fit the heat pipes. Good thermal contact between adapters and pipes is ensured by using a small amount of heat paste and one or more little steel clips to have adequate contact pressure. The links are made of copper, but in the cases where dissipation is low and thus a higher thermal resistance is allowed, they could be made of aluminium to save weight.

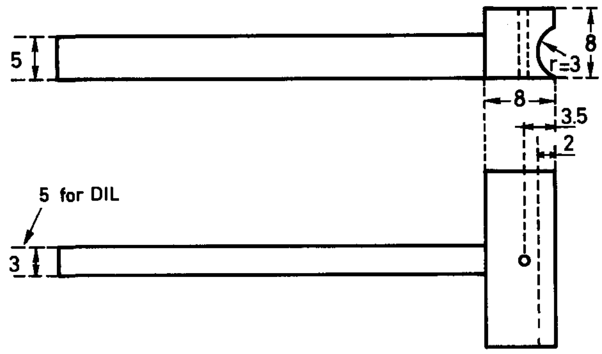


Fig. 7 Adapter for resistance networks or DIL

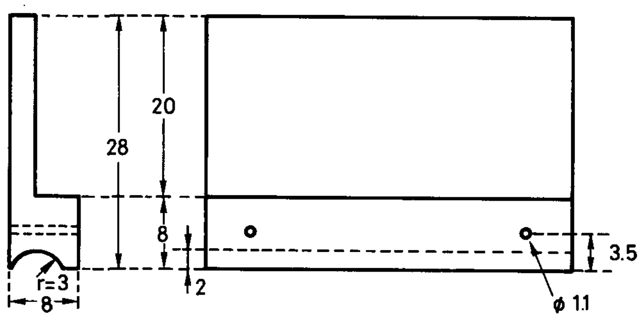


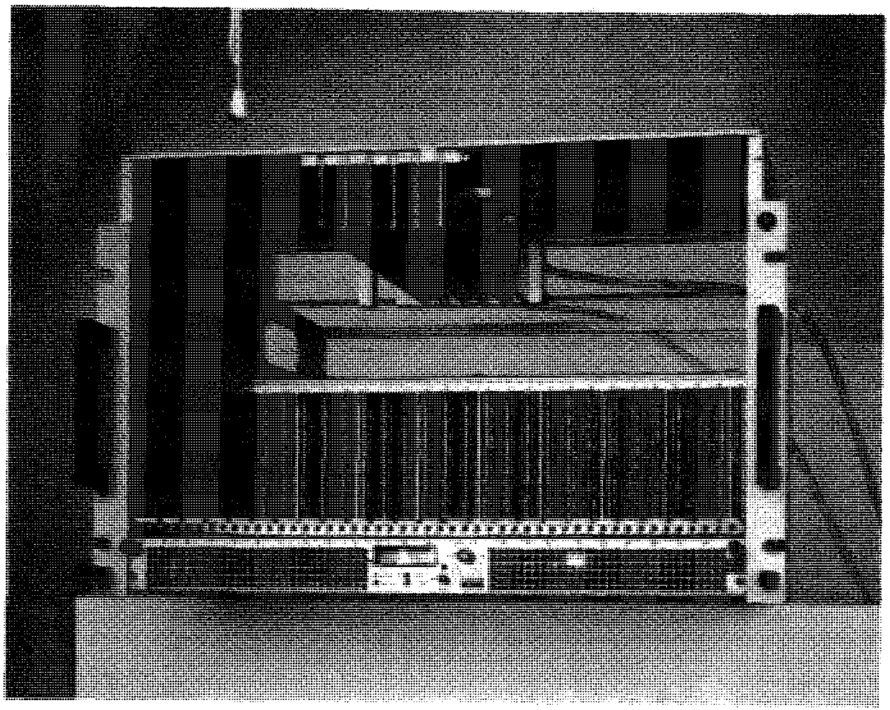
Fig. 8 Adapter for power transistors

6. TEST SET-UP

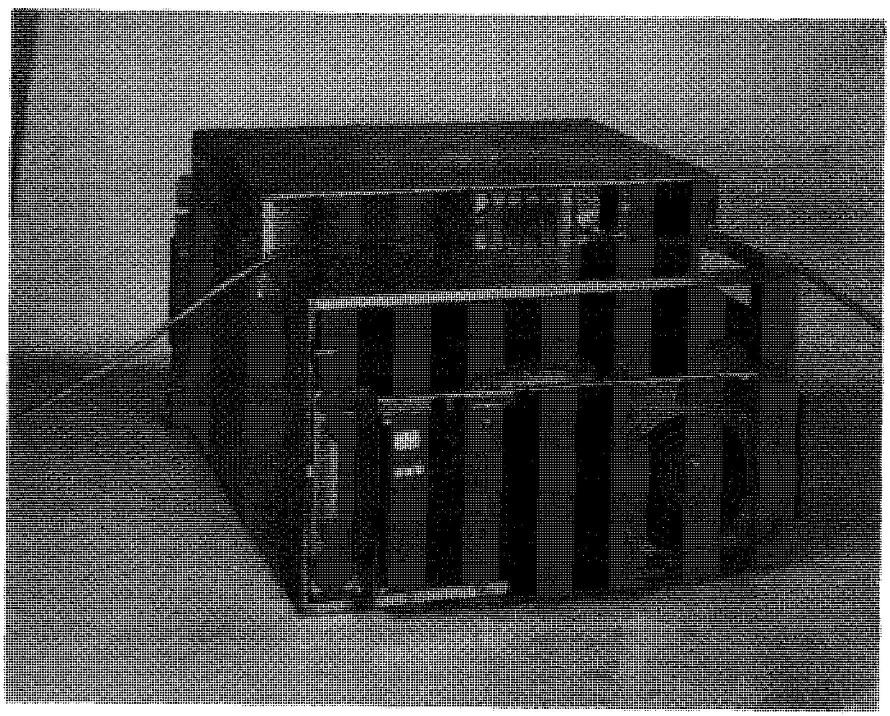
As mentioned earlier, a standard single-width CAMAC unit was used in the test. In view of the dimensions of the printed circuit board (182 × 285 mm), it was decided to equip the unit with three heat pipes, each serving to cool a surface of roughly 180 × 90 mm. From the point of view of heat capacity, two heat pipes ought to be more than adequate, but in that case the length of the adapter pieces has to be increased, and the thermal resistance increases in consequence. Also, the radial heat flow of the heat pipes becomes fairly high in this case, which adds to the temperature drop at the interface link → heat pipe.

