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AEROSPACE INFORMATION REPORT

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COOLING OF MODERN AIRBORNE ELECTRONIC EQUIPMENT

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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 12, 14 and 15 for those readers who may appreciate its inclusion.

2. 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 AIR 64 for the draw through method and in AIR 728 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 9. 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.

3. 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.

4. 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.

4. (Cont'd)

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 enviroment. (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.

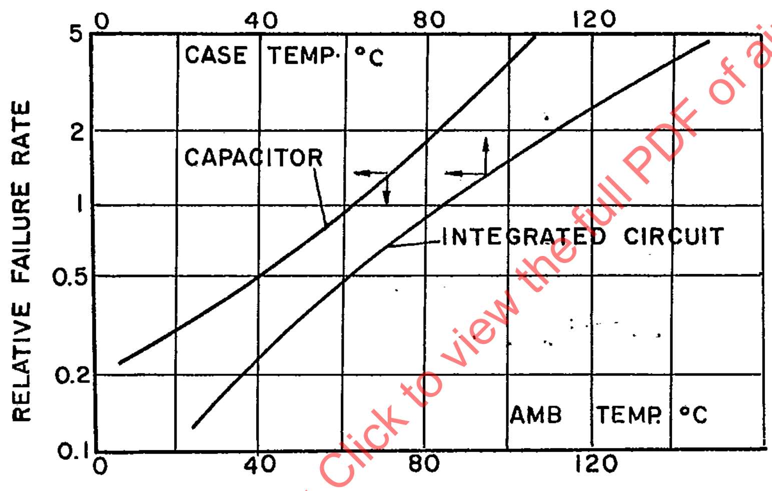


FIGURE 1. Trend of Failure vs Temperature

5. THE COOLING PROCESS

5.1 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.

5.2 Basic Laws: The processes illustrated in Fig. 2 are amenable to mathematical description and prediction. Table I summarizes applicable formulae for convenient reference. Typical property data can be found in Tables II and III.

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.

5.3 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.

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

5.5 Forced Flow: Heat flow by convection can be greatly increased by moving a cooling medium with external power over prepared surfaces as shown in Fig. 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.

5.6 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 III.

5.7 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.

5.8 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:

		COOLANT			
		AIR	LIQUIDS		
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

5.9 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/sec (4.6 to 6.1 m/s) in the plumbing. Table III gives values of rate of flow in a 1-inch (25.4-mm) tube based on this limit and consideration of the fluid viscosity.

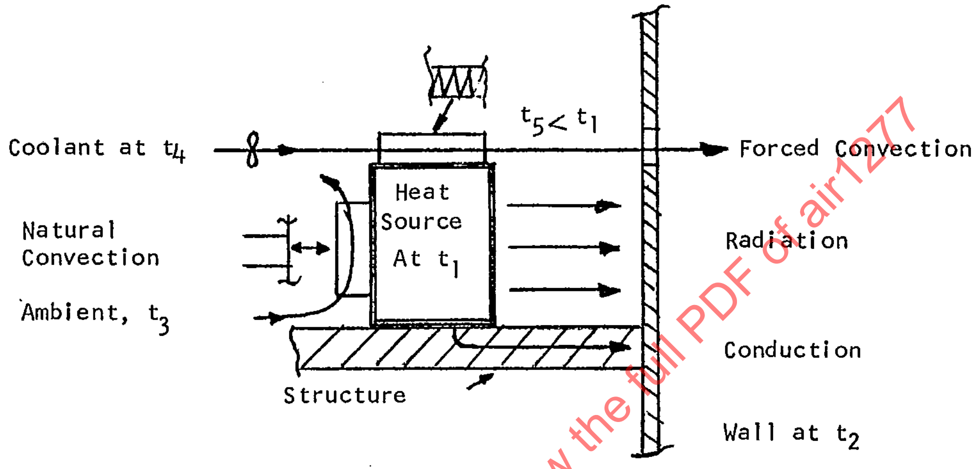


FIGURE 2. Combined Modes of Cooling

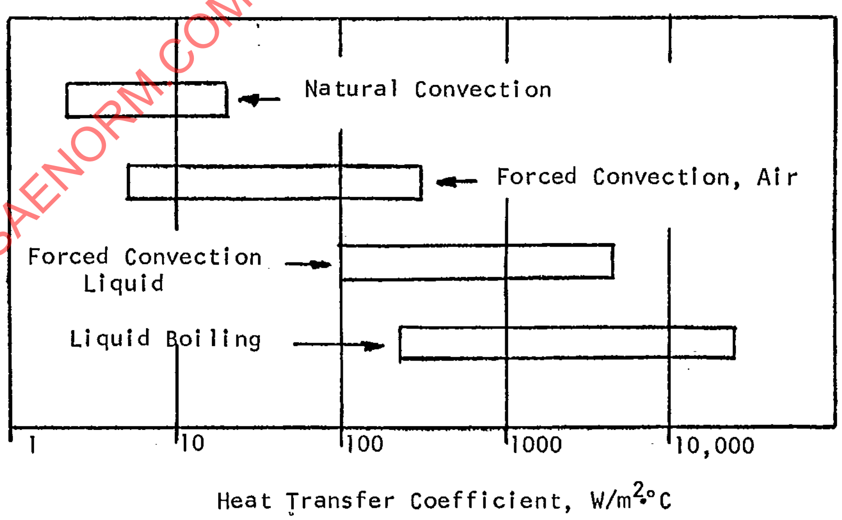


FIGURE 3. Comparative Rate of Heat Transfer per Unit of Area and Temperature Difference

5.10 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 ($Btu/hr\text{-}ft^2\text{-}F$ or $W/m^2 \cdot {}^oC$). The place of the various modes on this scale is shown in Fig. 3 in $W/m^2 \cdot {}^oC$. Radiation has not been included in Fig. 3 because of its different nature and application.

5.11 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 (^oC) 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/1b-F (0.63 kJ/kg·^oC) and the amount of residual cooling is 10% of normal, then time θ_e of emergency operation can be estimated as follows:

$$0.9\ Q\ \theta_e = G_e\ c_p\ \Delta t_B$$

or

$$\theta_e = 0.7\ \Delta t_B\ G_e/Q \quad \text{(s)}$$

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

$$\theta_e = 0.7 \times (5/0.5) \times 20 = 140 \text{ s} = 2.3 \text{ min.}$$

6. COOLANTS AND THEIR PROPERTIES

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

6.2 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).

6.3 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 III). 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.

6.4 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.

6.5 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 6.6 and 11.4).

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.

6.6 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

6.6 (Cont'd)

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.

7. STORED HEAT SINKS

7.1 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 III shows, about 60 Btu/1b (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.

7.2 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 8). 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.

7.3 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.

7.4 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.

7.5 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.

8. REFRIGERATION

8.1 General: The aircraft's ECS includes mechanical refrigeration and some of the sinks in Section 7, 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.

8.2 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.

8.3 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.

9. THERMAL CONTROL

9.1 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.

9.1 (Cont'd)

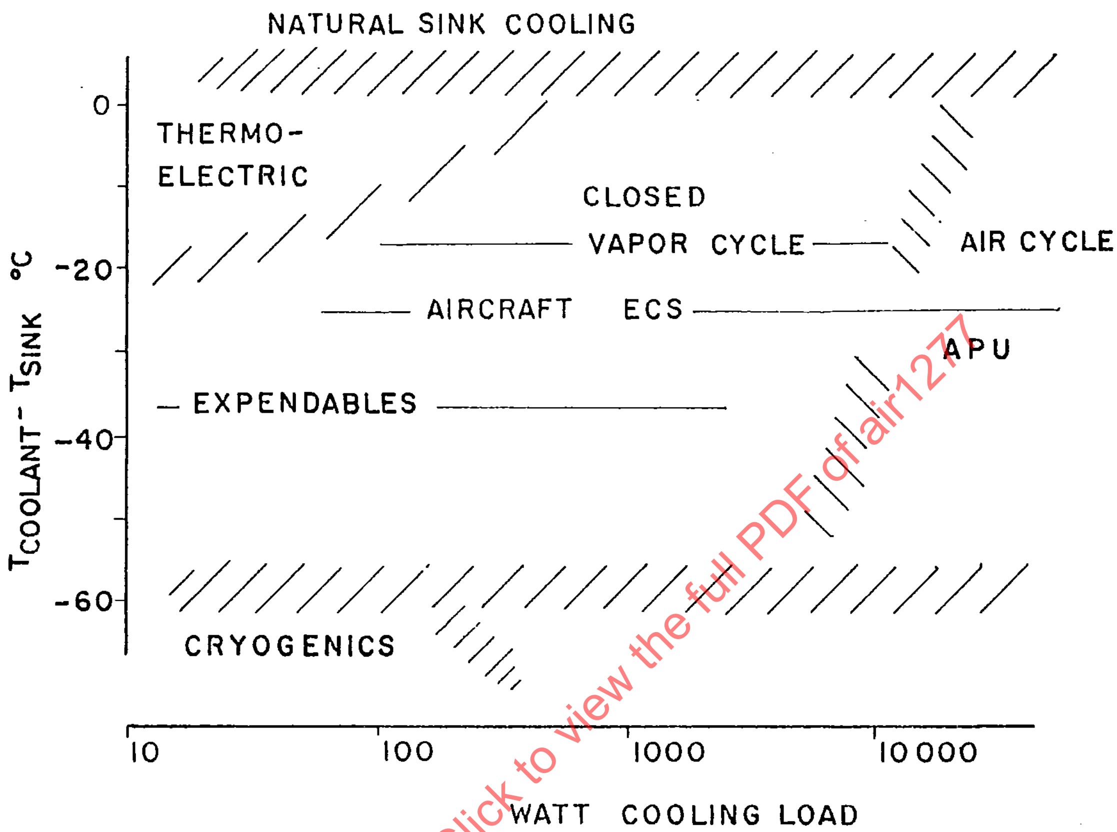


FIGURE 4. Application Trend for Various Cooling Methods as a Function of Load and Temperature Level Below Sink

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 12.5 and Table V.

9.2 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 Fig. 10 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 5.8, but resulting temperature rise must be investigated before the total required flow is determined.

9.2 (Cont'd)

If equipment is installed in a "closed loop" system where coolant recirculates, such as in a remote equipment bay (Fig. 13), 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.

9.3 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.

9.4 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 10 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.

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 V).

Bypass of coolant flow may be employed at the heat exchangers of individual equipment or in a system. Figure 13 gives an example of bypass control around the coolant exchangers of the loop.

9.5 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.

9.6 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).

10. APPLICATION OF COOLING TO EQUIPMENT

10.1 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 5(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 Fig. 5(b), which is a form of solid to fluid heat exchanger. Figure 5(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 5(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.

