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DESIGN MANUAL

OF

METHODS OF FORCED AIR COOLING ELECTRONIC EQUIPMENT

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DESIGN MANUAL OF METHODS OF FORCED AIR COOLING ELECTRONIC EQUIPMENT

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DEPARTMENT OF THE NAVY
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This publication, "Design Manual of Methods of Forced Air Cooling Electronic Equipment", NAVSHIPS 900,194 (C.A.L. Inc. report HF-845-D-19) is the fifth in a series of publications prepared for the Bureau of Ships by Cornell Aeronautical Laboratory Incorporated. This series of publications is intended to guide designers of electronic equipment in the use of convection, conduction and radiation heat transfer methods.

The other publications in the series are: "Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment", NAVSHIPS 900,189 (C.A.L. Inc. report HF-710-D-10); "Manual of Standard Temperature Measuring Techniques, Units and Terminology for Miniaturized Electronic Equipment", NAVSHIPS 900,187 (C.A.L. Inc. report HF-845-D-2); "Guide Manual of Cooling Methods for Electronic Equipment", NAVSHIPS 900,190 (C.A.L. Inc. report HF-710-D-16); "Design Manual of Natural Methods of Cooling Electronic Equipments", NAVSHIPS 900,192 (C.A.L. Inc. report HF-845-D-8); and the sixth and last of this series, which will be available approximately November 1958, "Design Manual of Methods of Liquid Cooling Electronic Equipment", NAVSHIPS 900,195 (C.A.L. Inc. report HF-845-D-9).

These publications are written for electronics personnel who are not well versed in thermodynamics. Information contained therein may be used in part or entirety in the preparation of other government publications.

Errors found in this publication (other than obvious typographical errors) should be reported to the Electronic Publications Section of the Bureau of Ships.

All Navy requests for copies of these publications should be directed to the nearest Bureau of Supplies and Accounts Forms and Publications Supply Point. Other requests should be directed to the Government Printing Office.

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Captain, U. S. Navy
Assistant Chief of Bureau for Electronics

I ABSTRACT

This Manual was prepared under the sponsorship of the Department of the Navy, Bureau of Ships, to aid electronic engineers in the thermal design of reliable electronic equipment. Principles and techniques to be followed in the design of electronic parts, assemblies, and equipments are outlined herein to permit heat transfer by forced convection to be reliably achieved. This Manual is an end product of Contract NObsr 63043 and supplements the previous Cornell Aeronautical Laboratory Manuals HF-710-D-16 (NAVSHIPS 900,190), HF-710-D-10 (NAVSHIPS 900,189), HF-845-D-2 (NAVSHIPS 900,187), and HF-845-D-8 (NAVSHIPS 900,192).

II INTRODUCTION

This Manual is the fifth publication of a series on the thermal design of reliable military electronic equipment. It has been prepared specifically to assist electronic engineers in the design of shipboard and ground-based equipment using forced-air cooling. Since the majority of future equipments will be cooled by forced air as part of the heat flow path to the ultimate sink, pertinent design data covering many configurations are presented in detail.

The necessity for adequate cooling to achieve reliability in military electronic equipment has been discussed in the other manuals of this series and in current technical literature. It cannot be overemphasized that reliability can be obtained only if the electronic, thermal, and mechanical designs are all well executed. The thermal design is fully as important as the circuit design.

The thermal design of ground-based and shipboard electronic equipment is emphasized in this Manual. No consideration has been given to non-steady-state heat transfer or operation at high altitudes. This Manual does not supercede NAVSHIPS 900,190 in whole or in part. Only minor theoretical material necessary for continuity has been duplicated between the two Manuals. In most cases methods of computation have been simplified by the use of nomographs and empirical algebraic equations. This Manual has been deliberately written at a technical level such that engineers without heat transfer background can design acceptable equipments. It outlines procedures to temper the designers' judgment and lead to the best practical solutions. When specific recommendations are given, care must be taken in adopting such recommendations so that they do not conflict with the requirements of the contracting activity.

Design formulas presented herein are usually simplified approximations which are adequate for their intended purpose. Electronic thermal design is not, at this writing, a precise mathematical science. The design tolerances are reasonably wide not only because of the somewhat limited knowledge of the thermal characteristics of all combinations of electronic heat sources, but primarily because of the normal variations in heat dissipation and flow between electronic heat sources of the same type. It is impractical to design a cooling system to an accuracy significantly greater than the electrical tolerances of the heat producing electronic parts used in the associated equipment.

It is recommended that shipboard electronic equipment be designed so that the heat dissipated is efficiently transferred into the sea. In general, this waste heat should not be rejected into the air in the compartment, or space, housing the equipment for later transfer to a ventilation or air conditioning system intended for personnel comfort. Consequently, it is necessary to short circuit the compartment air in the transfer of the heat from the source to the ultimate sink (the sea) by absorbing the heat in liquid cooled chassis or air-to-water-cooled heat exchangers within or adjacent to the equipment cabinet. Design techniques for forced-air cooling of heat sources will remain the same whether the air is exhausted into the compartment or passed through a water-cooled exchanger. However, the overall air-pressure drop may be slightly increased because of the exchanger and its associated ducts. Design data pertinent to such heat exchangers will be presented in a companion Manual, "Methods of Liquid Cooling Electronic Equipment", CAL Report No. HF-845-D-9.

We wish to mention particularly the productive cooperation we have received during this program from government agencies, universities, and industries. References of source materials are listed in the bibliography, together with reference numbers at pertinent locations in the discussion. The terminology used herein is presented in Cornell Aeronautical Laboratory Report No. HF-845-D-2. Symbols are listed in Appendix A. Design equations are identified by (D. E.).

Work on heat transfer in electronic equipment is continuing at this Laboratory. Future publications will supplement this Manual. Appendix C consists of a summary of reports already written and also planned under this program. Detailed information in these matters can be obtained from Mr. Rodney Hall, Code 817C3, Bureau of Ships, Washington 25, D. C. Comments are solicited.

This Manual is not to be construed as an endorsement of any commercial products mentioned.

III THEORY

A. General

Heat or thermal energy is transferred from one region to another by virtue of temperature difference. Although the basic laws of heat transfer are analogous to the electrical laws, transmission of heat is more complicated. There are three modes of heat transfer: conduction, convection, and radiation. In forced-air cooling, convection is the predominant mode of heat transfer, although some heat is also transferred simultaneously by conduction and radiation. Convection involves the transfer of heat by the mixing motion of the particles of a fluid.

Natural or free convection is the transfer of heat due to fluid motion resulting from differences in density caused by temperature differences. Forced convection is the transfer of heat due to motion of the fluid resulting from pressure differences produced by mechanical means.

1. The Mechanism of Convection

Convection is a complex process involving fluid mechanics as well as simple heat or energy transfer. When a mass of fluid moves with respect to a solid surface, the viscosity and "wetting ability" require that the fluid velocity be zero at the surface and increase continuously with increasing distance from the surface until the free stream velocity is reached (see Fig. 3-1). In laminar, or smooth, flow the fluid can be thought of as moving in layers or laminae, each moving faster than the one under it, and slipping on each other. Heat from the surface must pass by conduction through these layers of slowly moving fluid before it can be carried away by the rapidly moving fluid.

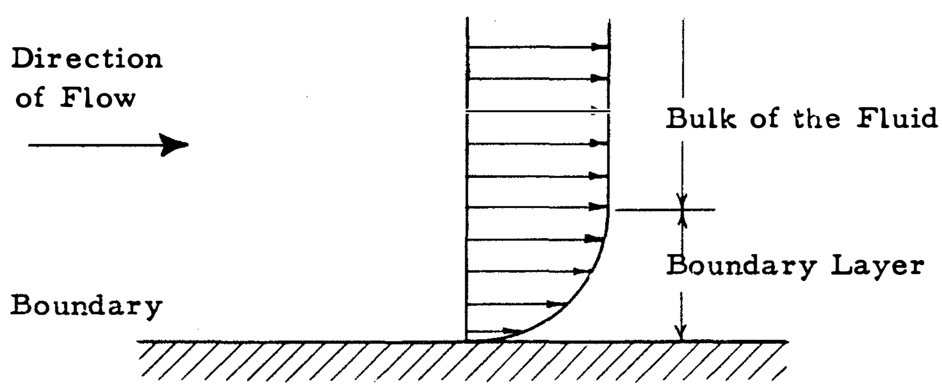
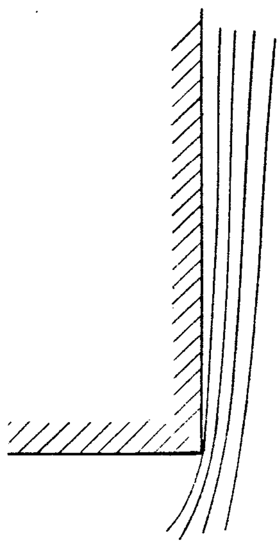
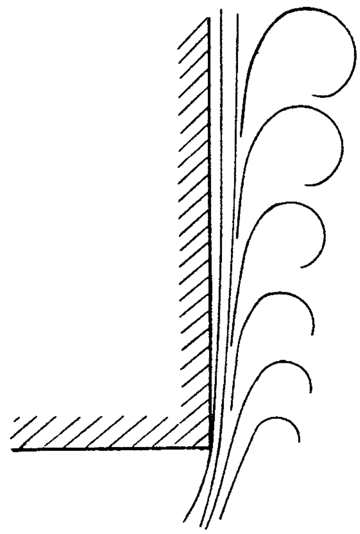


Figure 3-1 - Velocity Profile

If the free stream velocity is increased, as by a blower, a critical velocity is reached, depending on viscosity, density, and geometry of the surface, at which the laminae are broken up. Fluid particles then move in swirls and eddies and the local or point velocity is irregular (see Fig. 3-2(b)). This is called turbulent flow. Since the fluid velocity at the boundary must be zero, there is still a thin layer of laminar flow under the turbulence.



(a) Laminar Flow Lines



(b) Turbulent Flow with Laminar Sub-Layer

Figure 3-2

This laminar sub-layer is very much reduced in thickness when turbulent flow occurs. It is generally believed that this reduction in thickness reduces the resistance offered to heat flow. As soon as heat reaches the turbulent fluid it is rapidly carried away by the mixing due to velocity fluctuations. Increasing the free stream velocity increases the turbulence and reduces the sub-layer thickness. There seems to be no actual discontinuity in the flow, but turbulence gradually increases and the boundary layer gradually decreases in thickness as the force applied to move the fluid increases.

Turbulence is a measure of the steadiness of velocity. If dV is the mean fluctuation of vector velocity V, turbulence is defined by the ratio dV/V. The condition for the advent of turbulence is generally specified by a value of Reynolds number, which is defined on page 10. The velocities produced by natural convection are often not high enough to cause turbulence, but with forced convection, the velocity can be made as high as desired. Laminar or turbulent flow may occur with both natural convection and forced convection. In forced-air cooling of electronic equipment, the flow is usually turbulent. Increasing the velocity of the fluid flow decreases the thickness of the boundary layer through the increased turbulence and this results in a decreased resistance to heat transfer across the boundary layer, which resistance may be an order of magnitude less than that obtained with free convection. Thus, forced convection is a more effective method of heat removal than free convection.

2. Basic Equation

The basic equation for convection is:

$$q_c = h_c A_s \Delta t_s \qquad (1)$$

where:

q_c is the rate of heat transfer by convection

h_c is a convection coefficient of heat transfer

A_s is the surface area

Δt_s is the temperature difference between the surface and the main fluid stream (usually mean temperature difference),

The value of h_c, the convection coefficient, is influenced by many factors, including properties of the fluid, flow conditions, configuration, and surface characteristics. When a resistance analogy is applied to convection, the thermal resistance is $\frac{1}{h_c A}$.

3. Convection and Radiation

When convection and radiation occur simultaneously (as with electron tubes) the resistance analogy may be utilized by considering a fictitious radiation coefficient, such that the convection and radiation coefficients are additive. Then the radiation coefficient is defined as:(Ref. 20):

$$h_r = \frac{q_r}{A_s \Delta t_s} = \frac{\sigma_s F_e F_a (T_s^4 - T_a^4)}{t_s - t_a} \qquad (2)$$

where:

h_r is the radiation coefficient of heat transfer

q_r is the rate of heat transfer by radiation

A_s is the area of the surface

Δt_s is the temperature difference between the surface and the main fluid stream

t_s is the surface temperature of the hot part

t_a is the temperature of the surrounding air

Thus the total heat transferred from a surface is:

$$q = q_c + q_r = (h_c + h_r)\, A_s\ \Delta t_s = h\, A_s\ \Delta t_s \qquad (3)\ (D.\,E.)$$

where:

h is the combined coefficient of convection and radiation

$h = (h_c + h_r)$

This method "lumps" the two modes and thus simplifies the calculations. The heat exchanged between a multiplicity of electronic heat sources by radiation is sometimes difficult to predict analytically because of the uneven temperature distribution and irregularity of shapes.

