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Cooling of Modern Airborne Electronic Equipment

FOREWORD

Changes in this revision are format/editorial only.

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1. SCOPE:

This document contains information on the cooling of modern airborne electronics, emphasizing the use of a heat exchange surface which separates coolant and component. It supplements the information contained in AIR64 for the draw through method and in AIR728 for high Mach Number aircraft. Report contents include basic methods, characteristics of coolants, application inside and outside of the “black box” use of thermostatic controls to improve reliability and system design. Characteristics of typical cooling components are treated sufficiently to permit selection and to estimate size and weight.

While emphasis is placed herein on equipment cooling, section 10 dealing with thermal control of the environment, reminds the reader that some equipment will require heating for start up from a cold condition or as a means to control temperature within narrow limits (e.g. in a crystal oven).

Property data and constants are also tabulated. All numerical values are given in British and SI units.

1.1 Purpose:

This document is intended as an aid in the early layout and specification of the cooling of airborne electronic equipment. The material presented herein is directed toward an understanding of the cooling requirements, the selection of suitable means, the estimating of its physical impact on size, weight and power and the interfaces that exist with other disciplines.

Some basic familiarity with the subject is presupposed and it is understood that subsequent detail design will avail itself of the specialist who has at its command the hardware experience and analytical background necessary for that task. Analytical background somewhat beyond these objectives has been added in sections 13, 14 and 15 for those readers who may appreciate its inclusion.

2. REFERENCES:

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2.1 List of Symbols:

TABLE 1 - List of Symbols

Symbol	Unit British	Unit (SI)	Meaning
A, B, C, H	inch, ft	(m)	linear dimensions
A_s	ft	(m^2)	area for heat transfer
A_c	ft^2, $inch^2$	(m^2)	free flow area in a conduit or exchanger
A_{fr}	ft^2, $inch^2$	(m^2)	frontal area facing flow
A_o	-	-	constant in equation
AC	amp	(A)	alternating current
AU	Btu/hour-F	(W/°C)	overall heat transfer per unit of temperature difference
BP	F	°C	boiling point
C	Btu/hour-F	(W/°C)	product Wcp
c_p	Btu/lb-F	(J/kg·°C)	specific heat
d	inch	(mm)	thickness, fin
DC	amp	(A)	direct current
D_h	ft	(m)	hydraulic diameter $= 4A_c/P = 4A_cL/A_s$
E	-	-	effectiveness of heat exchanger
f	-	-	Fanning friction factor $= (\Delta pg\rho D_h/L)(A_c/W)^2$
F	-	-	combined radiation coefficient
g	$ft/second^2$	(m/s^2)	gravity constant 32.2 (9.81)
G_h	lb	(kg)	weight of heat exchanger
G_e			weight of equipment
h	$Btu/hour\text{-}ft^2\text{-}F$	$(W/m^2\cdot°C)$	coefficient of heat transfer, fluid to solid
$K_{1,2}$	-	-	constants in equations
k	Btu/hour-ft-F	$(W/m^2\cdot°C)$	thermal conductivity
L	ft	(m)	flow length for fluid
m	-	-	ratio of pressure drop on two sides of heat exchanger

TABLE 1 - List of Symbols (Continued)

Symbol	Unit British	Unit (SI)	Meaning
N_p	-	-	Prandtl number = $C_p\mu/k$
P	ft	(m)	perimeter of flow conduit touched by fluid flow
p	psi, PSF	(Pa)	static pressure in fluid
Δp	inch.H_2O, psi	(Pa)	pressure differential in fluid flow between two points
Q	Btu/min, Btu/hour	(W)	heatflow per unit of time ($Q \times \theta$ Btu, (J) is heat energy)
Q/A_c	Btu/hour-ft^2	(W/m^2)	heat flux
R	F hour/Btu	(°C/W)	thermal resistance
R_e	-	-	Reynold Number = $WD_h/A_c\mu$
S	Btu/hour-ft^2-$°R^4$	($W/m^2 \cdot K^4$)	universal radiation constant
T	°R	(K)	absolute temperature
t, Δt	F	(°C)	temperature, temperature difference
U	Btu/hour-ft^2-F	($W/m^2 \cdot °C$)	overall heat transfer coefficient between more than two elements
V	ft^3	(m^3)	volume of heat exchanger matrix
V_o	-	-	volume constant in heat exchanger equation
W	lb/min, lb/hour	(kg/s)	massflow of fluid
α, β	-	-	powers in equation for heat exchanger volume
Δ	-	-	differential
ε	-	-	emissivity
η	-	-	efficiency percent
θ, θ_e	second, min	(s)	time, emergency operating time
μ	lb/hour-ft	(Pa·s)	fluid viscosity
ρ	lb/ft^3	(kg/m^3)	density
σ	-	-	mean density/standard density

2.2 Symbols Not Used But Common in Literature:

TABLE 2 - Symbols Not Used But Common in Literature

Symbol	Unit British	Unit (SI)	Meaning
G	$lb/hour\text{-}ft^2$	$(kg/h \cdot m^2)$	massflow per unit area
j	-	-	Stanton number = h/GC_p
N_u	-	-	Nusselt number = hD_h/k
Units:			( ) denotes SI unit
Btu			British thermal unit for heat energy (International Tables)
(°C)			degree centrigrade temp.
ft, ft^2, ft^3			feet, length, area, volume
hour (h)			hour, time
Hz			Hertz, oscillations per second
inch, inch H_2O			inch length, inch water column pressure
(J)			Joule = W·s unit of energy
(kg)			kilogram mass
(°K)			degree Kelvin, absolute temperature = °C + 273
lb			pound mass
(m), (m^2, m^3)			metre length (area, volume)
(mm), (km)			milli - kilo-metre (10^{-3}, 10^3)
minute			minutes time
(Pa)			Pascal = Newton/m^2, unit of pressure
psi, psia			pounds per square inch pressure (a = absolute)
PSF			pounds per square foot
°R			degree Rankin, abs. temp. = F + 460
rpm			revolutions per minute
second (s)			second, time
SCFM			standard cubic foot per minute based on density of 0.075 lb/ft)
(W)			Watt, unit of power

TABLE 2 - Symbols Not Used But Common in Literature (Continued)

Symbol	Unit British	Unit (SI)	Meaning
Subscripts			
1, 2 etc.			sides 1 and 2 of heat exchanger, various temperatures etc.
a			air, (Wa)
amb			ambient condition
b			boiling, (h_b)
B			margin, (Δt_B)
c			cold (t_c), condensing (h_c), "free", (A_c)
e			equipment, (G_e)
fr			frontal, (A_{fr})
h			heat exchanger (G_h), hydraulic, (D_h) hot, (t_h)
ℓ			liquid, (W_ℓ), cp_ℓ
m			mean, (ρm)
o			constant (v_o), boiling (p_o)
s			surface, (A_s)
st			standard, (ρst)
w			water, (t_w)

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11. Saltzman, A. R., Closed Loop Thermal Process Control of Airborne Electronic Equipment, J. Aircraft Vol. 3, No. 3, June 1966, pp. 233/249

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Section 10

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4. BACKGROUND:

Equipment cooling in current commercial aircraft employs primarily the “draw through” method covered in References 1 and 7. The equipment is located in a rack and ambient cooling air is drawn through it in a controlled amount, exiting through plenums in the rack. Cooling air is thus in direct contact with the component parts.

Assuming reasonable internal air distribution, clean and dry air and suitable component locations, this method is simple and efficient. However, in many instances one or more of these conditions may not exist where sophisticated or densely packaged units must be cooled. It becomes necessary to separate component parts or subpackages from direct coolant contact by some form of intermediate heat transfer surface. Increased demands on equipment reliability and the demand on cooling capacity from the aircraft may require narrow temperature limits rather than limiting maximum temperature only.

The increased cooling demands may call for the use of an intermediate heat transfer surface, active control, closed equipment bays and separate refrigeration sources. The cooling system becomes more complex and new components are introduced on which information is difficult to obtain.

5. THERMAL ENVIRONMENT AND RELIABILITY:

Thermal environment strongly influences performance, life and reliability of electronic equipment. The rise of the rate of failure relative to an arbitrary ambient or case temperature is shown in Figure 1 for a passive and an active component. Case temperature rising above ambient level to dissipate the internally generated heat is the important value in the active component.

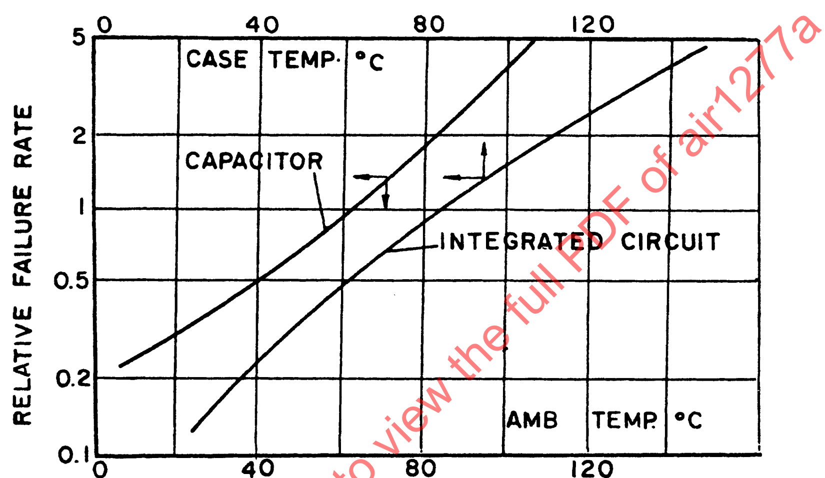


FIGURE 1 - Trend of Failure Versus Temperature

Humidity encountered during flight or in extended storage has been found to be a major source contributing to unreliable performance of the electronic equipment. It produces corrosion from galvanic action and fosters microbiological growth. Moisture absorption will lead to physical and electrical changes which may interfere with the equipment's function. The design of the cooling system must protect all equipment surfaces from such effects. Interfaces within the aircraft between the cooling system, water separators ducts and controls should function to assure a minimum moisture environment. (Literature No. 2)

Cyclic variation of the thermal environment induces cyclic stresses in equipment which may induce mechanical failure due to fatigue. A complete cooling system will therefore attempt to hold the thermal environment resistant by use of automatic control subject to a trade-off with mission needs and economics.

6. THE COOLING PROCESS:

6.1 Heat Flow:

A hot component maintains its temperature if it loses heat at the same rate as it is produced. For this to happen, a flow path to a medium of lower temperature (the coolant) must exist, and the resistance to heat flow along this path must be controlled. The heat flow path may take one or more of these forms:

a) Conduction through a body or structure
b) Convection with a moving medium
c) Radiation of energy through space

Figure 2 illustrates the possible combined existence of the above forms by a heat-producing unit at temperature t_1. Heat flows by conduction through the structure or by radiation to the wall at a lower temperature t_2. Heat is also dissipated into the surrounding space at the lower temperature t_3 by thermally induced circulation of air at the unit's surface. These processes occur simply by the existence of the elevated temperature of the unit.

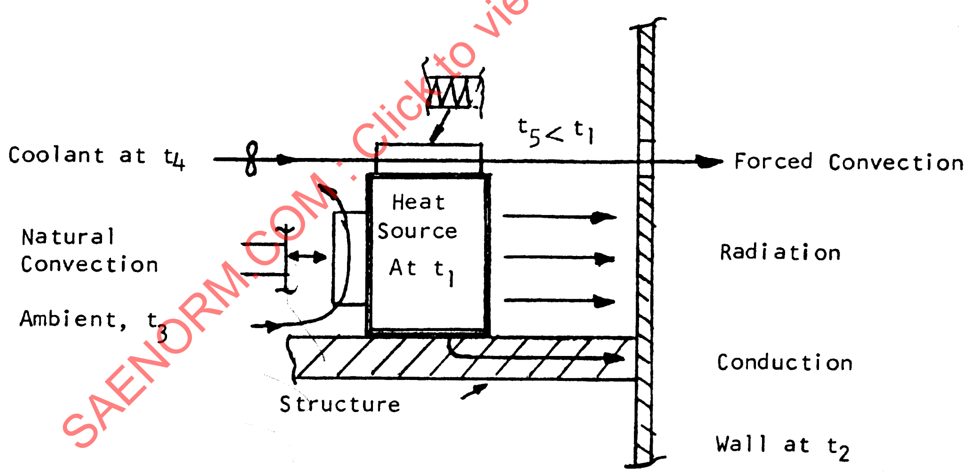


FIGURE 2 - Combined Modes of Cooling

6.2 Basic Laws:

The processes illustrated in Figure 2 are amenable to mathematical description and prediction. Table 3 summarizes applicable formulae for convenient reference. Typical property data can be found in Tables 4 and 5.

An elementary, but sometimes overlooked consequence of these laws, is the fact that a coolant cannot be brought to a higher temperature than that of the cooled component. This places a (theoretical) minimum on the amount of coolant to be supplied.

When applying the various modes of cooling, certain general rules should be considered. Adherence to such rules will give a maximum allowable temperature rise for the coolant and will thus minimize the required coolant flow. This is quite important in ground cooling where the entering air temperature may well be 120 °F (49 °C) or higher.

6.3 Conduction:

Use a short heat flow length, large sectional area, minimal air gaps and materials with a high coefficient of thermal conduction. Since such construction tends to increase weight, structural members should be employed for conducting heat as much as possible.

6.4 Convection:

Naturally induced convection works best on vertical surfaces with coolant flow having free access along the surface.

6.5 Forced Flow:

Heat flow by convection can be greatly increased by moving a cooling medium with external power over prepared surfaces as shown in Figure 2. It is this application of forced flow which yields a powerful means to effective cooling.

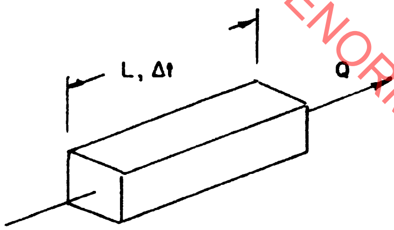
Forced convection requires the least external power if areas facing the flow are kept as large as practical, and if a large surface area for heat exchange is provided.

Flow volume and power required for cooling with air increase rapidly as the density of the air is reduced. When cooling with air at high altitude the effect is partially offset as long as the air becomes available at progressively lower temperature.

TABLE 3 - Guide Values for Various Modes of Heat Transfer

Process and Basic Law

Conduction



$Q = kA\Delta t/L$

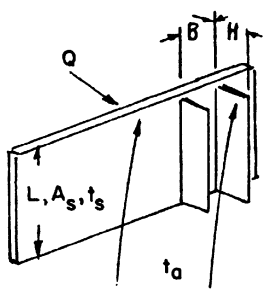
Typical Values of Controlling Factors

Values of Thermal Conduction Constant k at 68F (20°C)

	Btu/h-ft-F	(W/m·°C)
Copper	220	381
Aluminum	103	178
300 Stainless Steel	9	15.6
400 Stainless Steel	13	22.5
Oils	0.08	0.14
Insulators	0.06/.3	0.1/0.5
Beryllia	100	173

Contact Resistance	Btu/h-ft²F	(W/m²·°C)
Metal to Metal, k/l	500/2000	3000/12000

Convection, Natural Draft



$D_h = 4BH/(2H + B)$

For Air Near Room Temperature and Pressure:

$Q=0.28\ A_s\ (t_s - t_a)^{1.25}\ (1/L)^{0.25}(\rho/\rho_{st})^{0.5}$ Btu/hr.