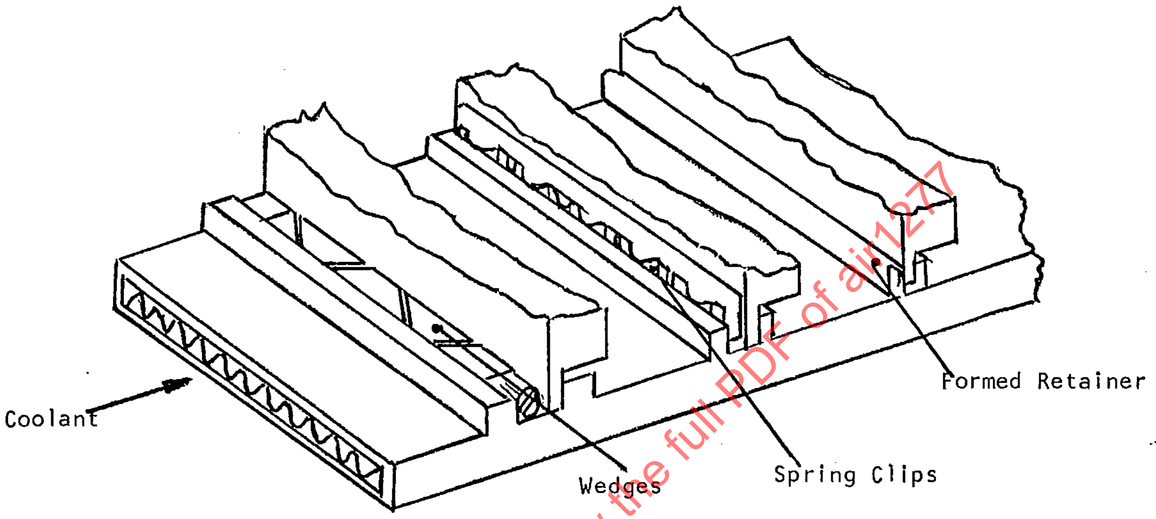
Whenever joints occur along the path of heat flow, an additional heat resistance is present. Depending on practical requirements, spring loading, wedging or heat conductive pastes can be used to bridge the air gaps and minimize joint resistance.

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 Fig. 6. Maintainability and reliability are affected by this approach, and for these reasons, some users may not approve of it.

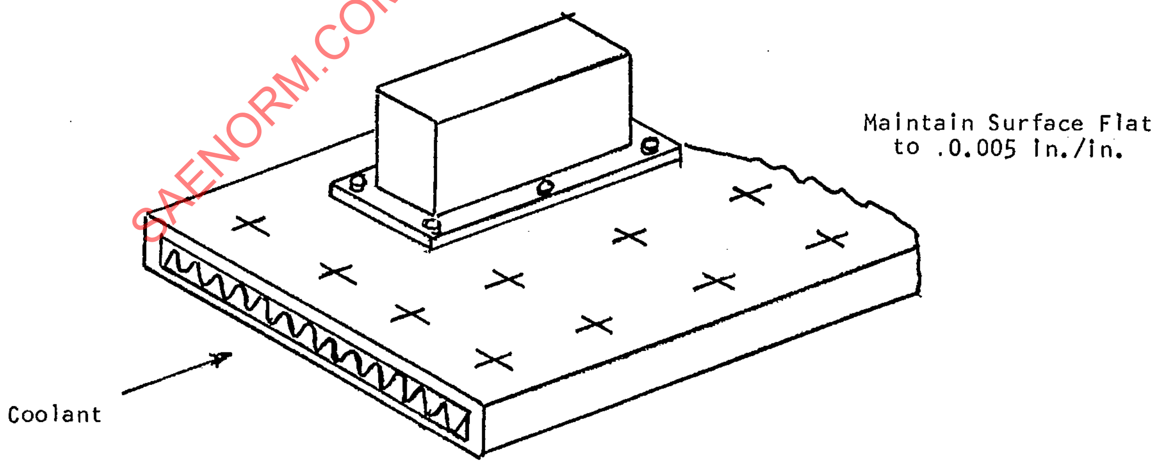
10.2 Transfer to the Coolant: The simplest method of cooling equipment is to arrange the various "black boxes" on shelves, in racks or panels with spacing sufficient to permit free circulation of compartment air around them. Air circulates because of the natural draft produced by the temperature difference 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 I.

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.

10.3 Forced Circulation: Controlling the airflow by external means is a logical step to achieve increased heat transfer. Figure 7 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 14 illustrates the use of the formulae given in this publication to estimate the ventilation effect and verifies the gain in heat transfer capacity.

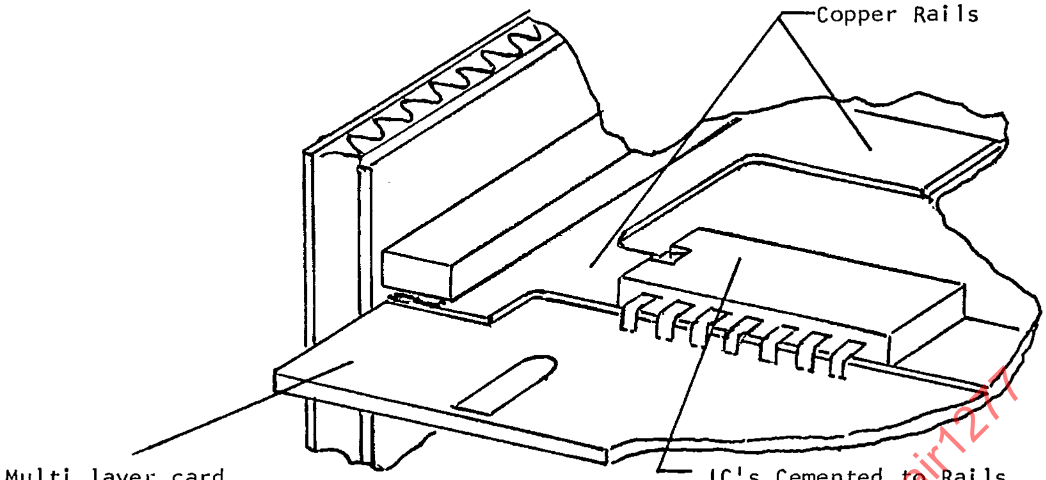


(a) Retaining Circuit Modules with Thermal Conduction

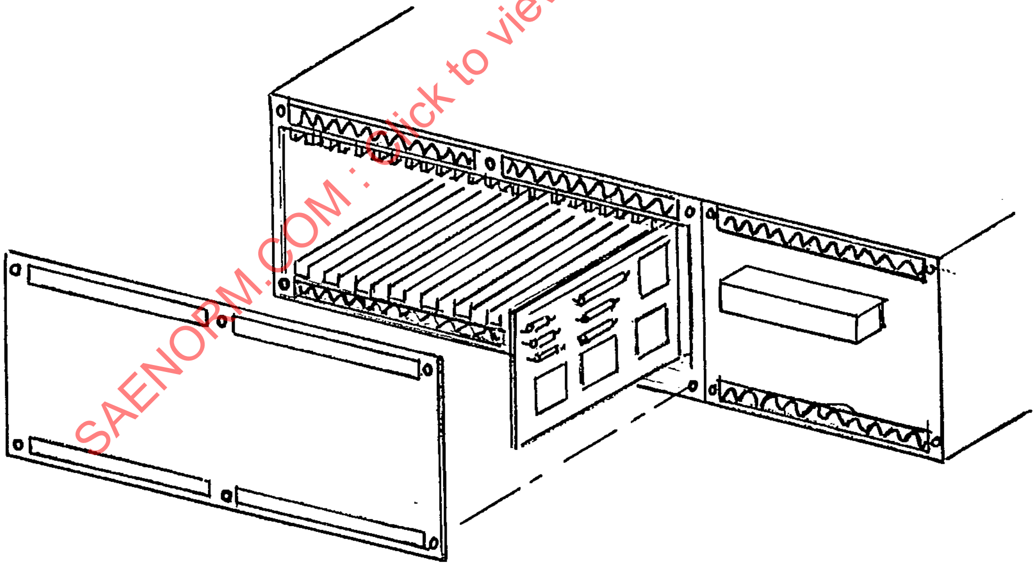


(b) Module mounted on Bolt Grid

FIGURE 5



(c) Thermal Shunt Rail



(d) Air Cooled L R U

FIGURE 5 (Continued)

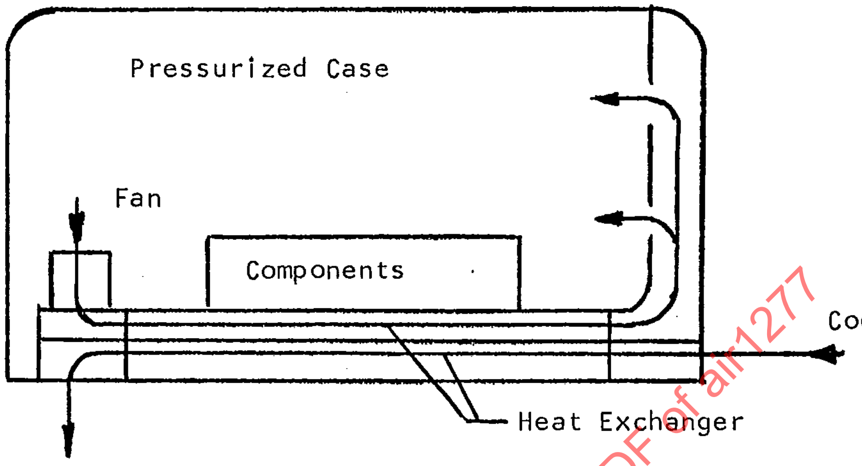


FIGURE 6. Unit with Internal Circulation

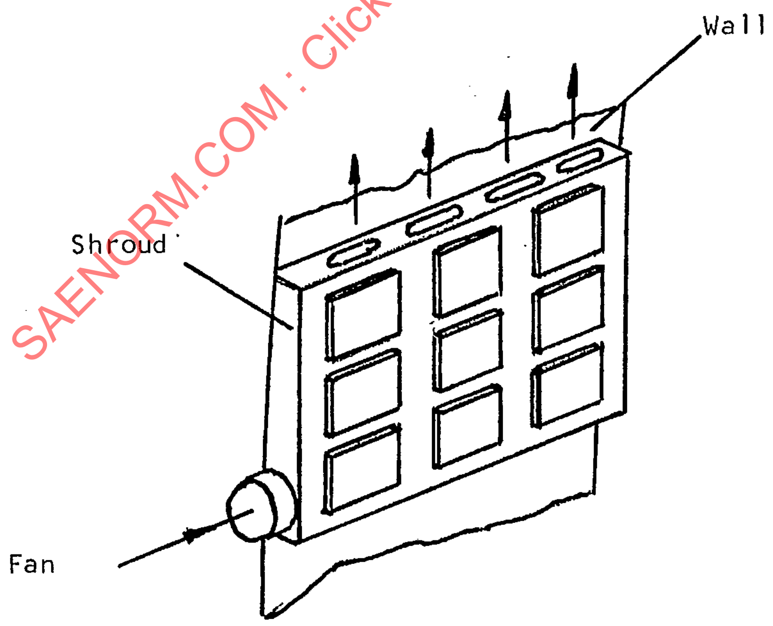


FIGURE 7. Display Panel with Cooling

10.4 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 10.3. 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 12.

