21.7 COOLING BARE TUBES UNDER NATURAL CONVECTION CONDITIONS

The analysis of the heat transferred from a bare electron tube under natural convection conditions entails a trial-and-error procedure because the heat transfer coefficient is dependent on the temperature difference to the $\frac{1}{4}$ or $\frac{1}{3}$ power and the radiation involves temperatures to the fourth power. Under these conditions radiation plays a very significant role. Basically, in this type of problem, one is given a heat dissipation and must assume a value for the bulb temperature. The heat transferred by both natural convection and radiation is then calculated and added together to determine whether the total is equal to the heat dissipated.

The worksheet given as Table 21.1 can be used to excellent advantage in making these calculations. Note that four temperatures are assumed. This will usually permit at least three good points to be obtained for a graphical plot such as the one in Fig. 21.12.

Problems of this type are frequently encountered in analyzing natural convection cooling of all types of electronic equipment. The surface temperature is a function of natural convection and radiation, which in turn are governed by the temperature difference between surface and surroundings. Indeed, one may write

$$q_c = hS\theta \tag{21.6}$$

for convection,

$$q_r = \sigma S F_e F_A (T_S^4 - T_R^4) \tag{21.7}$$

for radiation, and

$$q = q_c + q_r \tag{21.8}$$

as the total heat dissipation.

The numbers in Table 21.1 and the curves in Fig. 21.12 are a solution to the problem of a vertically mounted miniature vacuum tube $2\frac{1}{2}$ in (6.35 cm) high and $\frac{3}{4}$ in (1.91 cm) in diameter dissipating 4.75 W in a still-air environment at 26.7°C. It is presumed that a negligible amount of heat is conducted out of the tube through the electrical leads, and the tube envelope emissivity is specified as $\epsilon = 0.80$. If the tube is considered as a somewhat small body in a large enclosure, the emissivity

TABLE 21.1 Calculations for Heat Transfer from Vacuum Tube

$S = \pi dL = \pi(0.0191)(0.0635) = 0.00381\ m^2$
Significant dimension for natural convection: $L = 0.0635$ m
Constant in natural convection relationship: $C = 0.55$ (Table 6.9)

$q = 4.75$ W, $e = 0.80$, $p = 1.00$

Item	Dimensions	How computed	Trial I	Trial II	Trial III	Trial IV
Natural convection						
t_S	°C	Assume	82.2	93.3	104.4	115.5
t_R	°C	Given	26.7	26.7	26.7	26.7
θ	°C	$\theta = t_S - t_R$	55.5	66.6	77.7	88.8
$\theta/2$	°C	Compute	27.8	33.3	38.8	44.4
t_f	°C	$t_f = \frac{1}{2}(t_S + t_R)$	54.5	60.0	65.5	71.1
k @ t_f	W/m °C	Fig. 6.10	0.0284	0.0287	0.0291	0.0294
a @ t_f	m^{-3} °C^{-1}	Fig. 6.9	6.357×10^7	5.975×10^7	5.498×10^7	5.149×10^7
θ/L	°C/m	Compute	874.0	1048.8	1223.6	1398.4
$(\theta/L)^{1/4}$	$(°C/m)^{1/4}$	Compute	5.44	5.69	5.91	6.12
$(ap^2)^{1/4}$	$(m^{-3}\,°C^{-1})^{1/4}$	Compute	89.29	87.92	86.11	84.71
h	W/m^2 °C	$h = Ck(\theta/L)^{1/4}(ap^2)^{1/4}$	7.58	7.90	8.15	8.38
q_c	W	$q_c = hS\theta$	1.59	2.00	2.41	2.83
Radiation						
T_S	K	$T_S = t_s + 273$	355.2	366.3	377.4	388.5
T_R	K	$T_R = t_r + 273$	299.7	299.7	299.7	299.7
$T_S/100$	K	Compute	3.55	3.66	3.77	3.89
$T_R/100$	K	Compute	3.00	3.00	3.00	3.00
$(T_S/100)^4$	K^4	Compute	159.18	180.03	202.87	227.81
$(T_R/100)^4$	K^4	Compute	80.68	80.68	80.68	80.68
$(T_S/100)^4 - (T_R/100)^4$	K^4	Compute	78.50	99.35	122.19	147.13
q_r	W	$5.67\epsilon S[(T_S/100)^4 - (T_R/100)^4]$	1.36	1.72	2.11	2.54
q	W	$q = q_c + q_r$	2.95	3.72	4.52	5.37

factor is $F_\epsilon = \epsilon = 0.80$ and the arrangement factor is $F_A = 1.00$. As a margin of safety, the surface area contributed by the top of the tube is ignored.

Figure 21.12 shows that in this application, because the value of the coefficient of heat transfer in free convection is low, convective heat transfer is not very much more significant than radiation. The radiation is, of course, governed by the arrangement factor and the tube envelope emissivity. For this reason, natural convection cooling designs that ignore radiation are likely to be unduly pessimistic.

