Liquid Cooling of Power Thyristors

FOREST B. GOLDEN, MEMBER, IEEE

***Abstract*—A method of determining a given heat exchanger's thermal properties from the characteristics of the cooling medium is discussed. A test method is presented with an example which helps to clarify the results. It is concluded from running tests using three to four different liquids that heat exchanger thermal performance data can be derived for liquids having a wide range of liquid properties.**

Introduction

INCREASINGLY, designers are turning to liquid cooling to solve the heat-transfer problems associated with high-power thyristors. Frequently, the coolant selected is one other than water due to a number of reasons, such as the freezing limitation of water. When a liquid other than water is chosen, the designer is faced with the problem of estimating a given heat-sink's performance. This problem arises from the current manufacturer's practice of rating commercially available heat-sink performance only in terms of water as the liquid medium. Performance differs markedly when liquids other than water are used.

Previous writers have given formulas for heat-transfer properties of heat sinks used in high-power thyristor equipment [1]. However, the formulas suffer from deficiencies when viewed from the perspective of the equipment designer. First, they are only useful for heat sinks employing a particular liquid flow passageway such as a tube. Second, the formulas yield a heat-transfer coefficient in terms of Btu/(ft^2·h·°F) for the liquid interface only. The equipment designer needs a relationship which will yield thermal resistance R_θ in °C/W for liquid to sink. The third, and most fundamental area, is one of perspective. One feels existing formulas are given to aid the heat-sink designer. A vacuum exists concerning the establishment of a mathematical or graphical relationship for a given completed heat-sink design that would be useful to the heat-sink user—the solid-state equipment designer. It is the equipment designer who must apply the sink to a specific requirement with regard to thermal resistance, liquid selection, liquid flow rate, pressure drop, etc. The purpose of this paper then is to report on a method developed by the author that overcomes these three historical limitations.

Method

To illustrate both the problem with which one is faced, and the solution chosen by the author, a particular heat sink has been chosen as being representative of the kind now found in the industry. Cross sections of the liquid flow passages are shown in Fig. 1. The passageways are indicated by the white areas.

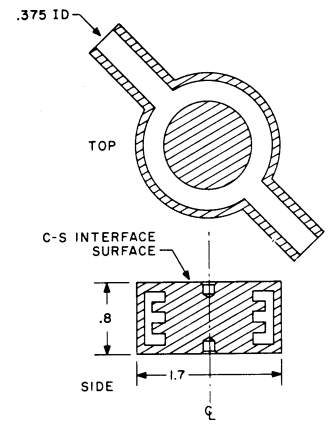


Fig. 1. Cross sections showing liquid passages of G6 heat sink.

The first decision to be made was one of basic approach to the problem, i.e., analytical versus empirical. The latter was taken for the following reasons. The literature [2], [3] abounds with formulas for calculating liquid-film, heat-transfer coefficients as a function of Reynolds and Prandtl numbers, the thermal conductivity of the liquid, and the physical dimensions of the particular geometry, i.e., tube, cylinder, wall, etc. However, with the geometry of this exchanger (and others) having odd-shaped passageways coupled with nonuniform entrance and exit flow passages, formulas, while adequate, require information not generally available and not easily obtained. Specifically, the liquid velocity distribution is unknown due to uneven path lengths for liquid flowing in the center of the sink. This is shown in Fig. 1 by the side cross-sectional view and for liquid flowing either near the top or bottom of the sink. Couple that problem with the unknown but very critical question of type of liquid flow that is occurring, i.e., laminar or turbulent, and one can see how problems multiply. For liquid flowing in a smooth pipe, the break point between laminar and turbulent flow is well known. It lies between a Reynolds number of 2100 and 3000. It varies by an order of magnitude for other geometries and is not readily available. It is obtained by experimental means for sinks having irregular geometries such as that shown by Fig. 1. Since turbulence enhances the heat-transfer coefficient by a factor of 2/1, it is extremely important. Still another important consideration is the component of thermal resistance due to the copper as opposed to the liquid film. The effective copper path varies with the film resistance, for, as the liquid-film thermal resistance increases, the copper sink surface in contact with the liquid will approach a more uniform temperature distribution. It follows that the effective thermal path through the

Paper TOD-71-178, approved by the Power Semiconductor Committee of the IEEE Industry Applications Society for presentation at the 1970 IEEE Industry and General Applications Group Annual Meeting, Chicago, Ill., October 5–8, 1970. Manuscript released for publication December 30, 1971.