10.5 Special Cooling Arrangements:

10.5.1 Liquid Cooling: As seen in Table II, 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.

10.5.2 Phase Change: Replacing conduction, or a circulating fluid, with evaporation and condensation of a liquid, results in the highest possible heat transfer. Figure 8 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 III and also Reference 8 and Literature 1.)

10.5.3 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 Fig. 9 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.

10.5.3 (Cont'd)

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.

10.5.3.1 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.

10.5.3.2 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.

10.6 Cooling on the Ground:

10.6.1 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.

10.6.2 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 hr)
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.

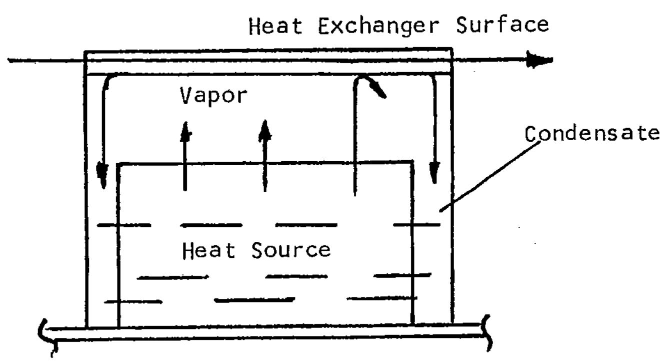


FIGURE 8.

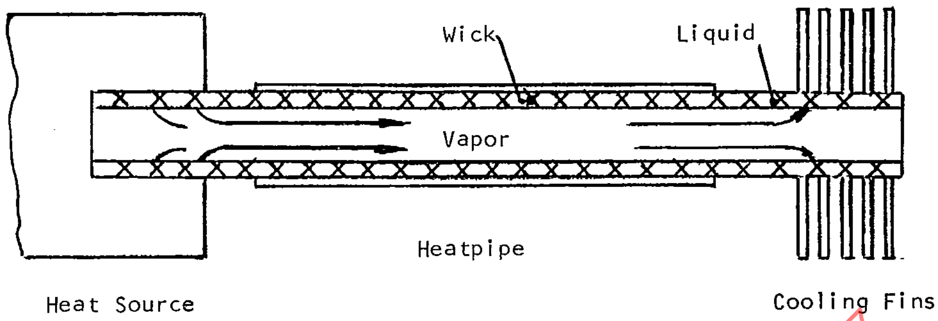


FIGURE 9.

11. COOLING SYSTEMS APPLICATIONS

11.1 Air Cooled Equipment on a Commercial Transport: Figure 10 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 can not 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.

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^{o}F (15.6^{o}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.

11.1 (Cont'd)

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.

11.2 Air Cooled System on an Unmanned Flight Vehicle: Figure 11 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 sec) 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 11 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 disolves 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.

11.3 Radar Cooling Loop: The use of a liquid cooling loop is shown in Figure 12, 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.

11.3 (Cont'd)

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.