4. Temperatures

Equation (1) is deceptively simple. The temperature difference must be very carefully defined because in practical cases the temperature varies from point to point both on the surface and throughout the fluid. In forced convection, the calculations can be simplified through the use of an average surface temperature, t_s. The bulk fluid temperature t_b presents a more difficult problem. When flow is confined to tubes, t_b may be taken on the axis. For unconfined flow, the free stream temperature may be used. Jakob (Ref. 17) recommends the "cup mixing temperature", that is, the temperature of the fluid in a short section normal to the direction of flow when uniformly mixed adiabatically. In forced-air cooling of a box, such as an electronic assembly, t_b can be taken as the average of exhaust and inlet fluid temperatures. The actual method of measuring a given Δt involving the above t_s and t_b, will be specified for the equations given.

5. Weight Rate of Flow

The amount of thermal energy absorbed by air is given by the following equation:

$$q = mc_p \, \Delta t_b = 7.62 \, m \, \Delta t_b \qquad (4)\ (D.E.)$$

where:

m is the weight rate of flow of air in pounds/min.

c_p is the specific heat in watt-min./(lb.)(°C)

Δt_b is the air temperature rise in °C

q is the heat rate in watts

For air, m is given by:

$$m = Q \times \rho \qquad (5)$$

where:

Q is the volumetric flow rate in cu. ft./min.

ρ is the air density in lbs./cu. ft.

The flow rate in cfm is equal to the average air velocity times the net cross-sectional area normal to the directional flow, or

$$Q = \frac{V \, A_{c_{net}}}{144} \quad (\text{cfm}) \qquad (6)\ (D.E.)$$

where:

V is the velocity of the flow

$A_{c_{net}}$ is the net cross-sectional area normal to the direction of flow

The density of air is given by the following:

$$\rho = 1.5 \, \frac{p}{t + 273} \qquad (7)$$

where:

p is the absolute pressure in lbs./sq. in. (or barometric reading in in. of mercury/2.036)

t is in ^{o}C

It is important to note that Equations (1) and (4) are the two basic equations to be used for calculating the amount of heat transferred by forced convection.

6. Mathematical Treatment

Differential equations can be written for fluid flow (the Navier - Stokes equations, Ref. 17) and for the convective heat flow. Integration subject to the boundary conditions is so difficult, however, that only a few simple configurations have been solved analytically.

A practical method of computing the amount of heat removed by forced convection is by dimensional analysis. Convective heat transfer involves ten variables:

h_c is the coefficient of convective heat transfer

k is the thermal conductivity of the fluid

c_p is the specific heat of the fluid at constant pressure (at the velocities normally used, the pressure is practically constant)

ρ is the density of the fluid

μ is the absolute or dynamic viscosity of the fluid

β is the coefficient of volumetric expansion of the fluid

t, T is the temperature in ^{o}C or ^{o}K respectively

g is the acceleration due to gravity

*L, D is the characteristic linear dimension corresponding to the diameter of circular cross-section parts

V is the velocity of the fluid

G is the mass velocity ($= V\rho = \frac{\text{lbs.}}{\text{min.-ft.}^2}$)

Four dimensionless products of these variables can be derived to reduce the number of variables from ten to four. This greatly simplifies experimental studies and leads to practical design information. The four products, named for famous investigators, are:

$$Nu = \frac{h_c L}{k} \quad \text{Nusselt number}$$

$$Re = \frac{VL\rho}{\mu} = \frac{LG}{\mu} \quad \text{Reynolds number}$$

$$Pr = \frac{c_p \mu}{k} \quad \text{Prandtl number}$$

$$Gr = \frac{L^3 \rho^2 g \beta \Delta T}{\mu^2} \quad \text{Grashof number}$$

*Note:

In the dimensionless parameters listed above, "L" and "D" can be used interchangeably, since all that is required is a linear dimension. "L" is generally used to represent length, width, height or distance, and "D" is used primarily to represent diameter. The particular problem under consideration dictates which symbol shall be used.

Any one of these products can be expressed as a function of the other three. Since h_c, the convection coefficient, is required for use in Equation (1), the functional form is written as:

$$Nu = \phi\,(Re,\ Pr,\ Gr) \tag{8}$$

The Reynolds number involves fluid velocity and applies only in forced convection when the fluid is mechanically driven. In free convection the Reynolds number is not involved and:

$$Nu = \phi\,(Pr,\ Gr) \tag{9}$$

The Prandtl number is a characteristic of the fluid, and for gases it varies slightly over wide ranges of environment. For air, the Prandtl number decreases 3.5% from 40°C to 175°C. It can therefore be considered constant in most electronic design applications.

The Grashof number involves the fluid acceleration due to heating, ($g\,\Delta t\,\beta$), and applies only in free convection. Therefore, in forced convection:

$$Nu = \phi\,(Re,\ Pr) \tag{10}$$

Dimensional analysis gives no clue as to the form of the function. As a result, this must be found experimentally.

Equation (10) is represented by a single family of curves and is relatively easy to apply. It is generally agreed that the functional form is:

$$Nu = C' Re^n \; Pr^a \tag{11}$$

where C', n and a are constants whose values depend on the units used and the geometry of the system.

Because the Prandtl number for air in the temperature range of interest to most electronic designers is essentially constant, the equation for forced-air cooling takes the form:

$$Nu = C \; Re^n \tag{12}$$

The equation can also be expressed as:

$$h = \frac{k}{D} \; C \left(\frac{DV\rho}{\mu}\right)^n = \frac{k}{D} \; C\left(\frac{DG}{\mu}\right)^n \tag{13}$$

As stated earlier, the value of h_c, the convection coefficient, depends upon several factors. Some factors (properties of the fluid, velocity of flow) appear in the dimensionless parameters (Nu, Re), while others (configuration, spacing of components, type of flow) determine the magnitude of the constants C and n. Forced convection is, therefore, subdivided into flow inside of ducts and tubes of various cross-sections (circular, rectangular, annular) and flow outside of components of various cross-sections. The flow outside of components can be subdivided further into flow parallel and normal (crossflow) to the axis of the components. Components in crossflow cooling may be arranged either staggered or in-line.

7. Relationship between Free and Forced Convection

There exists an interrelation between free and forced convection, which can be proven theoretically to be:

$$Nu = C''(Gr)^{\frac{n}{2}} \; (Pr)^a \tag{14}$$

Hence, the square root of the Grashof number can be considered a special case of the Reynolds number. This theoretical result is in satisfactory agreement with the results of experiments.

8. Electronic Parts

Complete design information has not been developed for the convective cooling of all combinations of electronic heat sources. However, basic heat transfer theory and data on idealized shapes have been found applicable with some modifications. The problem of cooling electronic equipment is made complex by the hot spots on the surfaces of the heat sources resulting from an uneven distribution of the internally generated heat. In crossflow cooling, the variation of hot spot temperatures along the surfaces of heat-producing parts is affected by the basic configuration and the part spacings. The small size of electronic heat-producing parts results in "end-effects" which tend to raise heat transfer coefficients, thus decreasing their thermal resistance. Further, the heat transfer coefficients for forced-air cooled electron tubes are also higher than the common convective coefficients, because the former include radiation coefficients, while the latter are corrected for pure convection.

B. Flow Within Tubes or Pipes

Figure 3-3 shows an axial section of a round pipe. The wall temperature, t_w, is constant at any section normal to the axis. Fully turbulent flow causes thorough mixing and results in a nearly

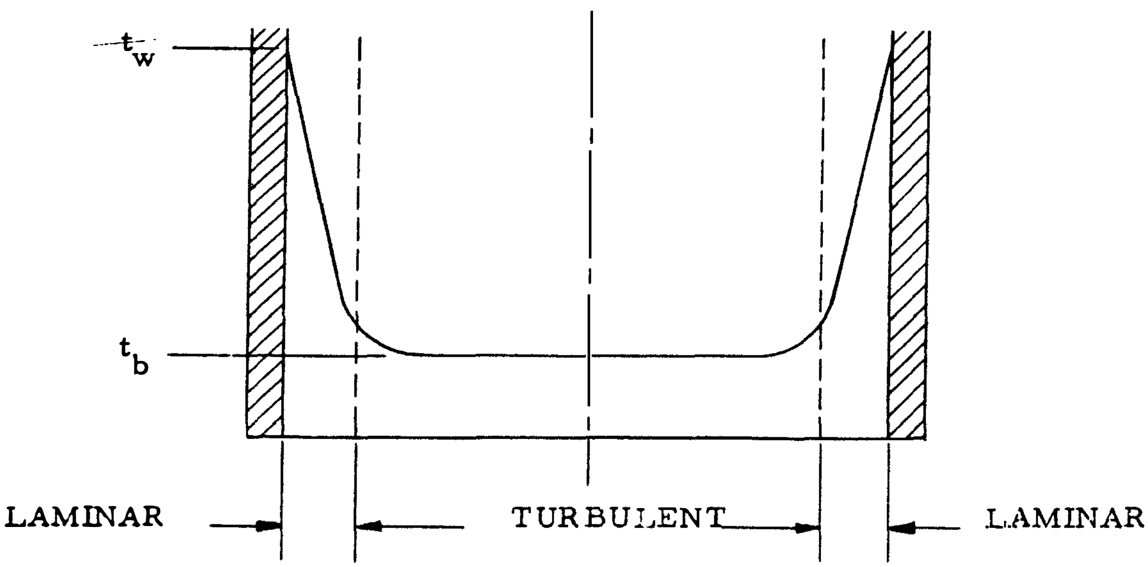


Figure 3-3 - Fluid Temperature Distribution for Flow within a Round Pipe

constant bulk temperature, t_b, in the bulk of the fluid. In the thin laminar film adjacent to the wall there is a steep and nearly uniform temperature gradient, so that the film temperature, t_f, may be taken as an average value, as in Equation (15):

$$t_f = \frac{t_w + t_b}{2} \qquad (15)$$

If heat is to be exchanged, the temperatures must of course vary axially along the tube. Since the heat transfer coefficient, h_c, is a function of fluid properties which vary with temperature, h_c will have a changing value along the tube. Evidently it becomes necessary to evaluate h_c locally, compute the heat flow in a differential element and integrate over the tube length. Formal integration of the functions is impossible, however, so the tube is treated by sections short enough so that temperature changes are relatively small and average values may be used.

For Reynolds numbers greater than 10,000 (fully turbulent flow), L/D ratio of the tube greater than 60, and with fluids having viscosities less than twice that of water, two equations have been found applicable.

$$h = 0.023\,(c_p)_b\, G_{max} \left(\frac{c_p\ \mu}{k}\right)_f^{-0.67} \cdot \left(\frac{D\,G_{max}}{\mu_f}\right)^{-0.2} \qquad (16)\ (D.\ E.)$$

Equation (16) is based on film temperature.

$$h = 0.023\ \frac{k_b}{D} \left(\frac{D\,G_{max}}{\mu_b}\right)^{0.8} \cdot \left(\frac{c_p\ \mu}{k}\right)_b^{0.4} \qquad (17)\ (D.\ E.)$$

Equation (17) is based on bulk temperature.

The subscripts b and f indicate values at bulk and film temperatures respectively. G_{max} indicates the largest value of mass velocity, normally occurring at the point of minimum cross-section.

Using Pr = 0.69 and c_p = 7.62 watt-min./lb.-^{o}C for air these equations become

$$h = 0.224\ \mu_f^{0.2} \cdot \frac{G_{max}^{0.8}}{D^{0.2}} \qquad (18)\ (D.\ E.)$$

$$h = 0.0198\ \frac{k_b}{(\mu_b)^{0.8}} \cdot \frac{G_{max}^{0.8}}{D^{0.2}} \qquad (19)\ (D.\ E.)$$

Heat exchangers to be used in the design of forced-air cooled equipment will, usually, be of air-to-water type. Because of limitations, discussed in a later manual, the water temperature rise will be limited to 5°C maximum (from 35°C to 40°C). Because of this small temperature range, the physical properties of water can be assumed to be constant. Based upon 38°C, and Equation (17), both heat transfer coefficient equations take the following simplified form:

$$h = 0.036 \frac{G_{max}^{0.8}}{D^{0.2}} \qquad \text{(20)(D. E.)}$$

For ducts of non-circular cross-section, the equivalent diameter D_e for use in these equations is defined by:

$$D_e = \frac{4 \text{ x cross-sectional area}}{\text{cross-sectional perimeter}} = 4 \text{ x hydraulic radius} \qquad (21)$$

C. Flow of Gases Parallel to Plane Surfaces, Wires and Cylinders

1. Turbulent Flow

The film coefficient for turbulent air flowing parallel to smooth plane surfaces is given by (Ref. 18):

$$h = 0.055\left(\frac{k}{D}\right)\left(\frac{DV\rho}{\mu}\right)^{0.75} \qquad \text{(22)(D. E.)}$$

Here D is the length of the surface and is limited to two feet, even if the length of the surface is greater (Ref. 18).

