SYSTEM ASPECTS OF THERMOELECTRIC COOLERS FOR HAND HELD THERMAL VIEWERS

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SUMMARY

Low power TE cooling systems are achievable for portable Hand Held Thermal Viewing Systems. However, it is important to minimize the heat load on the top surface of the TE cooler since the heat load relates directly to the input power of the cooler. Further, the TE cooler design must be integrated with the systems design of the thermal path between the base of the sealed package and the heat sink, the design of the heat sink, and the selection of an optimum power control system. With these factors in mind during the design phase, low powered TE coolers can be the most effective solution to cooling detectors in portable Hand Held Thermal Viewers.

INTRODUCTION

Hand Held Viewing Systems using infrared detectors are now being used in a wide variety of Military and Commercial applications. Where night time passive viewing or day time detection of thermal hot spots are required, infrared viewing systems are now available that will meet these requirements. Thermoelectric coolers (TE) are used to cool the infrared detector in one class of portable viewing systems that are battery operated. To minimize the number of batteries required over an 8-10 hour operation, the total power required by the viewer must be held to a minimum. Power is required for both the system electronics and the thermoelectric cooling system. Minimizing the power to the thermoelectric cooling system will contribute to the total system power reduction. The purpose of this paper is to emphasize those factors that are essential design considerations in minimizing power required by the thermoelectric cooling system in portable viewing systems.

HAND HELD VIEWING SYSTEM

The typical components in a thermoelectrically cooled Hand Held Viewing System are shown in Figure 1. The object being viewed is projected through an optical path and focused onto an infrared detector. The infrared detector is mounted onto a thermoelectric cooler and this assembly is mounted in a sealed package. The sealed package contains a window through which the infrared detector can view the object. The output from the infrared detector goes through a signal processor and a thermal image is displayed on a cathode ray tube (CRT). A battery provides power to the overall system. The sealed package containing the infrared detector and thermoelectric cooler is mounted to a heat sink.

SEALED TE COOLER PACKAGE

The sealed package contains the infrared detector and thermoelectric cooler. The total power required for the thermoelectric cooler is directly related to the total heat load on the top surface of the thermoelectric cooler called the cold plate. Therefore, it is important during the design phase to ensure the total heat load is held to a minimum. The total heat load consists of the sum of those heat loads that effect the top surface of the thermoelectric cooler. They include the bottom surface of the detector substrate if it over-hangs the area of the thermoelectric cold plate. Individual heat loads on the TE cooler consist of the following:

1. Radiation load,
2. Convective load if in an atmosphere such as nitrogen, zenon, or other dry gas,
3. Conductive load from lead wires to the infrared detector, or thermistor wires used to monitor or control the temperature, and
4. Electrical bias power dissipated by the detector.

The radiation load and convection load relate to the size of the cold plate of the thermoelectric cooler. As the size of the cold plate surface decreases, the heat load decreases, and in turn the input power to the thermoelectric cooler decreases. The importance of reducing the size of the detector substrate cannot be over-emphasized because of its significant impact on the size of the cold plate which relates to the input power of the thermoelectric cooler. As an example, when the cold plate of the thermoelectric cooler is at 195°K and the hot surface of the thermoelectric cooler is at 300°K (27°C) the input power to the thermoelectric cooler is decreased by about .05 watts for every milliwatt of heat load reduction.

The heat dissipated at the hot surface of the thermoelectric cooler into the base of the sealed package is the sum of the heat load on the cold surfaces plus the electrical input power to the thermoelectric cooler. Typically the heat load on the cold surfaces are small and the heat to be dissipated at the hot surface is essentially the input power to the thermoelectric cooler. Although the heat loads on the cold surfaces are small they are the driving force that determines the amount of heat to be dissipated at the hot surface.

One method of decreasing the radiation load on the top surface as well as some of the other cold surfaces of the cooler is to use a radiation shield. Depending on the design one or more radiation shields could be used. The radiation shield or shields could be mounted to one or more of the intermediate stages of the thermoelectric cooler. An example of a single shield would be to mount it on the second stage of a 4-stage cooler. The shield would reduce the heat load on the top 2 stages. The amount of heat load added to the second stage due to the radiation shield would be accounted for in the design of the TE cooler and is small in comparison to the heat pumping capacity of the second stage.

HEAT SINK PERFORMANCE

The heat dissipated by the thermoelectric cooler is absorbed by the metallic base of the sealed package. For minimum input power to the cooler, the temperature rise of the base must be held to a minimum. As the base temperature rises, additional input power is required by the cooler to achieve the same cold plate temperature. As more input power is required the temperature of the base rises. If not properly accounted for in the design, this condition can result in thermal runaway when in actual use.