The author is with the Semiconductor Products Department, General Electric Company, Auburn, N.Y. 13021.

LIBERTY EXHIBIT 1030, Page 1

Authorized licensed use limited to: Robert Durham. Downloaded on January 18,2023 at 02:24:56 UTC from IEEE Xplore. Restrictions apply.

copper to the case–sink interface surface will change. This complicates the calculation. Due to the preceding reasons, plus the need for tests to establish the validity of formulas should a mathematical technique be used, it was decided to use correlated test data. By this method, a mathematical relationship was established; the reliance on an analysis based on sink geometry was precluded.

If correlated data are used, the test points must be chosen with some care in order to satisfy the derived relationship for the liquid characteristics in the range of application interest. From dimensional analysis, Kraus [4] shows that

$$\frac{hD}{k} = M\left(\frac{\rho VD}{\mu}\right)^{a}\left(\frac{C_p\mu}{k}\right)^{1-c} \tag{1}$$

This equation, known as the Nusselt equation, relates the Nusselt number (hD/k) to the product of the Reynolds numbers $DV(\rho/\mu)$ and the Prandtl number $(C_p\mu/k)$. The preceding variables are defined in Appendix I. The Nusselt number is a liquid-film heat-transfer grouping, while the Reynolds and Prandtl numbers are, respectively, fluid flow and fluid thermal property groupings.

It is apparent from Fig. 2 that the total thermal drop R_θ, for the heat sink is composed of two variables consisting of the relatively fixed copper path and the highly variable film interface. The copper component can be approximated as a fixed value, while the film interface value is inversely proportional to h. From the Nusselt equation,

$$\frac{1}{h} = \frac{1}{M/Dk\,(\mathrm{Rey})^{a}P_R{}^{1-c}}. \tag{2}$$

Letting $D/M = B$, such that B incorporates the conversion constant needed to convert $1/h$ to °C/W, and dropping from both sides of the equation the units of feet, then (1) yields

$$R_\theta{}' = \frac{B}{k(\mathrm{Rey})^{a}P_R{}^{1-c}} \tag{3}$$

and sink thermal resistance can be expressed as

$$R_\theta = R_{\theta\mathrm{Cu}} + R_\theta{}'. \tag{4}$$

Focusing attention on the variables of the $R_\theta{}'$ term, it is apparent that the thermal resistance is inversely proportional to the product of three factors: 1) the liquid thermal conductivity k; 2) the Reynolds number, which in turn is proportional to the ratio of ρ/μ times VD, which are, respectively, a liquid property and a fluid velocity factor proportional to gal/min; and 3) the Prandtl number, which consists entirely of liquid properties. To be sure of obtaining the proper correlation, the pertinent liquid variables were reconciled by testing the heat-sink performance using four liquids having a wide spread in characteristics. The four liquids chosen are shown in Fig. 3. In order to further expand the range of usefulness, the liquid-film temperature was controlled and ranged from 70°F to above 160°F.

Table I illustrates the range of the two other pertinent fluid variables used for test purposes. The cube root of the Prandtl number is shown on the assumption that (3) is a linear straight-line logarithmic function and therefore can be defined by fixing any one of the three variables and deriving the remaining two. Customarily, the Prandtl number exponent is set equal to 1/3.