11.4 Multi-Loop Cooling System: Figure 13 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.

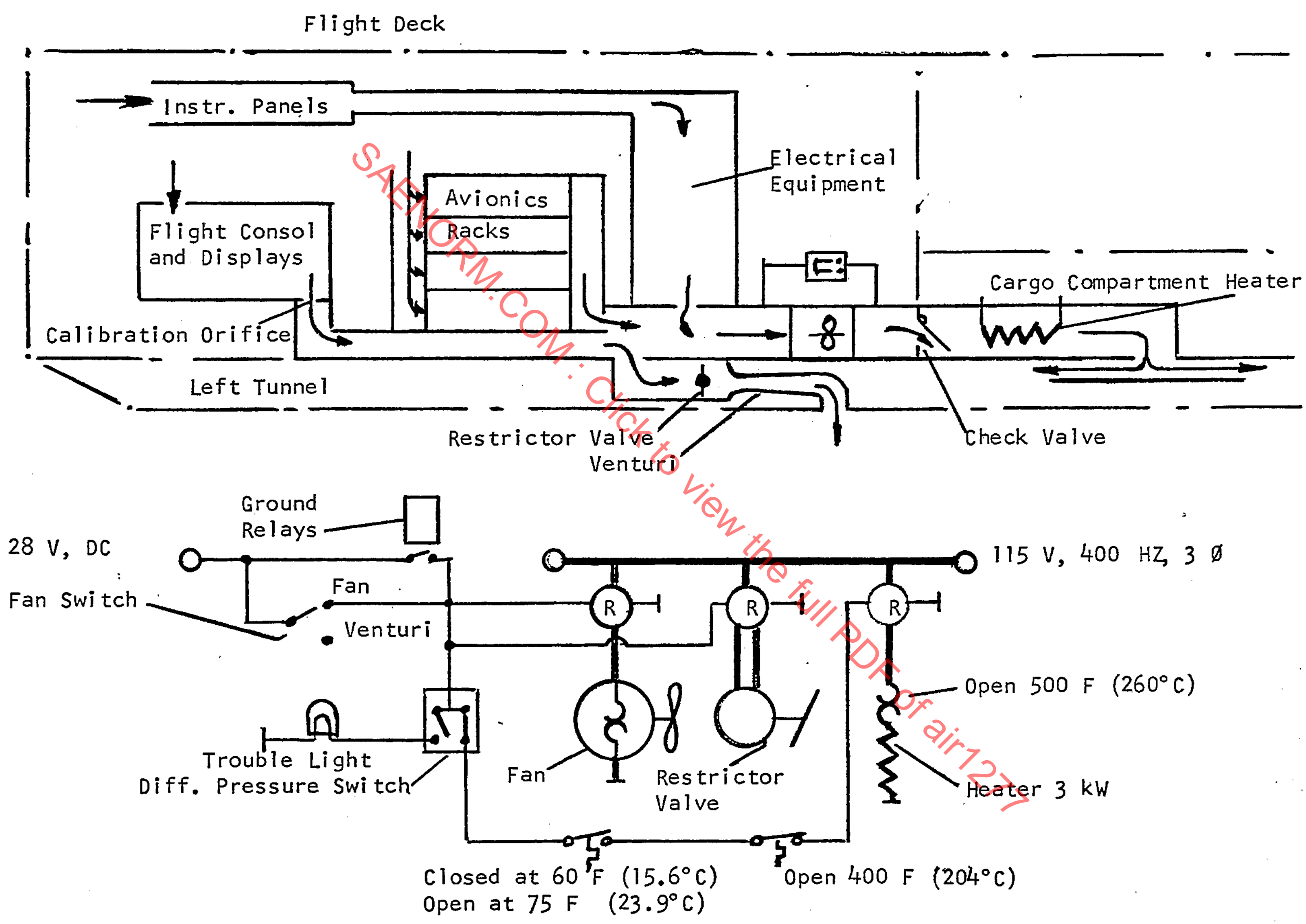


FIGURE 10. Air Cooled System, Commercial Transport

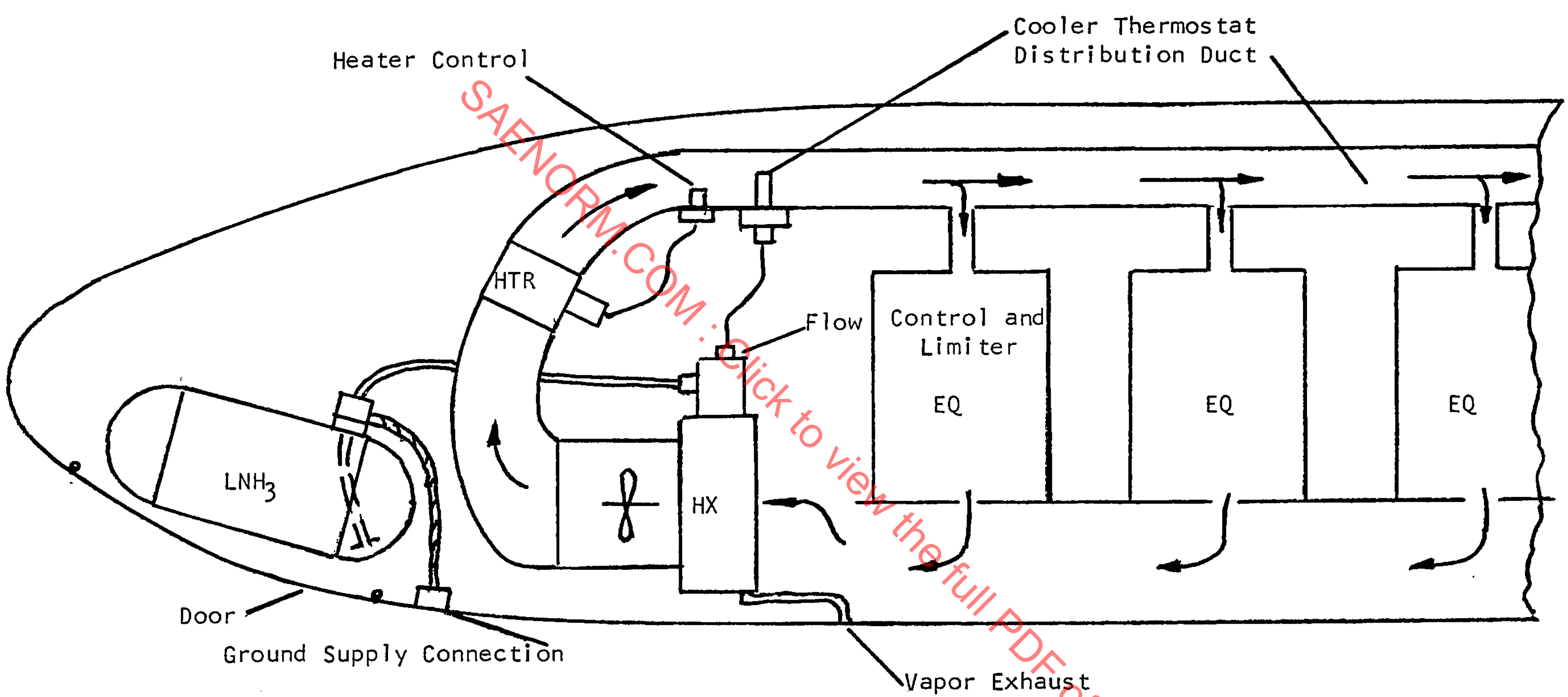


FIGURE 11. Evaporative Cooling System

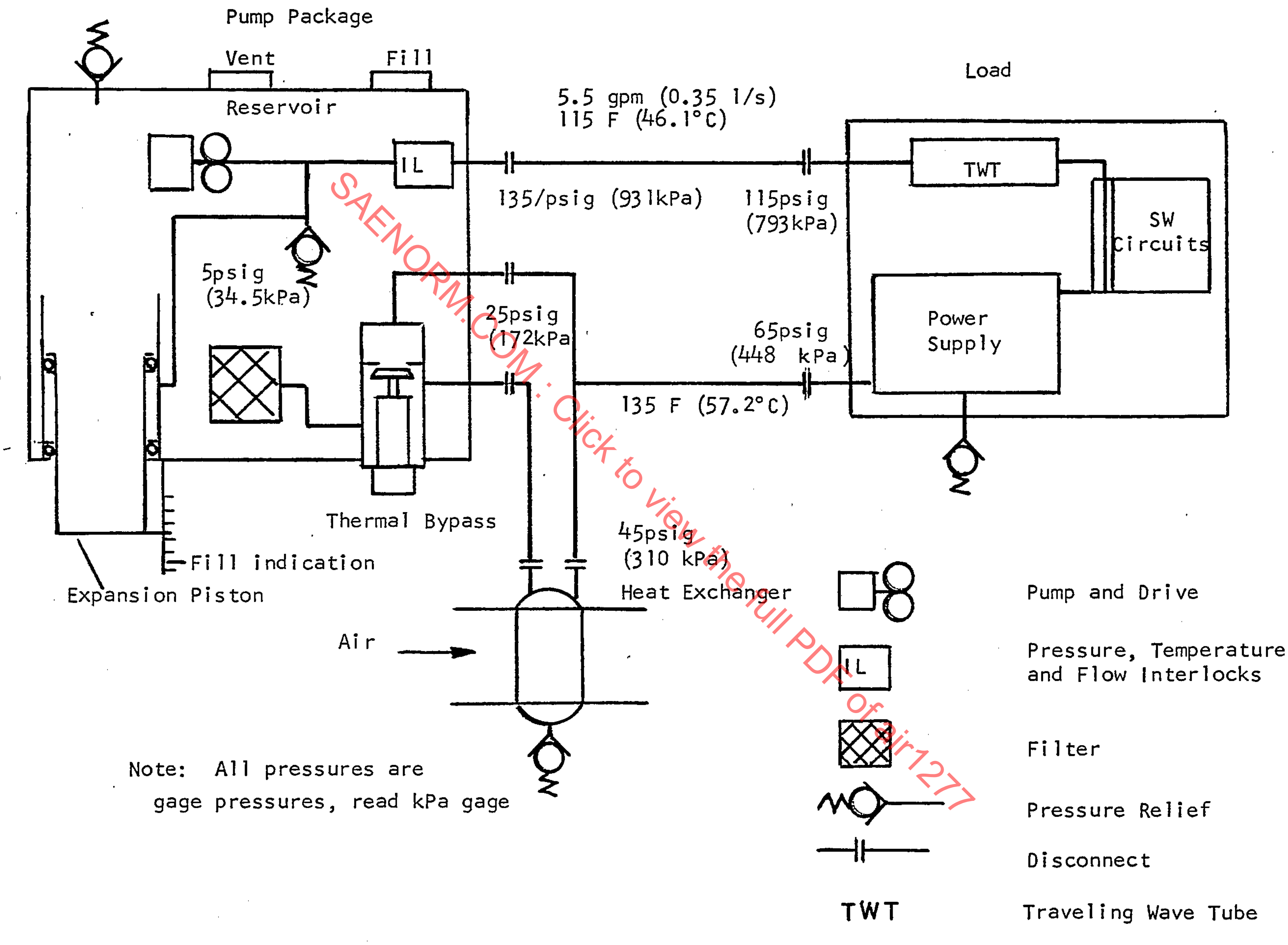


FIGURE 12. Radar Cooling Loop

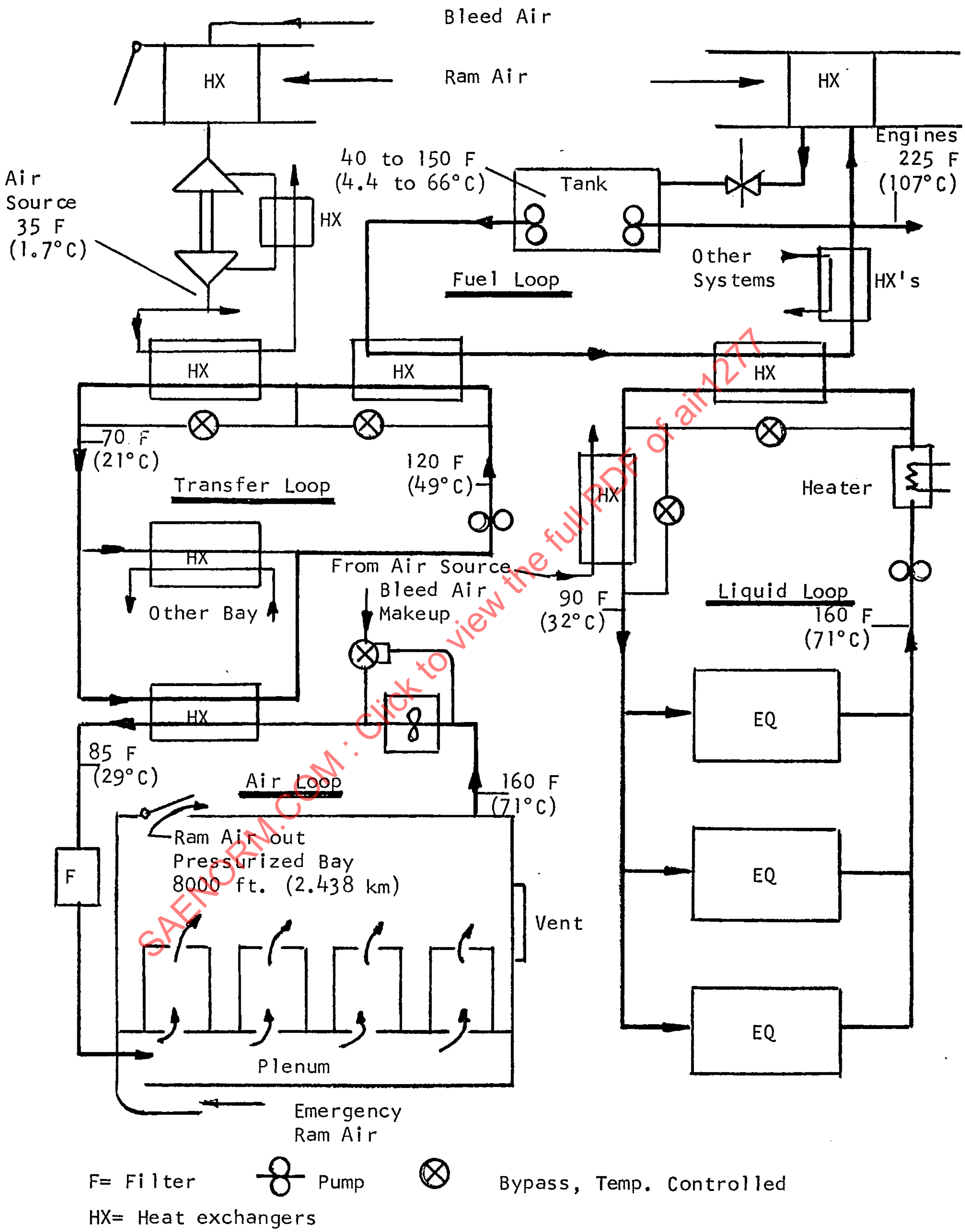


FIGURE 13. Multiloop Cooling

11.5 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 Fig. 14 using for illustration a power transistor as the "equipment".

Figure 14a refers to the cooling system of 11.1. 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 Fig. 12. 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 14c shows possible conditions relating to the multi loop cooling system of Fig. 13. 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 5.2).

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.

11.6 Thermal Resistance: The expressions $R = \Delta t/Q$ which are shown in Fig. 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).

11.7 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:

Equipment Supplier

Input Watt
Dissipated Watt
Type Coolant
Coolant Flow vs. Temperature
Pressure Drop vs. Flow
Control Requirements
Absolute Limit Temperatures
Thermal Design Margin
Thermal Time Constants
Contamination Tolerance
Materials in Contact with Coolant
Safety Interlocks
Ground Check Out

Airframe Manufacturer

Available Coolants and Capacity, Supply Pressure and Pressure Differentials
Flows vs. Temperature
Transients and their Duration
Extremes of Environment
Contaminants
Duty Cycle
In-Flight Check Out
Applicable Standards for Dimensions and Tests
Level of Reliability
Ground Cooling Provision
Ground Operating Time
Thermal Test Requirements