The ideal situation would occur when the temperature of the base of the sealed package is the same as the ambient temperature.

Installation of a fan to move air over the heat sink would minimize heat sink rise. However a fan adds additional weight and power, and thus is not practical for a light-weight viewer. With the constraint of a totally passive or natural convection heat sink the thermal path to ambient must be carefully analyzed to achieve a minimum temperature rise between the base of the sealed package and the ambient temperature. The distance the heat must travel, the type of material through which the heat must travel, the mechanical interface losses, the warm surfaces exposed to the ambient, and the orientation of these warm surfaces must be considered. Although the heat sink is the primary surface for dissipating heat, other surfaces in the thermal path exposed to the ambient also contribute. The thermal path is shown in Figure 2.

It should be remembered that the thermoelectric cooler is not the only source of heat generated internal in the viewer. Heat is also generated by the electronics in the signal processor, etc. The heat sink must accommodate this heat in addition to the thermoelectric cooler power. However, for this analysis, this source of heat was not included in the calculation. Final designs must take this added heat into account.

The rise in heat sink temperature above ambient is roughly linear with heat dissipated. The performance of a heat sink is determined from this linear ratio referred to as its thermal resistance in °C/watt. A heat sink with low thermal resistance will have a low temperature rise for a given heat dissipation. Therefore it is desirable to obtain a heat sink with a minimum thermal resistance.

Figure 3 shows the amount of power that can be dissipated for heat sinks with various thermal resistances at an ambient temperature of 40°C. The point of origin of the various thermal resistance is the ambient temperature. The thermal resistance for heat sinks in the range of interest vary from 1 to 6°C/watt. For a point of reference, a heat sink with a thermal resistance of 6°C/watt is a flat plate about 4.0 inch x 4.0 inches (10cm x 10cm) sq. A heat sink with a thermal resistance of 1°C/watt is a 4.0 inch x 4.0 inch (10cm x 10cm) flat plate with fin spacing and size optimized for nature convection. A typical 4.0 inch x 4.0 inch (10cm x 10cm) finned heat sink for the viewer application would have a thermal resistance of about 1.7°C/watt. For example, a heat sink with a thermal resistance of 1.7°C/watt dissipating 4 watts at an ambient temperature of 40°C would rise to 46.8°C as shown in Figure 3.

THERMOELECTRIC COOLER PERFORMANCE

Figure 4 shows the relationship between the power required to achieve a desired detector temperature as a function of the heat sink temperature. The curves represent loci of optimally designed 6-stage thermoelectric coolers as opposed to the performance of a given cooler design. It is clear that the optimum cooler power increases as the temperature of the detector gets colder or, as the ambient temperature gets warmer.

The power requirements are based on the assumption that the total heat load of 25mw is on the cold plate of the TE cooler. Of this total load, 8mw is considered to be a radiant heat load. The radiant load for this example is based on a detector substrate that is .15 inch x .15 inch (3.8mm x 3.8mm)sq. Conservatively the emissivity of the substrate is assumed to be 1.0. The atmosphere inside the sealed package is considered to be a vacuum. The balance of the heat load, 17mw, consists of the detector bias power, the heat conducted through the detector wires, and the heat conducted through the thermistor wires used to control the temperature of the detector if so desired.

HEAT SINK/TE COOLER INTEGRATION

The system design begins when the heat sink resistances shown in Figure 3 is superimposed on the TE cooler design shown in Figure 4 which results in Figure 5. An example of the power required for a specific set of conditions would be as follows: With an ambient temperature of 40°C, a resistance of 1.7°C/watt, and a detector temperature of 195°K, the input power would be 1.9 watts when the total heat load is 25mw.

The power required for ambient temperatures of 25°C, 40°C, and 55°C are shown in Figure 6. Each of the 3 lines represents the same heat sink with a thermal heat sink resistance of 1.7°C/watt. Other ambient temperature thermal resistance lines can be drawn parallel to those already shown beginning at the desired ambient temperature.

PERFORMANCE ANALYSIS

The optimum TE cooler design for any given set of conditions may lead to a cooler that is not practical to fabricate. For example, the optimum design may specify 6-1/2 pellets in a stage. However, TE coolers cannot be fabricated with 1/2 pellets. Therefore the design must be modified to accept either 6 or 8 pellets. Other geometric constraints are also applied and a final design chosen that comes nearest to the locus of the optimum cooler curve.