21.8 TUBE SHIELDS

The glass envelope of an electronic tube is a hindrance rather than an aid to the flow of heat from the tube. The glass is a very poor heat conductor and is virtually opaque to thermal radiation at temperatures below 400°C. Because the glass absorbs nearly all of the heat radiated by the contained tube elements, and because it is such a poor conductor, a hot spot occurs on the tube as shown in Fig. 21.13 [7]. This hot spot usually appears opposite the plate of the tube where the dissipation is the greatest. The severe temperature gradient between the hot spot and relatively cool ends of the tube causes a thermal stress that eventually leads to the failure of the glass itself.

The hot spot also causes evolution of gas from the inner surface of the glass envelope. The rate of gas evolution is proportional to the tube bulb temperature. This is another reason for hot spot elimination; gassy tubes will not operate with any degree of predictability.

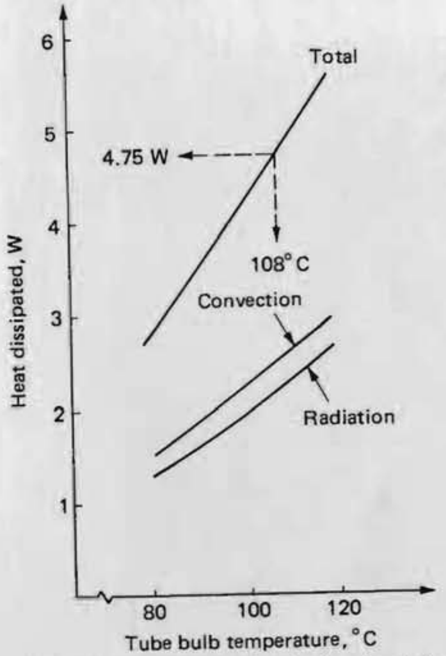

FIG. 21.12 Graphical solution for tube bulb temperature. Points for curves are obtained from calculations in Table 21.1.

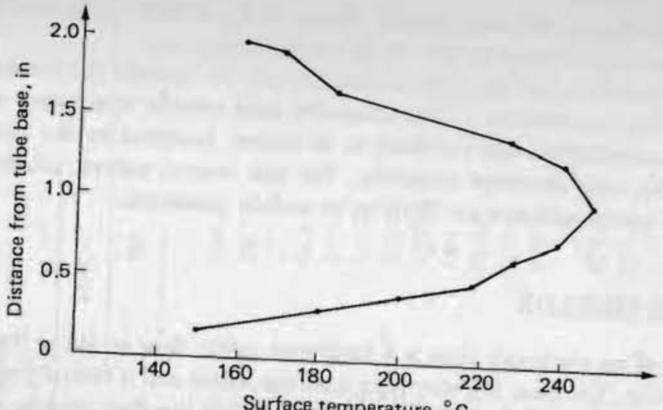

FIG. 21.13 Tube bulb temperature as a function of distance from the base of 6AQ5 vacuum tube. (From data provided by the International Electronic Research Corp., Burbank, Calif.)

Several organizations have developed tube shields that have solved the hot spot problem to a considerable degree. These tube shields have provided an efficient means of transferring heat by conduction to heat sink or chassis, reduce the severity of the temperature gradient between the hot spot and tube ends, provide a greater surface area for heat dissipation by convection, and serve as the tube mounting in severe shock and vibration environments.

Figure 21.14 shows some subminiature tube shields developed and marketed by

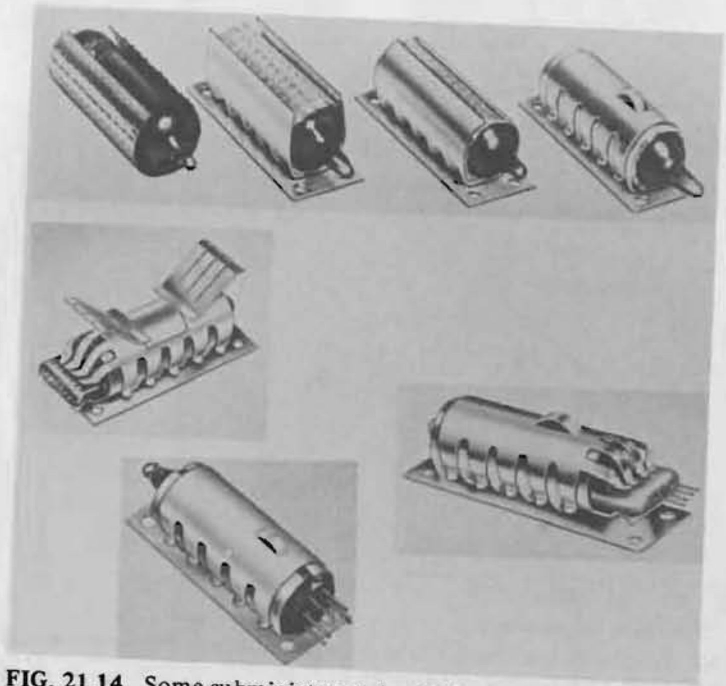

FIG. 21.14 Some subminiature tube shields. (Courtesy of International Electronic Research Corp., Burbank, Calif.)

the International Electronic Research Corporation (IERC). Note the silver liner between the tube envelope and the tube shield assembly.