The dissipating elements of the set-up are as follows:

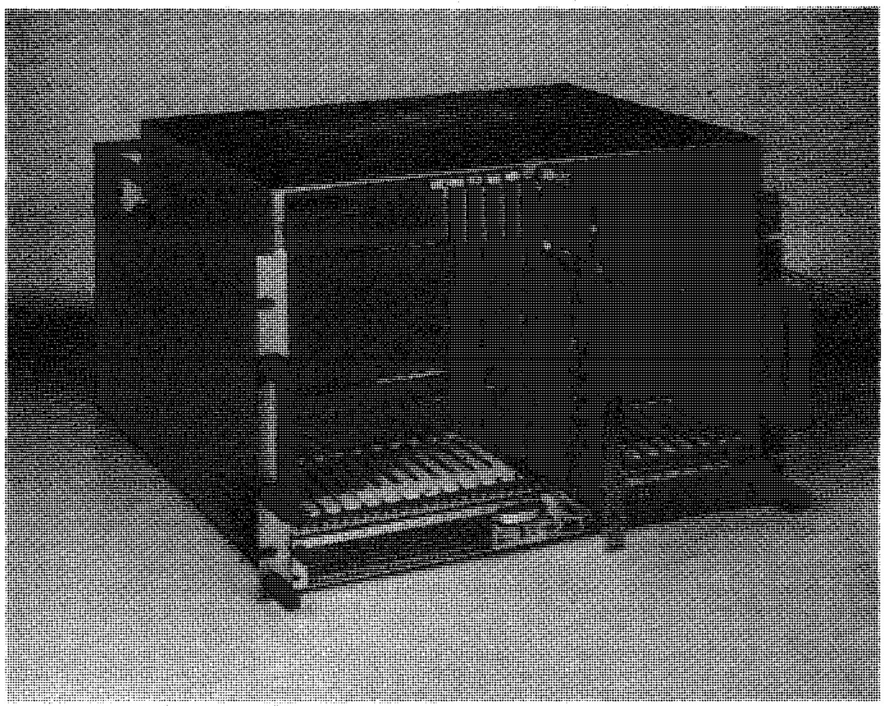
i) Power supply circuits having in general a high dissipation per unit. These are mounted on flat copper adapters (see Fig. 8).



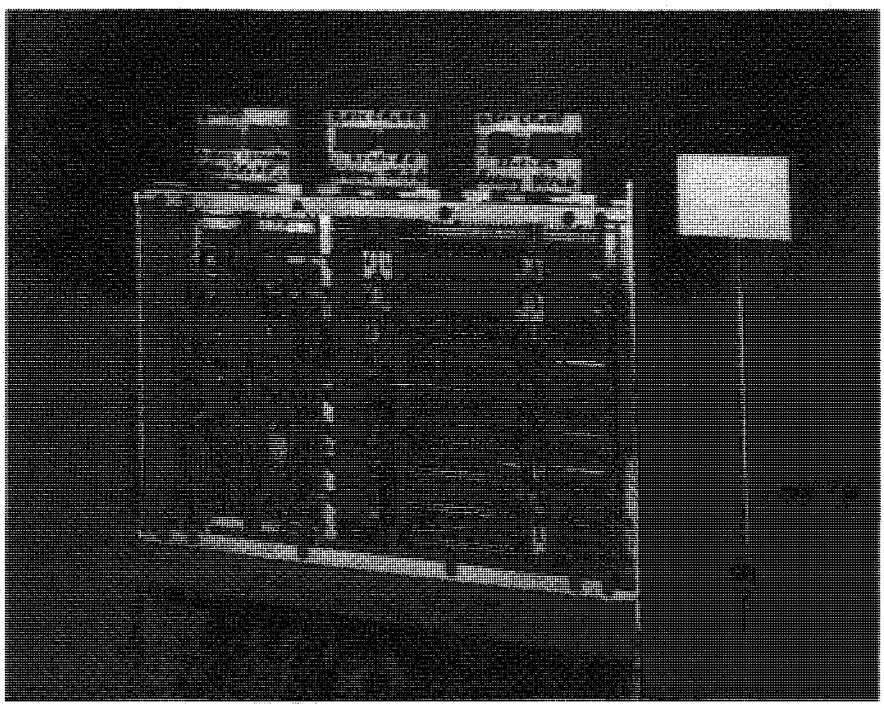
Crate with cooling strips



Water connections' cooling strips



Crate with prototype unit



Prototype and one heat pipe

ii) Integrated circuits with a comparatively low dissipation per unit. The dual in-line package is most suitable for mounting on a copper adapter of square cross-section (5 × 5 mm, see Fig. 7).

iii) Low-power transistors of 300 mW each in a plastic TO_{92} package, mounted with the flat side in contact with the side of the square adapter and using heat paste.

iv) Carbon resistors, 0.25 W, at a dissipation of 300 mW each, straddling the square adapters and using heat paste.

v) Resistance networks in a single in-line package, mounted in contact (but without heat paste) with the sides of a 3 × 5 mm copper adapter, using a spring-steel clip to provide adequate contact pressure.

vi) Low dissipating, free-mounted, resistance networks (50 Ω load resistors for the MECL circuits).

The heat load is distributed over the three pipes in the following way:

i) Back-end heat pipe, serving to cool the power supply circuits with a total dissipation of about 49.2 W. It should be noted that the dissipation of the one -5.2 V stabilizer has been chosen to be equivalent to a stabilizer of -5.2 V, 10-12 A that is powered from a non-stabilized voltage of 8-8.5 V.

ii) Centre heat pipe serving to cool 10 integrated circuits (ICs), 8 low-power transistors, and a number of resistors and resistance networks. Total dissipation, about 22.7 W.

iii) Front-end heat pipe serving to cool 6 ICs and 28 resistance networks. Total dissipation, about 48.6 W.

Details are given in Appendix IV.

The dissipation of the entire unit is about 120 W. From the fact that the centre heat pipe operates at less than half of the load of the other two, it is evident that the system as such could handle even more power.

7. TEMPERATURE MEASUREMENTS

At a number of places in the system described above, the temperatures were measured and the resulting junction or component temperatures were calculated. Details of the measurements and calculations are given in Appendix IV.

7.1 Temperatures of the cooling strip

Cooling water in	:	17.5 °C
" " out	:	23.7 °C
Average temperature at the surface	:	27.6 °C

In order to avoid condensation problems, it is advisable to have the surface temperature of the cooling strip at or slightly above ambient temperatures.

7.2 Working temperatures of heat pipes, and thermal resistances to the cooling strip

	Pipe temperature T_p	Thermal resistance pipe → sink $R_{th\ p \to s}$
Back-end pipe	45.4 °C	0.36 °C/W
Centre pipe	36.8 °C	0.41 °C/W
Front-end pipe	44.8 °C	0.35 °C/W

7.3 Working and junction temperatures of voltage stabilizers

i) -5.2 V, 1.85 A, consisting of a standard three-terminal voltage stabilizer MC7905.2 and a power transistor MJE3055, mounted together on a flat heat link. The temperature of the link was 66.8 °C.

Dissipation of the MC7905.2 = 6.4 W.

Thermal resistance junction → sink, $R_{th\ j \to s}$ = 5.6 °C/W, which gave a junction temperature of about 103 °C.

Dissipation of the power transistor = 33 W.

$R_{th\ j \to s}$ = 2 °C/W :: junction temperature ∿ 133 °C.

In both cases 150 °C is allowed, so there is still some safety margin.

ii) ±15 V, 160 mA. Standard three-terminal stabilizers, both dissipating about 1.6 W and mounted on one flat heat link.

Link temperature: 51 °C, $R_{th\ j \to s}$ = 5.6 °C/W, which gave a junction temperature of about 60 °C.

iii) A series pass diode mounted on a flat heat link, dissipation about 4.5 W.

Link temperature 55 °C, $R_{th\ j \to s}$ ∿ 9.2 °C/W, which gave a junction temperature of about 97 °C.

iv) -2 V, 0.5 A stabilizer mounted on the same heat link as the series pass diode; dissipation ∿ 2 W. Approximate junction temperature 67 °C.

7.4 Junction temperature of the integrated circuits

The ICs used were all MC10136P with a dissipation of 650 mW each (loaded). The worst-case value of the thermal resistance of the plastic DIL package was 70 °C/W [2]. This meant a junction temperature of about 46 °C above mounting base temperature.