The heat transferred, from a cylinder to a fluid which flows parallel to its axis, will not be much different from that between a plane plate and a fluid in parallel flow, provided that the diameter of the cylinder is large. For cylinders of small diameter the eddies occurring in turbulent flow may be great compared with the cylinder diameter and the flow will become more similar to that perpendicular to the axis (Ref. 17).

Heat transfer coefficients for turbulent flow of air parallel to flat plates and cylinders are expressed by (Ref. 17):

$$Nu = 0.0280 \, (Re)^{0.8}; \text{ for } Pr = 0.71 \qquad \text{(23)(D. E.)}$$

or

$$Nu = 0.0272 \, (Re)^{0.8}; \text{ for } Pr = 0.69 \qquad \text{(24)(D. E.)}$$

2. Laminar Flow

For laminar flow of air the heat transfer coefficients are related to other variables by (Ref. 17):

$$Nu = 0.592 \, (Re)^{0.5}; \text{ for } Pr = 0.71 \qquad (25)(D.E.)$$

or

$$Nu = 0.586 \, (Re)^{0.5}; \text{ for } Pr = 0.69 \qquad (26)(D.E.)$$

3. Electron Tubes

Reference 10 reports that for forced convection from any electron tube, with air flow upward and parallel to the axis of the tube, surrounded by a reflective flow baffle, a general equation is obtained as follows:

$$Nu_b = 0.313 \, (Re_b)^{0.6} \qquad (27)$$

(Region tested was for Bulk Reynolds Numbers, Re_b, between 130 and 8000.)

Investigations at this Laboratory with air flow horizontal and parallel to the axis of subminiature tubes, mounted inside insulated white cardboard ducts, indicate that:

$$Nu_b = 0.337 \, (Re_b)^{0.55} \qquad (28)$$

(Region studied was for Bulk Reynolds Numbers, Re_b, between 700 and 5000.)

The Nusselt and Reynolds numbers were based upon the tube diameter. The inlet bulk air temperature was used in Equations (27) and (28) for the evaluation of the properties of air.

D. Crossflow

1. Flow of Air Across Single Wires and Cylinders

The equation for the film coefficient across a single wire or cylinder takes the form:

$$\frac{hD}{k_f} = C \left(\frac{DV\rho}{\mu_f} \right)^n \qquad (29)(D.E.)$$

TABLE I

(From References 17 and 19)

CONSTANTS FOR USE IN EQUATION (29) FOR ROUND CYLINDERS

Reynolds Number $Re = \frac{VD\rho}{\mu}$	C	n
0.4 - 4	0.891	0.330
4 - 40	0.821	0.385
40 - 4,000	0.615	0.466
4,000 - 40,000	0.174	0.618
40,000 - 400,000	0.0239	0.805

The constants C and n are dependent on the magnitude of the Reynolds number and are given in Table I. The air properties should be determined at the mean of the arriving air and surface temperatures (film temperature). The convective coefficient, h, is based on the difference in these temperatures. The arriving air velocity is the reference velocity.

2. Flow of Air Across Cylinders of Non-Circular Cross-Section

The equation for the film coefficient for air flowing normal to the axis of cylinders of non-circular cross-section, is similar in form to that for cylinders of circular cross-section. The diameter of a circular tube with exposed surface equal to that of the non-circular cylinder should be used as the characteristic length in Equation (29). The constants C and n depend on the orientation of the cylinder with respect to the direction of air flow, and are given in Table II.

Note that the constant C for the flow parallel to a diagonal of square cross-section is approximately twice that for flow parallel to a side. In general the non-circular profiles offer better heat transfer than the circular ones.

TABLE II

(From References 17 and 19)

CONSTANTS FOR USE IN EQUATION (29) FOR CYLINDERS OF NON-CIRCULAR CROSS-SECTION

Cross-Section (investigator A - Reiher, B - Hilpert)	Description of Cross-Section	Description of Flow	Re Range	C	n
(A)	square	perpendicular to front face	2,500 - 8,000	0.160	0.699
(B)	square	perpendicular to front face	5,000 - 100,000	0.092	0.675
(A)	square	parallel to diagonal	2,500 - 7,500	0.261	0.624
(B)	square	parallel to diagonal	5,000 - 100,000	0.222	0.588
(B)	hexagonal	parallel to diagonal	5,000 - 100,000	0.138	0.638
(B)	hexagonal	perpendicular to face	5,000 - 19,500	0.144	0.638
(B)	hexagonal	perpendicular to face	19,500 - 100,000	0.035	0.782
(A)	thin, flat plate	perpendicular to flat	4,000 - 15,000	0.205	0.731
(A)	elliptical	parallel to major axis	2,500 - 15,000	0.224	0.612
(A)	elliptical	parallel to minor axis	3,000 - 15,000	0.085	0.804

The influence of mechanical disturbances of the flow at the surface may be seen from Table III. Constants C and n for air flow normal to cylinders with fins and grooves are given in Table III (Ref. 17).

TABLE III

VARIATIONS OF "C" AND "n"

Surface Disturbance	Re	C	n
None	400 - 6000	0.350	0.560
Longitudinal fin 0.1D thick, on front of tube	1000 - 4000	0.248	0.603
12 longitudinal grooves 0.07D wide	3500 - 7000	0.082	0.747
Same, with burrs	3000 - 6000	0.0368	0.86

As much as 50% increase in the average coefficient of heat transfer was obtained by some experimenters when the turbulence was increased by passing the air, flowing over a single pipe, through a grid. Similar effects were found with spheres.

3. Flow of Air Over Spheres

For air flow over spheres, the film coefficient is:

$$\frac{hD}{k_f} = 0.37\left(\frac{DG}{\mu_f}\right)^{0.6} \qquad (30)(D.E.)$$

for Reynolds numbers, Re_f, between 17 and 70,000 (Ref. 20)

4. Flow of Air Across Banks of Tubes or Cylinders

The relationship between Nusselt number and Reynolds number ($Nu = CRe^n$) for crossflow cooling of banks of tubes, depends upon the number of rows and upon the longitudinal and transverse spacings. These factors affect the turbulence of the air stream, and, therefore, the heat dissipation by convection.

Note:

A bank of tubes is a two-dimensional array, consisting of longitudinal rows and transverse rows. A longitudinal row consists of several tubes in a straight line parallel to the direction of flow, whereas a transverse row consists of several tubes in a straight line perpendicular to the direction of flow.

a. Correlation of Data

The correlation of data for the flow of air across banks of cylinders of circular cross-section takes the same form as Equation (29). The Reynolds number, $DV\rho/\mu$, may be written as DG/μ, where G is the mass velocity in pounds per hour-sq. ft. of flow cross-section. The value of G in the correlation is that obtained at the narrowest cross-section between the vacuum tubes, whether or not the minimum area occurs in the transverse or diagonal openings between tubes. Thus, Equation (29) is used in the following form:

$$\frac{hD}{k_f} = C\left(\frac{DG}{\mu_f}\right)^n \qquad (31)$$

The constants, C and n, depend on the distance between the tubes, the tube diameter and the configuration, i.e., staggered or in-line (see Fig. 3-4).

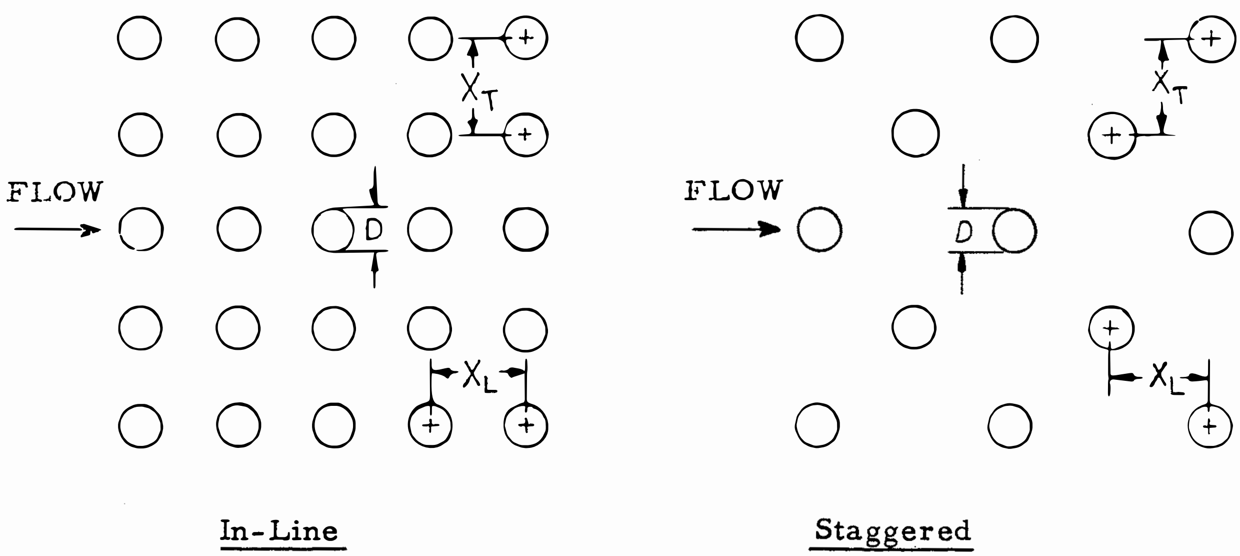


Figure 3-4 - Tube Banks with Tubes In-Line and Staggered

For forced-air cooling of banks of tubes, the values of C and n in Equation (31) depend upon the following parameters:

$$S_T = \frac{X_T}{D} \; ; \; S_L = \frac{X_L}{D}$$

where:

X_T is the transverse center to center distance between the heat sources in inches.

X_L is the longitudinal center to center distance between the heat sources in inches.

D is the heat source diameter in inches.

b. Effect of the Number of Transverse Rows of Tubes on the Heat Transfer Coefficient

When calculations for flow across fewer than four transverse rows are based upon the data for flow across more than four transverse rows, appreciable error may result. The tube surface temperature could run higher than predicted, since there is a decrease in the heat transfer coefficient. For 2 to 4 transverse rows, calculations should be based upon data for a single transverse row which will yield conservative results; or calculations based upon the data for more than four transverse rows should be reduced by an appropriate factor as shown in Table IV.

In general, increasing the number of transverse rows of tubes or pipes in the direction of air flow produces increased turbulence toward the rear of the tube bank. Hence, the heat transfer coefficients are greater for the tubes in the rear. References 19 and 20 give values of C and n obtained by Grimmison, to be used in Equation (31) for banks of tubes or pipes 10 transverse rows deep.

TABLE IV

RATIO OF MEAN FILM HEAT TRANSFER COEFFICIENT FOR n TRANSVERSE ROWS DEEP TO THAT FOR 10 TRANSVERSE ROWS DEEP

n	1	2	3	4	5	6	7	8	9	10
Ratio for Staggered Rows	—	0.73	0.82	0.88	0.91	0.94	0.96	0.98	0.99	1.00
Ratio for In-Line Rows	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99	1.00

However, these values are not directly applicable to electronic components and are, therefore, replaced by the data obtained by Robinson for cylinders with configurations and heat dissipations similar to those of electron tubes.

Robinson has found that heat transfer coefficients for in-line and staggered arrangements of prismatic heat sources are the same for the third to second last transverse row and slightly lower for the first, second and last transverse rows (for cylindrical parts the first three transverse rows have lower heat transfer coefficients). Turbulators could be placed in front of the parts with low heat transfer coefficients to increase turbulence, which in turn will increase the rate of heat transfer.

c. Electronic Parts

(1) Cylindrical Heat Sources

Robinson (Ref. 8, pp. 67-71) expresses the Nu_f; Re_f relationship in the form:

$$Nu_f = CF\, Re_f^{\,n} \tag{32)(D. E.}$$

where:

F is a configuration factor, dependent upon the cylinder geometry.