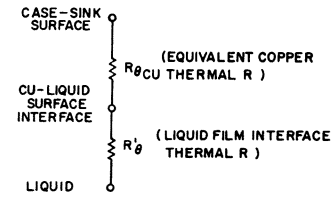


Fig. 2. Equivalent thermal resistance path for liquid-cooled sink

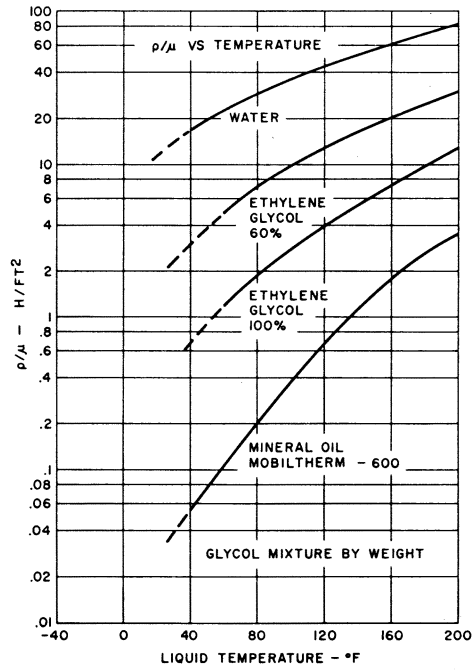


Fig. 3. Liquid properties versus temperature.

TABLE I
PROPERTIES OF TEST FLUIDS

Liquid (Temperature Range is 50–200°F)	k (Btu/ft·h·°F)	$(P_r)^{1/3}$ (Dimensionless)
Water	0.347–0.392	2.1–1.24
Ethylene–glycol, water 60 wt %	0.222–0.215	3.9–2.05
Ethylene–glycol, 100 wt %	0.178–0.132	6.5–3.05
Mineral oil Mobiltherm 600®	0.070–0.065	16.7–4.65

® Registered trademark of Mobil Research & Development Corporation, New York, N.Y.

DESCRIPTION OF TEST METHOD

A. Power Measurement

Thermal power was supplied by an electric hot plate operating at approximately 375°F. Heat was transferred from this hot plate through a calibrated copper thermal shunt. The thermal shunt used six thermocouples in three planes to ensure accurate measurement of temperature

LIBERTY EXHIBIT 1030, Page 2
Authorized licensed use limited to: Robert Durham. Downloaded on January 18,2023 at 02:24:56 UTC from IEEE Xplore. Restrictions apply.

drop. The shunt was calibrated by measuring the temperature rise of water through the heat exchanger for very low water flow rates such that the temperature differential of inlet–outlet water was substantial. In order to calibrate the water flow rate the bucket method of measurement was used. The calibrated value differed from the calculated by 10 percent. The average of the calculated and calibrated values was 9.5 W/°C, and this value was used. This value was high enough to yield large differences in shunt temperature readings, typically in the order of 25°C. The shunt contacted the sink with a 1-in diameter surface. Sink temperatures were monitored by the use of two thermocouple probes: one placed near the center of the interface surface, and the other close to the outer edge of the interface. This use of thermocouples turned out to be significant as there was a marked differential in temperature across the interface. Had only a single thermocouple been used, depending on its physical placement, it could have resulted in either ultra conservative or highly optimistic results. Again, the average of the two readings were used. The reason for the difference in temperatures can be seen from a glance at Fig. 1. The center of the sink has a longer thermal path through the copper from film to case–sink surface than the outer periphery, thus producing a nonuniform temperature gradient across the case–sink interface surface.