The temperature of the heat links on which the ICs were mounted varied from 39 °C to 50.6 °C. As a consequence, the junction temperatures of the individual ICs varied from 85 °C maximum to 97 °C maximum. For ceramic packages, these values were considerably lower owing to the lower thermal resistance of the package.

It is clear that since the noise margin depends, amongst other things, on the differences in junction temperature between chips, a heat-pipe system, with its inherently small differences in temperature, has definite advantages over an air-cooled system, where the temperature differences can be considerable.

7.5 Temperature of resistance networks

The resistance networks were purposely overloaded (overload 73%) and the surface temperatures were measured. The hottest resistor network reached 92 °C at an adapter temperature of 70 °C. During these temperature measurements the components and adapters were carefully insulated with cotton wool in order to avoid cooling by the ambient air. With a better heat link, the temperature of the networks could probably be lowered by 10 °C or more.

8. ADVANTAGES OF HEAT-PIPE COOLING

i) It allows power levels that would be impossible to handle in an air-cooled system.

ii) It is silent.

iii) Owing to the absence of air flow there is no accumulation of dust in the units.

iv) It is almost maintenance free.

v) It puts no demands on air-conditioning equipment.

vi) Because of a more uniform temperature the noise margin is improved.

vii) Perhaps the biggest advantage is that it will allow on-board stabilization, with its freedom from voltage drops at connectors and in power supply leads. Moreover, it allows the power supplies to be separated from the crate, since a slight voltage drop in the connecting cable is acceptable. This would make the crates much easier to handle and could save valuable rack space.

9. DISADVANTAGES

A definite disadvantage of the system is that it puts certain constraints on the circuit layout. It will be necessary to leave a free space for the heat pipes on the board, and to orient the components in such a way that they fit the adapters on the heat pipe. Also, care would have to be taken to mount components with a high dissipation in the immediate vicinity of a heat pipe. The board space lost because of the heat pipes is between 10% and 15%. Another point is the necessity for a cooling water supply. For small, temporary installations, ordinary tap water might do, but for larger installations a separate supply of demineralized water would be advisable.

As far as prices are concerned, it is still too early to be able to make an estimate. Much would depend on the quantities that would be required. Since the materials for a heat pipe as described cost less than SF 2.50 and there are no particular difficulties in manufacturing, the price in series production could be low.

It is doubtful whether a heat-pipe system would be more expensive per kilowatt of installed power than an air-cooled system, especially if one takes into account the cost of air-conditioning.

10. PRACTICAL PROBLEMS (AND POSSIBLE IMPROVEMENTS)

One problem is that of alignment. In the prototype used there are three heat pipes. In order to ensure that all three adapters fit the cooling strip to within, say, 0.05 mm, care has to be taken that the cooling strip is straight to within 0.025 mm and that the alignment of the three blocks also is good to within 0.025 mm. A misfit of this magnitude can be compensated for by silicon grease. Although it is quite possible to achieve this precision, great care is needed in mounting. A system with heat pipes that have a certain flexibility would definitely be better in this respect.

Measurements of the temperature profile of the heat pipes showed that the maximum heat load of the pipes used is considerably lower than that calculated. Although this is not a problem in the present set-up with its limited heat load, it is something that deserves attention, and the explanation may be that since the channels in the copper block are not lined with a wick, the working fluid returns by gravity. At the joint of the heat pipe and copper block, the returning liquid encounters a jet of vapour, which has a tendency to blow the liquid back into the channels of the block, thereby disturbing the liquid inventory of the pipe and causing partial dry-out of the wick with local overheating as a result. Effectively the entrainment limit of the pipe is considerably lowered. Another effect that results is the partial blocking of the channels by liquid, which increases the thermal resistance.

A possible way of avoiding this effect is to replace the copper block by a condenser that is entirely hollow, so that in the region where the returning liquid enters the wick it is protected from the vapour jet by a thin metallic lining (see Fig. 9). Added advantages of this modification are the considerable reduction in the weight of the condenser, and a construction that lends itself better to series production.

In the prototype, use has been made of heat links to conduct the heat from the components to the heat pipes.

An alternative solution, using conventional card layout, is possible. This approach would be to cover the component side of the printed circuit board with a 0.5 mm copper sheet on to which the components are mounted in a way that ensures good thermal contact between

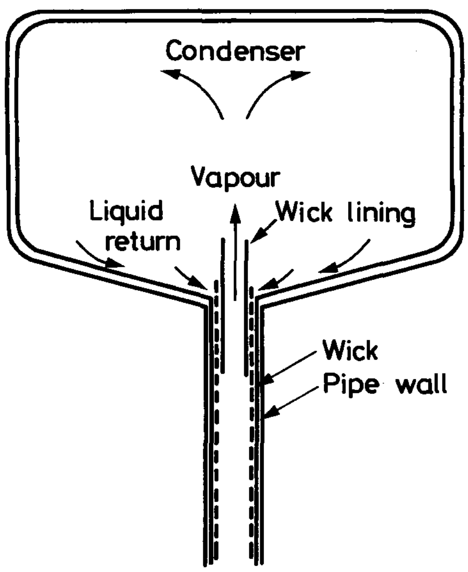


Fig. 9 Improved condenser design

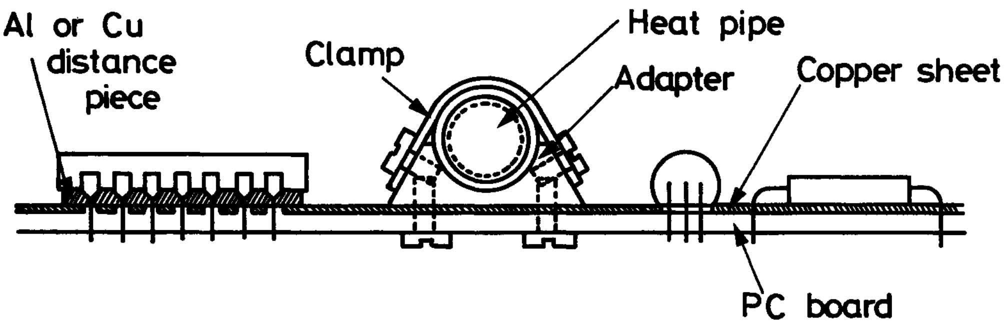


Fig. 10 Copper sheet heat conduction

the components and the sheet. The heat pipes would have to be mounted on the sheet with the aid of an adapter that minimizes the thermal resistance between the sheet and the heat pipe (see Fig. 10). An added advantage would be that the sheet provides an excellent ground plane.

The calculation given in Appendix V shows that for a board the size of a CAMAC unit there would be a maximum dissipation of about 80 W. For 1 mm sheet material this value nearly doubles.

Of course these values are subject to variation because of hot spots that are the result of highly dissipating components, obstructions in the heat flow path by rows of holes, etc.; but at least they give an indication of the capacity, and it is certain that power levels of several times the maximum possible with air cooling are within easy reach.

Another possible improvement is in the diameter of the heat pipes. Because of dimensional constraints, a diameter of 6 mm was chosen, but it is obvious that a larger diameter gives a lower thermal resistance at the pipe interface.

The use of a grooved heat pipe would make for a very simple and thus cheap construction with an acceptable thermal resistance. However, because of the need for special extrusions, this is only interesting for large series. An example of the calculation is given in Appendix III.