By applying the factor, F, all data were correlated with McAdams' (Ref. 20) data for crossflow cooling of single cylinders in a free air stream. Values of C and n as determined by Robinson are given in Table V, and expressions for F are given below.

TABLE V

VALUES OF C AND n FOR USE WITH EQUATION (32)

(Re_f) ave.	C	n
1,000 - 6,000	0.409	0.531
6,000 - 30,000	0.212	0.606
30,000 - 100,000	0.139	0.806

Note:

These values give a curve of Nu vs. Re which falls on McAdams curve for a single cylinder in a free air stream.

(1.) For single cylinder in free stream: $F = 1$

(2.) For single cylinder in duct: $F = \left(1 + \sqrt{\frac{1}{S_T}}\right)$

(3.) For in-line cylinders in a duct:

$$F = \left\{1 + \sqrt{\frac{1}{S_T}}\right\} \cdot \left\{1 + \left[\frac{1}{S_L} - \frac{0.872}{S_L^2}\right] \cdot \left[\frac{1.81}{S_T^2} - \frac{1.46}{S_T} + 0.318\right] \cdot \left[Re_f^{\left(0.526 - \frac{0.354}{S_T}\right)}\right]\right\}$$

(4.) Staggered cylinders in a duct:

$$F = \left\{1 + \sqrt{\frac{1}{S_T}}\right\} \cdot \left\{1 + \left[\frac{f_1}{\sqrt{S_L}} - \frac{f_2}{S_L}\right] Re_f^{\;0.13}\right\}$$

where:

$$f_1 = \frac{15.50}{S_T^2} - \frac{16.80}{S_T} + 4.15$$

$$f_2 = \frac{14.15}{S_T^2} - \frac{15.53}{S_T} + 3.69$$

S_T and S_L are defined on page 20. . Experiments performed at CAL on the crossflow cooling of banks of electron tubes are in general agreement with those of other investigators. For example, the investigation of crossflow cooling of 5902 subminiature tubes mounted in an insulated white cardboard duct showed that:

$$Nu_f = 0.15\ (Re_f)^{0.697} \tag{33}$$

for $S_T = 2.49$ and $S_L = 1.33$. The region tested was for Reynolds numbers, Re_f, between 800 and 6000. When the above equation is plotted, the curve coincides with Robinson's curve for $S_T = 2.333$ and $S_L = 1.250$.

The following equation was determined for crossflow cooling of 805 tubes:

$$Nu_f = 0.067\ (Re_f)^{0.778} \tag{34}$$

for $S_T = 1.57$ and $S_L = 1.46$. The region tested was for Reynolds numbers, Re_f, between 2500 and 15,000. This equation, when plotted, falls between Robinson's and McAdams curves for the corresponding S_T and S_L values. Here, higher radiation losses were incurred, because of poorer insulation and the higher temperature level. As presented in Fig. 3-5, there is a very good agreement between the experimental results achieved at this Laboratory and those of Robinson for a single tube in a duct.

(2) Prismatic Heat Sources

For the crossflow cooling of prismatic electronic components, with the side of the components perpendicular and parallel to the direction of flow, Robinson derived the following relationships:

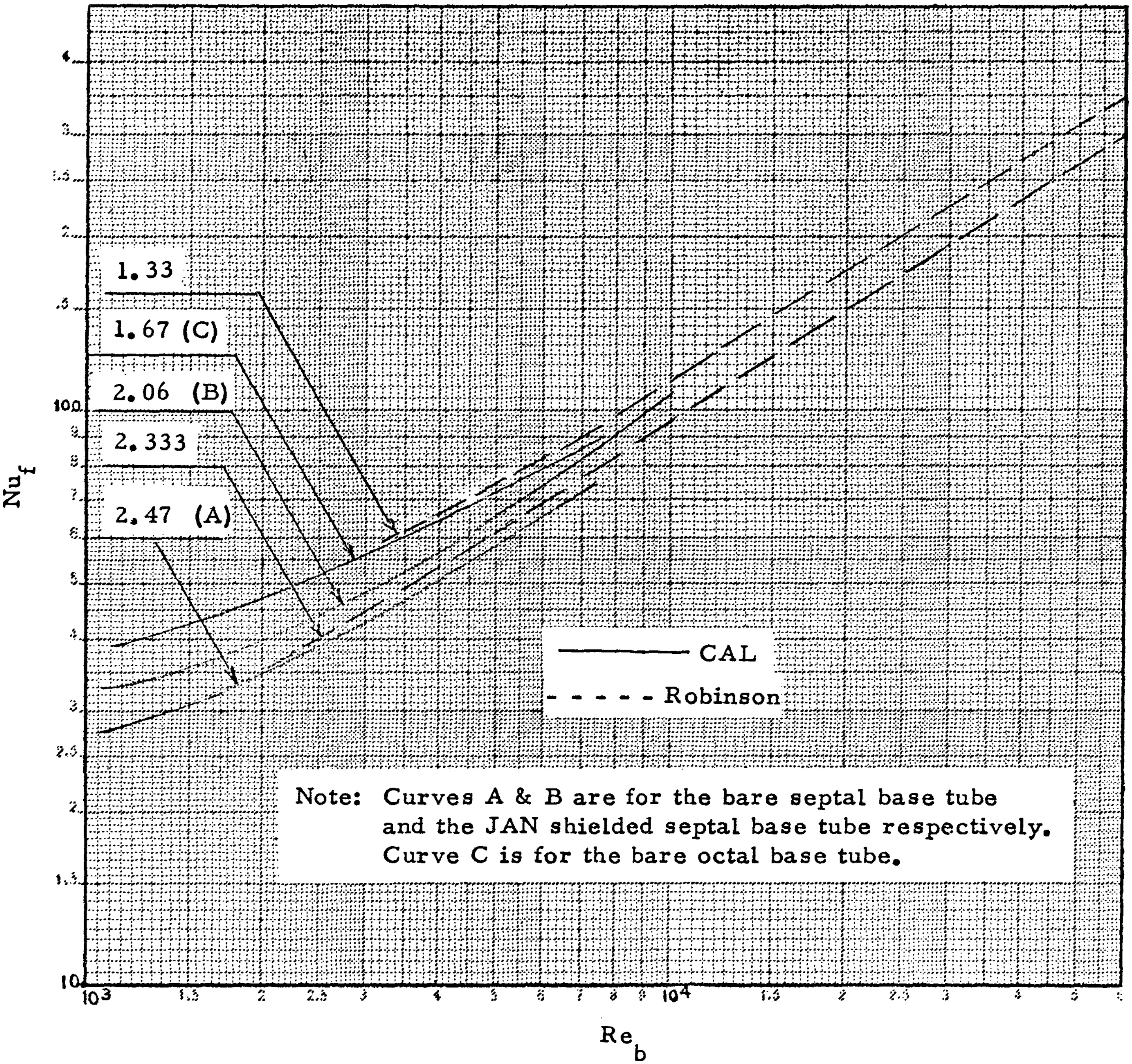


Fig. 3-5 Crossflow Data Correlation For a Single Tube.

For a single prism in a duct:

$$Nu = 0.446 \left\{ Re_o \cdot \frac{1}{\frac{1}{6} + \frac{5A_n}{6A_o}} \right\}^{0.57} \qquad (35)(D.E.)$$

(For Reynolds number, Re_o, between 2500 and 8000.)

where:

Re_o is the Reynolds number based upon gross duct flow area and prism side dimension

A_o is the gross duct flow area

A_n is the net duct flow area

In the free air stream and with the characteristic dimension $D_e = \frac{4d}{\pi}$ (diameter of a tube of equal surface area), the above equation reduces to:

$$Nu = 0.495\ (Re)^{0.57} \qquad (36)(D.E.)$$

Comparing Equation (36) with Jakob's data, the maximum difference is only 12% in the given range of Reynolds numbers.

For the in-line arrangement an equation was not determined experimentally because of the non-linearity of several of the measured curves and differences in their slopes. (However, a plot is presented by Robinson in Reference 12, page 47.)

For staggered arrangement the equation takes the form:

$$Nu = 0.446 \left[1 + 0.639 \left(\frac{S_T}{S_{T_{Max.}}} \right) \left(\frac{D}{S_L} \right)^{0.172} \right] \cdot \left[Re \frac{1}{\frac{1}{6} + \frac{5A_n}{6A_o}} \right]^{0.57} \qquad (37)(D.E.)$$

where:

$S_{T_{Max.}}$ - refers to maximum value of S_T when there are different values in the same configuration.

No data are available on forced-air cooling parallel to the diagonals of the prismatic electronic parts, although according to Jakob, better cooling should be achieved.

E. Design Considerations

1. Design Methods

The actual design methods for the cooling of electronic heat sources, which appear in Chapter VI, are based on the equations presented in this chapter. Most of the methods depend purely on a mathematical analysis of the problem at hand. However, a method evolved at this Laboratory involves some empirical relationships, in the form of h vs. Re curves. This system presents a rather simple method for several basic types of cooling configurations. It will be explained fully in Chapter VI.

2. Z - Factor Approach

Since the formulas for the heat transfer coefficients for air flow across banks of tubes are complicated, and the application of these formulas is time consuming, Robinson devised equations and charts to be used instead. These charts incorporate some new concepts, primarily one called the heat dissipation factor, Z. A complete discussion and the charts to be used can be found in Reference 9.

A series of investigative studies were conducted to determine the effectiveness of these charts as design material. As a secondary result, some additional information, shown in Table VI, was discovered to be applicable.

Even though this method was designed to eliminate laborious calculations, its merits do not warrant total acceptance. This method may, however, be considered a reasonably reliable alternate to the design methods presented in Chapter VI, and can be used for the few cases not covered.

TABLE VI

SUPPLEMENTARY DATA FOR USE WITH Z - FACTOR APPROACH

Tube Type	l_e, IN.
6AQ5	0.8
6L6 (Metal)	2.8
6L6-G	2.7*
6L6-GBY	2.9*
805	6.8
5902 (Sub.-Ass'y.)	1.1

*Use l_e = 2.8 as an average value

Since it is quite difficult to determine l_e accurately, the values are only approximate. They yield, however, satisfactory results when applied to charts and equations for crossflow forced-air cooling.

Design equations and curves, based on the heat dissipation factor, Z, for cylindrical heat sources are also applicable to forced-air cooling of prismatic parts. The results will be conservative since the heat transfer coefficients are higher (resistance to heat flow lower) for non-circular shapes because of the increased turbulence.

3. Surface Temperature Variation

The equations for the film coefficient of air under forced convection all presume an isothermal surface. However, electronic parts, such as tubes, resistors and enclosing boxes, do not have isothermal surfaces, and considerable surface temperature variation may exist. This temperature variation is not only a function of the internal structure of a part such as a vacuum tube, for example, but may vary with the manner in which the part is cooled. Since the heat transfer rate is a function of the difference between surface and air temperature, an average temperature must be used. If the maximum or hot-spot temperature is used, an erroneously high calculated heat transfer rate will result.

There are no exact rules by which the average temperature of a heated part or container can be calculated. However, a reasonable estimate is usually satisfactory. Information on this matter is included in Chapter VI. Reference 10 also presents some data related to the temperature distribution over the surface of four electron tube types cooled by several methods.

It is possible to obtain an average temperature by direct experiment. Thermocouples can be placed on the surfaces of the component, and an average temperature can be calculated on an area-weighted basis. Such a test should be conducted with whatever cooling method is to be used.

4. Finned Surfaces

The effectiveness of forced-air cooling usually can be increased by providing extended surfaces or fins over which the air is directed. Fins provide additional heat transfer by conducting heat outwardly from the body to which they are attached, thus effectively increasing the surface area of the heat source. The additional heat transfer provided by fins, more than offsets the slight increase in resistance due to the metallic heat flow path of the fins. The mathematics of fins is too involved to be presented here. Excellent treatment of extended surfaces is given in Reference 17.

There are several important general factors to consider:

(a) The extended surface should be of a metal with a high thermal conductivity.

(b) The fins should be either integral with the part or bonded to the part in a good metal-to-metal contact so that there is a minimum of contact resistance.

(c) Short, rather thick fins are more effective than long thin ones. The temperature drop from the base to the tip of a long fin may be appreciable, and tends to make the fin less effective.

(d) Where applicable, weight and space requirements should also be considered.