Dimensions of F, ft, ft², $\rho_{st} = 0.0765$ lb/ft³

$Q=1.9\ A_s\ (t_s - t_a)^{1.25}(1/L)^{0.25}\ (\rho/\rho_{st})^{0.5}$ W

Dimensions of °C, m, m², $\rho_{st} = 1.225$ kg/m³

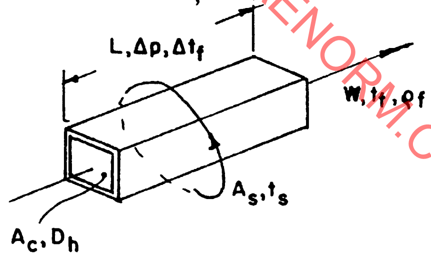
For finned heat sinks let:

$L/D_h < 11\cdot(L(ft))^{1/2}$ or $20\cdot(L(m))^{1/2}$, and $4 < L/D_h < 20$.

Air temperature rise $\approx 1/3\ (t_s - t_a)$

TABLE 3 - Guide Values for Various Modes of Heat Transfer (Continued)

CONVECTION, FORCED



$dQ = h\ (t_s - t_f)\ dA_s = Wc_p\ \Delta t_f$

$\Delta p = 4\ f\ (L/D_h)(W^2/A_c^{\,2}\ 2g\ \rho_f\ )$

$= f\ (A_s/A_c)(W^2/A_c^{\,2}2g\ \ \rho_f\ )$

dQ = Heat transfer per differential dA_s of surface A_s

h = Coefficient of Heat Transfer

f = Friction factor

Dh = Hydraulic Diameter

g = Acceleration constant

See also list of Symbols

Coefficients h and f depend on a dimensionless grouping

$R_e = D_hW/A_c$ called Reynold Number and a grouping of fluid properties $N_p = C_p\mu/k$ called Prandtl Number.

Flow is said to be laminar it $R_e < 2000$ and turbulent if $R_e > 2000$.

For laminar flow in smooth ducts:

$$hD_h/k \geqslant 1.9\ \ R_e^{1/3}\ \ (D_h/L)^{1/3}\ \ N_p^{1/3}\ \ \text{(dimensionless)}$$

$$f \sim 16/R_e\ \ \text{(Fanning Friction Factor)}$$

For turbulent flow in smooth ducts:

$$hD_h/k \approx 0.023\ R_e^{0.8}\ N_p^{0.4}$$

$$f \approx 0.046/R_e^{0.2}$$

For "wavy" or "serrated" finned surfaces typical of compact-extended surface type exchangers:

$$500 < R_e < 10.000$$

$$hD_h/k\ =\ (0.15\ \ldots\ 0.3)\ R_e^{0.6}\ N_p^{\cdot 1/3}$$

$$f\ =\ (0.4\ \ldots\ 0.8)\ R_e^{-1/3}$$

NOTE: These relations must not be applied to surfaces with variable flow area such as flow over banks of tubes or through beaded or dimpled tubes.

TABLE 3 - Guide Values for Various Modes of Heat Transfer (Continued)

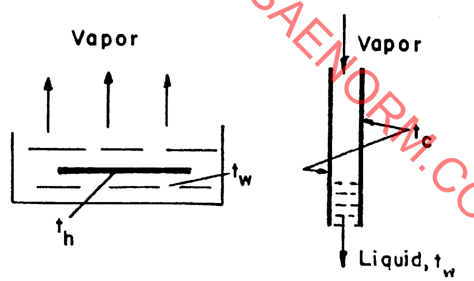
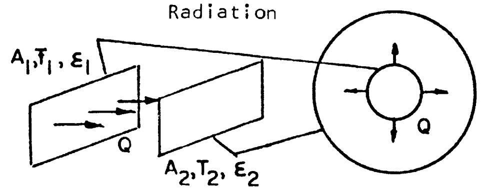
Boiling and Condensing  Δt = t_h - tw or t_c - tw $Q = h_b A_h \Delta t = h_c A_c \Delta t$	For submerged boiling in water, where p_o is sea level pressure and $0.2 < p_f/p_o < 1$, 9 (5) $< \Delta t <$ 40 (22) F (°C): $h_b = 4.3\ (80)\ \Delta t^2\ (p_f/p_o)^{0.5}$ Btu/hr - ft² F, (W/m²·°C) Evaporative Transfer to Freons and Ammonia in forced flow h_b = 160 ... 250 Btu/h-ft²-F = 910 ...1420 (W/m²·C) Condensing of pure vapors when Re < 2000 $h_c\ (\mu^2/k^3 g \rho^2)^{1/3} = 1.8\ Re^{-1/3}$, use consistent units. All property values are those of the condensed liquid, g = gravity constant
Radiation  Parallel Surfaces if $A_1 = A_2$ | Sphere or Cylinder if $A_2 \gg A_1$ $F_{1,2} = 1/(1/\varepsilon_1 + 1/\varepsilon_2 - 1)$, $F_{1,2} = \varepsilon_1$ $Q = \sigma F_{1,2} A_1 (T_1^4 - T_2^4) \approx 4 \sigma F_{1,2} A_1 T_1^4 \Delta t$ T = absolute temperature $\Delta t = (T_1 - T_2) \leq 0.1 T_1$	Radiation depends in a complex manner on the geometry of the case. Details may be found in Ref (2) Two simple cases are shown to the left: Constant $\sigma = 0.178\ 10^{-8}$ Btu/hr-ft²-R⁴ $= 5.7\ 10^{-8}$ (W/m² K⁴) Typical values of Emissivity ε are: Metals: polished 0.05; matte 0.2; oxidized 0.8 Ceramic: 0.9 Dull Black Paint: 0.8

TABLE 4 - Properties of Coolants

General Properties at 68F (20°)	Unit	Air	Fuel (JP5)	Water	60/40 ** Eth.Gl.Water	Syn. Ester (Coolanol 25)	Fluorochemical (FC75)
Specific Heat C_p	Btu/lb-F	0.24	0.48	1.0	0.74	0.44	0.25
	(J/kg·°C)	(1005)	(2010)	(4187)	(3098)	(1842)	(1047)
Thermal Conductivity k	Btu/hr-ft-F	0.015	0.079	0.35	0.22	0.077	0.08
	(W/m·°C)	(0.026)	(0.137)	(0.61)	(0.38)	(0.133)	(0.138)
Density ρ	lb/ft^3	0.075	49	62.4	67.4	56	109
	(kg/m^3)	(1.201)	(785)	(1000)	(1080)	(897)	(1746)
Prandtl Number	$C_p \mu/k$ -	0.71	9.7	7.1	40.4	86	11.9
Transfer Properties							
Heat Transfer for 100F (55.5°C) rise	Btu/lb	24	48	100	74	44	25
	(kJ/kg)	(55.8)	(112)	(232.6)	(172)	(102)	(58)
Relative Transfer Co-efficient	-	1	20	90	30	10	30
Viscosity at -40F (-40°C)	lb/hr-ft	(0.037)	35		392	200	20
	m Pa·s *	(0.015)	(14.5)	-	(162)	(83)	(8.3)
Flashpoint	F	-	>140	-	240 ***	>325	non-flammable
	(°C)		(60)		(116)	(163)	
Useful range of temperature	F	-	-65/280	35/212	-50/320	-70/330	-70/700
	(°C)		(-54/138)	(2/100)	(-45/160)	(57/166)	(-57/371)
Rate of flow for 1 inch (25 mm) Line	lb/min	-	200/400	200/400	100/200	80/160	90/180
	(kg/s)		(1.5/3)	(1.5/3)	(0.75/1.5)	(0.6/1.2)	(0.7/1.4)
Lubricity		-	Poor	Poor	Fair	Fair to Good	Poor. Carbon Face Seals Required.

* m Pa·s = 10^{-3} × Pa·s = Centipoise, ** by weight = 58% by volume, *** 100% Ethylene Glycol

TABLE 5 - Properties of Sinks

On Board Properties	Unit	Ram Air
Temperature Rise	F,(°C)	0.2 (460 + t amb) M^2 (F), (0.2 (273 + t amb)M^2) (°C) M = Mach No.
Pressure Rise	P, ambient	$\Delta p = ((1 + 0.2M^2)^{3.5} - 1) \times P$ amb., Δp system $\approx 0.6\ \Delta p$
Temp. vs. Altitude	F, ft(°C, m)	See Reference 2 and MIL-STD-210B
Pressure vs. Altitude		

Boiling Heat Transfer:+	Boiling Point at P_o, F, (°C)	Lat. Heat at BP Btu/lb (kJ/kg)	Max. Flux at BP, Btu/hr-ft2 (kW/m2)	Density lb/ft3 (kg/m3)	Boiling Temp. vs. Pressure (°C)
Water Po=14.7 psia (101 kPa)	212 (100)	970.3 (2257)	275000 (868)	59.8 (958)	$(10.8 - 2.89 \log_{10} Po/P)^2 - 17.6$*
L. Ammonia P_o=18.3 psia (126 kPa)	-20 (-29)	583.6 (1357)	Estimated: 120,000 (378)	38.7 (620)	$(9.8 + 3.51 \log_{10} P/P_1)^2 - 81.3$**
L. Nitrogen Po= 14.7 psia	-320 (-195.6)	86 (200)	35000 (110)	50.2 (804)	P_1 = 100 psia (687 KP_a)
FC 75 Po= 14.7 psia	215 (102)	37.8 (87.9)	140000 (440)	96 (1539)	-
Solid CO_2 P_o= 14.7 psia	-109.8 (-78.8)	271 (630)***	-	87.5 (1402)	-

Waxes	Melting Point F (°C)	Heat of Melting Btu/lb (kJ/kg)	Density lb/ft3 (kg/m3)	+ Submerged Boiling from Plate
Paraffin	126.3 (52.4)	63 (147)	54.2 (868)	* ±.5(°C) 10 to 115 (°C)
P - Toluidine	104 (40)	72 (167)	65.5 (1049)	** ±1.5 (°C) - 54 to 60 (°C)
Diphenye Methane	79.2 (26.3)	45 (105)	62.4 (1000)	*** Sublimation to 68F (20°C)

6.6 Evaporation:

The boiling of a liquid at the surface of a component is a form of convective transfer which yields a very high heat flux, but the flux has a limit beyond which component temperature shows a large increase as a blanket of vapor forms at the surface. The limit value ("burnout flux") must not be exceeded. Values are given in Table 5.

6.7 Radiation:

Heat loss by radiation is proportional to the 4th power of the absolute temperature of the radiating surface. If the temperature difference to the surroundings is less than about 200 °F (93 °C), radiation need be considered only when it combines with natural convection. A surface which is several hundred degrees centigrade above its surroundings, such as a power resistor may be, is predominantly cooled by radiation. Where radiation heat transfer is desired a surface with a high emissivity should be provided, such as metal oxide or black paint.

6.8 Coolant Flow:

Specifying the amount of coolant flow per kW of dissipation as a function of supply temperature may have been pre-empted by the airframe manufacturer. It is necessarily a compromise and may therefore vary substantially depending on the particulars of a case. For a rational approach assume, conservatively, that coolant is supplied near 80 °F (26.7 °C), equipment or component surface temperature is limited to 160 °F (71 °C) and that the coolant may leave at 125 °F (51.7 °C). See also Reference 1. This results in the following target values:

TABLE 6

	Coolant Air	Coolant Liquid	Coolant Liquid	Coolant Liquid
Specific Heat, Btu/lb-F	0.24	0.2	0.5	1.0
kJ/kg·°C	1.0	0.84	2.1	4.2
Flow per kW lb/min	5.27	6.32	2.53	1.26
kg/s	0.04	0.048	0.019	0.0095

6.9 Pressure Loss:

Pressure loss in a system varies approximately with the 1.8th power of flowrate and inversely with the density of the cooling medium. This explains the benefit of using liquid coolant for high and concentrated loads, especially in remote locations. Extraneous losses occurring in ducts or piping, at sudden changes of flow area and in fittings and bends should be kept at a reasonable ratio to the loss in exchangers where the useful work is done (say 10 to 25%). See Reference 2 for specific design information. Additional limitations on flow velocity arising from compressibility of air used as a coolant, suggests that the Mach number be held <0.05. However, where space is at a premium Mach numbers between 0.3 and 0.6 have been used. In a liquid system, the effects of erosion and "water hammer" suggest a limit of about 15 to 20 ft/second (4.6 to 6.1 m/s) in the plumbing. Table 5 gives values of rate of flow in a 1-inch (25.4-mm) tube based on this limit and consideration of the fluid viscosity.

6.10 Selection of Cooling Process:

The different modes of heat transfer cover a wide range of capacity for removing heat and may be primarily selected on that basis. Transfer capacity is characterized by the sustained "heat flux" which depends on the amount of heat removed per unit of time, temperature difference and transfer surface area (Btu/hour-ft^2-F or W/m^2·°C). The place of the various modes on this scale is shown in Figure 3 in W/m^2·°C. Radiation has not been included in Figure 3 because of its different nature and application.

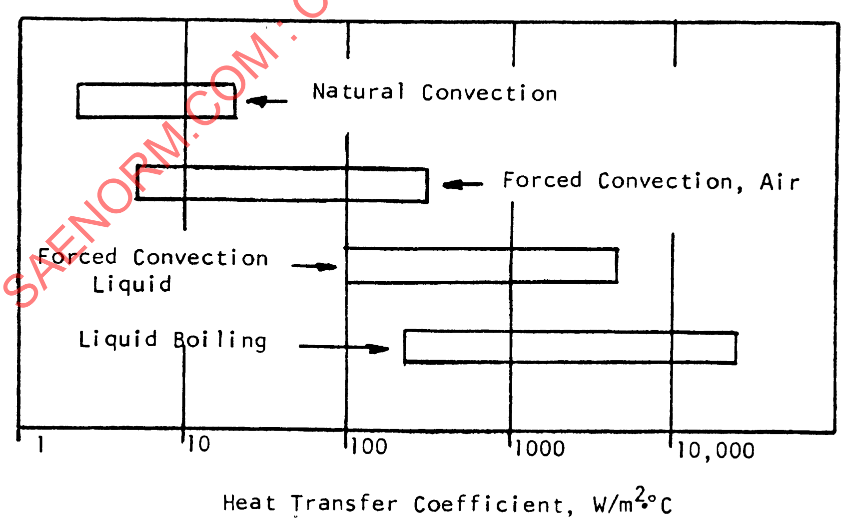


FIGURE 3 - Comparative Rate of Heat Transfer per Unit of Area and Temperature Difference

6.11 Loss of Cooling:

Equipment can be expected to operate for some time after loss of normal cooling before failure occurs. The time depends on the temperature margin Δt_B available during normal cooling, the ratio of power dissipation Q to equipment weight G_e, its material, its thermal homogeneity or absence of "hot spots", and the average amount of cooling by conduction, convection and radiation which remains.