B. Liquid Handling

The main problem to be solved is that of providing a means of accurately measuring the liquid flow in a closed-loop system for liquids having widely different viscosities. The approach used was to use a constant-volume variable-speed pump and allow it to do double duty, that is, perform the pumping function while at the same time meter the liquid flow rate. A Randolph Model 610 pump was used for this purpose. However, this introduced the additional problem of pressure and consequent flow pulsations in the liquid flow. This was overcome by a combination of two techniques. First, we used a few feet of Tygon® tubing, which served to dampen the pulsations while doubling as a heat exchanger when placed in a water or water ethylene–glycol bath. Second, six tee sections were used for additional damping. The tees were placed in the tubing 18 in apart, with each tee extended vertically 3 ft, capped at the top, and air filled. The tees perform a double function, for, while they served to filter the pump's pulsations, they also provided an expansion tank function as well as an outlet for entrapped air bubbles in the liquid. Liquid temperature was measured at the liquid inlet and outlet by placing thermocouple probes in the liquid flow passageway of the sink. Differential pressure drop across the sink was monitored by use of a mercury manometer. Liquid temperature was maintained constant at elevated temperatures by placing the Tygon tubing in a bucket with hot tap water circulating through the bucket. For low-temperature operation the bucket was replaced by a small foam cooler which was filled with an ethylene–glycol water solution and cooled as needed by applying dry ice to the ethylene–glycol mixture.

® Registered trademark of the Plastics and Synthetics Division, Norton Company, Akron, Ohio.

Test Results

A total of 38 test runs were made using the four liquids at flow rates from 0.15 to 3.4 gal/min and film temperatures from 70 to 160°F. A listing of a single test run is given in Appendix III.

From the calculated values in Appendix III, the desired correlation plots can be obtained. Before discussing the results, the derivation of $R_{\theta Cu}$ and the use of two different liquid temperatures need clarification. $R_{\theta Cu}$ is approximated as a constant, and its value is taken from the results of the plot of R_θ versus flow rate (GPM) for water as the test liquid shown in Fig. 4. Fig. 4 shows the total heat-sink thermal resistance asymptotically approaching a value of approximately 0.04°C/W, thus indicating a value for the copper component of 0.04°C/W. While all 38 test values of R_θ found using the four different test fluids could have been plotted, it was only necessary to plot values for water—they most clearly define $R_{\theta Cu}$ because of the superior cooling capabilities of water.

The bulk liquid temperature properties were used for the pressure-drop correlation data because the liquid film subjected to heat transfer is not a controlling property of resistance to liquid flow. But for the thermal correlation plot the heat-transfer characteristics of the fluid are a first-order function of film as opposed to liquid bulk or average temperature, and therefore it is important to use the film temperature. This is the reason for having multiple values for the same liquid properties and, consequently, multiple values for the Reynolds number.

The test data for (3) is plotted in Fig. 5. As shown, it may be approximated by a straight line, from which (5) is derived by conventional techniques:

$$R_\theta' = \frac{4.3}{kP_R^{1/3}\mathrm{Rey}^{2/3}}, \qquad \text{°C/W} \tag{5}$$

$$R_\theta = [0.04 + R_\theta'], \qquad \text{°C/W}. \tag{6}$$

The pressure-drop correlation data are shown plotted in Fig. 6. The details of the derivation of (7) used to model the pressure-drop data are omitted for the sake of brevity:

$$\mathrm{Rey}^\beta = \alpha\left(\frac{\rho V^2}{\Delta P}\right). \tag{7}$$

In a similar manner as was done for (3) and (5), (1) can be made to fit the straight-line correlation plot by solving for α and β. The result is

$$\Delta P = 3.04\left[\frac{\rho V^2}{\mathrm{Rey}^{0.576}}\right] \times 10^{-8}, \qquad \text{lb/in}^2. \tag{8}$$

In reviewing the result as shown in Figs. 5 and 6, it appears that a given heat sink can be modeled for liquids other than water using a single equation over a wide Reynolds number range. The greater data scatter of Fig. 6

LIBERTY EXHIBIT 1030, Page 3
Authorized licensed use limited to: Robert Durham. Downloaded on January 18,2023 at 02:24:56 UTC from IEEE Xplore. Restrictions apply.