11. CONCLUSIONS

The main object of this report is not to present a ready solution for all the cooling problems of modern electronics, but to show that it is very well possible to construct plug-in units that are cooled by heat pipes and that are capable of handling a far higher power level than is possible with air cooling.

The present prototype effectively handles 120 W power dissipation in a unit that was designed for 20 W maximum, without any of the components overheating.

It has been shown that the heat transfer capacity of this system can more than handle the dissipation that results from maximum density of standard components on any card. Doubtless the system can be improved, but already it provides a workable basis for the design of cooling systems.

Acknowledgements

The author wishes to thank Dr. B.D. Hyams for his support, his keen interest and many valuable suggestions, and Mr. Louis Bonet for his help in making the prototypes and, out of his invaluable experience, improving on them.

* * *

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2) Motorola Semiconductor Products, Inc., MECL, Systems Design Handbook, 2nd Edition (1972).

APPENDIX I

CALCULATIONS FOR HEAT PIPE No. 1

Characteristics

Material		:	Stainless steel
Outer diameter	(OD)	:	6 mm = 6×10^{-3} m
Inner diameter	(ID)	:	5 mm = 5×10^{-3} m
Evaporator length	(L_e)	:	170 mm = 0.17 m
Adiabatic length	(L_a)	:	20 mm = 0.02 m
Total length ($L_t = L_e + L_a$)		:	0.19 m
Wick		:	Five layers stainless steel mesh = 0.5 mm
Mesh pitch	(D)	:	0.5 mm = 5×10^{-4} m
wire diameter	(d)	:	0.05 mm = 5×10^{-5} m
Working temperature		:	40 °C = 313 K
Working fluid		:	Methanol

Properties of methanol at 313 K

Liquid density	(ρ_ℓ)	:	890 kg/m³
Liquid viscosity	(μ_ℓ)	:	0.456×10^{-3} kg/ms
Surface tension	(σ)	:	21.85×10^{-3} N/m
Latent heat of vaporization	(λ)	:	12×10^5 J/kg
Vapour density	(ρ_v)	:	1.5 kg/m³
Vapour viscosity	(μ_v)	:	1×10^{-5} kg/ms

Calculation of maximum pumping pressure P_{pm}

Capillary radius: $$r_c = \frac{D}{2} = \frac{5 \times 10^{-4}\ \text{m}}{2}$$

Maximum capillary pressure: $$P_{cm} = \frac{2\sigma}{r_c} = \frac{2 \times 21.85\times10^{-3}\ \text{N/m}}{2.5 \times 10^{-4}\ \text{m}} = 174.8\ \text{N/m}^2 .$$

Normal hydrostatic pressure:

vertical position, $\Delta P_\perp = 0$

horizontal position, $\Delta P_\perp = \rho_\ell g ID = 890 \times 9.81 \times 5\times10^{-3}\ \text{N/m}^2 = 43.65\ \text{N/m}^2$.

Axial hydrostatic pressure:

vertical position, $= \rho_\ell g L_t \sin\psi = 890 \times 9.81 \times 0.19\ \text{N/m}^2 = 1659\ \text{N/m}^2$

horizontal position, $= 0$

Maximum effective pumping pressure: $P_{pm} = P_{cm} - \Delta P_{\perp} + \rho_{\ell} g L_t \sin\psi$

P_{pm} (vertical) $= 174.8 - 0 + 1659\ \mathrm{N/m^2} = 1834\ \mathrm{N/m^2}$

P_{pm} (horizontal) $= 174.8 - 43.65 + 0\ \mathrm{N/m^2} = 131.15\ \mathrm{N/m^2}$

Calculation of liquid frictional coefficient F_{ℓ}

Wick cross-sectional area: $A_w = \dfrac{\pi(d_i^2 - d_v^2)}{4} = 7.07 \times 10^{-6}\ \mathrm{m^2}$

Wick crimping factor: $S \simeq 1.05$

Wick porosity: $\varepsilon = 1 - \dfrac{\pi S d}{4D} = 1 - \dfrac{\pi \times 1.05 \times 5\times10^{-5}}{4 \times 5\times10^{-4}} = 0.9175$

Wick permeability: $C_w = \dfrac{d^2\varepsilon^3}{122(1-\varepsilon)^2} = \dfrac{25\times10^{-10} \times (0.9175)^3}{122(1 - 0.9175)^2} = 2.326 \times 10^{-9}\ \mathrm{m^2}$.

Liquid frictional coefficient: $F_{\ell} = \dfrac{\mu_{\ell}}{C_w A_w \rho_{\ell} \lambda} = \dfrac{0.456 \times 10^{-3}}{2.326\times10^{-9} \times 7.07\times10^{-6} \times 890 \times 12\times10^5}$

$= 25.97\ \mathrm{N/m^2/W{\cdot}m}$.

Vapour frictional coefficient F_v

Vapour core cross-sectional area: $A_v = \dfrac{\pi d_v^2}{4} = \dfrac{\pi \times 16\times10^{-6}}{4} = 12.57 \times 10^{-6}\ \mathrm{m^2}$.

Vapour core hydraulic radius: $r_{hv} = 2\ \mathrm{mm} = 2 \times 10^{-3}\ \mathrm{m}$.

Drag coefficient: $f_v Re_v = 16$

$$F_v = \frac{(f_v Re_v)\mu_v}{2A_v r_{hv}^2 \rho_v \lambda} = \frac{16 \times 1\times10^{-5}}{2 \times 12.57\times10^{-6} \times 4\times10^{-6} \times 1.5 \times 12\times10^5} = 0.884\ \mathrm{N/m^2/W{\cdot}m} .$$

Capillary heat transport factor

$(QL)_{c\,max}$ (vertical) $= \dfrac{P_{pm}}{F + F_v} = \dfrac{1834}{25.97 + 0.884}\ \mathrm{W{\cdot}m} = 68.3\ \mathrm{W{\cdot}m}$

$(QL)_{c\,max}$ (horizontal) $= \dfrac{131.15}{25.97 + 0.884}\ \mathrm{W{\cdot}m} = 4.88\ \mathrm{W{\cdot}m}$

$Q_{c\,max}$ (vertical) $= \dfrac{(QL)_{c\,max}}{\frac{1}{2}L_e + L_a} = \dfrac{68.3}{0.085 + 0.02}\ \mathrm{W} = 650\ \mathrm{W}$

$Q_{c\,max}$ (horizontal) $= \dfrac{4.88}{0.085 + 0.02}\ \mathrm{W} = 46.48\ \mathrm{W}$.