The many factors involved made general predictions difficult. Assuming, however, that a meaningful value for the margin of temperature difference Δt_B (°C) can be stated, that good thermal design provides a reasonably uniform distribution of temperature, that the average specific heat of the material is 0.15 Btu/lb-F (0.63 kJ/kg·°C) and the amount of residual cooling is 10% of normal, then time θ_e of emergency operation can be estimated as follows:

$$0.9\, Q\, \theta_e = G_e\, c_p\, \Delta t_B \qquad \text{(Eq.1)}$$

or

$$\theta_e = 0.7\, \Delta t_B\, G_e/Q\ \text{(s)} \qquad \text{(Eq.2)}$$

Example: An LRU weighs 5 kg, dissipates 0.5 kW and can rise 20 °C before failure. Then:

$$\theta_e = 0.7 \times (5/0.5) \times 20 = 140\ \text{s} = 2.3\ \text{minutes} \qquad \text{(Eq.3)}$$

7. COOLANTS AND THEIR PROPERTIES:

7.1 Coolant:

Defined as any means which carries heat from the equipment overboard or to a conditioning device.

7.2 Heat Sink:

Implies a device or means to absorb heat from a component. The term may thus be applied to a coolant. If the coolant flows overboard, it becomes the "ultimate sink".

In a more specific sense, the term is applied to a device which extends a component's surface in contact with a coolant (transistor heat sink).

7.3 Ram Air:

Air taken aboard the aircraft by a scoop and released after being heated by the heat produced in electronic equipment is a basic coolant and heat sink.

Conditions of pressure and temperature of ambient air vary with climate and altitude. Standard values are defined by the U.S. Standard Atmosphere 1962 and extremes by MIL-STD-210. Tables may be found in Reference 2. Air taken aboard must be decelerated, whereby essentially all of its kinetic energy is converted into temperature rise in proportion to flight Mach Number (see Table 5). A rise in static pressure also results, which in part is available to force the flow of air through the ducts and equipment.

Characteristic performance of inlets, exhausts and duct work may be found in Reference 2.

Ram air contains varying amounts of moisture at low altitude. This may cause problems due to condensation, if equipment is cold at high altitude and is then subjected to warm moist air at low altitude.

7.4 Cabin Air:

Most modern aircraft have an Environmental Control System (ECS) which provides heating, cooling and pressurization air for the cabin. A cabin exhaust flow exists which is available in full or in part for equipment cooling. Since this flow exists under controlled conditions of temperature and pressure, it is an ideal source for cooling, and probably the most widely used. As long as the normal ECS flow is sufficient to supply the equipment's requirements, there is no penalty to the aircraft.

Electronic equipment is generally located in the pressurized area although this is not a necessary requirement. Cabin exhaust air contains smoke and dust particles which may cause troublesome accumulations on the heat transfer surfaces. Filtering and restricting air flow to minimum requirements may be used as remedies.

Alternately, the ECS may furnish air to the equipment in a separate path in parallel to the cabin flow. Such air is sometimes furnished from a point in an air cycle ECS where temperature and pressure control are not as ideal as in the cabin. At lower altitudes up to approximately 24,000 ft (7.3 km), the delivery temperature is usually maintained above 35 °F (1.7 °C) to prevent freezing of the free moisture in the air. At higher altitudes, such control is usually not required because of the low ambient humidity. If minimum temperature control is not used at higher altitudes, sudden changes (90 °F) (50 °C), in delivered air temperature may occur.

In more recently developed equipment, the trend is to control the delivered air to within ±10 °C (±18 °F) of a selected value at all times.

7.5 Fuel:

At supersonic speed or in low level transonic flight, ram air is too hot to remove equipment heat. In this case, the aircraft's fuel supply can be used as a heat sink.

Generally, sensible fuel heat only is available since the engine fuel control system is not able to handle vaporized fuel. Maximum temperature rise of the fuel is limited by safety and onset of decomposition on hot exchange surfaces to a range of 158 to 252 °F (70 to 122 °C).

Because the fuel supply is a primary aircraft system and of a hazardous nature, heat is transferred to specially designed exchangers located in the fuel tanks or in existing fuel lines. The electronic equipment is cooled by an intermediate liquid transfer loop (see 7.6 and 12.4).

Peak heat sink capacity is not necessarily limited by engine fuel consumption, since excess fuel flow required for cooling may be returned to the tanks.

Fuel may serve as an "ultimate" sink or as a coolant. If the airplane mission includes a substantial part of subsonic flight, the wing tanks can reject fuel heat to the ambient air, thus making the air the final heat sink.

Whenever fuel is considered as a heat sink for electronics, it should be remembered that available capacity will likely be shared with other systems such as hydraulic or engine oil.

7.6 Transfer Fluids:

Liquid-cooled equipment is generally served by using a fluid which is an intermediate between the electronics and the ultimate heat sink (fluid is circulated between equipment and sink). Fluids are selected for one or more of the following qualities:

Pumping power per unit of cooling
Low temperature properties
Vapor pressure
Dielectric constant
Electrical insulation resistance
Temperature stability
Chemical stability
Fire resistance
Compatibility with component materials

In order to obtain good heat transfer at a low pumping power, the fluid must have high specific heat and a low Prandtl number ($c_p\mu/k$). There is no one fluid that excels in all desirable properties. Best heat transfer is obtained with water or anti-freeze solution, while synthetic fluids sacrifice some of this capability for a wider range of desirable properties.

8. STORED HEAT SINKS:

8.1 General:

Fluids may be carried on the aircraft, in addition to those available for normal operation, to handle temporary or peak heat loads; or more rarely, to do all of the electronics cooling.

The mass of the equipment itself may also be considered as a temporary sink. While this capacity is small, neglecting it can result in an oversized cooling system when high peak loads and short duty cycles exist.

Mass has sometimes been deliberately increased by the addition of melting substances. As Table 5 shows, about 60 Btu/lb (140 kJ/kg) are absorbed in the process of melting. Considerably more may be obtained in terms of heat per unit weight of sink, if suitable fluids are evaporated. the following liquids have been used and may be considered if circumstances warrant it.

8.2 Water:

Boiling water has been used to top off ram air cooling on air cycle refrigeration machines under high speed flight conditions. Water boilers may be used also for specific items of electronic equipment. Storage in capillary material is feasible for moderate quantities and provides self-regulation.

Water provides the highest practical latent heat per pound. Limitations are the high boiling point of 212 °F (100 °C) at sea level, and freezing with volume expansion. Freezing problems can be overcome by design or by providing heaters.

Corrosion problems can be critical if aluminum is used with water glycol solutions. Attention to the effects of dissimilar metals is required.

The utility of boiling water may be extended to low altitude if it is used as the heat sink in a mechanical refrigeration system (Section 9). Where efficiency is not a problem, such a system could simply be an ejector producing vacuum.

Evaporation of water into an air stream is another way to use its latent heat for cooling at temperatures below the normal boiling point of 212 °F (100 °C). The method is used in air cycle machines, where a part of the cooling is done by re-evaporating the water that has been condensed out of the bleed air.

8.3 Ammonia:

Liquid ammonia has been used primarily for the cooling of electronic equipment on unmanned flight vehicles. Ammonia provides refrigeration since its boiling point ranges downward from -28 °F (-33 °C) at sea level. Compared to water, stored weight per Btu is double and volume is four times. Since liquid ammonia must be stored under pressure, tankage weight must be considered, and is about 20% of the weight of the stored liquid.

Drawbacks of ammonia are toxicity (although the vapor makes itself known by odor long before concentrations are dangerous), nuisance to operating personnel, disposal and the fact that it attacks copper alloys.

8.4 Cryofluids:

Liquid nitrogen has been used on some commercial aircraft as a stored refrigerant in lieu of dry ice for purposes other than electronics cooling. Since it is now readily available, it may have application where low temperature must be maintained as in infra red sensor cooling or for special computer equipment. Helium and hydrogen are often available on rocket propelled vehicles and have cooling potential. However, their possible use is not within the scope of this AIR. Dry ice (solid CO) has an attractive capacity per lb of stored material, but efficient application is difficult.

8.5 Other Fluids:

The Freon refrigerants may be useful inspite of their low latent heat. They are rated non-flammable and near non-toxic. (Caution: Vapor in contact with open flame decomposes into toxic products; also avoid vapor accumulation in enclosed spaces). Freon 22 has a vapor pressure - temperature curve very similar to ammonia. Alcohols have attractive latent heat, but fire hazard and relatively high boiling point make them appear impractical.

9. REFRIGERATION:

9.1 General:

The aircraft's ECS includes mechanical refrigeration and some of the sinks in Section 8, inherently have this effect. There may also be cases where separate closed cycle refrigeration is specifically provided for electronics. The type used depends mainly on load and temperature difference as shown in Figure 4.

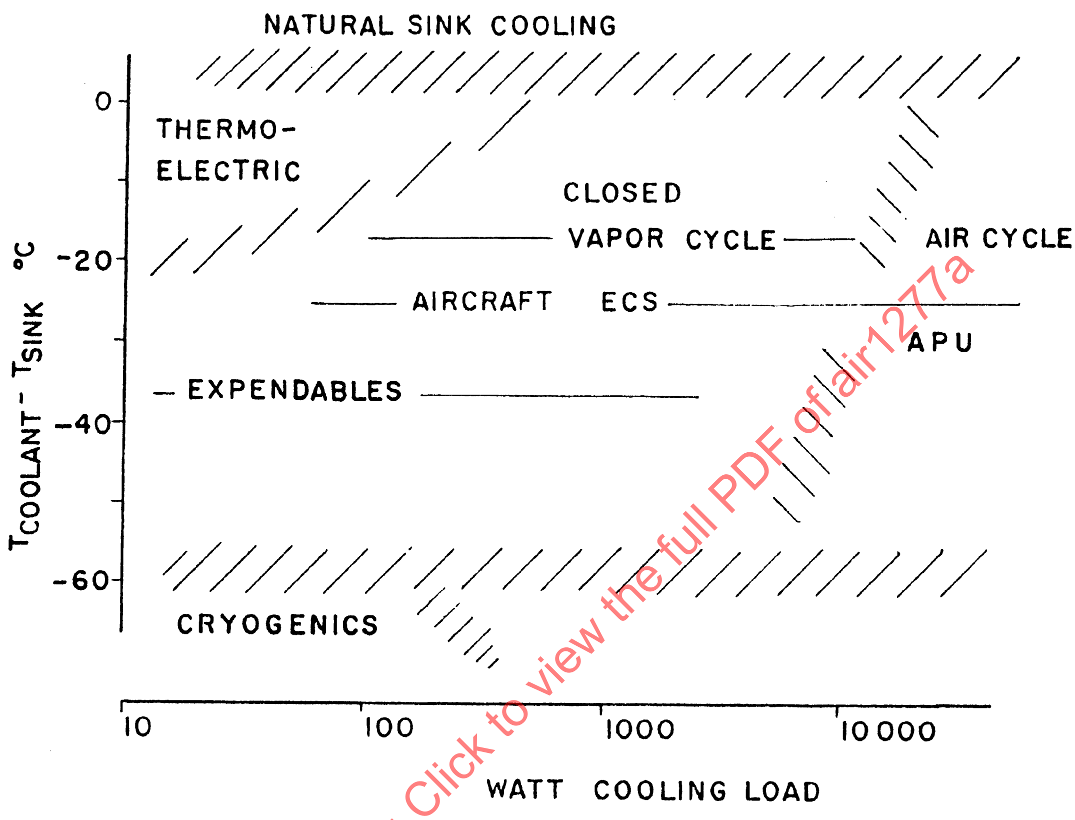


FIGURE 4 - Application Trend for Various Cooling Methods as a Function of Load and Temperature Level Below Sink

9.2 Thermo-electric Cooling:

Bismuth-Telluride semi-conductor devices are available which produce refrigeration, without moving parts, directly from the application of electric current. Efficiency is low (less than 10%) and substantial excess heat produced by the current input must be removed in addition to that produced by the cooled component. However, such devices may be justified for spot cooling inside an electronic assembly if it contains highly heat-sensitive parts with a relatively low share of the heat produced by the total assembly.

The cooling effect may be controlled by controlling current flow to the elements and may even be reversed to heating by reversing the current. It follows, that a source of direct current is required, and generally a special power supply must go with the application of thermo-electric cooling.

9.3 Mechanical Refrigeration:

For loads of a few hundred watt to several kilowatt, a vapor cycle system may find application for "add on's" or when an aircraft ECS is not available. Such use should be approached on a system basis considering trade-offs for power, weight, control and location. Location must consider access to electric or hydraulic drive power, and coolant or heat sink for the condenser.

If the demand for electronic equipment cooling is very large, a separate aircraft ECS using its own refrigeration machinery may be justifiable or even necessary. Such an approach simplifies the equipment supplier's problems and should be considered for large aircraft.

10. THERMAL CONTROL:

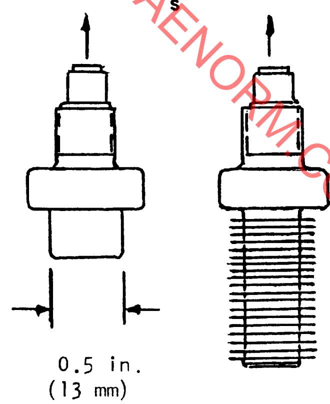
10.1 General:

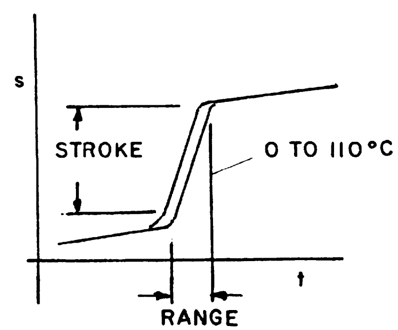
The objective of thermal control is to maintain the operating temperature of equipment within a specified band in spite of changes in duty cycle, environment and coolant supply temperature. When rapid changes occur, the automatic control system must follow, but it may exceed the steady state limits during such a transient.

Accurate control may be required to offset temperature sensitivity inherent in components, such as frequency drift of an oscillator, or a required level of reliability. Furthermore, its use may prevent excessive consumption of coolant and power. It is inherent in the function of a control that the system may have to heat or cool or do both. The environmental cabin control system of aircraft is a prime example of a thermal control system that encompasses a high level of refinement, since it interfaces directly with pressurization and human factors.