5. Comparisons

CAL test conditions differ from those of other investigators whose data and curves were used for comparison. The difference in results, however, is justified, even under similar conditions. Jakob cautions that in the equation, $Nu = CRe^n$, small differences in n, which occur in measuring the slope of a straight line, which, in bilogarithmic coordinates, represents the above equation, generally cause considerable changes in the constant C. Hilpert performed experiments on the flow of air of uniform velocity perpendicular to the axis of cylinders. When Nu vs. Re were plotted on bilogarithmic coordinates, the test points fell in a sequence of straight lines with bends at Re = 4, 40, 4000, 40,000. The bends obviously indicate changes in the character of flow. These bends were also found by other experimenters.

Jakob also states that when dimples were made in the flow duct, increased Nusselt numbers were noted (due to increased turbulence).

Since an electron tube is not as simple a geometric figure as the cylinders used in the above mentioned works, and radiation losses are lumped for miniature and subminiature tubes, CAL Nu vs. Re curves are approximate. However, they agree quite well with the published data. Increased radiation at low air flow rates account for high Nu values at low Re values. This is especially true of single tubes in metal ducts (septal, octal, noval). Cardboard ducts were used to decrease the losses to the surroundings in the 5902 assembly studies. Curves based on McAdams' data and Robinson's graphs (Ref. 8) for crossflow in-line cooling indicate that the Nusselt number for the 805 tube configuration should be greater than for the 5902 subminiature tubes.

However:

(1) the values obtained for the 805 tubes may be low, because of assumptions made in computing Nu based on total surface area of the tube and computing Δt_s from temperature readings on the upper half of the tube;

(2) Error in computing the average Δt_s might be quite large, since the value of Δt_s is only approximate for the 805's and 5902's;

(3) The duct was not insulated for the testing of the 805 tubes and the power level was higher, therefore, the amount of heat convected to the air was lower because of losses to the surroundings.

CAL curves fall, however, between the curves presented by McAdams and those presented by Robinson. Nu_f vs. Re_f curves for the single tubes agree well with Robinson's plots.

The performance curve for a JAN-shielded septal-base tube falls above the curve predicted by the theory (Section D-1-c-(1)), probably because more heat was lost by conduction to the surroundings, causing the actual heat transfer coefficient to be smaller than that calculated. However, the variation is not significant because low power levels are involved.

Robinson's Nu_f was based on the actual heat transfer coefficient h, which was taken as average of all the point heat transfer coefficients around the tube surface. To determine the heat transfer coefficient at any point heat flux and temperature difference must be known, as in Equation (38):

$$h = \frac{q/A}{t_s - t_b} \qquad (38)$$

By using cylinders with heating elements inside, Robinson could determine the heat flux by measuring temperature drop across a shell of known conductivity. This is not practical in the case of an electronic tube, therefore Nu_f must be based upon total heat dissipation.

IV FAN FUNDAMENTALS

A. Fan Types and Limitations

1. Definitions

A fan is a machine for applying power to a gaseous fluid. For the purposes of this Manual, a fan is an air-moving device.

Fans are classified as either blowers or exhausters. A blower is a fan used to force air under pressure across the surface or configuration to be cooled. The resistance to the air flow is imposed primarily upon the blower outlet. An exhauster is a fan used to withdraw air under suction across the surface or configuration to be cooled. The resistance to air flow is imposed primarily upon the exhauster inlet.

Blowers are recommended for the forced-air cooling of electronic equipment. Directing air under positive pressure usually provides more effective cooling than withdrawing air under suction.

2. General

Centrifugal and axial-flow fans or blowers are the two basic types. The type to be selected for a specific cooling problem is dependent on several factors, such as air flow and pressure requirements, efficiency, speed, space, the air ducting system, noise and fan characteristics.

Theoretically a fan can be used for cooling any system which has a pressure drop low enough so that the temperature rise of the air in the fan and motor does not cause the discharge air temperature to be equal to or greater than the temperature of the surfaces to be cooled. Temperature rise of air in the fan is due to the increase of kinetic energy. Since it is also necessary to cool the motor driving the fan, there is a temperature rise due to the motor cooling. Usually this rise is small under sea-level conditions. In practice there are other factors, such as weight, size, rotational speed, and power which must be considered.

Weight must be considered even in shipboard equipment. Fan size is usually limited by space. For very large external blade diameters, blade tip speeds will be high, possibly exceeding the safe design limits. High rotational speeds will shorten bearing and brush life. If the incorrect size and type of fans are selected for a particular system to be cooled, inefficient operation can cause excessive power consumption.

3. Centrifugal Fans

A typical centrifugal fan is shown in Fig. 4-1. The three important parts are: the housing, containing the air inlet and outlet; the rotor containing the fan blades or vanes; and the external driving motor. The air enters the housing normal to the side through a single or double inlet, i.e., at one or both sides of the center of the rotor, and is discharged in a direction perpendicular to the axis of rotation.

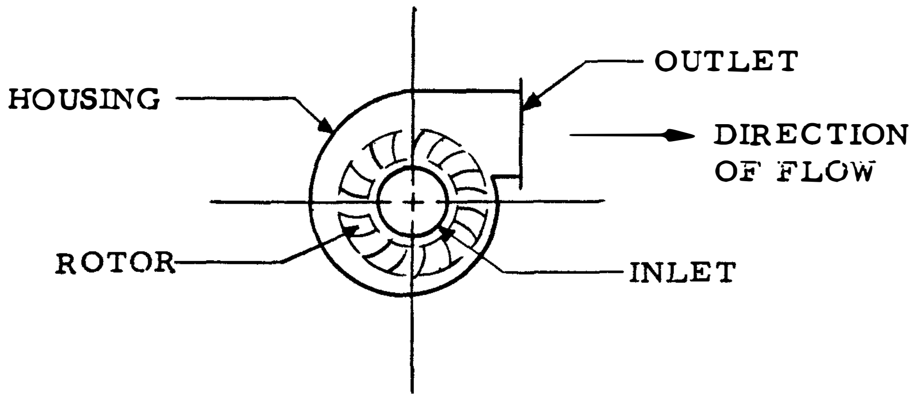


Figure 4-1 - Typical Centrifugal Fan

The centrifugal fan is generally used where high pressures are required. Centrifugal fans can have a single inlet or double inlet. For a given rotor speed and diameter, a double-inlet fan of the same type has greater air capacity and roughly the same pressure-producing ability as the single inlet unit. Centrifugal fans, capable of producing high air-pressure, have a special rotor construction which permits operation at the high peripheral blade speeds necessary for high pressure generation.

Centrifugal fans may be divided into three groups according to the shape of the blades: (1) forward-curved blades, (2) radial blades, (3) backward-curved blades.

Forward-curved blades curve away from the radial direction in the direction of rotor rotation. Backward-curved blades curve opposite to the direction of rotation. For a given rotor speed and diameter, the forward-curved blades have the greatest pressure-producing ability and the backward-curved blades the smallest. Hence, when fan size is a limiting factor, fans with forward-curved blades should be used. Fans with forward-curved blades have, however, relatively poor stability of operation and considerably less flexibility of control. Fans with backward-curved blades do not have the above disadvantages, and fans with radial blades are also considered satisfactory in this respect. In mechanical strength, the radial type centrifugal fan is superior to the other two types. Considering the advantages and disadvantages of the three types of centrifugal fans, the radial and forward-curved blade fans are best suited for cooling of electronic equipment. In most applications the forward-curved blade fan would prove the best because of its greater pressure producing ability for a given speed and size. However, care must be exercised to avoid the possibility of over-loading the drive motor, because of the inadvertant reduction in flow resistance. The radial type fan should be employed when appreciable pressure is required from a reasonably small unit. The drive and the load must have fairly constant control characteristics.

4. Axial-Flow Fans

Axial-flow fans are generally one of two types: the so-called propeller type, shown in Fig. 4-2, and a more efficient design shown in Fig. 4-3, commonly considered to be more truly an axial-flow fan.

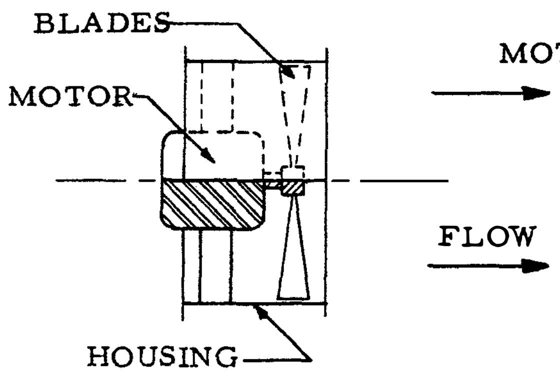


Figure 4-2

Propeller Type Fan

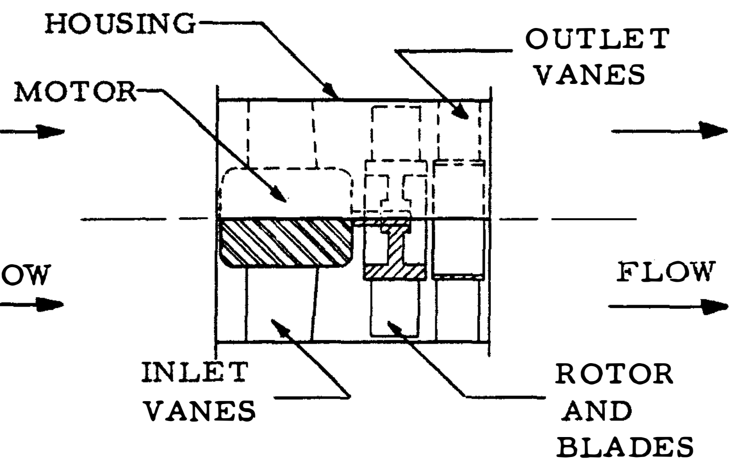


Figure 4-3

Axial-Flow Fan

There are several types of construction, but all are typified by the straight-through flow of air. The axial-flow fan is a more efficient design and is a higher pressure fan than the propeller type.

Propeller type fans are employed usually as circulating devices, while axial-flow fans are applicable to cooling of certain electronic equipments requiring high volume air flow at moderate to low system resistances. When appreciable increase in static pressure is required, several rotors are usually placed in series. Axial fans of this type are thus called multi-stage fans.

Axial fans are suitable for cooling of electronic equipment having low flow resistance and constant volumetric requirements and pressure performance. Because of their configuration, axial-flow fans can be installed in air ducts without change in flow direction.

5. Comparison of Centrifugal and Axial-Flow Fans

Centrifugal fans are better suited to the cooling of electronic equipment than axial-flow fans, because of their favorable proportions and control. However, the externally mounted motor requires cooling and occupies valuable space. Axial-flow fans are less desirable because of their inferior control characteristics, and the complicated multistage configurations needed to produce appreciable air pressure. Single stage units are no more complicated than centrifugal fans, but they deliver air at very low pressures. Because of their higher operating efficiency and greater volumetric capacity for comparable diameters, axial-flow fans deserve consideration. When designing for the consumption of minimum power, they are applicable to the cooling of large units, especially those dissipating 10 kw and over.

6. High-Performance Axial-Flow Fans

Since the major disadvantage of the axial-flow fans is the low pressure head, high performance one-and-two stage axial-flow fans were developed recently with increased pressure producing capabilities. Axial-flow fan design becomes difficult aerodynamically if high pressure rise is required at low flow rate. High pressure corresponds to high blade rotational speed (see laws of performance) and, since the motor speed is limited by the bearing and brush life, the blade rotational speed can be increased only by increasing blade diameter. When low flow rates are required, the radial blade length decreases very rapidly with the increase in blade diameter (see Fig. 4-4).

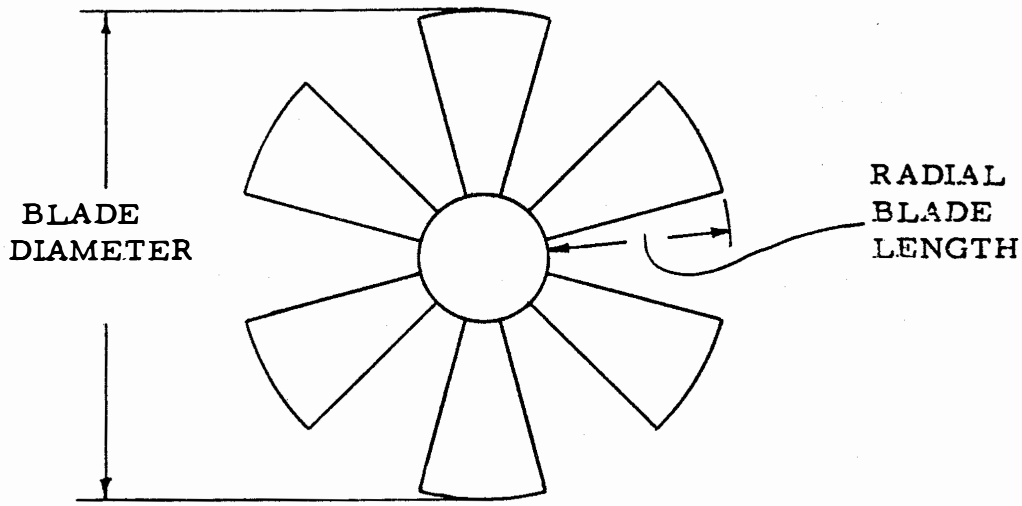


Figure 4-4 - Typical Fan Rotor with Radial Blades

A point can be reached where the increased losses (tip clearance, wall boundary layer) due to short blade lengths counteract the theoretical pressure increase due to the higher blade speed at the larger diameters.