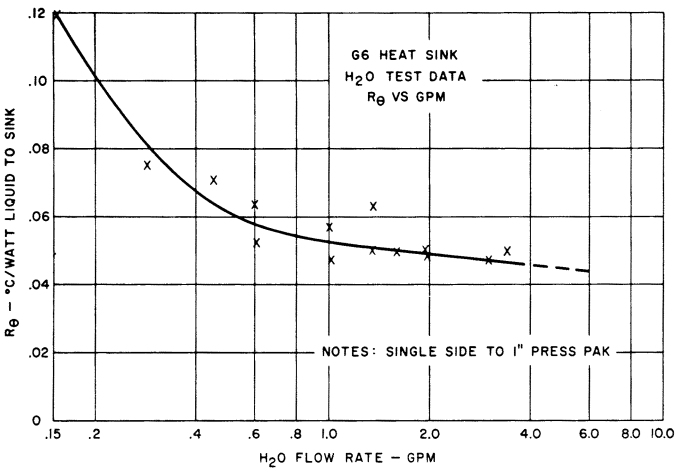


Fig. 4. Heat-sink thermal R versus water flow rate.

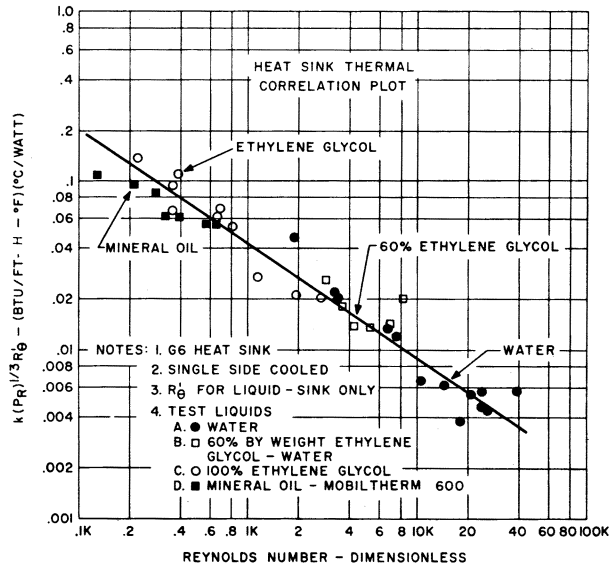


Fig. 5. Heat-sink thermal correlation plot.

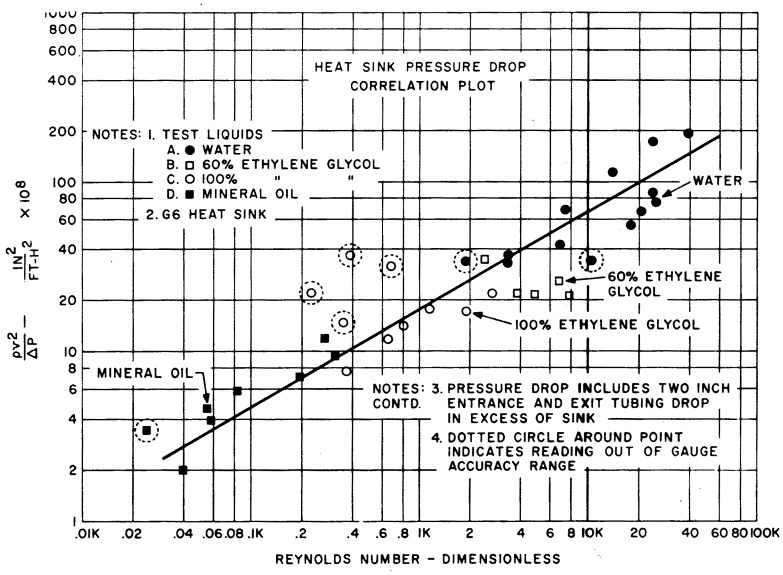


Fig. 6. Heat-sink pressure-drop correlation plot.

is partially due to the wider Reynolds number range and due to errors in pressure measurement. A curve fit could be made by using two straight lines with a break point near a Reynolds number of 6000. However, more testing would be needed to justify such a step. It is mentioned mainly to alert the reader to the fact that depending on the sink design, two straight lines may be called for to effectively model the sink, taking into account the turbulent and laminar flow regions.