Once the TE cooler is designed, its performance can be determined over a wide ambient temperature range. An example of the performance of a practical cooler over a wide temperature range is shown in Figure 7. Cooler No. 40 was designed at an ambient temperature of 40°C to pump a total heat load of 25mw at a detector temperature of 185°K. The locus of optimum designs is shown by the dotted line. Note that at the design point the optimum cooler and the practical cooler performance do not overlap for the reason mentioned above, but nevertheless have approximately the same performance. It is observed in Figure 7 that there is not a significant difference in power between a cooler designed for a specific ambient temperature and operated in an ambient temperature below the design point as compared to the locus of all optimum coolers designed at the lower ambient temperatures.

But consider what happens to Cooler No. 40 as the ambient temperature goes above 40°C. As the ambient temperature rises the amount of power required to maintain 185°K rises rapidly and thermal runaway would occur at about 42°C. With only a 2°C rise before thermal runaway, Cooler No. 40 would be a marginal design to meet the given cooling requirements.

One method to prevent marginal design is to design the cooler for a higher ambient temperature operation such as 55°C and then operate the cooler at any lower temperature than the maximum required ambient temperature of 40°C with only minor increase in input power. An example of a cooler designed at 55°C ambient is shown in Figure 8. Cooler No. 55 could rise as much as 15°C above the maximum required temperature of 40°C and the design would not be considered marginal. The cooler in effect will operate significantly above 40°C before the rapid rise in input power leads to thermal runaway. It just so happens that the geometrical constraints for this case were not severe resulting in very near optimum performance at the design point.

POWER CONTROL SYSTEMS

The performance of a TE cooler varies as the absolute temperature of the heat sink changes. As the temperature of the heat sink decreases the power required to maintain the same cold side temperature decreases. The decreasing power with decreasing heat sink temperature shown in Figures 7 & 8 assumes the power to the cooler has been minimized by adjusting voltage or current to achieve the desired temperature for each heat sink temperature. In effect the curve is based on true proportional control.

In practice the total power for the cooling system is the sum of the power to the cooler and the Power Control System. The power curves in this paper are related to the power to the TE cooler and do not include the power losses that would be associated with the Power Control Systems that may be considered for portable viewing systems.

1. Constant Voltage

Hand Held Viewing Systems are typically battery operated and the input voltage to the cooler is relatively fixed regardless of changes in the heat sink temperature. With this constant voltage applied directly to the cooler the performance of Cooler No. 55 will be shown in Figure 9. The impedance of all TE coolers decreases with temperature. Therefore, as the ambient temperature decreases the current and thus power to the cooler increases. The temperature of the detector will decrease initially, but eventually begin to increase due to Joule heat domination.

2. Constant Current

The Power Control System could contain a current regulator inserted between the battery and TE cooler. This technique would allow a constant current to be applied to the TE cooler as opposed to a constant voltage. With a constant current the "peak" current would not be exceeded and a decrease in heat sink temperature would always be accompanied by a decrease in the detector temperature. The power consumed when operating at a constant current as a function of heat sink temperature is shown in Figure 10.

3. Linear-Proportional

The Power Control System for linear-proportional control requires a linear control circuit to be inserted between TE cooler and battery. The linear-proportional system is achieved with series control transistors and the power to the TE cooler is smooth direct current. Although the power to the cooler is proportional, the power to the cooling system is still greater than the proportional power curve on Figure 10. The proportional power curve on Figure 10 assumes no losses in the Power Control System whereas in reality there are losses in the linear-proportional control elements. Linear-proportional control systems add a feature that may be desirable for specific applications. This type of control system incorporates a temperature control network which includes a sensor mounted to the cold surface of the TE cooler. The sensor provides a signal to the control circuit for constant control of the set point temperature.

Thermostatic

A thermostatic control network simply switches the voltage back and forth from full power to power off. When power to a TE cooler is switched off the heat pumping capacity of the cooler does not fall to zero. The cooler actually reverses the direction in which the heat is pumped. In effect there is a negative heat pumping or heat leakage when power is turned off.