B. Laws of Fan Performance

1. Theory

a. Definitions and Mathematical Relationships

Before the laws of fan performance can be presented, the quantities involved will be defined:

The static pressure (p_s) of a fluid is the compressive pressure existing in the fluid, and is the measure of the potential energy of the fluid.

The velocity pressure (p_v) of a fluid is the pressure corresponding to the average velocity determined from the volume of air flow and the cross-sectional area. Disregarding the units,

$$p_v = \frac{1}{2} \frac{\rho}{g} V^2 \qquad (39)$$

or, the velocity pressure is the measure of the kinetic energy of the fluid.

The total pressure (p_t) of the fluid is the sum of the velocity pressure and static pressure and it is a measure of the total energy of the fluid.

The volumetric flow rate of a fan must be calculated at outlet conditions, i.e., temperature, pressure, area, and velocity. The average velocity of the air at any point in the duct is determined from the velocity pressure readings, according to the following formula:

$$V = 1096.2 \sqrt{\frac{p''_v}{\rho}} \tag{40}$$

where:

ρ is the air density in lbs/cu. ft.

p''_v is the velocity pressure in inches of water

V is the average velocity of air in ft./min.

The air power (P_{air}) * is the fan power output, and it is the power supplied to air by the fan as mechanical work.

b. Air Horsepower for Centrifugal Fans

When the motor is mounted external to the ductwork, the power supplied to air is equal to air horsepower and may be expressed by the relationship:

$$P_{air} = P_{in} \times e_M \times e_{BL} \tag{41}$$

where:

P_{in} is the driving motor power input

P_{air} is the power dissipated in air

e_M, and e_{BL} are motor and blower efficiencies, respectively from the energy equation:

$$P \propto Hm \tag{42}$$

where:

H is the head, ft. of fluid

m is the weight rate of flow $\frac{\text{lbs.}}{\text{min.}}$

- - - - - - - - - - - - - - - - - -

* Air Horsepower in watts

Bernoulli's equation gives for the total pressure of a fluid:

$$p_t = \frac{\rho V^2}{2g} + p_s + \rho z \tag{43}$$

where:

> z is the vertical height of the point at which pressure is given.

Since head is pressure divided by density, the difference in head between outlet and inlet is:

$$H = \frac{p_{s_o}}{\rho_o} - \frac{p_{s_i}}{\rho_i} + \frac{V_o^2 - V_i^2}{2g} + z_o - z_i \tag{44}$$

where:

> Subscripts o and i indicate outlet and inlet, respectively.

For a fan z_o and z_i are practically equal. If the inlet and outlet ducts are of the same size and there is little compression; that is, if

$$\frac{\rho_i - \rho_o}{\rho_i} < 7\%$$

then

$$H = \frac{p_{s_o} - p_{s_i}}{\rho} \tag{45}$$

The air power is then:

$$P_{air} \propto \left(\frac{p_{s_o} - p_{s_i}}{\rho_o}\right) m = \left(p_{s_o} - p_{s_i}\right) Q_o \tag{46}$$

When the dimensional units are considered:

$$P_{air} = 0.116 \left(\frac{p_{s_o} - p_{s_i}}{\rho_o}\right) m = 0.116 \left(p_{s_o} - p_{s_i}\right) Q_o$$

or

$$P_{air} = 0.116\ \Delta p_s\ Q_o \qquad (46)(D.E.)$$

where:

P_{air} is in watts

$\left(p_{s_o} - p_{s_i}\right) = \Delta p_s$ is static pressure rise in the fan.

Q_o is the outlet volumetric flow rate.

c. Air Horsepower for Centrifugal Fans with Motor Overblow, and for Axial-Fans

When the fan motors are mounted axially with the fan or when motor overblow ducts are used (as in the case of cooling motors for centrifugal fans) the total power supplied to the motor may be considered to be dissipated in the air. If the density change of the air entering and leaving the motor-blower system is less than 7%, then:

$$P_{air} = P_{input} \qquad (47)(D.E.)$$

Since some of the heat developed by the motor will be lost by conduction through the motor base and by radiation and free convection from the casing, Equation (47) will give conservative results.

d. Air Temperature Rise Due to P_{air}

The power dissipated in air (P_{air}) must be added to the power dissipated by electronic parts when designing the heat exchangers in closed systems. Since, in the temperature and pressure ranges generally used in forced-air cooling, air may be assumed to be a perfect gas, the power dissipated in air can be expressed in terms of the weight rate of flow and air temperature rise as:

$$P_{air} = q = 7.62\ m\ \Delta t_b \qquad (48)(D.E.)$$

where:

P_{air} is the power dissipated in air, watts

q is the same as P_{air}

m is the weight rate of flow of air, $\frac{\text{lbs.}}{\text{min.}}$

Δt_b is the temperature rise of air, ^{o}C

$$(\Delta t_b = t_{b_o} - t_{b_i})$$

From the above equation the increase in air temperature can be calculated as:

$$t_{b_o} = t_{b_i} + \frac{P_{air}}{7.62m} \qquad (49)(D. E.)$$

It should be noted, however, that only the P_{air} as calculated by Equation (46) is the useful energy.

2. Similarity Relationships

It is exceedingly difficult to develop accurately, general analytical relationships for fans. There are, however, approximate dynamic similarity relations which help to indicate the relative performance of fans. These relations will aid in selecting fans for any particular application and will help to evaluate the adequacy of a given fan for a new application, when the performance of the fan is known.

The proportionalities of the dynamic similarity follow:

$$\Delta p \propto \frac{\rho D^2 N^2}{g}$$

$$Q \propto N D^3$$

$$HP \propto Q \Delta P \propto \frac{\rho D^5 N^3}{g}$$

$$m \propto \rho Q$$

where:

N is the fan rotational speed

D is some radial dimension, usually taken as impeller diameter

ρ is density of fluid at inlet condition

When any of the three quantities defined above is constant they, naturally, do not affect the proportionality.

For cooling of electronic equipment at sea level conditions, the density of air can be assumed to be constant.

When, for a centrifugal fan, D, N, and ρ are constant, then:

$$Q \propto B$$

$$HP \propto B$$

$$m \propto Q \propto B$$

$$\Delta P \text{ is constant}$$

where:

B is the impeller width

m is the weight rate of flow

3. Fan Characteristics

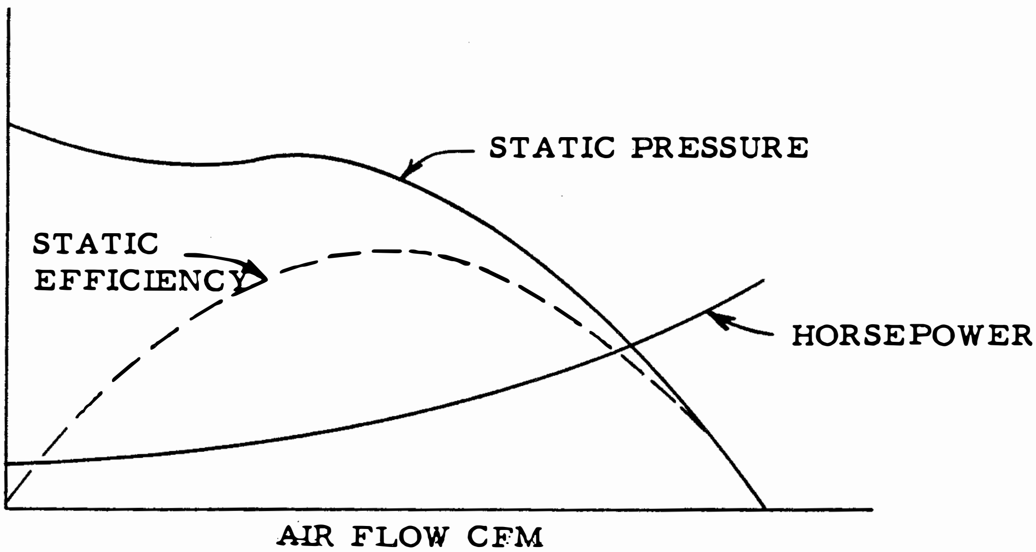


Figure 4-5 - Characteristic Curve of a Centrifugal Fan at Constant Speed

Figure 4-5 shows typical performance curves of a centrifugal fan running at constant speed. While there are various types of both centrifugal and axial-flow fans which have different characteristic curves, Fig. 4-5 is typical. With the outlet blocked there is no air flow and usually maximum static pressure. At maximum air flow or free delivery, the static pressure developed is zero. There is some point between these two extremes where efficiency is a maximum. Fans should operate at or near this point when power considerations are important. This requires that optimum fan type, particularly for size and speed, be selected to deliver the required air flow and static pressure. In other words, the fan must be well matched to the system requirements. It should be realized, however, that it sometimes may be difficult to evaluate system resistance prior to actual construction.

4. Interrelation Between System Resistance and Fan Performance

In order to show the relation between the duct system resistance characteristics and the fan performance, Fig. 4-6 is presented:

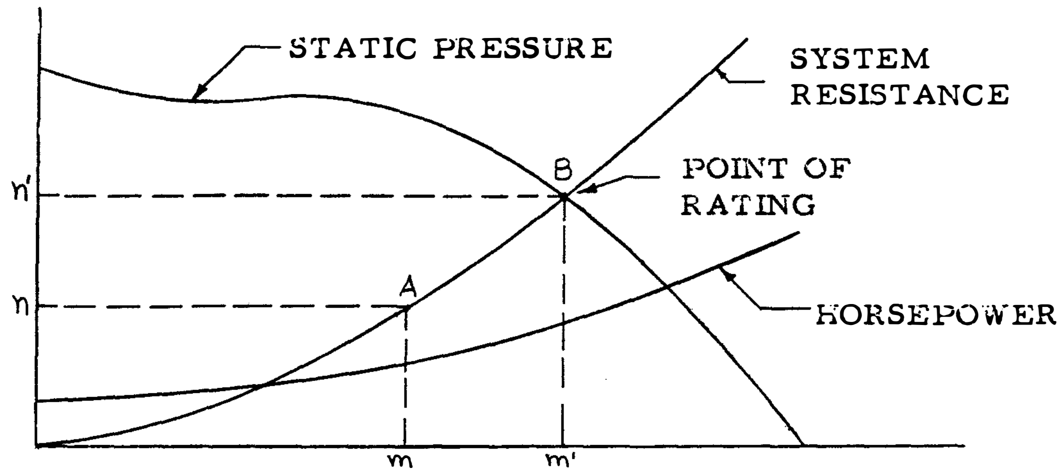


Figure 4-6 - Fan and Duct System Performance

First, consider the "system" resistance curve. This is the static pressure-air flow characteristic of the forced-air cooling system. For example, point "A" shows that, at m cfm air flow through the system, the required static pressure is n inches of water at a given air density. The pressure drop varies as the square of the air flow rate so that the system is a squared curve. The system rating point is that where the system resistance curve crosses the fan static pressure curve or, in this case, point "B". Thus, the air flow will be m' cfm at n' static pressure. It can be seen that it is important to select the fan so that its static pressure curve passes through or near the system resistance curve at the desired air flow and pressure near peak efficiency (if efficiency is important). Further, the speed and noise level must also be considered. Referring to the example in Fig. 4-6, if point "A" were desired but a fan was used having characteristics as shown, the air flow actually delivered would correspond to that at "B" and the horsepower would be necessarily high. The above situation can be remedied by judicious selection of fans and proper control of fan performance.