Sonic limit $Q_{s\,max}$

$$Q_{s\,max} = M_v A_v \rho_v \lambda \sqrt{\gamma_v R_v T_v} ,$$

where M_v = max. Mach number (= 0.2)

γ_v = specific heat ratio = 1.33

R_v = vapour constant = $\frac{\text{universal gas constant}}{\text{molecular weight}} = \frac{8314}{34}$

T_v = vapour temperature in degrees Kelvin .

$$Q_{s\,max} = 0.2 \times 12.57{\times}10^{-6} \times 1.5 \times 12{\times}10^{5} \sqrt{1.33\ \frac{8314}{34}\ 313} = 1443\ \text{W} .$$

Turbulence limit $Q_{t\,max}$

$$Q_{t\,max} = \frac{Re_v A_v \mu_v \lambda}{2r_{hv}} \quad \text{(where } Re_v = \text{Reynold's number} = 2300)$$

$$= \frac{2300 \times 12.57{\times}10^{-6} \times 1{\times}10^{-5} \times 12{\times}10^{5}}{2 \times 2{\times}10^{-3}} = 86.7\ \text{W} ,$$

Entrainment limit $Q_{e\,max}$

$$Q_{e\,max} = A_v \lambda \sqrt{\frac{\sigma\rho_v}{2r_{hs}}} \quad \text{(where } r_{hs} = D/2)$$

$$= 12.57{\times}10^{-6} \times 12{\times}10^{5} \sqrt{\frac{21.85{\times}10^{-3} \times 1.5}{2 \times 0.25{\times}10^{-3}}} = 122.12\ \text{W} .$$

Boiling limit $Q_{b\,max}$

First calculate the thermal conductivity K_e of the saturated wick, with

thermal conductivity of liquid methanol: $K_\ell \sim 0.20$ W/m K ,

" " " stainless steel: $K_w \sim 17.3$ W/m K ,

wick porosity: $\varepsilon = 0.9175$:

$$K_e = \frac{K_\ell[(K_\ell+K_w) - (1-\varepsilon)(K_\ell-K_w)]}{[(K_\ell+K_w) + (1-\varepsilon)(K_\ell-K_w)]}$$

$$= \frac{0.2[(0.2+17.3) - (1-0.9175)(0.2-17.3)]}{[(0.2+17.3) + (1-0.9175)(0.2-17.3)]} = 0.236\ \text{W/m K}$$

$$Q_{b\,max} = \frac{2\pi L_e K_e T_v}{\lambda\rho_v \ln (r_i/r_n)} \left(\frac{2\sigma}{r_n} - P_c\right) ,$$

where L_e is the evaporator length and r_n the nucleation radius of vapour bubbles. For carefully outgassed liquid, $r_n \sim 0.25 \times 10^{-6}$ m.

If we want to know the maximum power per centimetre of heat-pipe length ($L_e = 0.01$ m), $P_c \ll 2\sigma/r_n$ can be neglected, and we get

$$Q_{b\,max} = \frac{2\pi \times 0.01 \times 0.236 \times 313}{12{\times}10^{5} \times 1.5 \ln 5/4} \left(\frac{2 \times 21.85{\times}10^{-3}}{0.25{\times}10^{-6}}\right) = 2.02\ \text{W/cm} .$$

Thermal resistance of the heat pipe at the evaporator

The most interesting value is the thermal resistance per centimetre of heat-pipe length (L_e = 0.01 m)

for the pipe wall: $$R_{pe} = \frac{\ln r_0/r_i}{2\pi L_e K_w} = \frac{\ln 6/5}{2\pi \times 0.01 \times 17.3} = 0.1677 \text{ °C/W} ,$$

for the wick: $$R_{we} = \frac{\ln 5/4}{2\pi \times 0.01 \times 0.236} = 15.05 \text{ °C/W} .$$

Total thermal resistance per centimetre of heat-pipe length: $\sim$ 15.22 °C/W.

Measurements show that in reality this value is somewhat lower.

APPENDIX II

CALCULATIONS FOR HEAT PIPE No. 2

Characteristics

Material		:	Stainless steel
Outer diameter	(OD)	:	6 mm = 6×10^{-3} m
Inner diameter	(ID)	:	5 mm = 5×10^{-3} m
Evaporator length	(L_e)	:	170 mm = 0.17 m
Adiabatic length	(L_a)	:	20 mm = 0.02 m
Total length ($L_t = L_e + L_a$)		:	0.19 mm
Wick		:	Artery, single layer, 0.1 mm thick
Mesh		:	Phosphor bronze D = 85×10^{-6} m d = 50×10^{-6} m
Inner diameter of artery		:	1.5 mm = 15×10^{-4} m
Working temperature		:	40 °C = 313 K
Working fluid		:	Methanol

Properties of methanol at 313 K

Liquid density	(ρ_ℓ)	:	890 kg/m³
Liquid viscosity	(μ_ℓ)	:	0.456×10^{-3} kg/ms
Surface tension	(σ)	:	21.85×10^{-3} N/m
Latent heat of vaporization	(λ)	:	12×10^5 J/kg
Vapour density	(ρ_v)	:	1.5 kg/m³
Vapour viscosity	(μ_v)	:	1×10^{-5} kg/ms

Calculation of maximum pumping pressure P_{pm}

Capillary radius: $r_c = \frac{D}{2} = 0.0425 \text{ mm} = 425 \times 10^{-7} \text{ m}$

Maximum capillary pressure: $P_{cm} = \frac{2\sigma}{r_c} = \frac{2 \times 21.85 \times 10^{-3} \text{ N/m}}{425 \times 10^{-7} \text{ m}} = 1028 \text{ N/m}^2$.

Normal hydrostatic pressure:

vertical position, $\Delta P_\perp = 0$

horizontal position, $\Delta P_\perp = \rho_\ell g ID = 890 \times 9.81 \times 5 \times 10^{-3} \text{ N/m}^2 = 43.65 \text{ N/m}^2$.

Axial hydrostatic pressure:

vertical position $= \rho_\ell g L_t \sin\psi = 890 \times 9.81 \times 0.19\ N/m^2 = 1659\ N/m^2$

horizontal position $= 0$.

Maximum effective pumping pressure: $P_{pm} = P_{cm} - \Delta P_\perp + \rho_\ell g L_t \sin\psi$

P_{pm} (vertical) $= 1028 - 0 + 1659\ N/m^2 = 2687\ N/m^2$

P_{pm} (horizontal) $= 1028 - 43.65 + 0\ N/m^2 \sim 984\ N/m^2$

<u>Calculation of liquid frictional coefficient F_ℓ</u>

Artery permeability: $C_a = \frac{r^2}{8} = \frac{(0.75 \times 10^{-3})^2}{8} = 7.03 \times 10^{-8}\ m^2$

Artery cross-sectional area: $A_a = \pi(0.75 \times 10^{-3})^2 = 1.77 \times 10^{-6}\ m^2$

Artery liquid frictional coefficient: $F_{\ell a} = \frac{\mu_\ell}{C_a A_a \rho_\ell \lambda} = \frac{0.456 \times 10^{-3}}{7.03\times10^{-8} \times 1.77\times10^{-6} \times 890 \times 12\times10^5}$

$= 8.037\times10^{12} \times 4.27\times10^{-13} = 3.43\ N/m^2/W{\cdot}m$.

Permeability of the wick itself C_w:

Wick cross-sectional area: $A_w = L_e \times$ wick thickness $t_w = 0.17 \times 1\times10^{-4} = 17\times10^{-6}\ m^2$.

Wick crimping factor: $S = 1.05$

Wick porosity: $\varepsilon = 1 - \frac{\pi S d}{4D} = 1 - \frac{\pi \times 1.05 \times 5\times10^{-5}}{4 \times 85\times10^{-5}} = 0.515$

Wick permeability: $C_w = \frac{d^2\varepsilon^3}{122(1-\varepsilon)^2} = \frac{(5\times10^{-5})^2 \times (0.515)^3}{122(1 - 0.515)^2} = 1.19 \times 10^{-11}\ m^2$

Wick liquid frictional coefficient: $F_{\ell w} = \frac{\mu_\ell}{C_w A_w \rho_\ell \lambda} = \frac{0.456 \times 10^{-3}}{1.19\times10^{-11} \times 17\times10^{-6} \times 890 \times 12\times10^5}$

$= 4.94\times10^{15} \times 4.27\times10^{-13} = 2109\ N/m^2/W{\cdot}m$.