Pulse Width Modulation (PWM)

A pulse width modulation control system is an adaptation of the thermostatic method through more sophisticated circuitry and with the added capability of temperature control. Figure 11 shows the power required by the PWM control system when in the power "on" and power "off" mode. The instantaneous reversal in heat pumping is a result of the instantaneous nature of Peltier Cooling. That is, the distance between the two curves represents the Peltier heat pumping curve. The consequence of applying a PWM control system are depicted by power vs. time and heat pumping vs. time in Figure 11. The net area above the curve represents the total heat to be pumped at 25mw. The important feature of this type system is the fact that the duty cycle is still quite large even at the "best" ambient conditions. An unfiltered PWM control system is not effective as true proportional control due to the inefficiency of low net heat pumping at low temperatures under full applied voltage.

Switching Proportional

The switching proportional power control system is a essentially the same as a high frequency PWM method but with an output filter. The filter transforms the signal to the TE cooler into a steady, minimum level DC voltage to meet the control temperature with minimum power. The main advantage of this method, other than precision temperature control is the potential for minimum system power.

In general it is difficult to define the overall minimum power system which includes the TE cooler power and the Power Control System. The general characteristics of the various Power Control Systems are shown in Table 1. Each of the Power Control Systems are relatively small and light weight. Although the Thermostatic, Pulse Width Modulation, and Switching Proportional offer temperature control they do generate Electro Magnetic Interference which must be accounted for in the overall system design. A detailed analysis of those Power Control Systems that show most promise for the particular system requirements is necessary to determine which system provides minimum power.

CONCLUSIONS

Low powered thermoelectric coolers are a practical method of cooling infrared detectors in Hand Held Viewing Systems. The key to achieving low power is consideration of the overall system aspects in the initial design phase.

FIGURE 1:

TE COOLED VIEWING SYSTEM

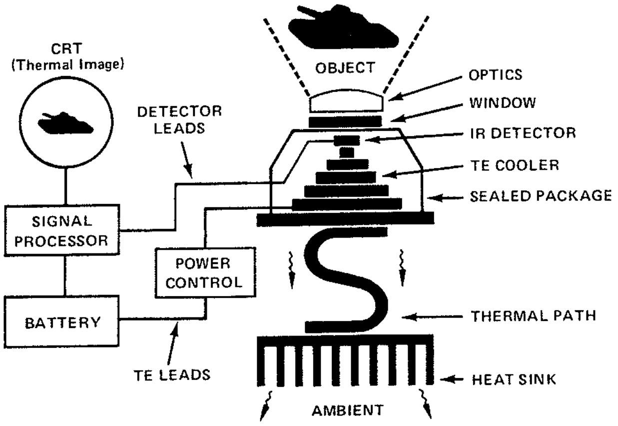

FIGURE 2:

THERMAL PATH

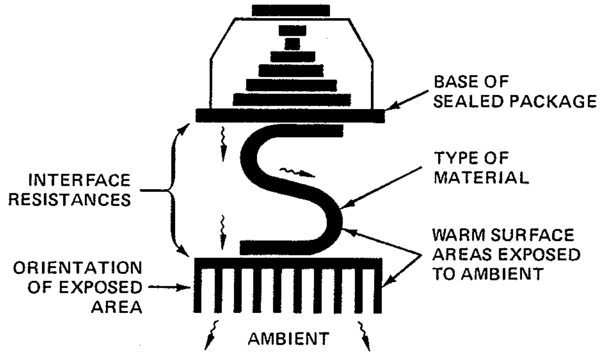

FIGURE 3:

HEAT SINK PERFORMANCE

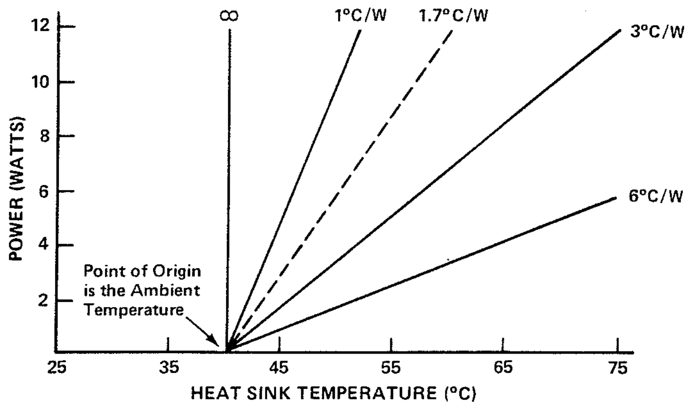

FIGURE 4:

VIEWER COOLER DESIGNS, 6-STAGES

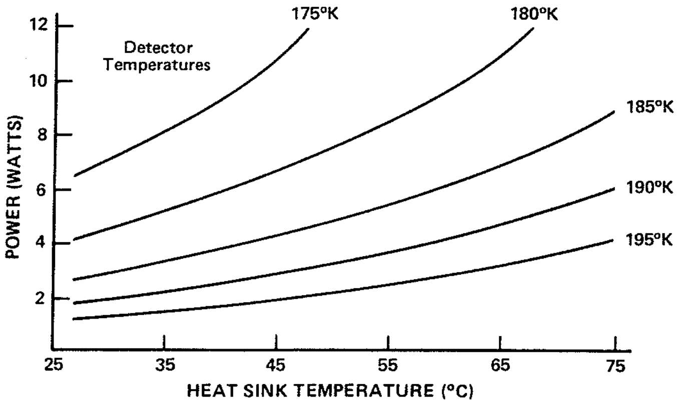

FIGURE 5:

TE COOLER/HEAT SINK SYSTEM

25 mW Total Heat Load

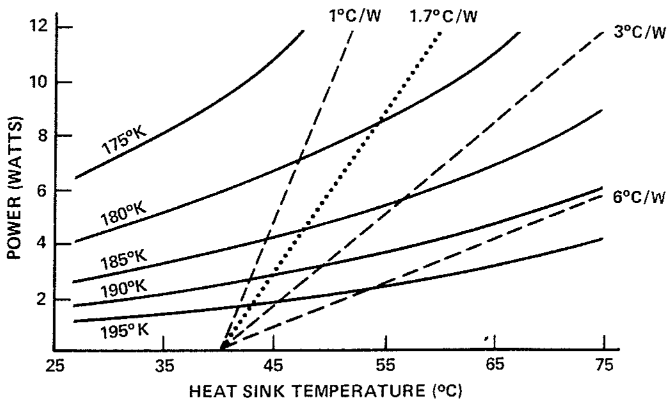

FIGURE 6:

EFFECT OF AMBIENT

25 mW Total Heat Load

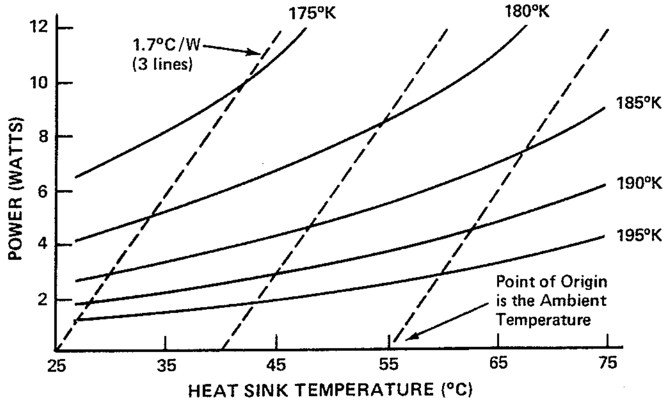

FIGURE 7:

COOLER No. 40 PERFORMANCE vs OPTIMUM DESIGN LOCUS

25 mW Total Heat Load, 185°K Cold Spot

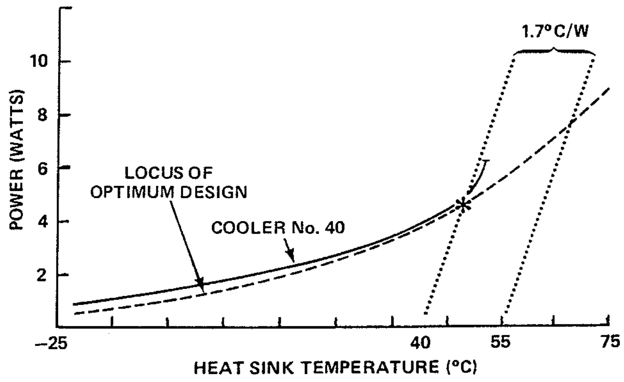

FIGURE 8:

COOLER No. 55 PERFORMANCE vs OPTIMUM DESIGN LOCUS

25 mW Total Heat Load, 185°K Cold Spot

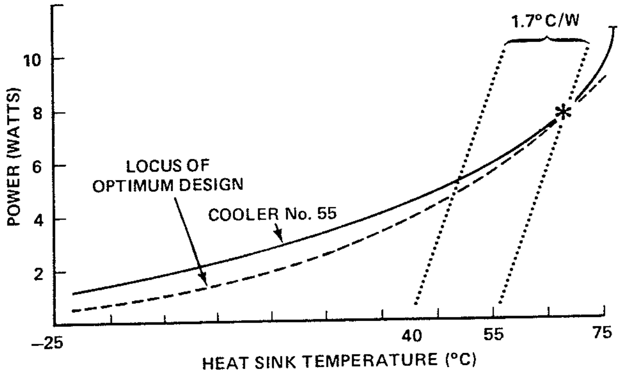

FIGURE 9:

COOLER No. 55 PERFORMANCE

Constant 5.5 Volts, 25 mW Total Heat Load

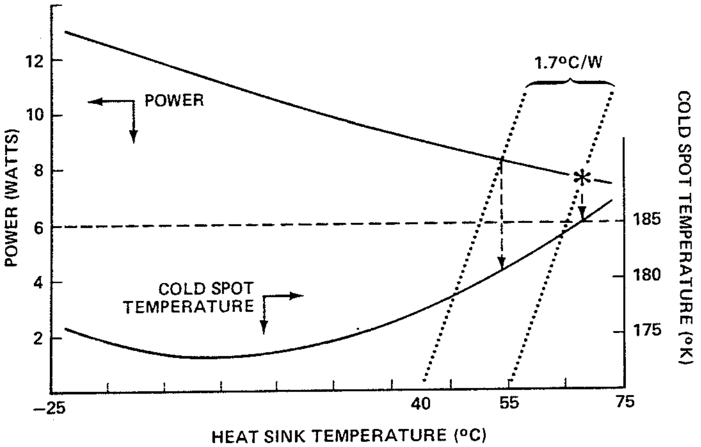

FIGURE 10:

POWER CONSUMPTION COMPARISON, COOLER No. 55

25 mW Total Heat Load, 185°K Cold Spot

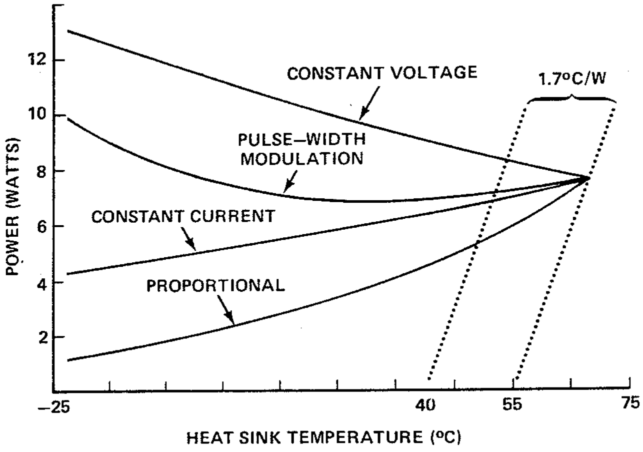

FIGURE 11:

PULSE-WIDTH MODULATION TEMPERATURE CONTROL, COOLER No. 55

185°K Cold Spot

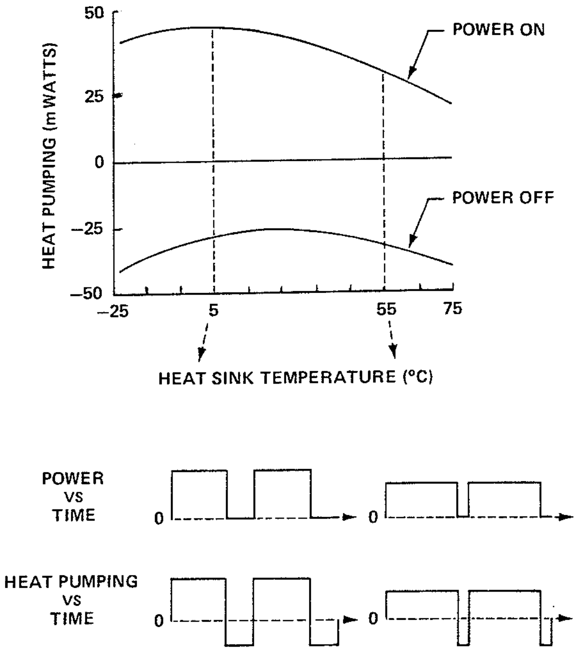

TABLE 1:

POWER CONTROL SYSTEMS

(Ambient Temperature Range of +50°C to −25°C)

	Power To Cooler	Power Control Eff.	Complexity	EMI	Cost	Temperature Control
Constant Voltage	H	None	None	None	None	None
Constant Current	M	L	L	None	M	None
Linear Proportional	L	L/M	M	None	M	H
Thermo-static	M	H	L	M	L	M
Pulse Width Modulation	M	H	M	H	M	H
Switching Proportional	L	M/H	H	H	H	H

L – Low M – Medium H – High