Conclusions

Performance data on liquid-cooled heat sinks can be derived that will allow the designer to accurately predict the performance of a given sink for liquids other than water having a wide range of liquid properties. Furthermore, a technique for deriving the relationship using readily obtained equipment is outlined.

It is recommended that plots having the format of Figs. 5 and 6 or a similar format be made available to the equipment designer together with charts that simplify the conversion of a given liquid flow rate in gal/min and the liquid propety ρ/μ a Reynolds number based on the heat sink in question. Space does not allow inclusion of this material in this text. Further work needs to be done to gather together the properties of the most commonly used cooling liquids and plot their properties versus temperature using a single set of units to facilitate comparison and using the aforementioned recommendation.

Although the author does not see any limitations in the use of the proposed equations or alternate plots, there is a limitation in the test method suggested in the form of the need to use flexible tubing for the metering pump with a corresponding limit in liquid system pressure. If a ball–disk meter were used capable of operating over a wide

LIBERTY EXHIBIT 1030, Page 4
Authorized licensed use limited to: Robert Durham. Downloaded on January 18,2023 at 02:24:56 UTC from IEEE Xplore. Restrictions apply.

viscosity range, a higher pressure and higher capacity centrifugal pump could be used in place of the metering pump. This then would allow for the use of high-pressure tubing and a greater range of Reynolds numbers could be obtained with a single test liquid, thus reducing the number of test fluids needed for the same range of liquid properties. A given liquid's properties could be varied over a greater range by increasing the temperature excursion to a greater range. This was not possible with the Tygon tubing due to pressure and temperature limitations.

Appendix I

Nomenclature of Variables in Nusselt Equation

h	Heat-transfer coefficient, Btu/ft²·h·°F.
μ	Viscosity, lb/h·ft.
C_p	Specific heat, Btu/lb·°F.
k	Thermal conductivity, Btu/ft·h·°F.
ρ	Density, lb/ft³.
D	Diameter, ft.
d	Diameter, in.
V	Velocity, ft/h.
$M,\alpha,(1-c)$	Constants, dimensionless.

Appendix II

Example

Assume that it is desired to single-side cool a pressure-mounted thyristor using GE 10-C[1] transformer oil at 40°C. Furthermore, assume a 1-gal/min oil flow rate. Using the subject heat sink shown in Fig. 1, the pressure drop across it as well as the sink thermal resistance is calculated next using Figs. 5 and 6 based on a 300-W thyristor dissipation. The given conditions are $P_{\text{disp}} = 300$ W, $T_{\text{inlet}} = 40$°C, the oil flow rate is 1 gal/min, and a G6 heat sink is used. The following calculations are made.

A. Pressure Drop

$$DV = 122 \quad \text{gal/min}/d \tag{9}$$

therefore,

$$\begin{aligned} DV &= 320, && \text{ft}^2/\text{h}\cdot \\ \rho &= 54, && \text{lb/ft}^3, \quad \text{at } 40°\text{C} \\ \mu &= 17, && \text{lb/h}\cdot\text{ft}, \quad \text{at } 40°\text{C} \end{aligned}$$

therefore,

$$\begin{aligned} \rho/\mu &= 3.19, && \text{h/ft}^3, \quad \text{at } 40°\text{C} \\ \text{Rey} &= DV(\rho/\mu) = 1020, && \text{at } 40°\text{C} \\ \rho &= 54, && \text{lb/ft}^3 \\ V &= (DV)/D \\ V &= 9.75k, && \text{ft/h} \\ \rho V^2 &= 51.3 \times 10^8, && \text{lb/ft}\cdot\text{h}^2. \end{aligned} \tag{10}$$

Referring to Fig. 6,

$$\left(\frac{\rho V^2}{\Delta P}\right) = 18 \times 10^8\big|_{\text{Rey}=1020}$$

therefore,

$$\Delta P = \frac{\rho V^2}{18 \times 10^8} = 2.85, \qquad \text{lb/in}^2.$$

Note that since the T inlet to outlet is less than 3°C from the relationship

$$T = 0.237\left(\frac{P}{\text{GPM } C_p}\right), \qquad °\text{C} \tag{11}$$

when P, GPM, ρ, and C_p are in units already given, then T_{bulk} or T_{av} is assumed equal to T_{inlet}.