If the vapour frictional coefficient is neglected, then the

Capillary limit $Q_{c\,max} = \frac{P_{pm}}{F_{\ell a}(\frac{1}{2}L_e + L_a) + F_{\ell w}(\frac{1}{4}\ \text{circumference wick})}$

$Q_{c\,max}$ (vertical) $= \frac{2687}{3.43(0.085+0.02) + 2109(\frac{1}{4}\pi \times 5\times10^{-3})} = \frac{2687}{0.36 + 8.28} = 311\ W$.

$Q_{c\,max}$ (horizontal) $= \frac{984}{0.36 + 8.28} = 113.9\ W$.

Sonic limit $Q_{s\,max}$

$$Q_{s\,max} = M_v A_v \rho_v \lambda \sqrt{\gamma_v R_v T_v}$$

where M_v = max. Mach number (= 0.2)

γ_v = specific heat ratio = 1.33

R_v = vapour constant = $\frac{\text{universal gas constant}}{\text{molecular weight}} = \frac{8314}{34}$

T_v = vapour temperature in degrees Kelvin .

$$Q_{s\,max} = 0.2 \times 15.83\times10^{-6} \times 1.5 \times 12\times10^{5} \sqrt{1.33 \frac{8314}{34} 313} = 1818.22 \text{ W} .$$

Turbulence limit $Q_{t\,max}$

Vapour core cross-sectional area:

$$A_v = \pi(2.40^2 - 0.85^2) = 15.83 \times 10^{-6} \text{ m}^2$$

Vapour core hydraulic radius:

$$r_{hv} = (2.40 - 0.85) \times 10^{-3} = 1.55 \times 10^{-3} \text{ m}$$

$$Q_{t\,max} = \frac{Re_v A_v \mu_v \lambda}{2r_{hv}} \quad \text{(where } Re_v = \text{Reynold's number} = 2300\text{)}$$

$$= \frac{2300 \times 15.83\times10^{-6} \times 1\times10^{-5} \times 12\times10^{5}}{2 \times 1.55\times10^{-3}} = 140.94 \text{ W} .$$

Entrainment limit $Q_{e\,max}$

$$Q_{e\,max} = A_v \lambda \sqrt{\frac{\sigma\rho_v}{2r_{hs}}} = 15.83\times10^{-6} \times 12\times10^{5} \sqrt{\frac{21.85\times10^{-3} \times 1.5}{0.085\times10^{-3}}} = 373 \text{ W} .$$

Boiling limit $Q_{b\,max}$

First calculate the thermal conductivity K_e of the saturated wick, with:

thermal conductivity of methanol: $K_\ell \sim 0.20$ W/m K ,

" " " phosphor bronze: $K_w \sim 200$ W/m K ,

" " " pipe wall material: $K_p = 17.3$ W/m K .

$$K_e = \frac{K_\ell[(K_\ell+K_w) - (1-\varepsilon)(K_\ell-K_w)]}{[(K_\ell+K_w) + (1-\varepsilon)(K_\ell-K_w)]}$$

$$= \frac{0.2[(0.2+200) - (1-0.515)(0.2-200)]}{[(0.2+200) + (1-0.515)(0.2-200)]}$$

$$= 0.2 \frac{(200.2 + 96.9)}{(200.2 - 96.9)} = 0.575 \text{ W/m K}$$

$$Q_{b\,max} = \frac{2\pi L_e K_e T_v}{\gamma\rho_v \ln(r_i/r_n)} \left(\frac{2\sigma}{r_n} - P_c\right) ,$$

where L_e is the evaporator length and r_n the nucleation radius of vapour bubbles. For carefully outgassed liquid, $r_n \sim 0.25 \times 10^{-6}$ m.

If we want to know the maximum power per centimetre of heat-pipe length (L_e = 0.01 m), $P_c \ll 2\sigma/r_n$ can be neglected, and we get

$$Q_{b\,max} = \frac{2\pi \times 0.01 \times 0.575 \times 313}{12\times10^5 \times 1.5 \ln (5/4.8)} \left(\frac{2 \times 21.85\times10^{-3}}{2.5\times10^{-7}} \right) = 26.9 \text{ W/cm} .$$

Thus the maximum radial heat flow is 26.9 W per cm pipe length, as far as the boiling limit is concerned.

<u>Thermal resistance of the heat pipe at the evaporator</u>

The thermal resistance per centimetre of heat-pipe length (L_e = 0.01 m) is

for the pipe wall: $$R_{pe} = \frac{\ln r_0/r_i}{2\pi L_e K_p} = \frac{\ln 6/5}{2\pi \times 0.01 \times 17.3} = 0.1677 \text{ °C/W} ,$$

for the wick: $$R_{we} = \frac{\ln 5/4.8}{2\pi \times 0.01 \times 0.575} = 1.130 \text{ °C/W} .$$

Total thermal resistance per centimetre of heat-pipe length = 1.3 °C/W.

APPENDIX III

CALCULATIONS FOR AN ALTERNATIVE HEAT PIPE

Calculations for an aluminium heat pipe with a grooved wick showed that at the working temperature of interest (40 °C) the boiling limit of an ammonia-filled heat pipe is fairly low (of the order of 2.5 W per cm pipe length). Therefore a calculation with methanol as a working fluid is given. Since methanol is incompatible with aluminium, it would be preferable to use copper as the pipe material.

Characteristics

Material		:	Copper
Outer diameter	(OD)	:	8 mm = 8×10^{-3} m
Groove depth	(δ)	:	0.5 mm
" width	(w)	:	0.5 mm
Number of grooves	(n)	:	18
Vapour core diameter		:	6 mm = 6×10^{-3} m = $2 \times r_{h_v}$
Evaporator length	(L_e)	:	170 mm = 0.17 m
Adiabatic length	(L_a)	:	20 mm = 0.02 m
Total length ($L_t = L_e + L_a$)		:	0.19 m
Working fluid		:	Methanol
Working temperature	(T)	:	40 °C = 313 K

Properties of methanol at 313 K

Liquid density	(ρ_ℓ)	:	890 kg/m³
Liquid viscosity	(μ_ℓ)	:	0.456×10^{-3} kg/ms
Surface tension	(σ)	:	21.85×10^{-3} N/m
Latent heat of vaporization	(λ)	:	12×10^5 J/kg
Vapour density	(ρ_v)	:	1.5 kg/m³
Vapour viscosity	(μ_v)	:	1×10^{-5} kg/ms

Calculation of maximum pumping pressure P_{pm}

Capillary radius: $r_c = w = 0.5 \times 10^{-3}$ m

Maximum capillary pressure: $$P_{cm} = \frac{2\sigma}{r_c} = \frac{2 \times 21.85\times10^{-3}\ \text{N/m}}{0.5 \times 10^{-3}\ \text{m}} = 87.4\ \text{N/m}^2 .$$

Normal hydrostatic pressure: $\Delta P_\perp = 0$

Axial hydrostatic pressure:

vertical position $= \rho_\ell g L_t \sin\psi = 890 \times 9.81 \times 0.19 = 1659\ \mathrm{N/m^2}$

horizontal position $= 0$

Maximum effective pumping pressure: $P_{pm} = P_{cm} - \Delta P_\perp + \rho_\ell g L_t \sin\psi$

P_{pm} (vertical) $= 87.4 - 0 + 1659\ \mathrm{N/m^2} = 1746.4\ \mathrm{N/m^2}$

P_{pm} (horizontal) $= 87.4\ \mathrm{N/m^2}$.