The theory and application of automatic control is the subject of an immense body of literature. This section must necessarily be limited to a few basic principles relevant to thermal control. A description of simple sensors and actuators which have found use in airborne thermal control will be found in 13.5 and Table 7.

TABLE 7 - Temperature Sensing and Control





Pellet Type Actuator

Heat operated combined sensor and power element using a wax fill.

Typical values:

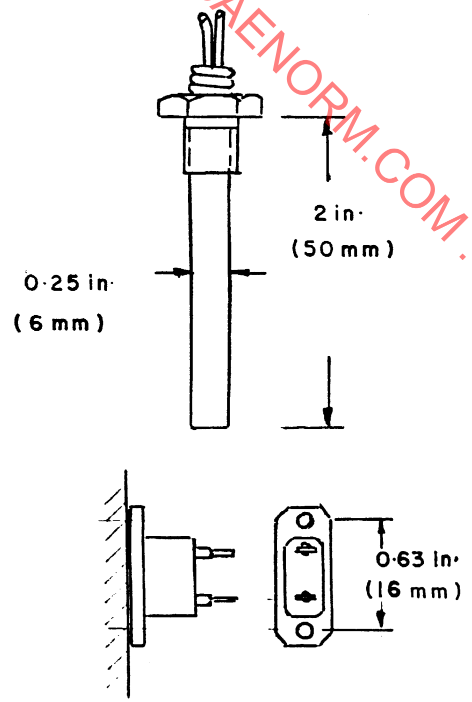
Stroke	(mm)	3	(0.1 in.)
Force	(kg)	10	(20 lb)
63% Response	(s)	22 to 50	
Differential	(°C)	5 to 10	(9 to 18 F)

Elements may be used in liquids or gases. Most commonly used for liquid bypass control.

Advantage: Simple and rugged.

Limitations: Fixed setpoint. Proximity of sensor and valve.

TABLE 7 (Continued)



Bimetal Type Thermostat

Tubular or disc type bimetallic sensor operating an electrical switch or for pneumatic applications a variable orifice.

Typical values:

Setpoint accuracy	(°C)	2 to 3	(3.5 to 5.5F)
Differential (°C)			
Slow make and break		0.5 to 1.5	(1 to 3 F)
Snap acting		2 to 15	(4 to 25F)
Current capacity (Amp)		1 to 7	
63% Response (s)		20 to 180	

Advantage: Freedom of location, may mount directly on component. Wide temperature range available.

Limitation: Generally fixed setpoint. Non-snap acting switches will have some bounce under vibration.

TABLE 7 (Continued)

Thermistor Type Sensor

Electrical semi-conductor with large temperature coefficient used in bridge circuit for on-off or proportional control.

Typical values:

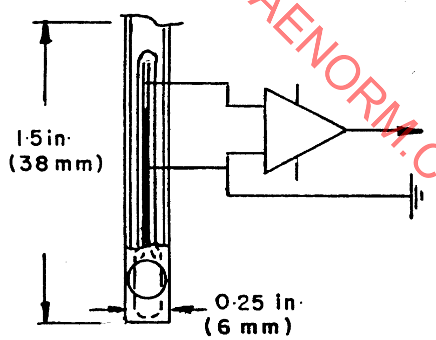
Nominal 77F, (25°C) resistance (Ohm). 100 to 100,000
Neg. temperature coefficient $\Delta R/_R$ 4%/°C (2.5%/F)
(Pos. " " 7%/°C)*, (4%/F)
63% Response time down to 2 sec. in still air at typical 0.7mW electrical load for 1 (°C) (1.8F) self-heating.

Advantage: Available in numerous forms for wide range of temperature.

Limitation: Requires electrical amplifiers and power supply.

*Doped Silicon temperature sensitive resistance elements are available from 10 to 10 kohm with a nearly linear resistance change of +0.7%/°C, -50 to 150°C (1.25%/F, -58 to 302F)

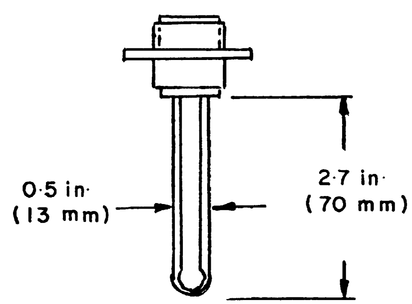
TABLE 7 (Continued)



Thermometer Type

Mercury column contact thermometer for on-off control. May have multiply contacts. Used direct or with (integral) solid state relay(s). More rugged in vibration than slow make and brake Bimetal type.

Typical values:	Setpoint accuracy (°C)	to ±0.5	(± 1F)
	Multiple Contact Band (°C)	> 5	(> 9F)
	Ambient Temp. Range (°C)	-40 to 285	(-40 to 545F)
	Vibration to 30g		
	Shock to 100g		



Resistance Thermometer

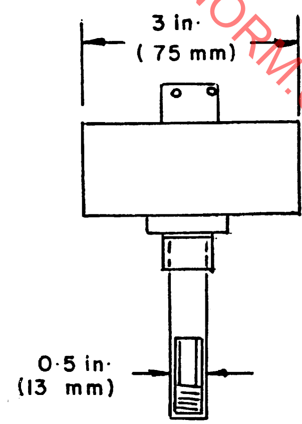
Wirewound sensor in suitable housing: used with electrical resistance bridge and amplifier.

Typical values:		
Nominal (25C, 77F) resistance		100 to 500 Ohm
Temperature Coefficient.	Platinum wire	0.39%/C (0.22%/F)
	Nickel wire	0.65%/C (0.36%/F)
Response time (63% of final) depending on construction and air flow rate.		0.2 to 20 sec.

Advantages: Stable. Can be used over wide range of temperature.

Limitation: Requires electrical amplifier.

TABLE 7 (Continued)



Cooling Effect Sensor

When mounted in a cooling air duct it senses when the combination of weight flow and temperature of cooling air is in the "adequate" or "inadequate" region.

The signal is used to adjust airflow or temperature to the adequate cooling region.

Advantage: Limits cooling air flow to required minimum.

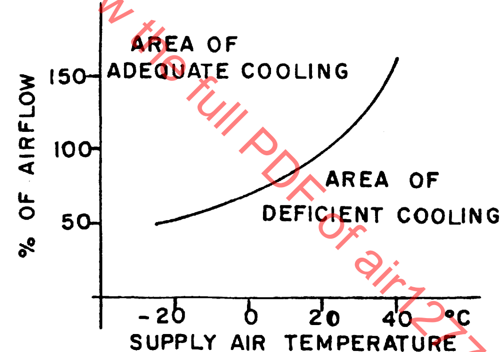
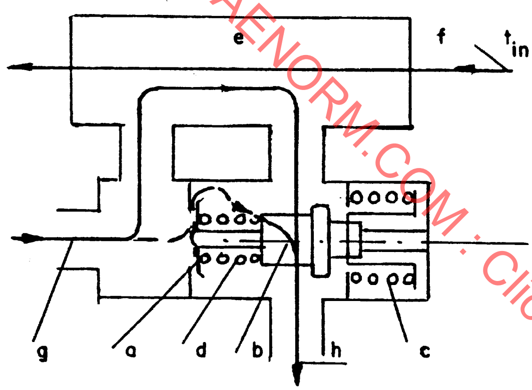
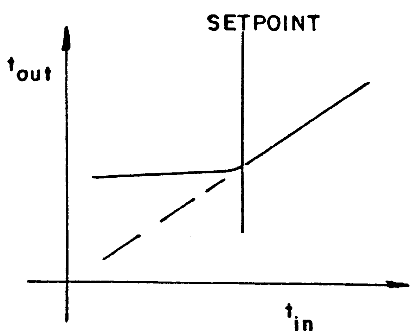


TABLE 7 (Continued)





Bypass Type Cooling Control

a. Poppet Valve
b. Pellet Element
c. Return Spring
d. Override and Pressure Relief
e. Heat Exchanger
f. Coolant inlet at t_{in}
g. Cooled fluid inlet
h. Cooled fluid out at t_{out}

Action:

If outflow temperature is above operating range pellet element closes valve against spring.

As outflow temperature falls through operating range spring returns element plunger and opens valve so that most fluid can bypass heat exchanger. Element must sense mixed outflow temperature. Complete valve elements with override and pressure relief features are commercially available.

Upper temperature limit for pellet type actuator ~ 177 °C (350F). Normal range full open to full closed 10 to 15 °C. (18 to 27F).

10.2 Level of Control:

The type and extent of controlled cooling depends on the sensitivity of the equipment to the thermal environment and on the way it is installed in the aircraft.

An "open loop" system as shown in Figure 5 where the air is supplied within narrow temperature limits, is metered essentially in parallel to single equipment, and dumped overboard after doing the cooling, requires a minimum of control effort. Supplying an adequate amount of coolant and distributing it correctly is all that is required. Target values are given in 6.8, but resulting temperature rise must be investigated before the total required flow is determined.

If equipment is installed in a "closed loop" system where coolant recirculates, such as in a remote equipment bay (Figure 6), thermal time constants and duty cycle of the various equipments must be considered to assure stable operating conditions. In a complex case, analysis by use of a thermal model may serve to uncover hidden problems.

10.3 Design Approach:

Detailed control system design is not within the scope of this AIR and therefore discussion is limited to some basic concepts.

Controls may be applied to coolant temperature, flow or pressure, or directly to the temperature of certain components. Applying control specifically to single equipment adds cost and complexity. If control can be derived adequately from other systems already on board, such as the cabin control system, this is preferred. When separate, specific controls must be used, tolerances should not be specified closer than needed. A more complex system would result, being more subject to (potentially dangerous) instability.

10.4 Limiting Coolant Flow:

Controlling the rate of coolant flow is common for ram air and it may occur either at the air intake or the air exhaust.

Figure 5 shows a type of control which holds coolant flow nearly constant over a wide range of flight altitude. Cabin exhaust air discharges through a venturi. At low altitudes flow is subsonic in the venturi and is approximately proportional to venturi exit area times the square root of the differential pressure between cabin and ambient. When this differential becomes 0.528 times cabin pressure or larger the flow in the throat of the venturi becomes sonic and is now proportional to throat area times cabin absolute pressure. This occurs at about 20,000 ft (6.1 km). Since cabin pressure is regulated, flow above 20,000 ft (6.1 km) is fairly constant. At lower altitude the flow will vary but its mean can be controlled by selecting throat to exit area ratio of the venturi.