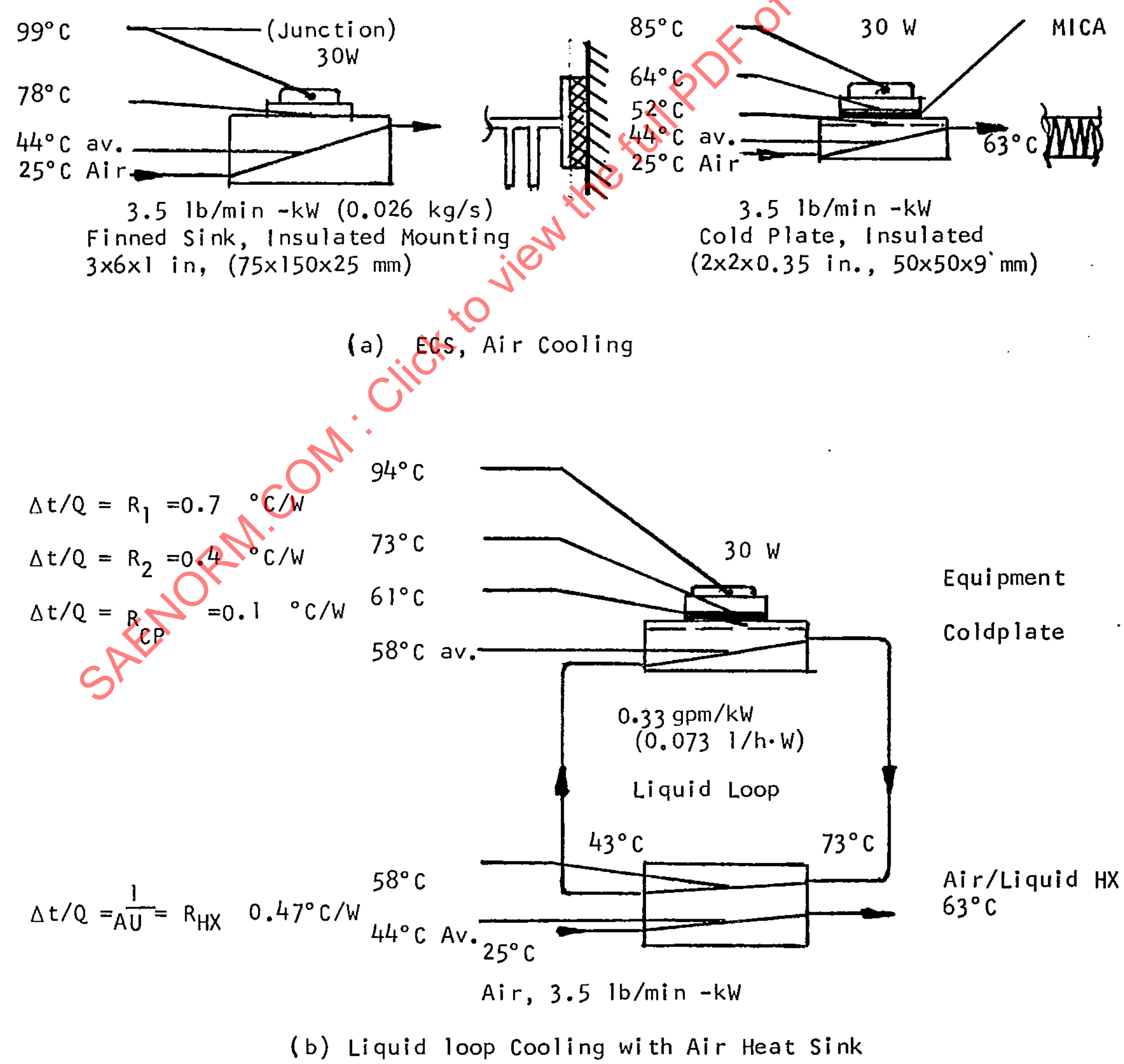


FIGURE 14. Temperature Patterns

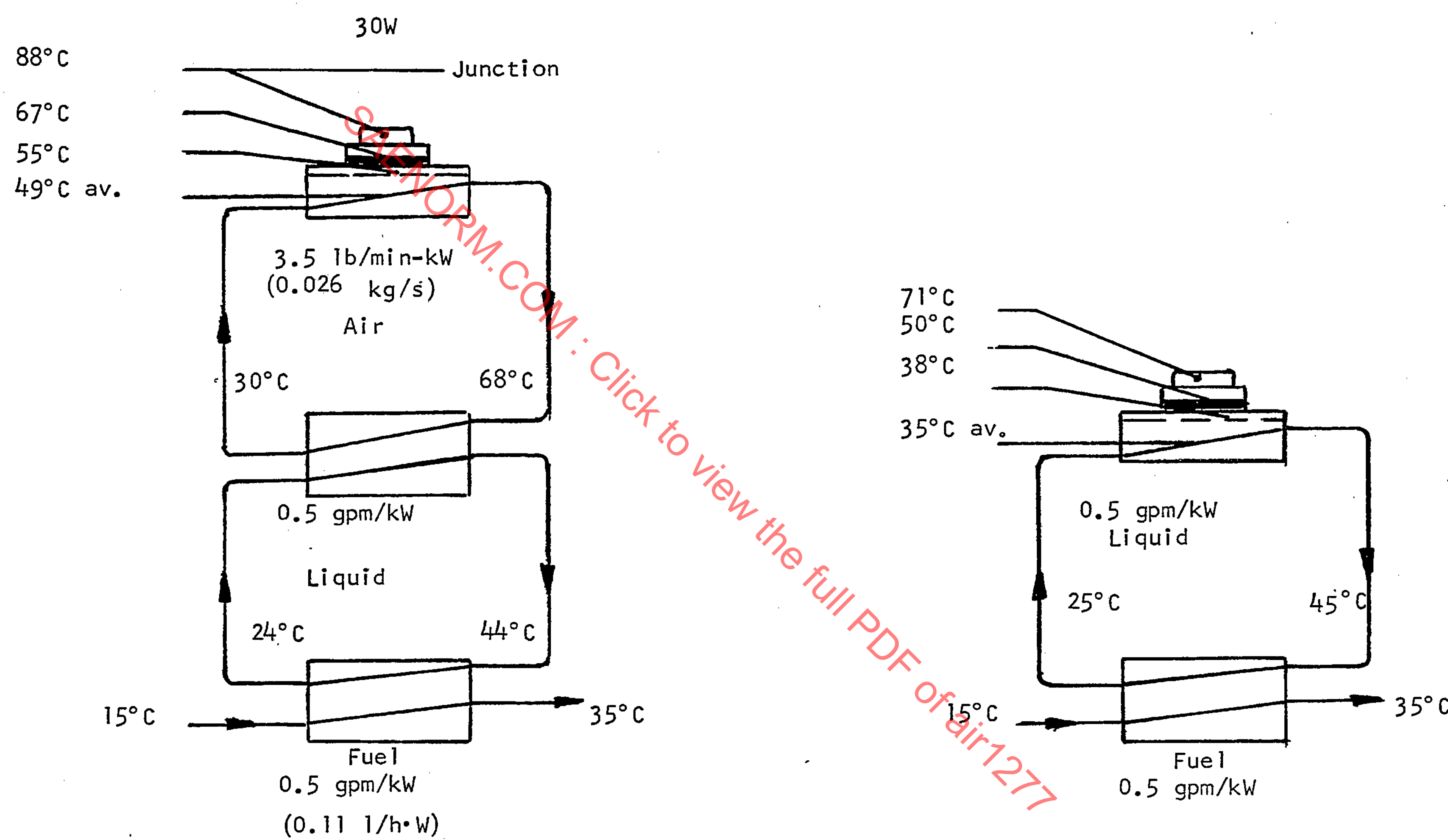


FIGURE 14(c). Multi Loop, Fuel Cooled Condition

12. COMPONENT CHARACTERISTICS

12.1 General: The material in this section is supported by Tables IV, V and Section 15. 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 13.2 give further information.

12.2 Heat Exchangers:

12.2.1 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 in. (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.

12.2.2 Finned Heat Sinks: For cooling power semi-conductors, a mounting surface with widely spaced fins (0.2 to 0.5 in. (5 to 13 mm) of considerable height (1 to 3 in. (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).

12.2.3 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 IV.

12.2.4 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 15 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.

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

$$E = \frac{\Delta t_{max}}{(t_1 - t_2)}$$

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 $\Delta t_{max} \rightarrow (t_1 - t_2)$, $E \rightarrow 1$.

12.2.6 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:

$$\delta \Delta p = \Delta p \times \rho m / \rho st$$

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

12.3 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 in. 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 Fig. 15. Figure 15 may therefore be used to estimate weight on the basis of power input.

Consider for example a 27 vdc blower that has 300 Watt input. Going across from 300 Watt to the efficiency curve gives 37%. This would be an "airpower" of 300 x 0.37 = 111 Watt. Thus if the blower were flowing 210 cfm (0.099 m^3/s) it would give a pressure rise of 4.5 in. H_2O (1120 Pa). Going across from 300 Watt to the curve labeled weight x rpm and down to the weight x rpm scale gives a reading of 6 x 10^4 lb x rpm (2.7 kg rpm). If the blower turns 7000 rpm the estimated weight is 6 x 10^4/7000 = 8.6 lb (3.9 kg). Volume as given in Fig. 15 becomes V = 8.6 (0.3 x 8.6 + 18) = 177 in.^{3} (2.9 x 10^3 m^3). (Note: Do not use the curves of Fig. 15 outside the given range).

If a blower must be used at low ambient pressure, it must be designed for the altitude condition. To avoid high power consumption at sea level a high slip or "altitude" motor may be used which reduces speed as torque is increased.

12.4 Pumps: A general review of liquid pumps appears in Reference 2. Liquid pumps may be of the dynamic (Centrifugal impeller) or of the positive displacement type.

Dynamic pumps depend on acceleration of the fluid and diffuser action to produce pressure head. Head per stage is thus limited by impeller tip speed, but the pump will not overload the prime mover when installed and it requires no relief valve. Because of its construction the centrifugal pump is less sensitive to poor lubricity of the fluid than a positive displacement device.

Positive displacement pumps produce pressure by mechanical force against the fluid system using some form of gear or vane as a piston. Pressure rise becomes essentially independent of flow and a relief around the pump should be provided.

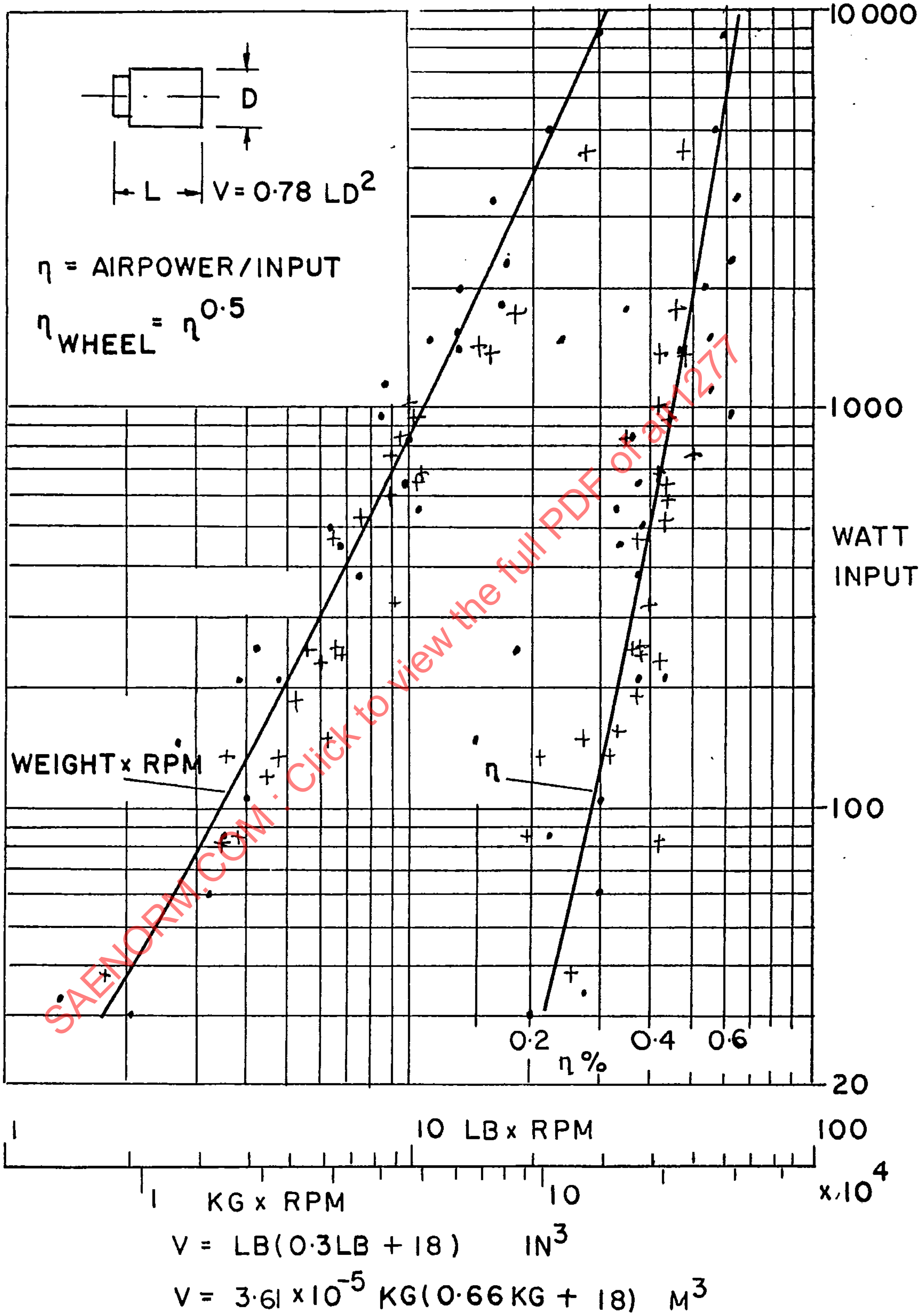


FIGURE 15. Blower Weight vs Power Input

12.4.1 Efficiency: Hydraulic efficiency of both types of pumps can be 80 to 90%, but in dynamic units, efficiency will drop substantially if a fluid with high viscosity must be pumped (when the flow Reynolds Number is low).