Calculation of liquid frictional coefficient F_ℓ

Mean radius of liquid flow passage

$$r_m = \frac{(d_v + \delta)}{2} = \frac{(6\times10^{-3} + 5\times10^{-4})}{2} = 3.25 \times 10^{-3}\ \mathrm{m}\ .$$

Wick cross-sectional area: $A_w = 2\pi r_m \delta$

$= 2\pi \times 3.25\times10^{-3} \times 0.5\times10^{-3} = 10.2\times10^{-6}\ \mathrm{m^2}$.

Wick porosity: $\varepsilon = \dfrac{nw}{2\pi r_m} = \dfrac{9 \times 10^{-3}}{2\pi \times 3.25\times10^{-3}} = 0.441$.

Groove aspect ratio: $\alpha = w/\delta = 1$.

Groove hydraulic radius: $r_h = \dfrac{2w\delta}{w + 2\delta} = \dfrac{2 \times 0.5\times10^{-3} \times 0.5\times10^{-3}}{0.5\times10^{-3} + 1\times10^{-3}}$

$= \dfrac{0.5 \times 10^{-6}}{1.5 \times 10^{-3}} = 0.333 \times 10^{-3}\ \mathrm{m}$.

Drag coefficient: $f_e Re_\ell = 14.25$

Wick permeability: $C_w = \dfrac{2\varepsilon r_h^2}{f_e Re_\ell} = \dfrac{2 \times 0.441 \times (0.333\times10^{-3})^2}{14.25} = 6.88 \times 10^{-9}\ \mathrm{m^2}$.

Liquid frictional coefficient: $F_\ell = \dfrac{\mu_\ell}{C_w A_w \rho_\ell \lambda} = \dfrac{0.456 \times 10^{-3}}{6.88\times10^{-9} \times 10.2\times10^{-6} \times 890 \times 12\times10^5}$

$= 6.084\ \mathrm{N/m^2/W\cdot m}$.

If the vapour frictional coefficient is neglected, then

$$Q_{c\,max}\ (\text{vertical}) = \frac{P_{pm}}{F_\ell(\tfrac{1}{2}L_e + L_a)} = \frac{1746.4}{6.084(0.085 + 0.02)} = 2734\ \mathrm{W}$$

$$Q_{c\,max}\ (\text{horizontal}) = \frac{87.4}{6.085(0.085 + 0.02)} = 136.8\ \mathrm{W}$$

Sonic limit $Q_{s\,max}$

$$Q_{s\,max} = M_v A_v \rho_v \lambda \sqrt{\gamma_v R_v T_v}\ ,$$

where M_v = max. Mach number (= 0.2)

γ_v = specific heat ratio = 1.33

R_v = vapour constant = $\frac{\text{universal gas constant}}{\text{molecular weight}} = \frac{8314}{34}$

T_v = vapour temperature in degrees Kelvin .

$$Q_{s\,max} = 0.2 \times \pi \times 9\times10^{-6} \times 1.5 \times 12\times10^{5} \sqrt{1.33 \times \frac{8314}{34} \times 313} = 3247\ \text{W} .$$

<u>Turbulence limit $Q_{t\,max}$</u>

Vapour core cross-sectional area:

$$A_v = \pi(3\times10^{-3})^2 = \pi \times 9\times10^{-6}$$

$$Q_{t\,max} = \frac{Re_v A_v \mu_v \lambda}{2r_{hv}} \qquad \text{(where } Re_v = \text{Reynold's number} = 2300)$$

$$= \frac{2300 \times \pi \times 9\times10^{-6} \times 1\times10^{-5} \times 12\times10^{5}}{6\times10^{-3}} = 130\ \text{W} .$$

<u>Entrainment limit $Q_{e\,max}$</u>

$$Q_{e\,max} = A_v\lambda\sqrt{\frac{\sigma\rho_v}{2r_{hs}}} = \pi \times 9\times10^{-6} \times 12\times10^{5} \sqrt{\frac{21.85\times10^{-3} \times 1.5}{2 \times 0.5\times10^{-3}}} = 194\ \text{W} .$$

<u>Boiling limit $Q_{b\,max}$</u>

First calculate the thermal conductivity K_e of the saturated wick, with

thermal conductivity of methanol: $K_\ell = 0.20$ W/m K ,

" " " copper: $K_w = 400$ W/m K ,

w_f = groove fin thickness = $\frac{2 \times \pi \times 3.25\times10^{-3} - 18.05\times10^{-3}}{18} = 6.34 \times 10^{-4}$ m

w = " width = 0.5×10^{-3} m

δ = " depth = 0.5×10^{-3} m

$$K_e = \frac{(w_f K_w K_\ell \lambda) + wK_\ell(0.185\ w_f K_w + \delta K_\ell)}{(w+w_f) \times (0.185\ w_f K_w + \delta K_\ell)}$$

$$= \frac{(6.34\times10^{-4} \times 400 \times 0.2 \times 0.5\times10^{-3}) + 0.5\times10^{-3} \times 0.2(0.185 \times 6.34\times10^{-4} \times 400 + 0.5\times10^{-3} \times 0.2)}{(0.5\times10^{-3} + 6.34\times10^{-4}) \times (0.185 \times 6.34\times10^{-4} \times 400 + 0.5\times10^{-3} \times 0.2)}$$

$$= \frac{25.36\times10^{-6} + 1\times10^{-4}(0.047)}{1.134\times10^{-3} \times 0.047} = 0.564\ \text{W/m K}$$

$$Q_{b\,max} = \frac{2\pi L_e K_e T_v}{\lambda\rho_v \ln r_i/r_v}\left(\frac{2\sigma}{r_n} - P_c\right)$$

where L_e is the evaporator length and r_n the nucleation radius of vapour bubbles. For carefully outgassed liquid, $r_n \sim 0.25 \times 10^{-6}$ m.

If we want to know the maximum power per centimetre of heat-pipe length ($L_e = 0.01$ m), $P_c \ll 2\sigma/r_n$ can be neglected, and we get

$$Q_{b\,max} = \frac{2\pi \times 0.01 \times 0.564 \times 313 \times 2 \times 21.85\times10^{-3}}{12\times10^5 \times 1.5 \times \ln 7/6 \times 2.5\times10^{-7}} = 6.99 \text{ W/cm} .$$

With an evaporator length of 17 cm this means a total boiling limit of about 118 W.

Thermal resistance of the heat pipe at the evaporator

The thermal resistance per centimetre of heat-pipe length ($L_e = 0.01$ m) is

for the pipe wall: $$R_{pe} = \frac{\ln r_0/r_i}{2\pi L_e K_w} = \frac{\ln 8/7}{2\pi \times 0.01 \times 400} = 5.3 \times 10^{-3} \text{ C/W} ,$$

for the wick: $$R_{we} = \frac{\ln 7/6}{2\pi \times 0.01 \times 0.564} = 4.35 \text{ °C/W} .$$

Total thermal resistance per centimetre of heat-pipe length = 4.36 °C/W. For a total evaporator length of 17 cm,

$$R_{th} \sim 0.26 \text{ °C/W} .$$

APPENDIX IV

DISSIPATION AND TEMPERATURES IN THE PROTOTYPE PLUG-IN UNIT

DISSIPATION

Heat load on back-end heat pipe (power supplies)

1 stabilizer	-5.2 V	1.85 A	(26.5 V_{in})		39.4 W
1 "	-15 V	160 mA	(26.5 V_{in})		1.84 W
1 "	+15 V	160 mA	(24 V_{in})		1.44 W
1 "	-2 V	0.5 A	(6 V_{in})		2.0 W
1 series pass diode, voltage drop ∿ 1 V at 4.5 A					4.5 W
				Total dissipation	∿ 49.2 W

The dissipation of the -5.2 V stabilizers was chosen to be equivalent to a stabilizer of -5.2 V, 10-12 A, powered from a non-stabilized voltage of 8-8.5 V.