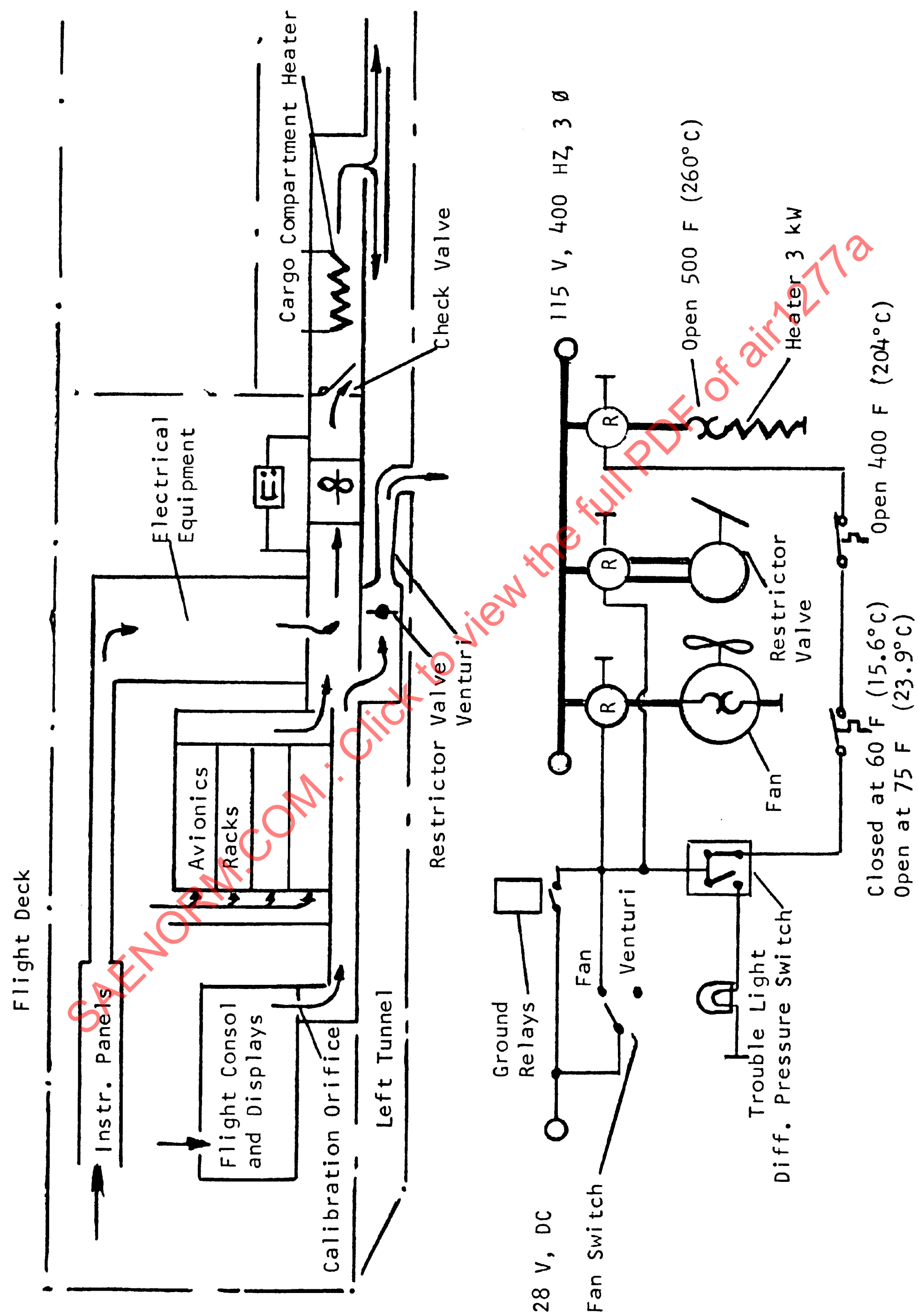


FIGURE 5 - Air Cooled System, Commercial Transport

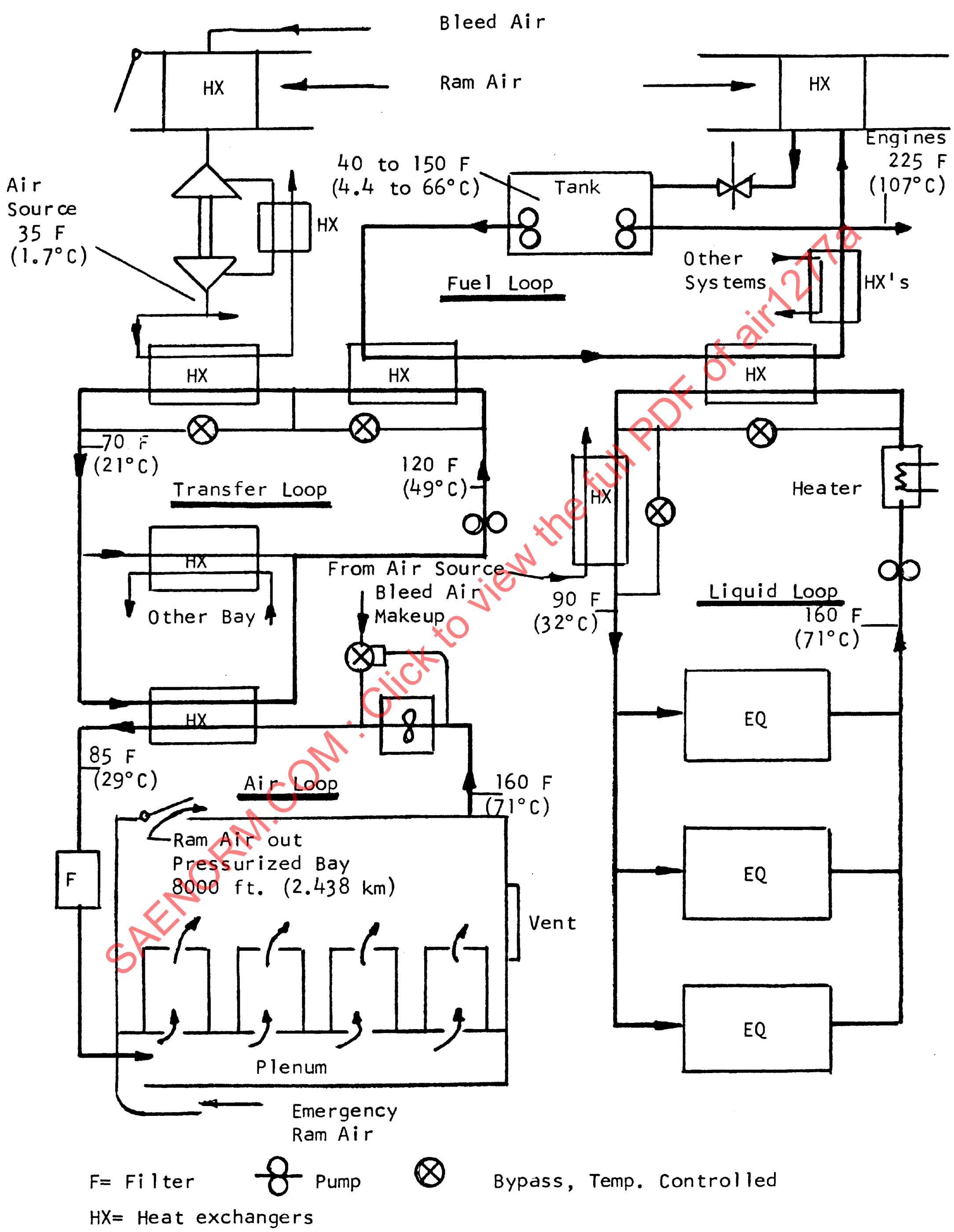


FIGURE 6 - Multiloop Cooling

10.4 (Continued):

With ram air, or air coming directly from an air cycle machine, the rate of flow may be adjusted downward as inlet temperature or load decreases. This may be done in two ways. One is to sense air exhaust temperature and maintain it between specified limits by varying flow. The sensor may be located judiciously in the air exhaust or near critical equipment. The other way is to use a "cooling effect" system. This system employs a resistance heating element and control circuit to maintain a constant heat transfer rate. The system can adjust coolant temperature and/or flow rate, however, control of air flow as a function of air temperature is the most common mode. As cooling air temperature increases or decreases, the sensor transmits a signal for increased or decreased flow to compensate for the temperature change. This type of control for aircraft applications is very attractive because it controls flow to the minimum required to satisfy a given cooling load. It, therefore, minimizes aircraft performance penalties (See Table 7).

Bypass of coolant flow may be employed at the heat exchangers of individual equipment or in a system. Figure 6 gives an example of bypass control around the coolant exchangers of the loop.

10.5 Temperature Regulation:

Temperature regulation implies control within narrow limits at all times of the supply temperature of coolant in a system or the actual temperature of a selected component.

An example of a closely controlled supply is the cabin temperature control system of commercial aircraft as applied to equipment conditioning. An example of direct control of component temperature could be the control of the current to the thermo-electric modules in a spot cooling application.

10.6 Stability:

Temperature regulation implies that the sensor is linked to the controlling element in a "closed loop". This is not possible without a time delay, making the control lag behind the original disturbance. The system may then become unstable, or "hunt" around a setpoint without ever finding it. Reference should be made to applicable literature for details. The following general observations may be made:

The tendency to oscillations or instability increases with increased demands on control accuracy (gain). It also increases with the number of elements which introduce time lags. Instability will not occur if one time lag can be made dominant without slowing system response beyond acceptable limits.

Sophisticated electronic type controls may be justified in order to manipulate speed of response, often by using multiple inputs in a functional relationship (logic).

11. APPLICATION OF COOLING TO EQUIPMENT:

11.1 Transfer Inside Equipment:

The "Draw Through" method of cooling relies on direct convection transfer inside the "Black Box". Today's circuit board assemblies may well be cooled in this way if suitable protection against moisture is applied. At high load density additional cooling can be provided by convection along the board to the retainers, to the board edge and the mounting structure to a special heat transfer surface or the case. Figure 7(a) illustrates methods for bringing heat out from circuit cards in this manner. Large components may be mounted directly on a "cold plate", as shown in Figure 7(b), which is a form of solid to fluid heat exchanger. Figure 7(c) illustrates a way to conduct heat from IC's along the cards by mounting the IC's on "shunt rails" of copper on the multi-layer cards.

Figure 7(d) illustrates a card assembly with cold plates which are an integral part of the assembly housing.

Accurate prediction of the temperature change along the conductive path can be quite difficult and may require a simulated network analysis.

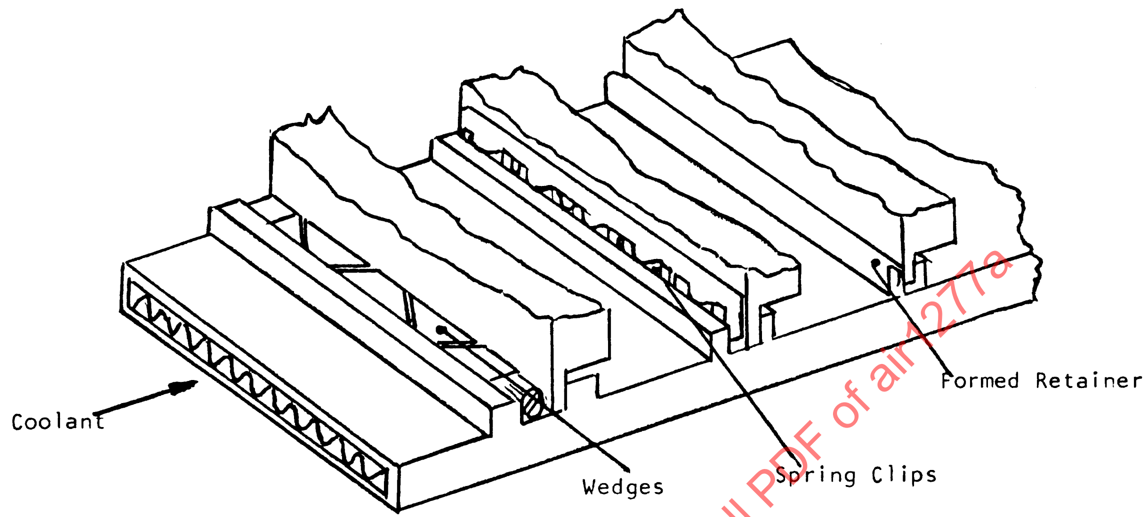
Whenever joints occur along the path of heat flow, an additional heat resistance is present. Depending on practical requirements, spring loading, wedging or heat conductive pastes can be used to bridge the air gaps and minimize joint resistance.

Large equipment having a variety of components may benefit from combining the conductive transfer with internal circulation of air or some other gas. This approach will only be effective if the equipment is pressurized to near sea level or above, either by sealing or by make up air. The cold plate used for conduction will then be expanded into a two-fluid heat exchanger as shown in Figure 8. Maintainability and reliability are affected by this approach, and for these reasons, some users may not approve of it.

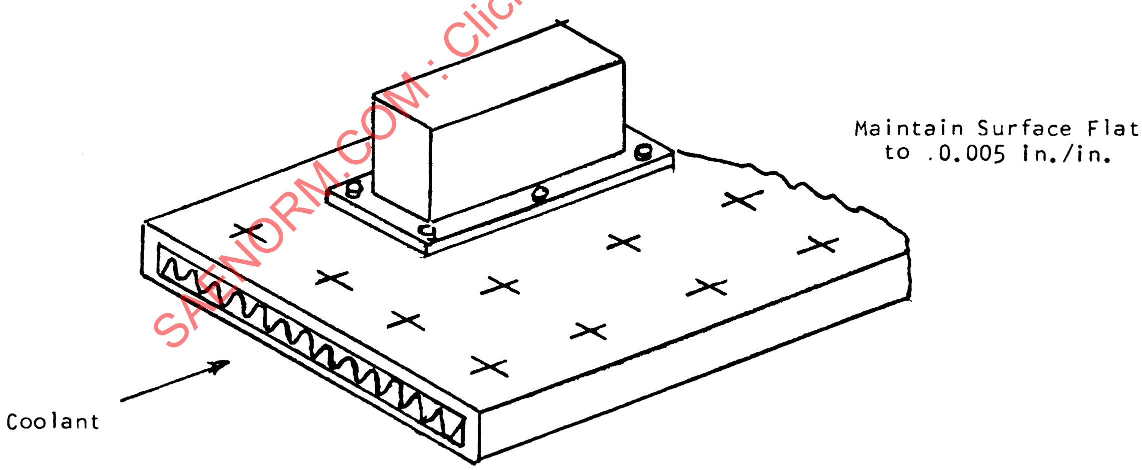
11.2 Transfer to the Coolant:

The simplest method of cooling equipment is to arrange the various "black boxes" on shelves, in racks or panels with spacing sufficient to permit free circulation of compartment air around them. Air circulates because of the natural draft produced by the temperature difference between the air and the equipment case. The equipment case may have widely spaced fins (transistor heat sinks) to increase the area. Data to estimate the effect of this cooling surface are given in Table 3.

Natural convection cooling provides a rather low heat flux, and even this may not exist if the installation impedes air circulation or adjacent equipment raises ambient levels. If this method is applied, actual in-use conditions must be considered.

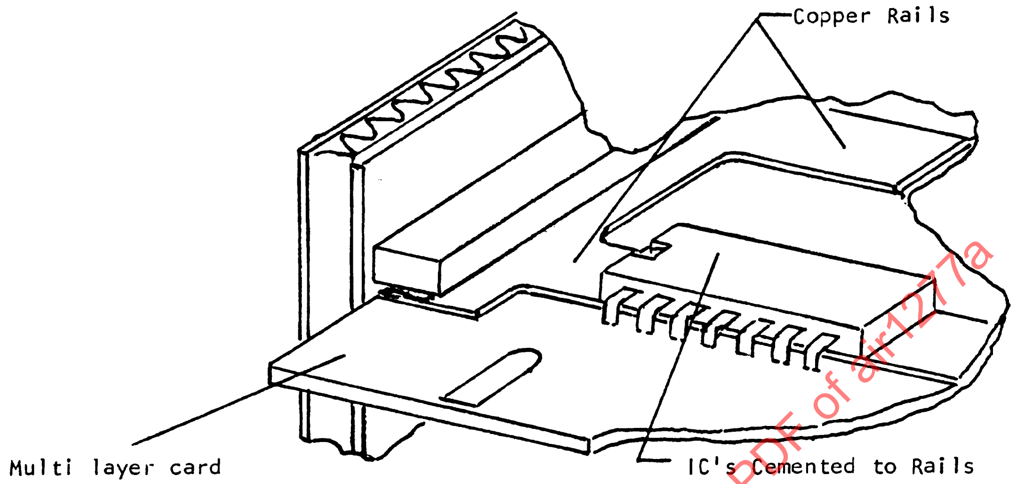


(a) Retaining Circuit Modules with Thermal Conduction

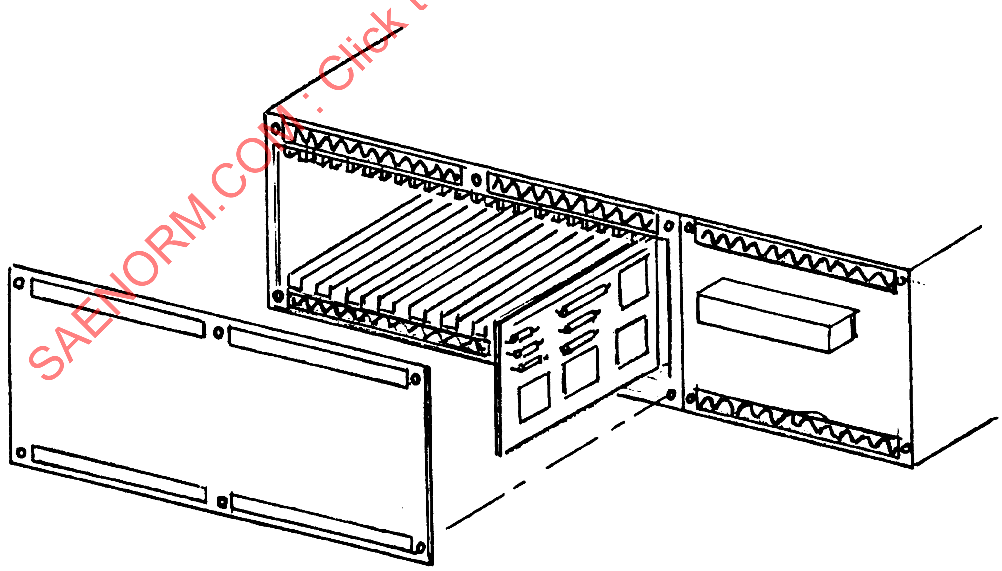


(b) Module mounted on Bolt Grid

FIGURE 7



(c) Thermal Shunt Rail



(d) Air Cooled L R U

FIGURE 7 (Continued)

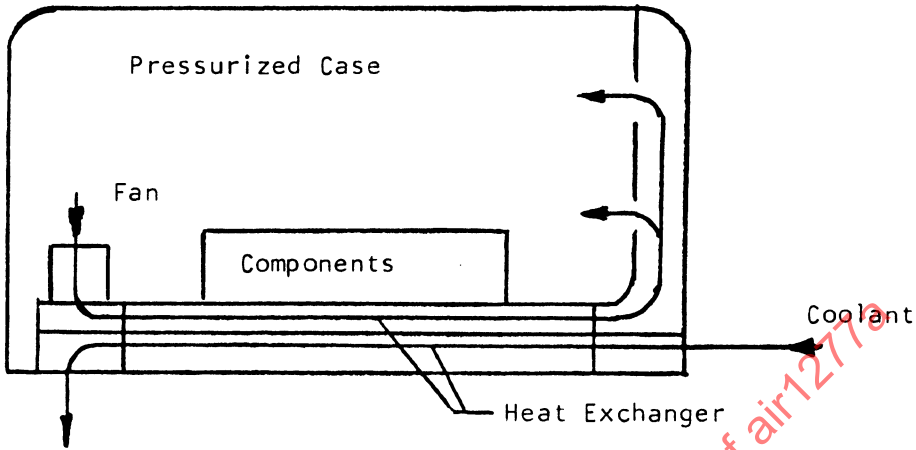


FIGURE 8 - Unit with Internal Circulation

11.3 Forced Circulation:

Controlling the airflow by external means is a logical step to achieve increased heat transfer. Figure 9 illustrates an instrument panel placed against a wall (or shroud). A fan blows air into the space between the wall and panel with simple precautions taken to control flow direction in a desired manner. A heat flux of 3 to 10 times that obtained by natural convection is possible. Section 15 illustrates the use of the formulae given in this publication to estimate the ventilation effect and verifies the gain in heat transfer capacity.