12.5 Temperature Sensors and Controls: Sensing temperature, and providing the desired characteristics and power output for a control may be done thermo-mechanically when the response required is in the order of 10 to 100 seconds.

If faster action is necessary, electronic pick-up and amplification will be needed. Table V describes a number of elements which are available and useful in airborne systems.

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14.0 APPENDIX TO SECTION 10.3

Fan Cooling Example

Consider an instrument panel as sketched in Figure 7.

The following assumptions may be made:

Panel size, ft^2	1 x 1	($0.305 \times 0.305\ m^2$)
A_h, Instrument surface area,* ft^2	2	($0.186\ m^2$)
A_c, Approximate free flow area straight up, ft^2	0.033	($0.003\ m^2$)
D_h, Average hydraulic diameter of flow passages ($\approx$ 2 x spacing), ft	0.05	(0.015 m)

Ambient pressure approximately sea level.

* Assume 9 square instruments 3.5 x 3.5 x 3 in. deep, (90 x 90 x 75 mm) offset about 0.3 in. (7.5 mm) from wall and vertical and rear surface effective only for convection.

Assume that heat load is equally distributed, (for simplicity only, not a necessary requirement). If it is permissible to have the surface of the cases go 50 F (27.8°C) above the (local) air temperature reference to Table I convection natural draft, gives:

$$Q/A\Delta t = 0.28 \quad (50/1)^{0.25} = 0.75\ \text{Btu/hr-ft}^2\ \text{F}$$

The convection heat rejection for the whole panel becomes:

$$Q = 0.75 \times 2 \times 50 = 75\ \text{Btu/hr} = \underline{22W}$$

Air temperature rises about 17 F (9.4°C) as air passes up the panel which makes the upper row of instruments run at a case temperature 67 F (37°C) above ambient. Heat flux is 0.75 Btu/hr-ft^2-F = 4.26 W/m^2·°C or within the low end of the range shown in Figure 3.

To obtain an approximation for radiant heat transfer assume that:

1. Surfaces facing front and rear only or about 2 ft^2 (0.186 m^2) contribute to radiation, since the other ones "look at each other".

2. Emissivity is 0.8 for radiating and absorbing surfaces.

3. Radiation occurs between parallel surfaces of equal area.

4. The sink surfaces are at 70 F (21°C) or 530 R (294 K) and the radiating surfaces average 580 R (322 K).

Table I gives for this case: $F_{1,2} = 1/(1/0.8 + 1/0.8-1) = 0.67 \approx 0.7$
$Q = 0.178 \times 10^{-8} \times 0.7 \times 2\ (580^4 - 530^4) = 85.4$ Btu/hr (25 W), which is of the same order as the free convection transfer. Total transfer is now 47 W.

For comparison assume now that the total heat load increases to 112W and forced air cooling shall keep the case temperature at 25 F (13.9°C) above local ambient. Radiation transfer decreases to about 12W at this lower case temperature, leaving 100W (341 Btu/hr) to forced convective transfer.

This is an increase in flux to $50/25 \times \frac{100}{22}$ or 9 times. The heat transfer coefficient required to achieve these conditions is $h = Q/\Delta t\ A_h$

$$= \frac{341}{25 \times 2} = 6.8 \text{ Btu/hr - ft}^2\text{- F} \quad (38.7 \text{W/m}^2\text{,°C}).$$

Referring to Table I, convection forced, turbulent duct flow, it is necessary to provide a flow "Reynold" number:

$$R_e = \left(\frac{hD_h}{0.023k}\right)^{1.25} \times N_p^{-0.5} \quad \text{and using}$$

$$k = 0.015 \text{ Btu/hr - ft - F}, \; Np = 0.71 \text{ from Table 2 for air}$$

$$R_e = \left(\frac{6.8 \times 0.05}{0.023 \times 0.015}\right)^{1.25} \times \frac{1}{(0.71)^{.5}} = 6553$$

which is indeed in the turbulent range.

From the same table, weight flow of air can be obtained as:

$$W = R_e \mu \; Ac/D_h = \frac{6553 \times 0.044 \times 0.033}{0.05} = 190 \text{ lb/hr} \; (= 0.024 \text{ kg/s})$$

(Consistent units must be used.)

The air temperature now rises only $\frac{341}{190 \times 0.24} = 7.5$ F (4.2°C).

To estimate flow losses for fan requirements assume (rather conservatively) that $f = 0.5/R_e^{1/3} = 0.027$, since the air is not flowing in a "smooth" duct.

Using the information from Table I and applying consistent units leads to:

$$\Delta_p = 4fL/D_h \; (1/2g\rho) = f \; (A_s/A_c)(W/A_c)^2 \times (1/2g\rho)$$

$$= 0.027 \; (2/0.033)(190/3600)^2 \; (1/0.033^2)(1/64.4 \times 0.075)$$

$$= 0.87 \text{ PSF } (17 \text{ Pa}).$$

Air flow in terms of "standard cubic feet per minute" becomes 190/60 x 0.075 = 42 SCFM. Pressure rise and flow required are well within the capacity of a 3 in. (75 mm) diameter 400 Hz propeller fan consuming 10 to 12 W of power in providing the cooling air.

Note that this example concerns itself with instrument case temperature only. To completely specify the instrument operating conditions, temperatures resulting inside the case must also be evaluated.

15.0 APPENDIX TO SECTION 12.2

Estimating Heat Exchanger Volume

15.1 Air to Air Heat Exchangers: The volume of the heat exchanger matrix or "core", that is the active part in which exchange takes place may be estimated using the curves shown in Figure 16.

The analytical and experimental background for these and the following curves is discussed in paragraph 15.4.

The core volume is given by:

$$V = V_o W_a / \left((\delta \Delta p_1)^{0.37} \, m^{0.15} \right) \quad \text{in.}^3$$

herein:

W_a : the smaller air flow, lb/min.

$\delta \Delta p_1$: the smaller pressure drop, in. H_2O.

referenced to a standard density of 0.0765 lb/ft^3 (1.225 kg/m^3)

δ : actual density of air at average operating temperature and

pressure of the related flow, lb/ft^3 times 13.07.

m : $\delta \Delta p_2 / \delta \Delta p_1 > 1$ the ratio of the reduced pressure losses

of the two flows.

V_o : as obtained from figure 16 as a function of the desired

effectiveness for W_a.

To obtain the volume in SI units; let

$$V = 0.0167 \, V_o W_a \left((\delta \Delta p_1)^{0.37} \times m^{0.15} \right) \quad m^3$$

herein:

W_a in kg/s, $\delta \Delta p_1$ in Pa

and δ = actual air density in kg/m^3 x 0.816

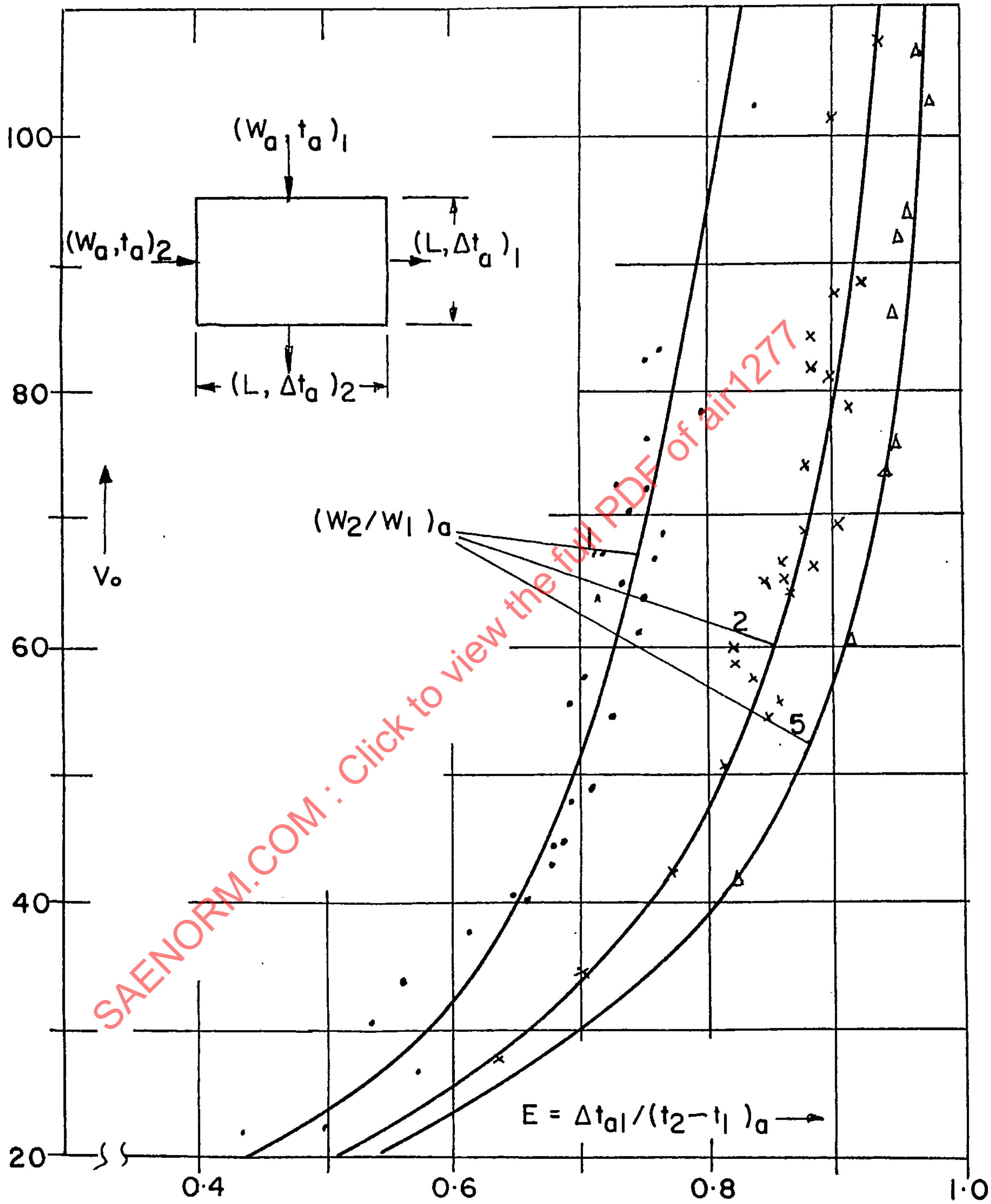

Figure 16. Air To Air Heat Exchanger Volume

For tube bundle type exchangers the obtained volumes should be multiplied by 1.30 to correct for the generally lower density of packaging in such units.