Heat load on centre heat pipe

10 integrated circuits MC10136P of ∿ 650 mW each		= 6.5 W
8 resistors 750 Ω at 15 V		= 2.4 W
8 transistors 2N3904 at 15 V 20 mA		= 2.4 W
4 resistance networks 7 × 150 Ω at -5 V		= 4.67 W
4 " " 7 × 150 Ω at +6 V		= 6.72 W
	Total dissipation	∿ 22.7 W

Heat load on front-end heat pipe

6 integrated circuits MC10136P of ∿ 650 mW each		= 3.9 W
8 resistance networks 7 × 150 Ω at -5 V		= 9.33 W
6 " " 9 × 150 Ω at -5 V		= 9 W
8 " " 7 × 150 Ω at +6 V		= 13.44 W
6 " " 9 × 150 Ω at +6 V		= 12.96 W
	Total dissipation	∿ 48.63 W

The total dissipation of the entire unit was about 120 W.

TEMPERATURES

The surface temperature of the cooling strip was 27.6 °C. The working temperature of the pipes and the calculated thermal resistances pipe → sink were:

	T_P	$R_{th\ p \to s}$
Back-end pipe	45.4 °C	$\frac{45.4\ °C - 27.6\ °C}{49.2\ W} = 0.36\ °C/W$
Centre pipe	36.8 °C	$\frac{36.8\ °C - 26.7\ °C}{22.7\ W} = 0.41\ °C/W$
Front-end pipe	44.8 °C	$\frac{44.8\ °C - 27.6\ °C}{48.6\ W} = 0.35\ °C/W$

Working and junction temperatures of the voltage stabilizers

i) -5.2 V, 1.85 A. Of this 1.85 A, about 0.3 A passed through the voltage stabilizer MC7905.2; the rest (1.55 A) was taken by a power transistor MJE3055. As the input voltage was 26.5 V, the dissipations were respectively:

MC7905.2: (26.5 V - 5.2 V) × 0.3 A = 6.4 W ,

MJE3055: (26.5 V - 5.2 V) × 1.55 A = 33.0 W .

The temperature in the centre of the heat link on to which the components are mounted was 66.8 °C.

The thermal resistances of the MC7905.2 were

Junction → case $R_{th\ j \to c} = 5\ °C/W$

Case → sink $R_{th\ c \to s} = 0.6\ °C/W$ which gives

Junction → sink $R_{th\ j \to s} = 5.6\ °C/W$.

The resulting junction temperature was

T_j = 66.8 °C + (6.4 W × 5.6 °C/W) = about 103 °C (maximum rating = 150 °C).

For the power transistor

$R_{th\ j \to c} = 1.39\ °C/W$

$R_{th\ c \to s} = 0.61\ °C/W$

$R_{th\ j \to s} = 2.0\ °C/W$,

T_j = 66.8 °C + (33 W × 2 °C/W) = about 133 °C (maximum rating = 150 °C).

ii) ±15 V, 160 mA; V_{in} = +24 V and -26.5 V, which gave a dissipation of about 1.6 W each. Both were mounted on the same heat link, which had a temperature of 51 °C. At $R_{th\ j \to s}$ of 5.6 °C/W, this gave T_j = 51 °C + (1.6 W × 5.6 °C/W) ∿ 60 °C.

iii) Series pass diode.

Voltage drop ∿ 1 V, current 4.5 A :: dissipation ∿ 4.5 W.

$R_{th\ j \to c} = 6.5\ °C/W$

$R_{th\ c \to s}$ (insulated mounting) = 2.7 °C/W

$R_{th\ j \to s} = 9.2\ °C/W$.

Link temperature = 55 °C; T_j = 55 °C + (4.5 W × 9.2 °C/W) ∿ 97 °C.

iv) -2 V, 0.5 A stabilizer, mounted on the same heat link as the series pass diode.

V_{in} = -6 V; dissipation (6 V - 2 V) × 0.5 A = 2 W.

$R_{th\ j \to s}$ = 5.6 °C/W :: T_j = 55 °C + (2 W × 5.6 °C/W) ∿ 67 °C .

<u>Junction temperatures of the integrated circuits</u>

The MC10136P was chosen for the following reasons: standard DIL package, (relatively) high dissipation per chip (∿ 650 mW loaded), easy availability, and a plastic package with fairly high thermal resistance (70 °C/W worst case) so as not to make things too easy.

Mounting was done by using heat paste between the IC and the heat link:

$\Delta T_{j \to s}$ = 0.65 W × 70 °C/W ∿ 40 °C .

The temperatures of the heat links varied from 39 °C to 50.6 °C, so the junction temperatures were, at maximum, between 85 °C and 97 °C.

APPENDIX V

SHEET HEAT CONDUCTION

Instead of heat links, a copper or aluminium sheet can be used to conduct the heat to the heat pipes. For a calculation of the maximum power we started with the following assumptions:

1) The heat production is evenly spread over the entire surface of the sheet at a rate of P_m W/cm².
2) The heat pipes have a spacing of 10 cm, so the maximum distance D_m from any point to the nearest heat pipe is 5 cm. The average distance D_a is consequently 2.5 cm.
3) The sheet thickness d = 0.5 mm = 0.05 cm.
4) The temperature difference at the interface sheet → heat pipe is 5 °C.
5) The maximum sheet temperature is 15 °C above the heat-pipe temperature, so the temperature gradient over 5 cm = ΔT_m = 10 °C max.
6) The inevitable holes in the sheet have no significant influence on the temperature behaviour.

Calculation

The thermal conductivity of copper: K = 4 W/cm °C

" " " " aluminium: K = 2.38 W/cm °C .

Thermal conductivity of a strip, width B, thickness d = KBd W·cm/°C.

Heat production in a strip of BD_m cm² = P = BD_mP_m W

$$\Delta T_m = \frac{PD_a}{KBd}\ °C = \frac{BD_mP_mD_a}{KBd}\ °C$$

$$P_m = \frac{\Delta T_m Kd}{D_mD_a}$$

For copper:

$$P_m = \frac{10\ °C \times 4\ W/cm\ °C \times 0.05\ cm}{5\ cm \times 2.5\ cm} = 0.16\ W/cm^2\ .$$

For aluminium:

$$P_m = \frac{10\ °C \times 2.38\ W/cm\ °C \times 0.05\ cm}{5\ cm \times 2.5\ cm} = 0.095\ W/cm^2\ .$$

For a board the size of a CAMAC unit this would mean a maximum dissipation of

$$18\ \text{cm} \times 28\ \text{cm} \times 0.16\ \text{W/cm}^2 \simeq 80\ \text{W in the case of a copper sheet}$$

and

$$18\ \text{cm} \times 28\ \text{cm} \times 0.095\ \text{W/cm}^2 \simeq 48\ \text{W for an aluminium sheet}\ .$$