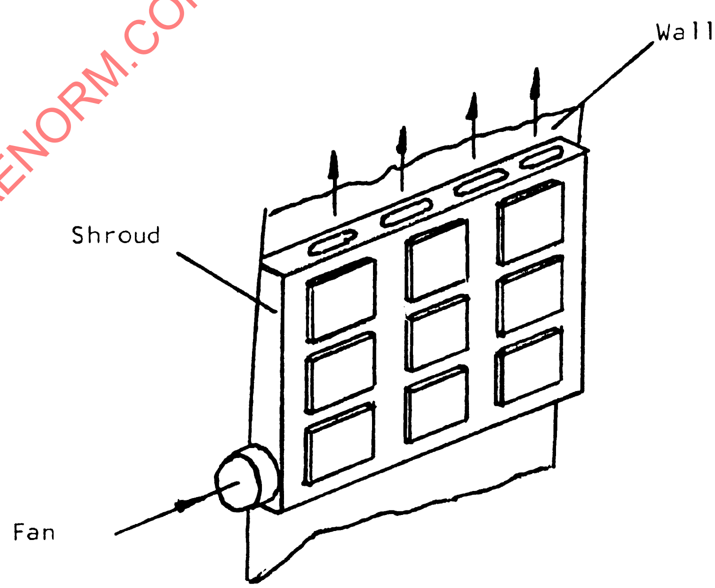


FIGURE 9 - Display Panel with Cooling

11.4 Heat Exchange Surfaces:

Equipment mounted in racks may be cooled by merely flowing air over the external surfaces of the boxes. The arrangement has sometimes been called "Area Cooling" and is very similar to the ventilated panel discussed in 11.3. Without increasing the outside area by fin extensions (extended surface), the cooling effect may be rather limited. Specifically designed surfaces for heat transfer outside or inside the case with controlled airflow are a better means of obtaining increased cooling with forced flow.

Such devices are generally known as heat exchangers when heat is transferred between two (or more) fluids, and as cold plates or heat sinks if the exchange is between hot components and a cooling fluid. Characteristics of interest are discussed in Section 13.

11.5 Special Cooling Arrangements:

11.5.1 Liquid Cooling: As seen in Table 4, thermal conductivity and density are much higher for liquids than for air. This difference promotes higher coefficients of transfer for a given amount of power. Rate of flow and size of piping to bring the flow to the equipment are also reduced, making liquid cooling a more favorable or even necessary method of cooling concentrated or remote loads. On the other hand, the liquid requires circulation in a return line, and is subject to loss by leakage. The equipment to be cooled must generally include heat exchangers which are specifically designed for this service and the transfer inside the modules must match the higher rate of heat flux.

11.5.2 Phase Change: Replacing conduction, or a circulating fluid, with evaporation and condensation of a liquid, results in the highest possible heat transfer. Figure 10 illustrates this principle with a power transformer. The case has a liquid cooled heat exchanger and is partially filled with a suitable fluid into which the hot core is submerged. The liquid boils off the hot surface and vapor is condensed again at the exchanger. Liquid returns by gravity. Capillary material may be used to contain and transport the liquid.

To be useful for such an application, the liquid must not only have a high latent heat, but also a manageable vapor pressure at the desired boiling temperature and acceptable insulation and dielectric properties. (See Table 5 and also Reference 8 and Literature 1.)

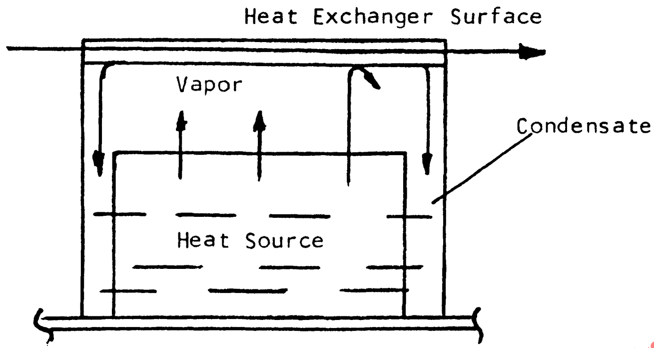


FIGURE 10

11.5.3 Heat Pipe: A logical extension of the boiling-condensing concept is the device which is now known as a "Heat Pipe".

A heat pipe is a device for conducting heat away from a source to a remote heat sink at a rate much higher than could be obtained by conduction in a metal path of similar dimensions.

The heat pipe shown in Figure 11 consists of a closed channel or tubing, which is filled with a measured charge of working fluid. The pipe also contains some form of capillary material or wicking, usually attached to the inner wall. Heat from the device to be cooled is transferred by conventional external means to one end of the pipe and evaporates the fluid charge from the wicking. Vapor pressure forces the resulting vapor through the tubing, and as it reaches the other end, the vapor is condensed by external cooling. The resulting liquid returns through the capillary material under the driving potential which is set up by the evaporation. Thus, a continuous cycle is maintained.

Since the transport of heat is done by phase change, involving the latent heat of vaporization, actual material movement is small compared to the heat flux carried, and the transport occurs nearly isothermal. For instance, if water is considered as the charge, it could transport about 1000 Btu/min for every lb/min (2326 kJ/kg) of water flow. Thus, a heat pipe may carry many times the heat that would be practical with a metal rod of equal dimensions.

Working fluids may range from cryogenics to liquid metals covering a wide range of temperatures with manageable working pressures. Metal screens, metal cloth, ceramic felt material and sintered powder have been used as capillary materials.

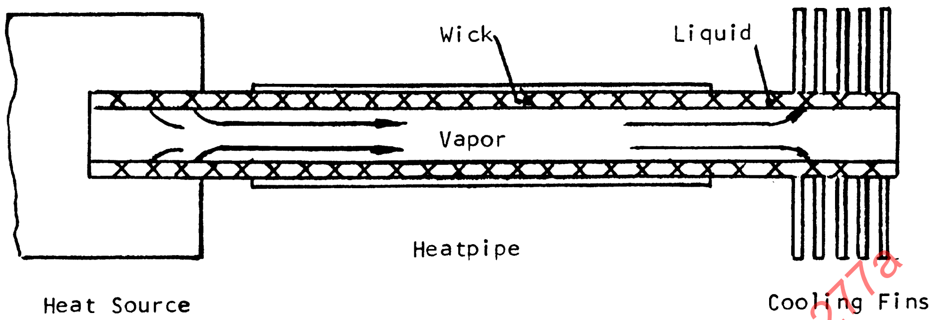


FIGURE 11

11.5.3 (Continued):

Heat pipes of several feet in length can be practical. In this form, they represent a very simple and very efficient transfer loop, useful in reaching difficult heat sources.

One should keep in mind that the problem of entering and removing the heat flux may severely limit the capability of such an installation. A study of applicable literature and consultation with manufacturers will aid in deciding how and where to use such a device, and if special features of the next sub-paragraphs are applicable.

11.5.3.1 Variable Conductance Heat Pipe (VCHP): This is a variation of the basic concept which varies conductance in response to heat load in order to achieve thermostatic control of the source temperature over a wide load range, without use of active (external) control elements. A non-condensible gas is introduced in the pipe and the condensing end is modified to achieve this effect. For details refer to Reference 9.

11.5.3.2 Diode Heat Pipe: As the name implies this variation has high conductance for one flow direction, but minimizes conduction if the temperature gradient along the pipe reverses. Conductance ratios in excess of 500:1 have been achieved. A variety of techniques may be used to achieve this feature, one of the most common being wick dry out. Details may be found in References 10 and 11.

11.6 Cooling on the Ground:

11.6.1 General: On the ground with engines off, equipment must still be operated for checkout or to maintain communication.

Available cooling capacity is often less than in flight. Therefore, the use and operating time of equipment should be carefully established. Equipment thermal time constants are of importance under such restrictions and should be known, to determine the extent of cooling required.

11.6.2 Means for Ground Cooling: Cooling may be supplied by one or more of the following:

a) Auxiliary fan (internal)
b) External air supply (may include refrigeration)
c) Aircraft fuel for a limited time (1 hour)
d) Other stored coolants
e) Onboard APU operating the ECS

Use of ambient air for cooling must consider that delivered temperatures will reach 125 °F (52 °C) due to sun load. Under such conditions, capacity is greatly restricted.

12. COOLING SYSTEMS APPLICATIONS:

12.1 Air Cooled Equipment on a Commercial Transport:

Figure 5 shows the arrangement of cooling with cabin exhaust air on a modern commercial jet aircraft and its integration with other needs of the aircraft. Besides cooling the electronic and electrical equipment, this system provides ventilation for the flight deck and can provide ventilation and heating for the cargo compartment as may be required for live cargo. Because of the ventilation flow, smoke produced by the equipment in case of a malfunction cannot penetrate into the cockpit area.

Equipment is located in consoles and racks, properly shrouded and ducted to provide the desired distribution of air flow. Any mode of air cooling can be employed for the electronic modules as long as pressure drops are matched between units. Calibration orifices establish the proper distribution between consoles and the electrical equipment section.

A study of the wiring schematic reveals interlocks and safety devices which prevent the continuation of a dangerous condition resulting from malfunction.

12.1 (Continued):

The system operates as follows:

When the airplane is pressurized, the electrical and electronic equipment cooling function is automatic. The size of vents, ducts and the venturi ensures sufficient airflow. The normal flow of air through the venturi is approximately 20 lb/min (0.15 kg/s) with the fan off. If cargo compartment heating is required in flight, the fan is turned on, and the venturi restrictor valve closes partially to limit flow through the discharge venturi to 5 lb/min (0.04 kg/s). If the cargo compartment thermostat is below 60 °F (15.6 °C), and the heater element temperature below maximum operating limits, the cargo compartment heater will turn on. Heated air is now directed through the diffuser, through the double floor of the cargo compartment, and exhausted to the right utility tunnel. The air flows with the passenger compartment exhaust air to be vented overboard through the outflow valve.

When the airplane is on the ground, and electrical power is available, the fan and the venturi restrictor valve circuits are energized regardless of the switch position. With the control switch in the venturi position, the fan relay circuit is completed through the ground control relay contacts. When the fan relay contacts close, the fan operates and the venturi restrictor valve is driven closed.

If a minimum pressure differential is not sensed across the fan, the differential pressure switch contacts remain closed and the "Fan Off" indicating light on the annunciator panel will come on. This light warns the operator that insufficient cooling air is flowing through the electrical and electronic equipment racks. If the airplane takes off with the switch in the venturi position, then the fan relay is de-energized by the ground control mechanism. The fan then stops and the venturi restrictor valve is driven open.

When the switch is in the fan position, whether the airplane is in flight or on the ground, the fan relay will always be energized, and the pressure switch and the warning light circuits will be armed through the contacts of the switch.

12.2 Air Cooled System on an Unmanned Flight Vehicle:

Figure 12 is a schematic of a cooling system which serves air cooled equipment by circulating air and cooling it with the use of an expendable heat sink. The sink used in this case is liquid ammonia stored in a pressure vessel on board the aircraft. Such a system has the advantage of providing a known total amount of cooling for a given mission with minimal requirements for mechanical cooling equipment. Operation is straight forward. Air is circulated by a fan through distribution ducting to the various black boxes and returns through the body of the aircraft into a heat exchanger and into the fan. Ammonia is supplied from the tank under its own vapor pressure to a flow metering system into the exchanger where it converts to vapor as it absorbs the heat rejected from the electronics and the structure. Vapor is vented overboard. It is quite practical to use a simple, fast cycling (1 to 15 second) on-off control system if the sensor is located in the distribution duct where air velocity is appreciable and the temperature is uniform. Maximum flow is restricted by the control to avoid spilling liquid coolant on initial cool-down.

Figure 12 also shows a heating system to assist in warm-up from a cold soaked condition. The heater is cycled at a lower temperature level than the cooler to avoid interference. If extended ground operation is required an external coolant supply can be added. Water cooling is feasible, without freezing problems because the ammonia dissolves in water. Care must be taken however, to use compatible materials (no nickel plating in contact with the mixture).

As in any air cooled system, static pressure in the recirculation loop affects capacity. This must be considered in the design.

Moisture in the air (initially present or introduced in flight) will condense on the heat exchanger surfaces and will form frost even when the ammonia is exhausted at sea level pressure, (-28 °F (-33 °C) boiling point) until the dew point is sufficiently lowered. If the frost is troublesome, it can be prevented by keeping the coolant pressure in the exchanger sufficiently high, at the expense of a larger heat exchanger. Filling of the supply tank is best done off the aircraft where it can be controlled by weighing. This assures that sufficient ullage is present for thermal expansion of the liquid. All required plumbing should be designed to avoid a potentially dangerous hydraulic lockup.