Tube shields for miniature and octal tubes are shown in Figs. 21.15 and 21.16. Also shown are spring liners that adapt themselves to variations in tube envelope contour to maintain maximum contact between envelope and shield.

Typical test results [8] on a 12BY7 thermatron tube dissipating 10 W are shown in Fig. 21.17. These data were provided by IERC and show tube bulb temperature as a function of air velocity. These are forced convection tests showing that when these tubes were equipped with JAN-type tube shields, the tubes actually ran hotter than with no shield at all.

The reader should not infer that a tube shield is a cure-all for problems of high tube bulb temperature. Tube shields are helpful only when selected judiciously. Figure 21.17 shows that the lowest tube bulb temperature is obtained *with* a tube shield. These test data indicate that some shields under natural convection conditions operate at a cooler temperature than a bare tube at a 150 m/min air velocity. Figure 21.18, also based on valid test data [9], shows that the lowest bulb temperature is obtained with no shield at all. This difference in results is due to the different tube shields used, showing that the designer must exercise care in the selection and utilization of the tube shield.

FIG. 21.15 Some miniature tube shields. (Courtesy of International Electronic Research Corp., Burbank, Calif.)

FIG. 21.16 Some octal tube shields. (Courtesy of International Electronic Research Corp., Burbank, Calif.)

21.9 FORCED CONVECTION COOLING OF VACUUM TUBES

21.9.1 Vacuum Tubes in Cross Flow

In Chap. 6, a reference was made to a correlation due to Hilpert [10] for air flowing across single cylinders:

$$\mathrm{Nu} = B(\mathrm{Re})^n \tag{21.9}$$

where the values of B and n depend on a range of values of the Reynolds number. These values were given in Table 6.1.

Robinson et al. [11] have proposed a modification to Eq. (21.9) when the cylindrical heat sources that are encountered in electronic equipment are considered:

$$\mathrm{Nu} = FB(\mathrm{Re})^n \tag{21.10}$$

The factor F is an arrangement factor depending on the cylinder geometry. This factor permits utilization of the correlation for the flow of air across a single cylinder. Values of F are given in Table 21.2.

Robinson et al. [11] have further proposed values of B and n that yield a curve identical to that obtained using Table 6.1. These values of B and n are in the more limited range of $1{,}000 \leqslant \mathrm{Re} \leqslant 100{,}000$ and are given in Table 21.3.

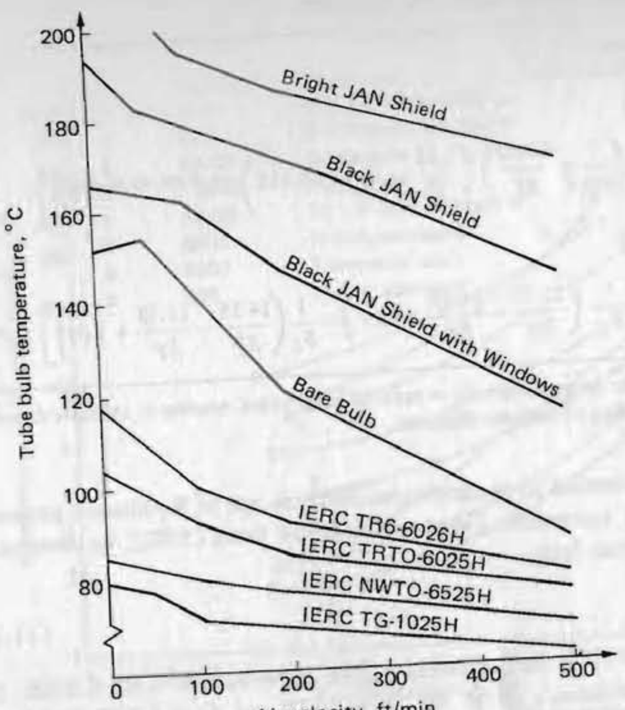

FIG. 21.17 Test results for a 12BY7 thermatron tube. (From data provided by International Electronic Research Corp., Burbank, Calif.)

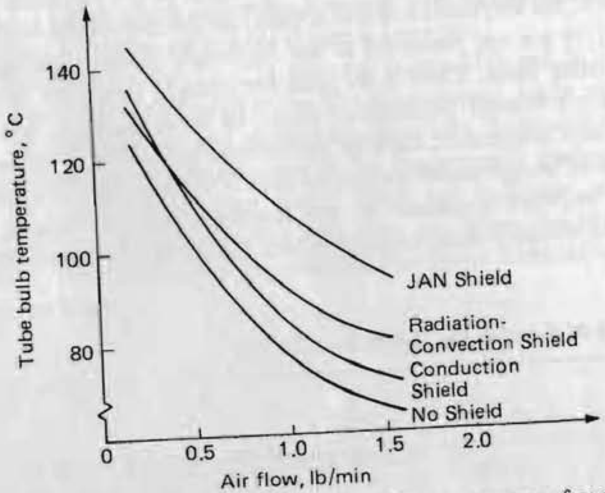

FIG. 21.18 Downwind bottom bulb temperature for seven pin miniature tubes in cross flow [9].

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