C. Noise

Noise level is an important factor in the selection of a fan for cooling electronic equipment. In general, a fan operating with high blade-tip velocity and developing a static pressure of more than 2" of water produces noticeable amount of noise in quiet surroundings. Static and dynamic balancing, vibration isolation, and streamlined flow design provide smoother performance with fewer vibrations, which also tend to reduce noise. Where low noise output is important, the manufacturer should be consulted. When the noise level is known for a particular fan speed, corrections for variations of fan speed not exceeding ± 10% may be made according to the following formula (for flat response observations):

$$\text{Db change} = 50 \log_{10} \frac{RPM_2}{RPM_1} \tag{50}$$

D. Selection of Fans

Fans are usually selected on the basis of the required air flow and the required static pressure. The required performance characteristics, i.e., air flow and static pressure, can be determined from the system power dissipation, the allowable temperature of the electronic parts and configuration of the flow ducts. Since several types of fans could probably provide the needed flow rate and pressure head, the problem is to select the fan best suited for a particular application. Where noise is of importance, the fan rotational speed will be limited by the acceptable level of noise. It is also limited by the blade speed.

The "specific speed" concept is quite helpful in selecting the proper type of fan, and is defined as:

$$N_s = \frac{N\sqrt{Q}}{(H_s)^{3/4}} \tag{51}$$

where:

N_s is the specific speed, rpm

N is the rotational speed, rpm

Q is the volumetric flow rate, cfm

H_s is the head developed, feet of fluid

This concept incorporates the parameters affecting the fan performance. All fans having the same proportions (but not necessary size) will have the same specific speed. Specific speed is defined here as the speed necessary to move 1 CFM of fluid against 1 ft. of fluid static pressure. Since specific speed is purely a relative thing, various units can be used in determining it. However, one should be careful to always use identical units when comparing two or more systems by means of specific speed. Specific speed does not change with change in fan speed (because the other parameters change accordingly). To determine specific speed, those values of Q and H_s are taken that correspond to the point of maximum static efficiency of the fan, where:

$$e_s = \frac{0.211\,(\Delta p_s)Q}{SHP} \tag{52}$$

where:

e_s is the static efficiency

Δp_s is the static pressure rise across fan (inches, water)

Q is the volumetric flow rate, cfm

SHP is the input shaft horsepower to fan, in watts, equal to electrical power input to motor times motor efficiency.

Knowing the specific speed, the proper type of fan can be chosen from Fig. 4-7. The fan or blower manufacturer can be of great help in proper fan selection and it is suggested that his advice on difficult fan problems be solicited.

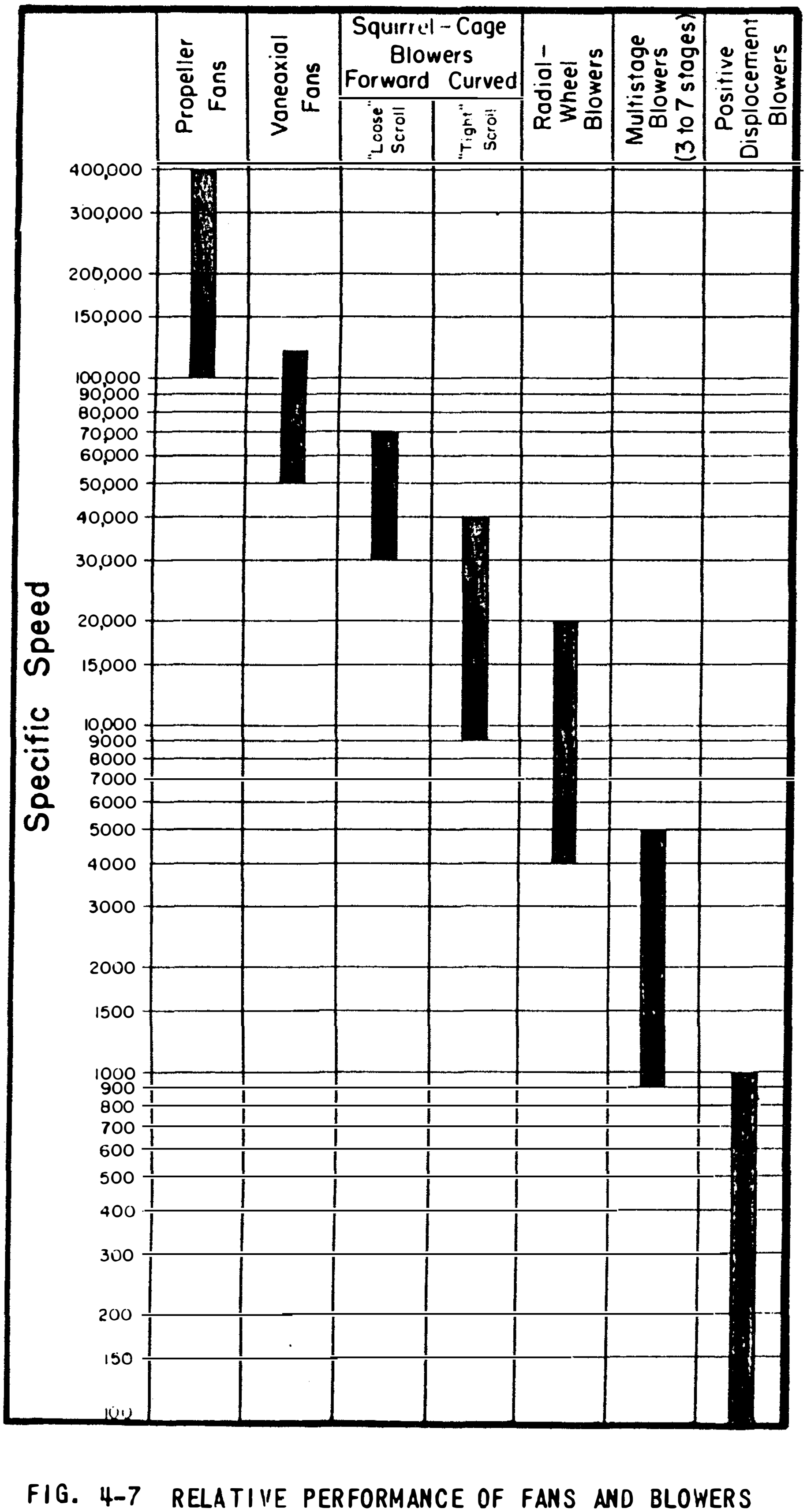


FIG. 4-7 RELATIVE PERFORMANCE OF FANS AND BLOWERS

V PRESSURE LOSSES AND THEIR DETERMINATION

A. Pressure Drop

Fluid flow through a duct or conduit is caused by a difference in energy along the conduit, the flow being from the higher energy level to the lower energy level. In steady flow, this difference in energy levels exists because there is present a resistance to the flow. This resistance can be divided into two types:

1. Viscosity, the resistance offered by a fluid to the relative motion of its parts, and

2. friction to the flow due to contact with the inner surface of the conduit.

Head loss or pressure drop is the name given to the difference in energy. In the absolute sense, friction results in the conversion of flow energy to some other form that is of no use in sustaining the flow, i.e., heat energy.

An electrical analogy for head loss is the voltage drop across a length of wire. Conduit friction is analogous to the ohmic resistance of wires. Thus, according to Ohm's Law, head loss and voltage drop are directly proportional to the flow rate and current respectively.

The head loss or pressure drop around a given cooling system must be known in order to select the proper fluid moving device, i.e., blower. The blower has to provide sufficient pressure head to result in the desired flow rate after all losses have been overcome.

1. Head Loss

Flow through a conduit may be either laminar or turbulent. It has been shown by experiments that, in laminar flow, head loss varies directly with the viscosity of the fluid and its velocity. However, for turbulent flow, head loss varies directly with the density of the fluid and the square of its velocity.

Generally, for most fluids and especially gases, laminar flow takes place at extremely low velocities, hence all commonly used equations apply to turbulent flow.

The equation most used in calculations of head loss is the Darcy-Weisbach Equation:

$$H_L = f \frac{L}{D} \frac{V^2}{2g} \qquad (53)(D.E.)$$

where:

H_L is the head loss - feet of fluid

L is the length of conduit - feet

D is the diameter of conduit - feet

V is the velocity of the flow - feet per second

g = 32.2 feet per second per second - $\frac{\text{Ft.}}{\text{Sec.}^2}$

f is the dimensionless parameter known as the friction factor

where:

$$f = \phi \left[\frac{\epsilon}{D}, \text{Re} \right] \quad (54)$$

where:

ϵ is the absolute roughness of the surface (ft.)

Re is the Reynolds number (dimensionless)

Note:
There are in existence two systems of noting friction factor. One of these is four times greater than the other. However, use of the larger f is becoming predominant, and it is this factor that is used in this Manual.

In generalized flow, the friction factor has been shown to be a function of the relative roughness, $\frac{\epsilon}{D}$ and the Reynolds number, Re. Relative roughness is the ratio of the absolute roughness of the conduit surface to the diameter of the conduit. The absolute roughness refers to the size of the small bumps and ridges which are inevitably present in commercial conduits. Excellent correlation between experiments using articially roughened conduits and experiments using natural conduit surfaces has been obtained. (Nikuradse's tests, etc.). Reynolds number is discussed elsewhere in this Manual.

The dependency of the friction factor on the two quantities stated makes it possible to show the relationship between friction factor and Reynolds number as a two dimensional plot, with a family of curves for each of which $\frac{\epsilon}{D}$ is constant. This plot (Fig. 5-1) is known as the Moody Diagram. Its use will be explained in the sample problems.

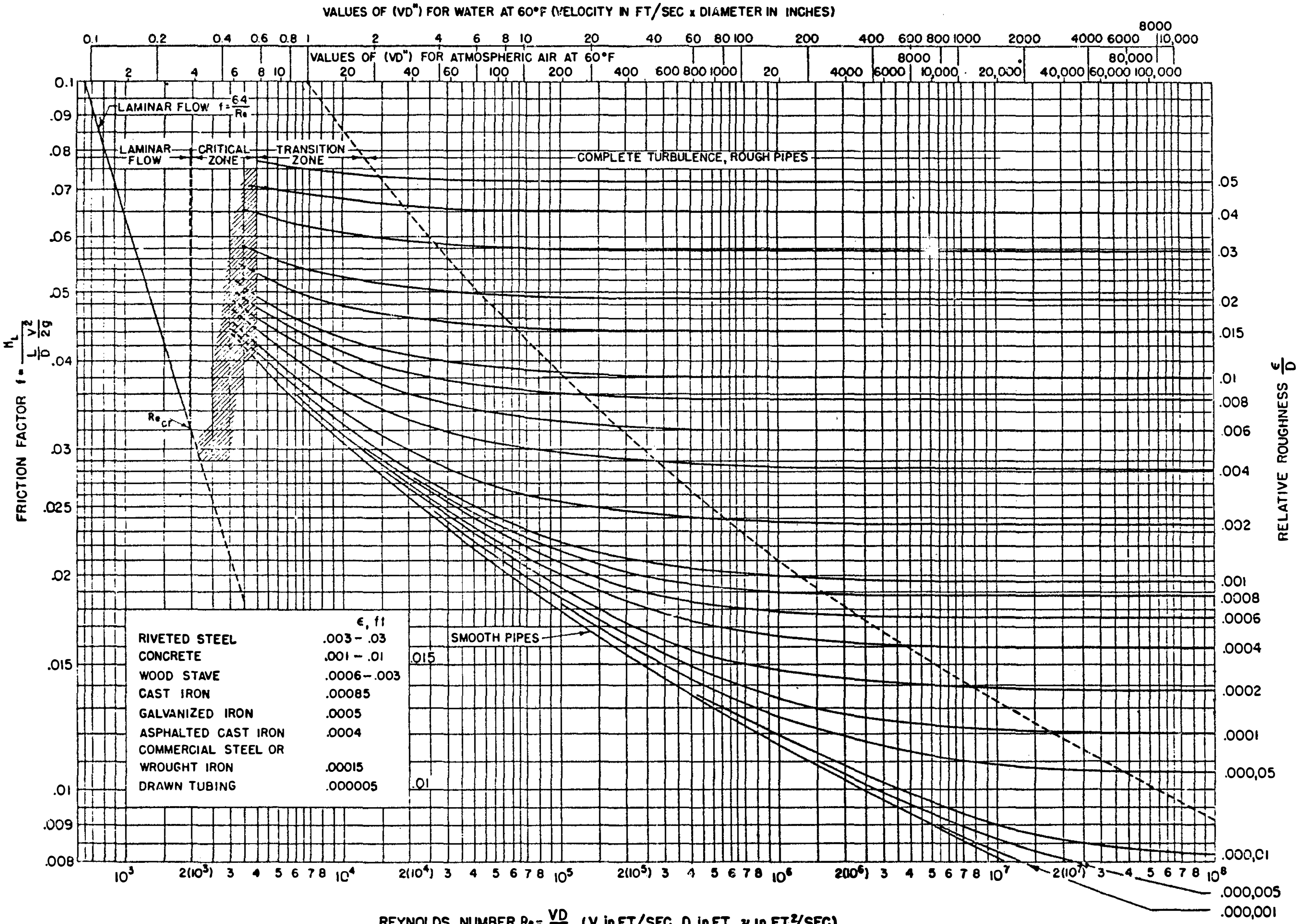


FIG. 5-1 MOODY DIAGRAM

For perfectly smooth conduit, the friction factor is a function of the Reynolds number only. The following equations have been empirically validated:

(a) Hagen-Poiseulle Equation

$$f = \frac{64}{Re} \tag{55}$$

valid only up to Re = 2000 (laminar flow)

(b) Blasius Equation

$$f = \frac{0.316}{(Re)^{0.25}} \tag{56}$$

valid up to Re = 100,000

(c) Prandtl Equation

$$f = \frac{0.184}{(Re)^{0.2}} \tag{57}$$

valid up to Re = 3,000,000

TABLE VII

ABSOLUTE ROUGHNESS CLASSIFICATION OF CONDUIT SURFACES FOR SELECTION OF FRICTION FACTOR, f, IN THE MOODY DIAGRAM.*

Commercial Surfaces (new)	Absolute Roughness - feet
Brass, Glass, Tubing (drawn)**	0.000005
Wrought Iron, Steel	0.00015
Galvanized Iron, Steel	0.0005
Cast Iron	0.00085
Concrete, Average	0.004
Riveted, Light, Steel	0.003
Riveted, Heavy, Steel, Brick	0.03

*The values given in Table VII are for clean conduits. Rusted or carbunculated conduit may have several times the roughness of clean conduit.