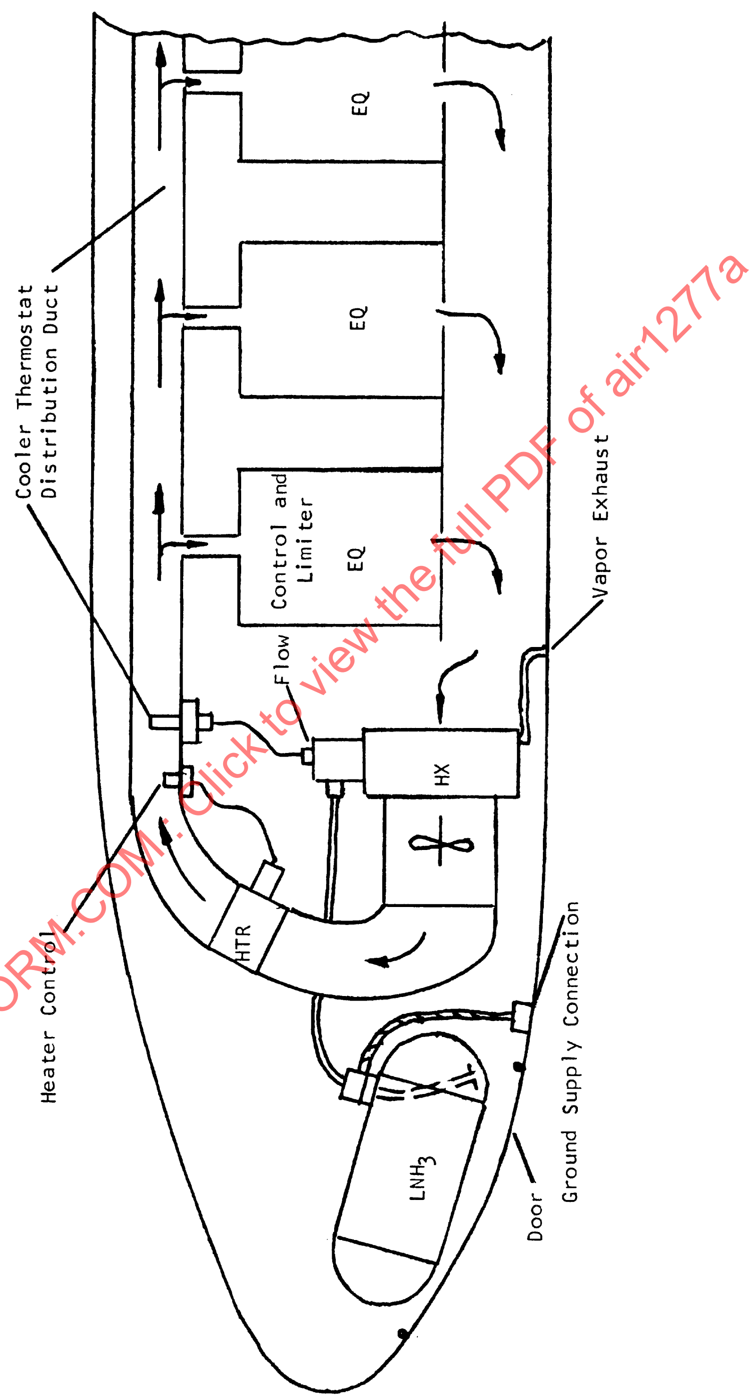


FIGURE 12 - Evaporative Cooling System

12.3 Radar Cooling Loop:

The use of a liquid cooling loop is shown in Figure 13, assuming that a high power radar must be cooled where the transfer rates obtainable with air would be inadequate.

The loop consists of a pump package and associated control and safety devices, a heat exchanger which transfers the rejected heat to air coming from a refrigeration package, tubing with quick disconnects and the radar set.

Operation is as follows:

Fluid is pumped from the reservoir through the supply line to the radar set. It passes through a traveling wave tube, cold plates which serve the switching circuits and then through a liquid filled high voltage power supply. The fluid then returns to the heat exchanger, through a filter to the reservoir. A valve bypasses fluid around the exchanger when the fluid temperature in the loop falls below the level requiring cooling.

Values for temperature and pressure levels are shown which are typical of an actual installation.

The system is completely sealed and fluid expansion with temperature is made possible by use of a "bootstrap" expansion piston which keeps a positive pressure on the suction side of the pump.

Entrapped air is a problem in a sealed system and it may be difficult to remove during the fill process. Air vents at top locations and design of components and plumbing which minimizes closed voids are used to overcome the entrapment problem.

Quick disconnects are shown which permit the removal of serviceable units from the aircraft plumbing without loss of fluid. The units must have fluid vents to prevent hydraulic lockup and over-pressure when detached from the loop's expansion provisions.

Disconnects and sliding seals are possible with the normal hydraulic type transfer fluids. Dielectric fluids like FC75 may be troublesome, since the low surface tension of these fluids makes substantial leakage possible at pin holes.

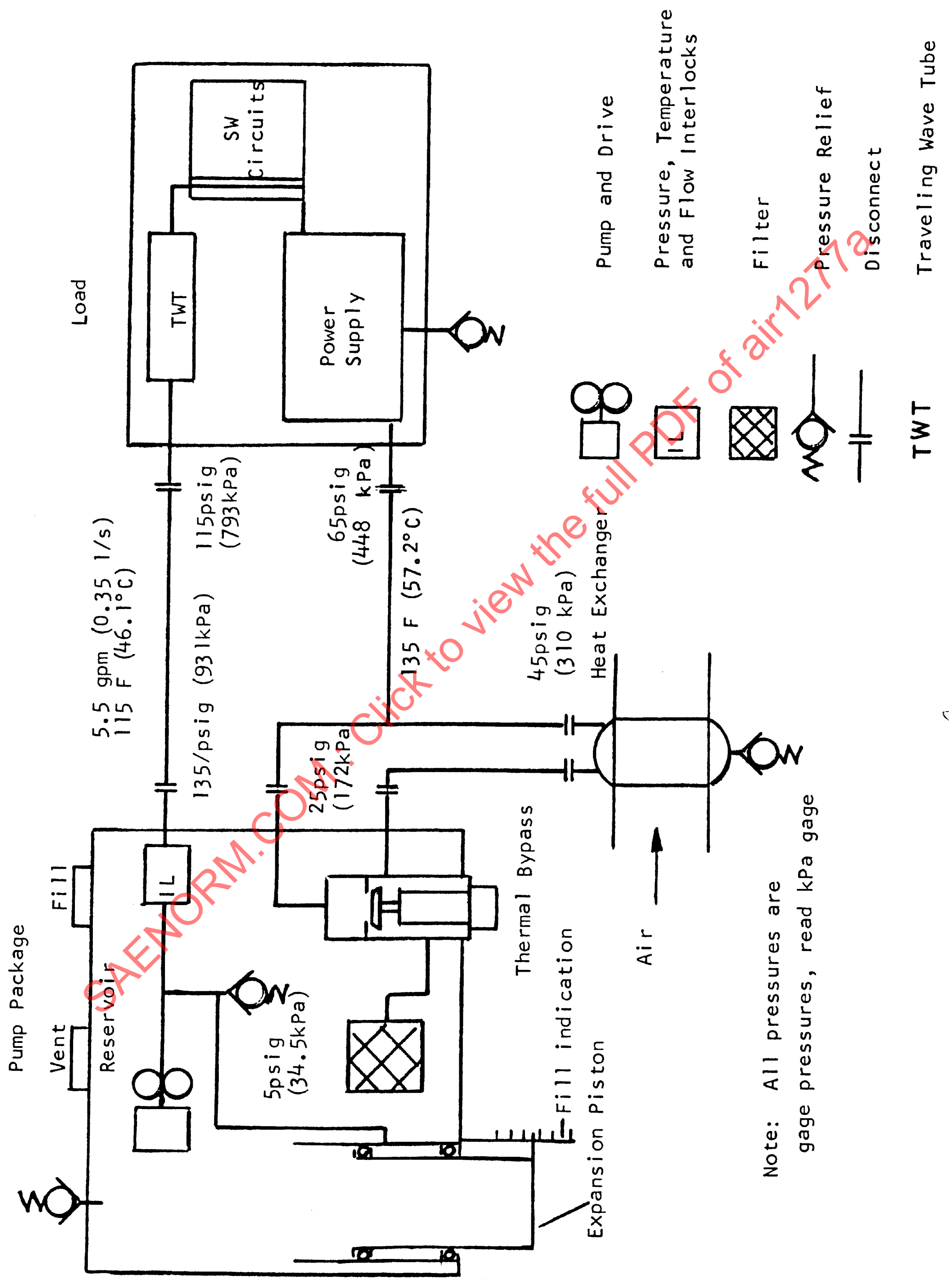


FIGURE 13 - Radar Cooling Loop

12.4 Multi-Loop Cooling System:

Figure 6 illustrates a system intended for use in a large (supersonic) aircraft with extensive equipment cooling requirements.

Heat sinks are provided by bleed air operated air cycle machines, fuel, and ram air for cooling under emergency conditions. The sinks are complementary during flight, with fuel being available at temperatures from 45 to 120 °F (7.2 to 48.9 °C) during about 60 to 70% of the flight time, and air cycle cooling taking over for the remainder.

An intermediate loop is used to transfer liquid coolant at 70 °F (21 °C) to several bays housing air-cooled avionics. Each bay is pressurized by bleed air to obtain efficient air cooling, and it includes an air circulation loop with fan, filter and heat exchanger. Ram air can be admitted for emergency cooling, should the normal system fail. Delivered air temperature is high enough to avoid problems with condensation of moisture. A separate coolant loop is provided to serve liquid-cooled equipment.

Supply temperatures are controlled by means of bypassing heat exchangers as necessary and by heating provisions to speed up cold start of liquid-cooled units. In either case, the units are supplied with a coolant which has constant rate of flow and temperature and is free of contamination. The system need only be balanced for pressure drop to obtain the desired share of cooling capacity for each equipment item.

12.5 Comparative Temperature Differences:

The systems shown in this section may be thermally compared by inspecting the temperature differences which occur along the path of heat transfer. This is shown in Figure 14 using for illustration a power transistor as the "equipment".

Figure 14a refers to the cooling system of 12.1. Two different methods of heat sinking the air cooled equipment are assumed for comparison. In the first, the transistor is mounted directly to a finned, forced cooled sink, which can be mounted through electrical insulation if required. In the second, the transistor is mounted to a cold plate heat sink and is electrically insulated by a mica washer. In spite of the extra drop through the washer, the overall drop is less because the cold plate is more efficient, though it is less voluminous. This is accomplished however, at the expense of added pressure loss in the cooling air flow.

Figure 14b shows conditions as they might exist in the air-cooled, liquid transfer loop of Figure 13. Addition of the extra liquid to air exchanger consumes about 14 °C temperature drop, but only 10 °C is reflected in the load, because the liquid-cooled cold plate will be more efficient than an air-cooled plate.