B. Thermal Resistance

Iteration 1): Assume $T_{\text{film}} = T_{\text{bulk}}$. Then

$$\begin{aligned} \text{Rey} &= 1020 \\ (P_r)^{1/3} &= 4.76 \\ k &= 0.0722, && \text{Btu/ft}\cdot\text{h}\cdot°\text{F} \\ k(P_r)^{1/3} &= 0.356, && \text{Btu/ft}\cdot\text{h}\cdot°\text{F}. \end{aligned}$$

Referring to Fig. 5,

$$R_\theta' k(P_r)^{1/3}\big|_{\text{Rey}=1020} = 0.041, \qquad \text{Btu}\cdot°\text{C/ft}\cdot\text{h}\cdot°\text{F}\cdot\text{W}$$

therefore,

$$R_\theta' = 0.115, \qquad °\text{C/W}.$$

From (4),

$$R_\theta = 0.155, \qquad °\text{C/W}$$

$$T_{\text{film}} = T_{\text{bulk}} + 1/2\,(P\cdot R_\theta') = 57.3, \qquad °\text{C}.$$

Iteration 2): Assume $T_{\text{film}} = 52$°C. Then

$$\begin{aligned} \rho/\mu &= 4.2, && \text{h/ft}^2 \\ \text{Rey} &= 1280 \\ k(P_r)^{1/3} &= 0.320, && \text{Btu/ft}\cdot\text{h}\cdot°\text{F}. \end{aligned}$$

From Fig. 5,

$$k(P_r)^{1/3}R_\theta'\big|_{\text{Rey}=1280} = 0.0355, \qquad \text{Btu}\cdot°\text{C/ft}\cdot\text{h}\cdot°\text{F}\cdot\text{W}$$

therefore,

$$R_\theta' = 0.111, \qquad °\text{C/W}.$$

The new film temperature is

$$T_{F_1} = T_{\text{bulk}} + 1/2\,(PR_\theta') = 56.7, \qquad °\text{C}.$$

By interpolating the results of iterations 1) and 2), it may be concluded that further iterations are not necessary. Rounding R_θ' to two places yields

$$R_\theta = 0.15, \qquad °\text{C/W}$$

from (4), for

$$R_{\theta\text{Cu}} = 0.04, \qquad °\text{C/W}.$$

[1] Properties (ρ, μ, k, C_p, $(P_r)^{1/3}$ at specified temperatures) of General Electric 10-C transformer oil are available from Transformer Department, 100 Woodlawn Ave., Pittsfield, Mass.

Authorized licensed use limited to: Robert Durham. Downloaded on January 18,2023 at 02:24:56 UTC from IEEE Xplore. Restrictions apply.