Allow for adequate space to bring and collect air to and from the heat exchanger core and consider that the velocity head in the approach tube will be essentially lost.

The weight of the complete exchanger may be obtained as:

$$G_h = 0.024 \text{ lb/in.}^3 = 664 \text{ kg/m}^3 \text{ for aluminum construction}$$

$$G_h = 0.048 \text{ lb/in.}^3 = 1329 \text{ kg/m}^3 \text{ for steel construction.}$$

Based on the volume obtained for the plate and fin type.

15.2 Air to Liquid Heat Exchangers: The assumptions made in 15.1 for general construction apply. In addition, air properties are taken at 68 F (20°C). The ratio of fin heights air to liquid is assumed as 5:1. Fins are of the "offset" type about 15 fins per inch.

The ratio of temperature difference on the air side has been centered at 0.70 of the overall difference. (See para. 15.4 for details).

A liquid side pressure loss of 5 to 25 psi (34.5 to 1724 kPa) is assumed and the liquid may make several passes through the unit.

The exchangers will cost less if $B/A \leq 1$. Volume and frontal area facing the air flow may be obtained from the curves of Figure 17 and the following equations.

$$V = ABC = (1.2 \pm 0.2)\ V_o W_a / \delta \Delta p^{0.22} \quad \text{in.}^3$$

$$A_{fr} = AB = (1.1 \pm 0.2)\ A_o W_a / \delta \Delta p^{0.4} \quad \text{in.}^2$$

where dimensions ABC are in inch, airflow W_a is in lb/min. and Δp is reduced pressure loss of the airflow in in. H_2O as explained in 15.1.

To obtain Volume and Frontal area in SI units let:

$$V' = (1.2 \pm 0.2)\ 7.3 \times 10^{-3}\ V_o W_a / (\delta \Delta p)^{0.22} \quad m^3$$

$$A_{fr'} = (1.1 \pm 0.2)\ 0.915\ A_o W_a / (\delta \Delta p)^{0.4} \quad m^2$$

Where W_a is in kg/s, $\delta \Delta p$ is in Pa, dimension ABC are in m, and δ is as explained in 15.1.

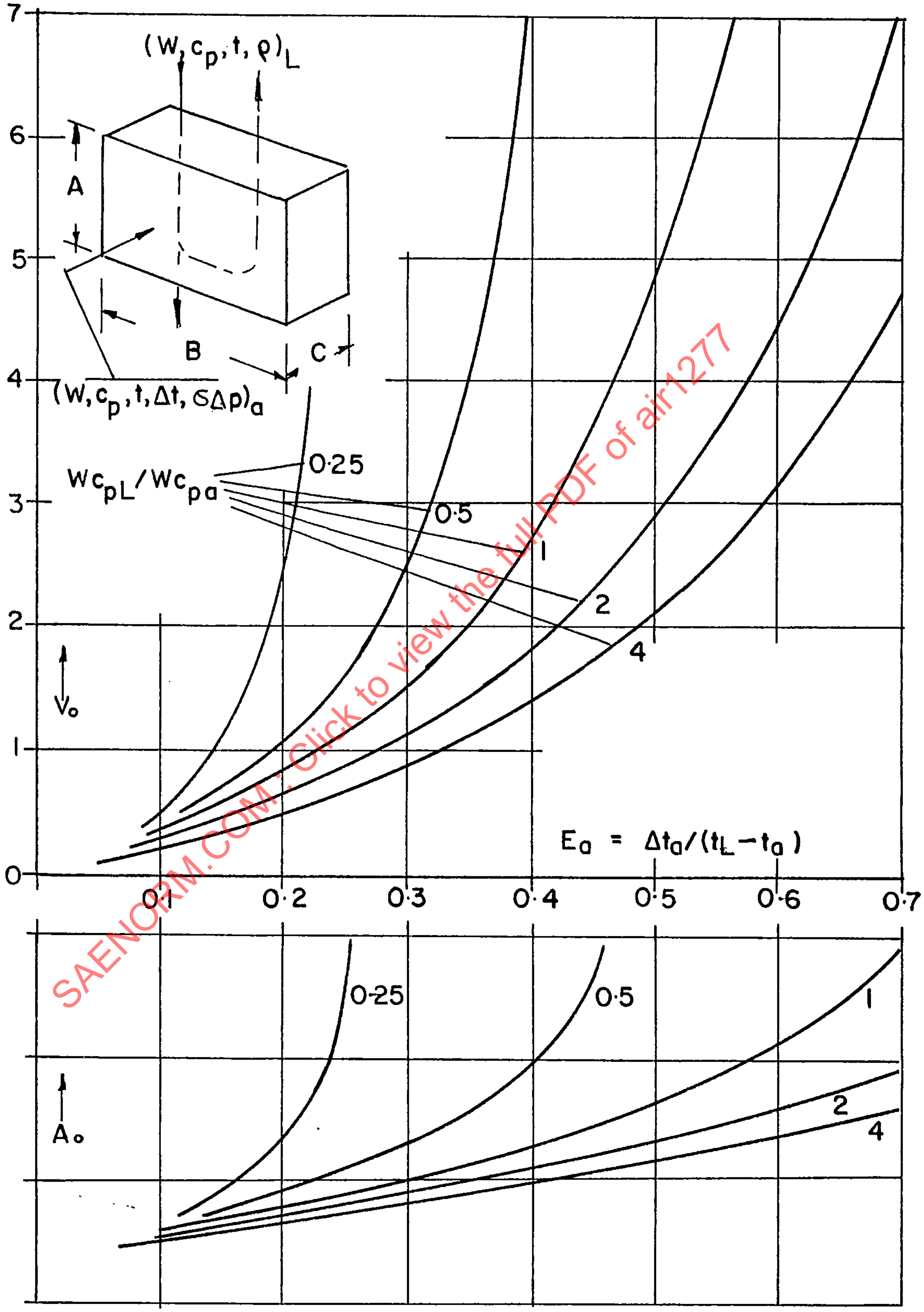

Figure 17. Air To Liquid Heat Exchanger
Volume and Frontal Area

The "wet" weight of the complete exchanger, including externals, in aluminum construction may be obtained from:

$$G_h = 0.1 \; V^{0.8} \; (1 + 0.004 \rho_L) \quad \text{lb}$$

Where the liquid density ρ_L is in lb/ft^3

and $$G'_h = V'^{\,0.8} \; (306 + 0.076 \rho_L) \quad \text{kg}$$

Where ρ_L is in kg/m^3

15.3 Liquid to Liquid Heat Exchangers: Liquid to liquid heat exchangers are mostly of tube and shell design either aluminum or steel depending on the requirements for pressure, corrosion and fire resistance. The higher pressure fluid usually flows through the tubes. Because of the many parameters that enter into design, the graph of Figure 18 can be a very general guide only.

The following assumptions are made:

AU/V is fixed at 20 Btu/hr-F-in.^{3} (6.44×10^5 W/°C·m^3). Tubing is 0.125 to 0.156 in. outer diameter (3.2 to 4 mm) and is beaded. Pressure loss on either side is assumed to be within 10 to 40 psi (69 to 276 kPa). With reference to Figure 18 for V_o find the shell volume from:

$$V = D^2 \; L \; \pi/4 = V_o C_1 \quad \text{in.}^3$$

Where C_1 is the product of W_1 and C_{p1}, flow in lb/min. and specific heat C_{p1} in Btu/lb-F and index (1) may be assigned to either side.

In SI Units:

$$V' = 0.518 \times 10^{-6} \; V_o \; C_1 \quad m^3$$

where C_1 as above but W_1 in kg/s and C_{p1} in J/kg·°C

Dry Weight: (Proof pressure of 400 psi (2758 kPa)

G_h = 0.035V, (Alum.) = 0.06V, (Steel) (lb)

G'_h = 970 V' (Alum.) = 1660 V' (Steel) (kg)

To obtain wet weight add one shell volume of fluid averaging density for the two sides.

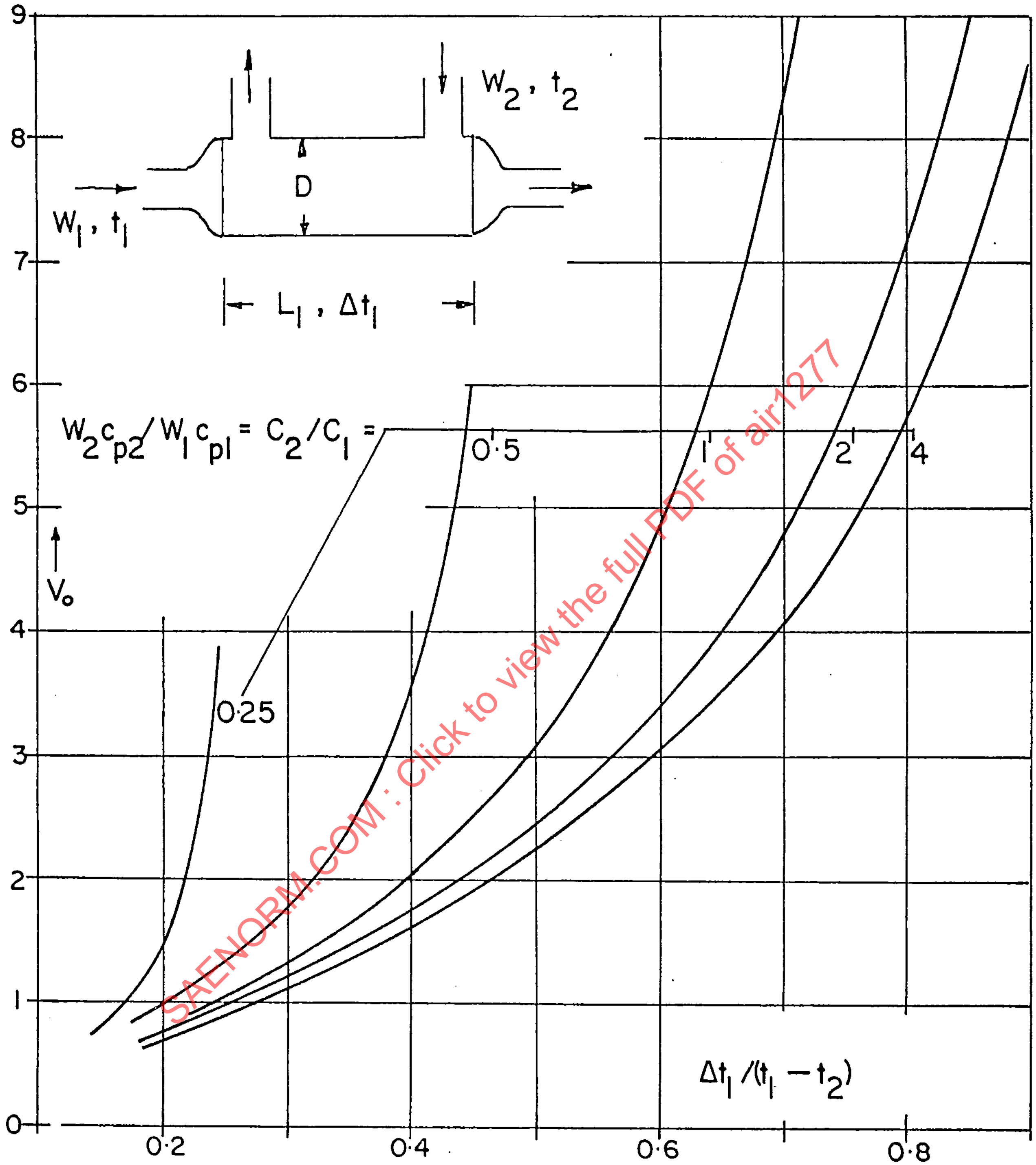

FIGURE 18

LIQUID TO LIQUID HEAT EXCHANGER VOLUME

15.4 Basis for Estimating Equations: It can be shown (Reference 12) that the performance of a typical turbulence promoting extended surface approaches the form:

$$V/A_s h = K_1 \left(A_s h / \rho \Delta p W_a C_p\right)^{\alpha}$$

$$A_{fr}/W_a = K_2 \left(A_s h / \rho \Delta p W_a C_p\right)^{\beta}$$

where the nomenclature is that of the list of symbols. For the range $1000 < Re < 10000$ constant values of α and β give results which are accurate enough for an estimating procedure and this covers a majority of applications. Dimensions and a surface which may be considered representative have been selected in Table 4 and the above relations then apply directly with the following (approximate) values.