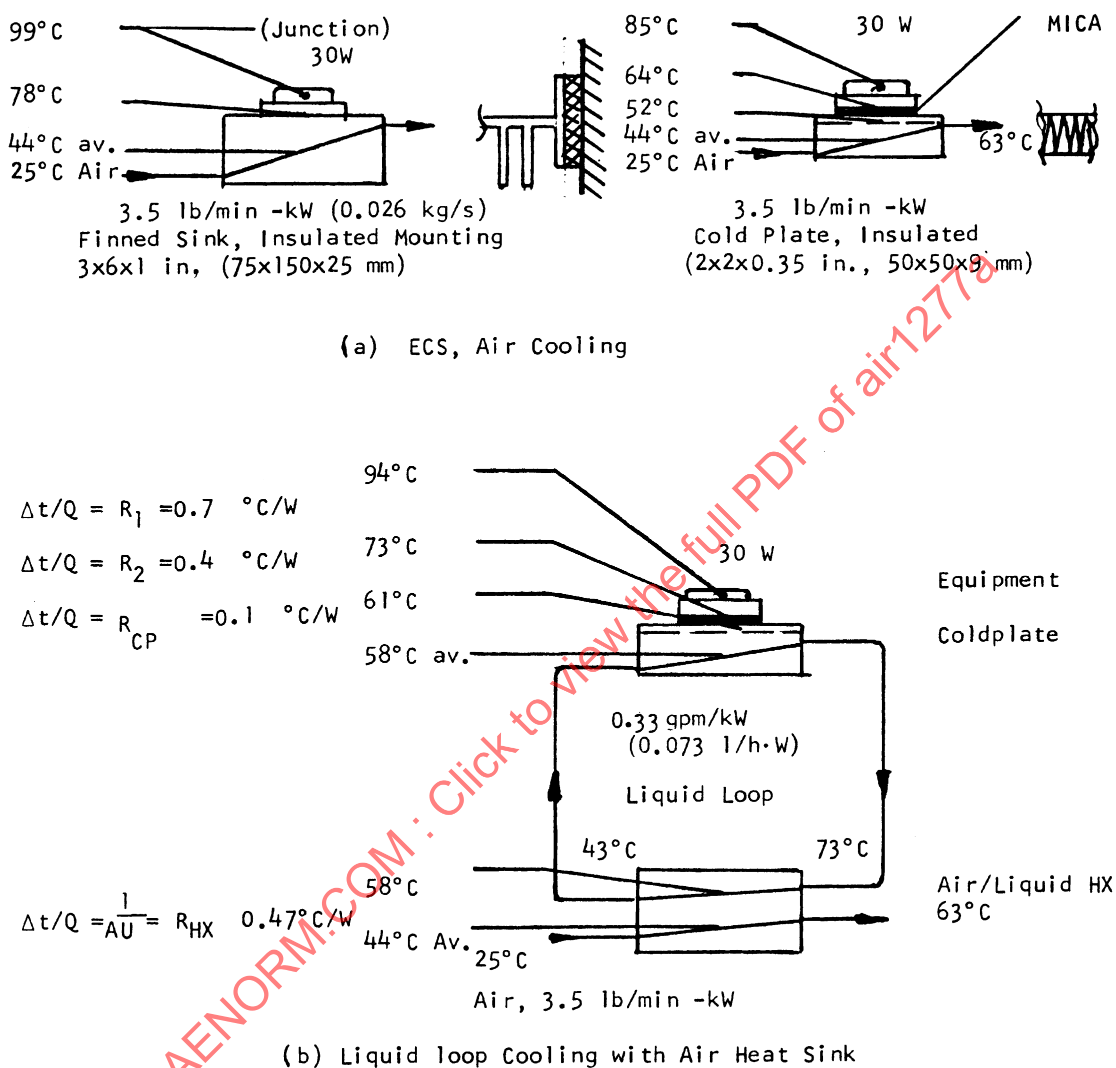


FIGURE 14 - Temperature Patterns

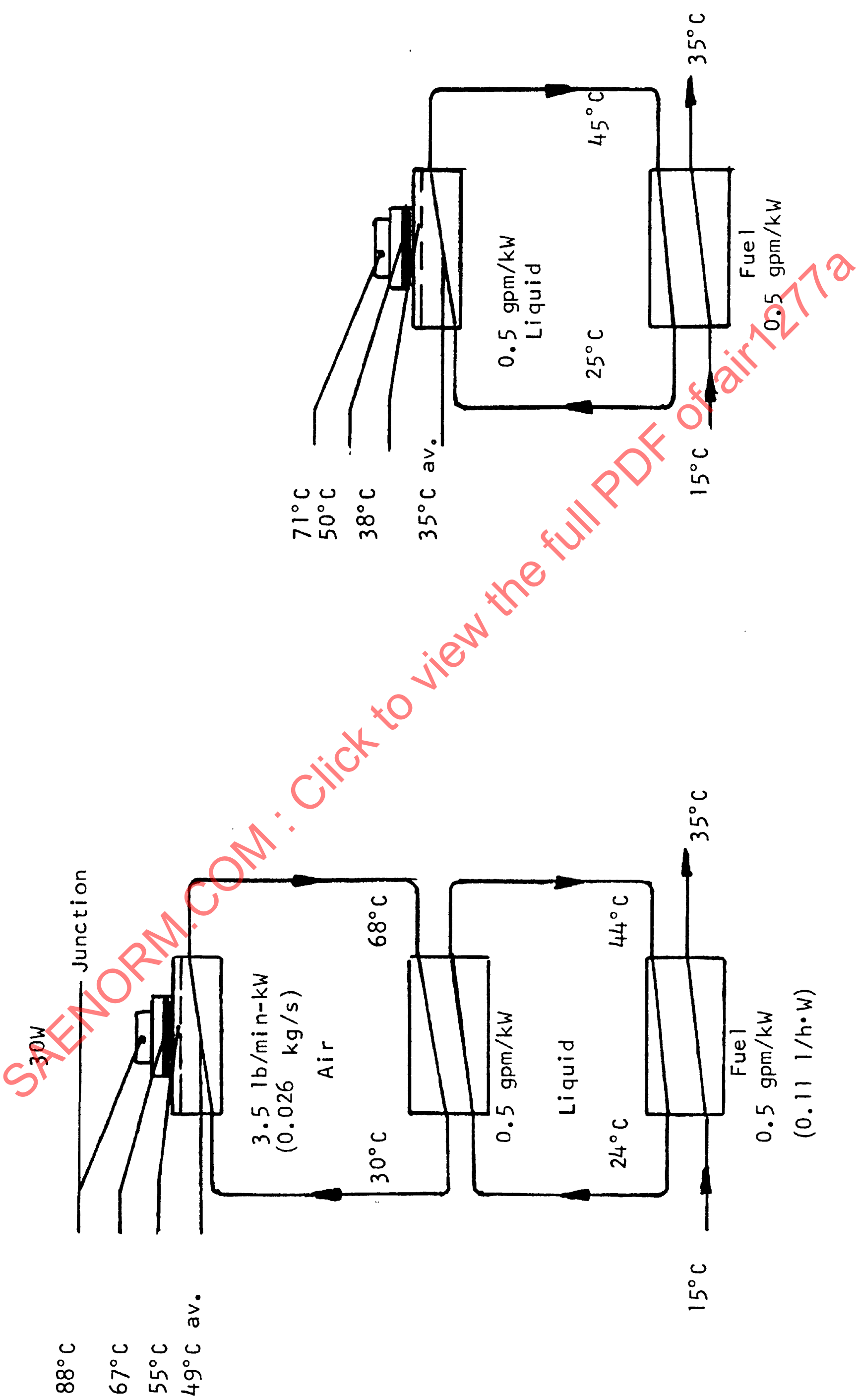


FIGURE 14 (Continued)

12.5 (Continued):

Figure 14c shows possible conditions relating to the multi loop cooling system of Figure 6. To avoid needless complexity, a flight condition has been assumed where all cooling is accomplished with fuel. Again, the effect of the additional step required for the remote air-cooled equipment is readily seen. Several facts are evident from these comparisons:

1) The temperature of the hot component must always be above the exit temperature of the coolant (Refer also to Section 6.2).

2) The greater the number of cooling circuits stacked in a system, the higher the total temperature difference is likely to be.

3) An increase in coolant flow per kW of load will help to reduce the overall temperature differential.

12.6 Thermal Resistance:

The expressions $R = \Delta t/Q$ which are shown in Figure 14b are known as the thermal resistances of the associated elements. The analogy to ohmic resistance is evident if Δt is thought of as voltage drop and Q as current flow. The concept is useful for network study, since the relation is linear for the commonly found modes of conductive and convective heat transfer. However, it should be remembered that $\Delta t/Q$ is not linear in natural convection, radiation and phase change. This has been strikingly demonstrated by the fact, that under certain conditions, a heat pipe can be made to act analogous to a rectifier diode. (References 10 and 11).

12.7 Interfaces:

The cooling of extensive electronics installations cannot be done in the most efficient way unless it is planned into the aircraft design as early as possible. Intended or foreseeable flight missions, safety regulations, carrier operations and aircraft needs must be collected to establish the approach. Equipment suppliers, airframe engineering, weights, logistics, standards and thermodynamics must join for an optimum solution creating interdisciplinary interfaces.

Today the black box or LRU (Line Replaceable Unit) is the typical installation module but it may well be replaced by the circuit card in the future. A total systems approach must at least consider wiring, connectors, cooling, packaging, power supplies and automated test equipment.

While many of these interfaces remain within the organization of the airframe manufacturer's system design, much information must pass between the electronic equipment manufacturer and the airframe manufacturer and/or user. The following may serve as a check list:

TABLE 8

Equipment Supplier	Airframe Manufacturer
Input Watt	Available Coolants and Capacity, Supply
Dissipated Watt	Pressure and Pressure Differentials
Type Coolant	Flows versus Temperature
Coolant Flow versus Temperature	Transients and their Duration
Pressure Drop versus Flow	Extremes of Environment
Control Requirements	Contaminants
Absolute Limit Temperatures	Duty Cycle
Thermal Design Margin	In-Flight Check Out
Thermal Time Constants	Applicable Standards for
Contamination Tolerance	Dimensions and Tests
Materials in Contact with	Level of Reliability
Coolant	Ground Cooling Provision
Safety Interlocks	Ground Operating Time
Ground Check Out	Thermal Test Requirements

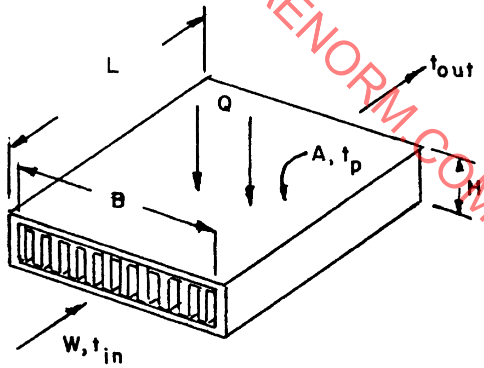
13. COMPONENT CHARACTERISTICS:

13.1 General:

The material in this section is supported by Tables 7, 9 and Section 16. It is intended for use as an aid in orientation, selection and for estimating size and weight. It is not intended for detail design. Reference 2 and the literature under Section 3 give further information.

TABLE 9 - Aircooled Cold Plate

Material: Aluminum $H \cong 0.5$ in. (13mm)



$B = 0.004 \; W_a(\Delta t/\Delta t_p \,\delta\, \Delta p)^{0.43}$ ft (lb/h, in. H_2O)

$L = 0.16 \,\delta\, \Delta p(\Delta t/\Delta t_p \,\delta\, \Delta p)^{0.83}$ ft (in. H_2O)

$B = 105 W_a(\Delta t/\Delta t_p \,\delta\, \Delta p)^{0.43}$ m (kg/s, P_a)

$L = 0.019 \,\delta\, \Delta p(\Delta t/\Delta t_p \,\delta\, \Delta p)^{0.83}$ m (P_a)

Weight $\cong$ 2.9 $(BL)_{ft^2}$ lb

$\cong$ 14.2 $(BL)_{m^2}$ kg

Q Btu/h, (W) Total heat input, (sum from both sides).

A ft^2, (m^2) Plate area, one side = B x L

W lb/h, (kg/s) Rate of airflow

t F, (°C) Local air temperature

Δt F, (°C) = $t_{in} - t_{out}$ = 4.17 Q/W, (F) = 10^{-3} Q/W, (°C)

t_p, F, (°C) Local plate temperature

Δt_p F, (°C) = $t_p - t$ At local heat flux Q/A

Δp in. H_2O, (P_a) Air pressure difference inlet to outlet

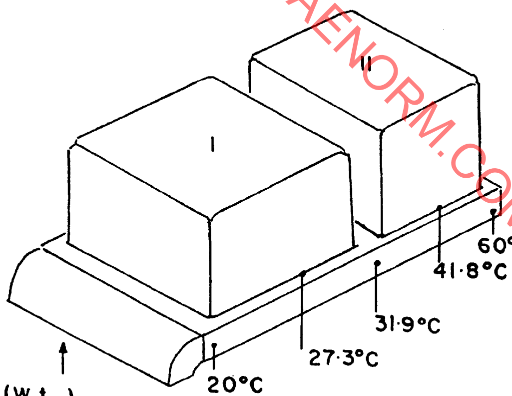
$\delta = \rho/\rho_o$ dimensionless. ρ = mean air density

$\rho_o = 0.0765$ lb/ft^3 = 1.225 kg/m^3

Any four of the above may be design parameters.

Equations are based on a surface having 17 fins/inch (0.67/mm) a channel height of 0.375 in. (9.5 mm), offset at 0.25 (6.5 mm) interval, a thickness of 0.008 in. (0.12 mm). Hydraulic diameter is 0.0078 ft (0.0024 m), surface per unit volume 461 ft^2/ft^3 (1512 m^2/m^3) and fin area/total area = 0.87.

TABLE 9 (Continued)



Assumed Values:

Load I	(W)	96.
Load II	(W)	224.
Contact Area I	(m^2)	0.06
Contact Area II	(m^2)	0.04
t air inlet	(°C)	20.
$\Sigma\Delta p$ permitted	(Pa)	186.
Minimum Plate Area to accommodate loads	(m^2)	0.11
Δt Air permitted	(°C)	40.

Example carried out in S.I. units

Calculated Values

Total Q_t (W)		320
Wa (kg/s)	$= Q_t/\Delta t\ C_p = 320/40 \times 1005$	0.008
B x L (m^2)	$= 2.78\ (40/\Delta t\ \ 174)^{1.26}$	
Δt_p (°C)		3.0
B (m)		0.28
L (m)		0.39
Weight (kg)	$= 14.2 \times (0.28 \times 0.39)$	1.55
Δt_{av} at Load I (°C):	$3 \times (96/0.06)(0.11/320) =$	1.65
Δt_{av} at Load II (°C):	$3 \times (224/0.04)(0.11/320) =$	5.8
t_{air} after Load I (°C):	$20 + 96/(1005 \times 0.008) =$	31.9

Average plate temperature at Load I

$\Delta t_{P1\ av} = 0.5\ (20 + 31.9) + 1.55 = 27.3$

and $\Delta t_{P2\ av} = 0.5\ (31.9 + 40) + 5.8 = 41.8$

If the physical configuration of the load fixes both B and L the pressure drop will vary. With the same assumptions as above but $B_2 = 0.35$; $L_2 = 0.314$ the pressure loss is approximately $\Sigma\Delta p_2 = \Sigma\Delta p_1\ (B_1/B_2)^2\ (L_2/L_1)$

or $\Sigma\Delta p_2 = 89.4$ (Pa). By writing down B/L and solving for Δt_p with the new $\Sigma\Delta p_2$ it follows that

$$\Delta t_{p2} = 3.4\ (°C)$$

13.2 Heat Exchangers:

13.2.1 General: Exchangers used for airborne electronics are generally constructed from aluminum alloys and use "extended surface", either in the form of finned channels between plates or as finned tubing. If exchange is between two liquids, a tube and shell design may be used. The exchangers are often termed "compact" because the transfer surface per unit of volume is several times that found in industrial equipment.

Because weight is a prime consideration, then gage materials of between 0.020 and 0.005 inch (0.5 to 0.13 mm) are quite common. Mechanical soundness and effects of corrosion (and sometimes erosion) must be considered when minimum weights are specified.

13.2.2 Finned Heat Sinks: For cooling power semi-conductors, a mounting surface with widely spaced fins (0.2 to 0.5 inch (5 to 13 mm)) of considerable height (1 to 3 inches (25 to 75 mm)) is practical with natural or moderately forced convection. Material thickness must match the chosen fin dimensions, since heat must flow from the components to the edge of the fin and causes a drop in temperature. This effect can be accounted for by calculating a "fin effectiveness" (Reference 2). For natural convection and aluminum fins the effect remains negligible as long as $(H/d^{0.5}) < 8$ (<40), where "H" is the height in flow direction and "d" the thickness of the fin (assumed to be constant), both measured in inch (mm).

13.2.3 Cold Plate: A cold plate is a type of compact exchanger which transfers heat by conduction from the components to a surface in contact with the coolant. It presents to the designer a surface of known thermal properties which is separated from the coolant and permits reasonably predictable design of the cooling system. The temperature between the mounting plate and the coolant varies in proportion to the local heat flux and the increase in coolant temperature as it traverses the plate. The cold plate may form a part of the equipment case. Estimates for size and weight may be obtained from Table 9.

13.2.4 Two-Fluid Exchangers: These units transfer heat from one fluid to another without mixing, and no leakage must therefore exist between the two sides. Brazing techniques are generally employed to secure this requirement.

The heat exchange may be gas to gas, gas to liquid or liquid to liquid. In special cases a liquid-vapor phase change may occur on one or both sides.

Many parameters enter into the design of these units, but for the purpose estimating size and weight, Section 16 serves as a guide. Inspection reveals two significant facts:

1) Size and weight may be traded to some extent for power consumed in moving the fluids.
2) Size and weight of an exchanger increase rapidly as its "effectiveness" enters into the area between 0.75 and 1.

13.2.5 Effectiveness: This item is often used to express the thermal behavior of the two-fluid exchanger:

$$E = \frac{\Delta t_{max}}{(t_1 - t_2)} \qquad \text{(Eq.4)}$$

Where Δt_{max} is the temperature change of the fluid experiencing the greater change as it passes the unit, and $(t_1 - t_2)$ is the absolute value of the difference of the entering temperatures of the fluids. It follows that as $\Delta t_{max} \rightarrow (t_1 - t_2)$, $E \rightarrow 1$.

13.2.6 Pressure Loss: Pressure difference required to force a fluid through an exchanger has the general form $\Delta p = C \times W^n/\rho m$ where $n \approx$ 1.6 to 1.8, W the rate of flow and ρm the density of the fluid at the mean fluid temperature and pressure. In gaseous flow the pressure loss is commonly given for a standard value of ρm so that:

$$\sigma\Delta p = \Delta p \times \rho m/\rho st \qquad \text{(Eq.5)}$$

It is important in specifying a permissible loss, to spell out clearly what is implied and what value is to be used for ρst.

13.3 Blowers:

Air blowers or "fans" for electronics cooling are generally of the axial flow type, but mixed flow or centrifugal units may occasionally be used for higher pressure rise capability at low flows. (Reference 2).

At pressure differentials of a few tenths of an inch of water column ($\approx$ 100 Pa), a simple propeller fan is adequate.

Pressure differentials of about 0.8 to 15 inch H_2O (200 to 3735 Pa) are obtained with axial vane designs. A survey of manufacturer's data on electrically driven fans has shown that some 80% of these units follow the curves of Figure 15. Figure 15 may therefore be used to estimate weight on the basis of power input.