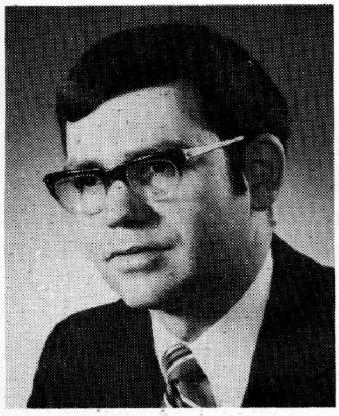
Appendix III

Mineral Oil Test Run #6

Flow rate = 0.55 gal/min.
Pressure drop = 4.6 lb/in^2.
Inlet temperature = 34°C.
ΔT inlet–outlet = 2°C.
T_1 = 122°C, the average temperature reading of the lower shunt thermocouples.
T_2 = 135°C, the average temperature reading of the middle shunt thermocouples.
T_3 = 149°C, the average temperature reading of the upper shunt thermocouples.
$T_3 - T_1$ = 27°C.
T_4 = 98.5°C, the average temperature reading of the sink interface thermocouples.
P = 0.256 kW, the power delivered to the sink, equal to $9.5(T_3 - T_1)$.
$T_4 - T_{\text{inlet}}$ (ΔT sink–liquid) = 64.5.
R_θ = 0.25°C/W.
T_{av} (liquid bulk) = 35°C, the average liquid (bulk) temperature between inlet and outlet.
T_F (liquid film) = 62°C, the liquid film temperature, equal to $T_{\text{av}} + 1/2\ (T_{\text{Cu}} - T_{\text{av}})$, where $T_{\text{Cu}} = T_4 - (P \times R_{\theta\text{Cu}})$; $R_{\theta\text{Cu}}$ = 0.04°C/W.
ρ (at bulk temperature) = 58 lb/ft^3.
ρ/μ (at bulk temperature) = 0.35 h/ft^2.
Rey (at bulk temperature) = 61.5.
ρ (at film temperature) = 56.5 lb/ft^3.
ρ/μ (at film temperature) = 1.2 h/ft^2.
Rey (at film temperature) = 210
V = 5.7 ft/h $\times 10^3$, based on liquid flow passage diameter of 0.375 in.
V^2 = 32 ft^2/h^2 $\times 10^6$, based on liquid flow passage diameter of 0.37 in.
DV = 175 ft^2/h, based on liquid flow passage diameter of 0.375 in. (D = passage diameter in feet, while d = passage diameter in inches).
k (at film temperature) = 0.068 Btu/ft·h·°C.
$(C_p\mu/k)^{1/3}$ (at film temperature) = 6.4.
R_θ' = 0.210 °C/W ($R_\theta' = R_\theta - R_{\theta\text{Cu}}$).
$[k \times (C_p\mu/k)^{1/3} \times R_\theta']$ (at film temperature) = 9.2×10^{-2} Btu·°C/ft·h·°F·W.
ρV^2 (at bulk temperature) = 18.5 lb/ft·h^2 $\times 10^8$.
ΔP (at bulk temperature) = 4.6 lb/in^2.
$V^2/\Delta P$ (at bulk temperature) = 4.0(in^2/ft·h^2) $\times 10^8$.

References

[1] E. L. Harris, Jr., "How to design water-cooled heat sinks for SCR's," *Contr. Eng.*, pp. 105, 109, Apr. 1965.
[2] M. Jacob, *Heat Transfer*. New York: Wiley, 1949.
[3] W. H. McAdams, *Heat Transmission*, 3rd ed. New York: McGraw-Hill, 1954.
[4] Allan D. Kraus, *Cooling Electronic Equipment*. Englewood Cliffs, N.J.: Prentice-Hall, 1965.



Forest B. Golden (S'59–M'65) received the B.S.E.E. degree from Northeastern University, Boston, Mass., in 1961 and attended graduate school at Rensselaer Polytechnic Institute, Troy, N.Y.

He is a Consulting Engineer with the Semiconductor Products Department, General Electric Company, Auburn, N.Y. He has been active in high-current industrial semiconductor applications for five years. His field of expertise is thyristor fusing. His most recent work has been in the area of resistance welding for automotive assembly applications. This has resulted in a new industry-wide rating concept. He is a frequent speaker at seminars involving high-power solid-state control.

Mr. Golden is a Registered Professional Engineer in the State of New York.

LIBERTY EXHIBIT 1030, Page 6

Authorized licensed use limited to: Robert Durham. Downloaded on January 18,2023 at 02:24:56 UTC from IEEE Xplore. Restrictions apply.