$$V = \begin{matrix}(2.43 \times 10^{-5}) \\ 0.186\end{matrix} A_s h \left(A_s h / \rho \Delta p W_a C_p\right)^{0.26} \qquad \begin{matrix}(m^3) \\ in.^3\end{matrix}$$

$$A_{fr} = \begin{matrix}(1.202) \\ 0.022\end{matrix} W_a \left(A_s h / \rho \Delta p W_a C_p\right)^{0.43} \qquad \begin{matrix}(m^2) \\ in.^2\end{matrix}$$

(where airflow W_a is in lb/hr (kg/sec).

The equations of Table 4 follow from this by using the equalities:

$$Q = A_s h \Delta t_p = W_a C_p \Delta t$$

using the nomenclature of Table 4, and herewith follows:

$$A_s h / W_a C_p = \Delta t / \Delta t_p$$

This leads to:

$$L \times B \times H = V = \begin{matrix}(0.0875) \\ 0.186\end{matrix} W_a \rho \Delta p \left(\Delta t / \Delta t_p \rho \Delta p\right)^{1.26} \cdot C_p \qquad \begin{matrix}(m^3) \\ in.^3\end{matrix}$$

and

$$B \times H = A_{fr} = \begin{matrix}(1.202) \\ 0.022\end{matrix} W_a \left(\Delta t / \Delta t_p \rho \Delta p\right)^{0.43} \qquad \begin{matrix}(m^2) \\ in.^2\end{matrix}$$

Using H = 0.5 in. (12.5 mm) cp = 0.24 Btu/lb-F

(1005 J/kg·°C) and dividing leads to:

$$B = \begin{matrix}(93.3) \\ 0.0036\end{matrix} W_a \left(\Delta t / \Delta t_p \rho \Delta p\right)^{0.43} \qquad \begin{matrix}(m) \\ ft\end{matrix}$$

$$L = \begin{matrix}(0.02) \\ 0.169\end{matrix} \rho \Delta p \left(\Delta t / \Delta t \rho \Delta p\right)^{0.83} \qquad \begin{matrix}(m) \\ ft\end{matrix}$$

For the purpose of the Table the constants have been adjusted to include an allowance for inlet and exit pressure loss.

This approach may be extended to the air to liquid exchanger if certain reasonable assumptions for the conditions on the liquid side are made. The single surface relation may then be rewritten in terms of the "overall transfer coefficient AU" to read:

$$Va/(AU) = 0.186\ ((AU)/a \partial \Delta p\ W_a C_p)^{\alpha}$$

where V expresses the total volume of the exchanger matrix. The value a is the ratio

$$a = (AU)/(A_s h)_{air}$$

A survey of actual cases suggests to use

$$a = 0.7\ (C_L/C_{air})^{0.2}$$

at least as long $0.25 < C_L/C_{air} < 4$.

The ratio $(AU)/W_aC_p$ = NTU is known as "Number of Transfer Units" and is a known function of effectiveness and C_L/C_{air}.

With the airflow given in lb/min (kg/sec), the volume follows now as:

$$V = \left[0.186 \times 60\ \ NTU^{1.26} / 0.7^{1.26} (C_L/C_{air})^{0.25} \right] \frac{W_a}{\partial \Delta p^{0.26}} \text{ in.}^3$$

$$V = V_o W_a / \partial \Delta p^{0.26} \text{ in.}^3$$

respectively $V = 9.1 \times 10^{-3}\ \ V_o W_a / \partial \Delta p^{0.26}\ \ m^3$

A useful correlation was obtained for a number of available cases by adjusting $\partial \Delta p^{0.26}$ to $\partial \Delta p^{0.22}$ and allowing a tolerance band by multiplying with (1.2 ± 0.2). Figure 19 shows the distribution of the data spread obtained in the survey. Curves for frontal area were derived in a similar manner.

For the case of the air to air exchanger the previous analysis has suggested a correlation of the form given in paragraph 15.1.To test its usefulness a survey of airborne heat exchangers as used in air cycle machines and pressurization systems was made. The survey included 15 heat exchangers from two manufacturers. 5 units were of stainless steel plate and fin construction; 4 units were compact tube bundles of stainless steel construction and 6 units were plate and fin type exchangers built in aluminum construction.

Data points obtained from the performance maps of these exchangers have been entered into figure 16. Their distribution attests to the usefulness of these curves.

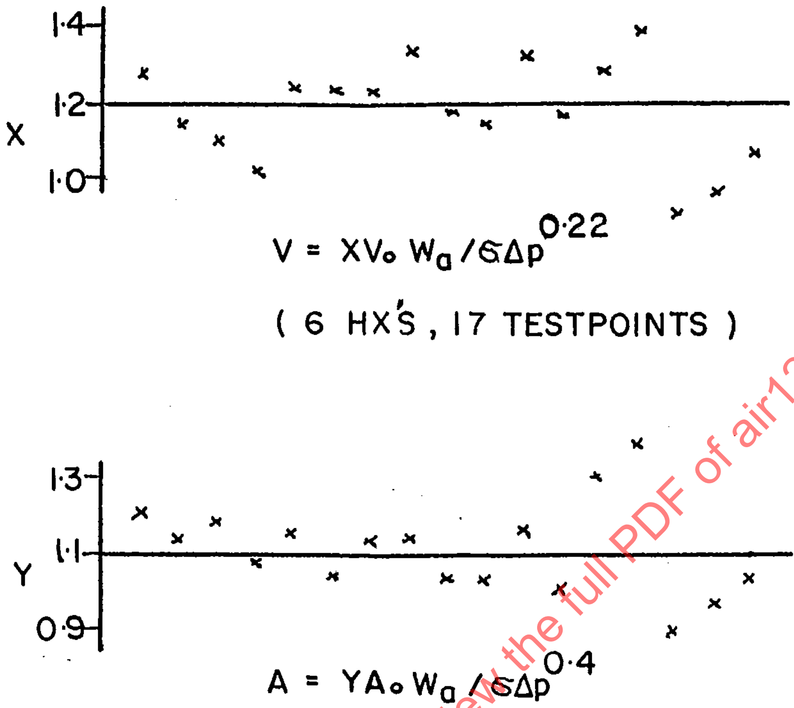

Figure 19. Data Spread, Air To Liquid Heat Exchangers

16.0 LIST OF SYMBOLS

Symbol	Unit: British	Unit: (SI)	Meaning
A,B,C,H	in., ft	(m)	linear dimensions
A_s	ft	(m^2)	area for heat transfer
A_c	ft^2, $in.^2$	(m^2)	free flow area in a conduit or exchanger
A_{fr}	ft^2, $in.^2$	(m^2)	frontal area facing flow
A_o	-	-	constant in equation
AC	amp	(A)	alternating current
AU	Btu/hr-F	(W/°C)	overall heat transfer per unit of temperature difference
BP	F	°C	boiling point
C	Btu/hr-F	(W/°C)	product Wcp
c_p	Btu/lb-F	(J/kg·°C)	specific heat
d	in.	(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 = $(\Delta p g \rho D_h/L)(A_c/W)^2$
F	-	-	combined radiation coefficient
g	ft/sec^2	(m/s^2)	gravity constant 32.2 (9.81)
G_h	lb	(kg)	weight of heat exchanger
G_e			weight of equipment
h	$Btu/hr\text{-}ft^2\text{-}F$	($W/m^2 \cdot °C$)	coefficient of heat transfer, fluid to solid

LIST OF SYMBOLS (Cont'd)

Symbol	Unit British	Unit (SI)	Meaning
$K_{1,2}$	-	-	constants in equations
k	Btu/hr-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
M	-	-	Mach number, ratio of flow velocity to local velocity of sound
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	in.H_2O,psi	(Pa)	pressure differential in fluid flow between two points
Q	Btu/min,Btu/hr	(W)	heatflow per unit of time (Q x θ Btu, (J) is heat energy)
Q/A_c	Btu/hr-ft^2	(W/m^2)	heat flux
R	F hr/Btu	(°C/W)	thermal resistance
R_e	-	-	Reynold Number = $WD_h/A_c\mu$
S	Btu/hr-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/hr-ft^2-F	$(W/m^2 \cdot °C)$	overall heat transfer coefficient between more than two elements
V	ft3	(m3)	volume of heat exchanger matrix
V_o	-	-	volume constant in heat exchanger equation

LIST OF SYMBOLS (Cont'd)

Symbol	Unit British	Unit (SI)	Meaning
W	lb/min, lb/hr	(kg/s)	massflow of fluid
α, β	-	-	powers in equation for heat exchanger volume
Δ	-	-	differential
ε	-	-	emissivity
η	-	-	efficiency percent
θ, θ_e	sec, min	(s)	time, emergency operating time
μ	lb/hr-ft	(Pa·s)	fluid viscosity
ρ	lb/ft^3	(kg/m^3)	density
σ	-	-	mean density/standard density

SYMBOLS NOT USED BUT COMMON IN LITERATURE:

Symbol	Unit British	Unit (SI)	Meaning
G	$lb/hr\text{-}ft^2$	$(kg/h\cdot m^2)$	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
hr (h)	hour, time
Hz	Hertz, oscillations per sec.
in., in. H_2O	inch length, inch water column pressure
(J)	Joule = W·s unit of energy
(kg)	kilogram mass

LIST OF SYMBOLS (Cont'd)

Units:	( ) denotes SI unit
(°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)
min.	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
sec (s)	second, time
SCFM	standard cubic foot per minute based on density of 0.075 lb/ft)
(W)	Watt, unit of power

Subscripts

1, 2 etc.	sides 1 and 2 of heat exchanger, various temperatures etc.
a	air, (W_a)
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)

LIST OF SYMBOLS (Cont'd)

Subscripts

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)