**Under brass and tubing is included all drawn material such as lead, block tin, aluminum, copper, steel and glass. Metal tubing 1/4" O.D. or less is often finished "no plug" draw, in which case it will not be smooth inside, but wrinkled, and must be considered as rough as cast iron.

2. Sample Problem I

Note:

This is not a practical problem, but is included to demonstrate the operations and principles involved in this type problem.

(For illustrating use of Darcy-Weisbach Equation and Moody Diagram)

Problem - 1000 cfm atmospheric air at 60° F flows through a 6" diameter galvanized duct 70 feet long. What is the head loss in inches of water?

Solution:

a. Area of duct $= \frac{\pi D^2}{4} = \frac{\pi}{4} \times \left(\frac{1}{2}\right)^2 ft^2 = \underline{\underline{\frac{\pi}{16}\ ft^2}}$

b. Velocity $= \frac{\text{flowrate}}{\text{area}}$, $V = \frac{Q}{A_c}$

$$V = 1000\ \frac{ft^3}{min} \times \frac{min}{60\ sec} \times \frac{16}{\pi\ ft^2}$$

$$= \frac{1000 \times 16}{60\ \pi}$$

$$= \underline{\underline{85.\ \frac{ft}{sec}}}$$

c. $\mu = 3.8 \times 10^{-7}\ \frac{lb\text{-}sec}{ft^2}$ (from tables)

Note:

This is based on the dimensional system where one pound-force is the basic unit, To convert to the pound-mass system, μ must be multiplied by g.

Hence:

$$\mu = 3.8 \times 10^{-7}\ \frac{lb\text{-}sec}{ft^2} \times 32.2\ \frac{ft}{sec^2}$$

$$= \underline{\underline{1.22 \times 10^{-5}\ \frac{lb}{ft\text{-}sec}}}$$

d. Since $pv = RT$

and $\rho = \frac{1}{v}$

then ρ can be found using p and T.

$$\rho = \frac{1}{v} = \frac{p}{RT}$$

$$= 14.7 \frac{lb}{in^2} \times \frac{144\ in^2}{ft^2} \times \frac{R^o}{53.3\ ft} \times \frac{1}{520\ R^o}$$

$$= \frac{14.7 \times 144}{53.3 \times 520} \quad \frac{lb}{ft^3}$$

$$= 0.0765 \frac{lbs}{ft^3}$$

e. Reynolds Number

$$Re = \frac{VD\rho}{\mu}$$

$$= 85 \frac{ft}{sec} \times 0.5\ ft. \times 0.0765 \frac{lbs}{ft^3} \times \frac{1}{1.22 \times 10^{-5}} \quad \frac{ft\text{-}sec}{lb}$$

$$= \frac{85 \times 0.5 \times 0.0765}{1.22} \quad \times 10^5$$

$$= 2.66 \times 10^5$$

f. ϵ for galvanized surface - 0.0005 ft.

$$\frac{\epsilon}{D} = \frac{0.0005\ ft}{0.5\ ft} = \underline{\underline{0.001}}$$

g. From Moody Diagram

$Re = 2.66 \times 10^5$, $\frac{\epsilon}{D} = 0.001$

$\underline{\underline{f = 0.021}}$

h. $H_L = f \dfrac{L}{D} \dfrac{V^2}{2g}$

$$= 0.021 \times \frac{70 \text{ ft}}{0.5 \text{ ft}} \times (85)^2 \frac{\text{ft}^2}{\text{sec}^2} \times \frac{\text{sec}^2}{2 \times 32.2 \text{ ft}}$$

$$= \frac{0.021 \times 70 \times 7200}{0.5 \times 64.4} \text{ ft. of air}$$

$$= \underline{\underline{329.0 \text{ feet of air}}}$$

i. $H_L = 329 \text{ feet of air} \times 0.0765 \dfrac{\text{lbs}}{\text{ft}^3 \text{ of air}} \times \dfrac{\text{ft}^3 \text{ of water}}{62.4 \text{ lbs}}$

$$= \frac{329 \times 0.0765}{62.4} \text{ ft of water}$$

$$= \underline{\underline{0.403 \text{ feet of water}}}$$

j. $H_L = 0.403 \text{ feet of water} \times \dfrac{12 \text{ in.}}{\text{ft.}}$

$$= \underline{\underline{4.85 \text{ inches of water}}} \qquad \underline{\underline{\text{ANSWER}}}$$

Note that since the fluid is air at 60°F, the VD" scale at the top of the Moody Diagram can be utilized. Instead of calculating the Reynolds number, the VD" product is used.

$$VD'' = 85 \frac{\text{ft}}{\text{sec}} \times 6'' = 510$$

$$\frac{\epsilon}{D} = 0.001$$

then $f = 0.021$

Note that this is the same f as that previously determined. This method provides somewhat of a shortcut if the air is at atmospheric pressure and 60°F.

B. Types of Additional Losses

1. "Minor" Losses

The head loss due to curves, elbows, meters and valves may be expressed in either of two ways: (1) as a constant times the velocity head, or (2) as being equal to the loss in a certain additional length of straight conduit. These values are known as loss coefficients and equivalent lengths, respectively.

The loss coefficient is a constant for a given type fitting and is identified by the symbol K.

$$H_L = K \frac{V^2}{2g} \qquad (58)(D.E.)$$

where:

H_L is the head loss, in feet of fluid flowing

K is the loss coefficient, or the sum of several loss coefficients

$\frac{V^2}{2g}$ is the velocity head, feet of fluid flowing

TABLE VIII (From Ref. 4)

LOSS COEFFICIENTS FOR SOME COMMON FITTINGS*

Fitting	K
Close Return Bend (180° bend)	2.2
Standard Tee	1.8
Standard Elbow	0.9
Medium Sweep Elbow	0.75
Long Sweep Elbow	0.60
45° Elbow	0.42

*The loss coefficients presented in this table are the resultant average of several experiments conducted by various sources. Truly reliable loss coefficients have not been fully established.

Through use of a simple relationship, the loss coefficient of a fitting can be converted to its equivalent length, i.e. the additional length of straight conduit whose head loss is the same as that of the fitting.

$$L_e = \frac{KD}{f} \tag{59)(D.E.}$$

then the head loss equation becomes:

$$H_L = f \frac{L_e}{D} \frac{V^2}{2g} \tag{60)(D.E.}$$

where:

L_e is the equivalent length, feet

K is the loss coefficient

D is the diameter of conduit, feet

f is the friction factor

Generally, "minor" losses may be neglected in those situations where they comprise less than 5% of the total head losses due to friction.* The friction factor, at best, is subject to approximately 5% error and it is impractical to compute values to more than two significant figures.

2. Changes in Duct Size

In duct systems a sudden change in cross-sectional area is identified as either a sudden expansion or sudden contraction.

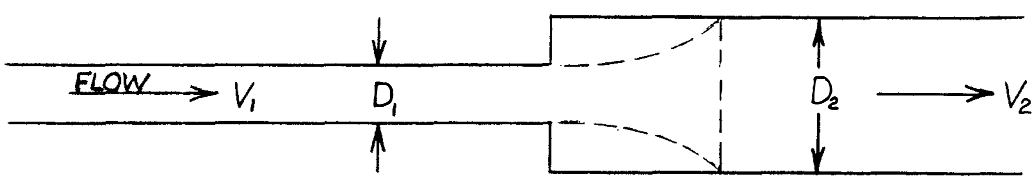


Figure 5-2 - Sudden Expansion

*In a multichassis equipment of rack-type construction having a central blower system, "minor" losses are likely to be significant and should not be neglected.

a. Sudden Expansion

When the flow enters a large cross-section from a small cross-section, the following equation is applicable:

$$H_e = K \frac{V_1^2}{2g} \qquad (61)(D.E.)$$

where:

H_e is the loss in feet of fluid due to expansion

$$K = \left[1 - \left(\frac{D_1}{D_2} \right)^2 \right]^2 \qquad (62)$$

if Section 2 is a reservoir then

$$\frac{D_1}{D_2} = \frac{D_1}{\infty} = 0$$

and $\underline{\underline{K = 1.0}}$

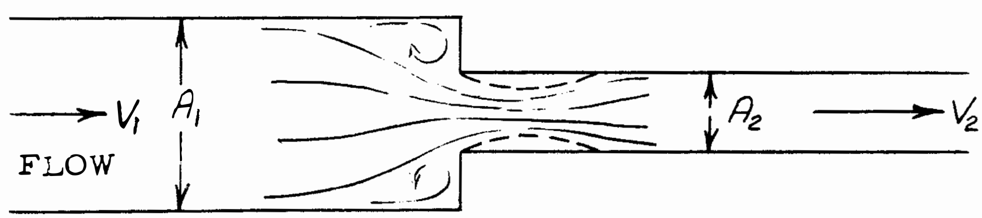


Figure 5-3 - Sudden Contraction

b. Sudden Contraction

With a sudden contraction, there is a contraction of the flow and a coefficient of contraction must be incorporated into the loss coefficient. The values presented are in close agreement with several references.

$$H_c = K \frac{V_2^2}{2g} \qquad (63)(D.E.)$$

where:

H_c is the loss in feet of fluid due to contraction

K is the loss coefficient, from Table IX

$\frac{V_2^2}{2g}$ is the velocity head

TABLE IX

LOSS COEFFICIENTS FOR SUDDEN CONTRACTIONS

Area Ratio	$\frac{A_2}{A_1}$	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
Loss Coeff.	K	0.46	0.41	0.36	0.30	0.24	0.18	0.12	0.06	0.02	0

(From Ref. 2)

Note that the velocities used in the equations for head loss due to sudden changes of cross-section are the velocities which exist in the duct having the smaller diameter.

c. Gradual Contraction

The loss in a gradual contraction, as shown in Fig. 5-4, is generally computed as:

$$H_L = 0.05 \frac{V_2^2}{2g} \qquad (64)(D.E.)$$

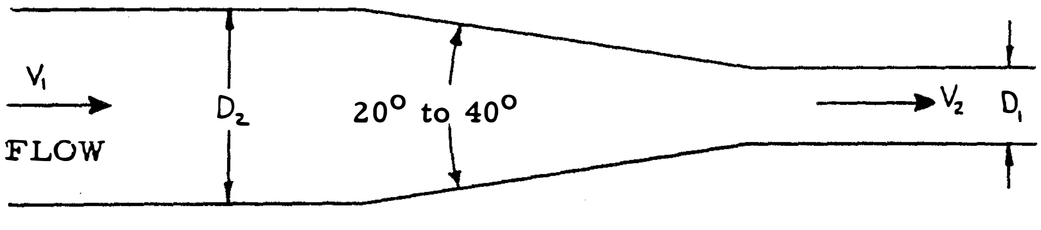


Figure 5-4 - Gradual Contraction

The loss coefficient, i.e., K = 0.05, is considered constant over the range of angles shown.

d. Gradual Expansion

The case of the gradual expansion is completely different from that of gradual contraction because the loss coefficient is directly dependent on the angle subtended by the sides of the expansion cone (see Figures 5-5 and 5-6).

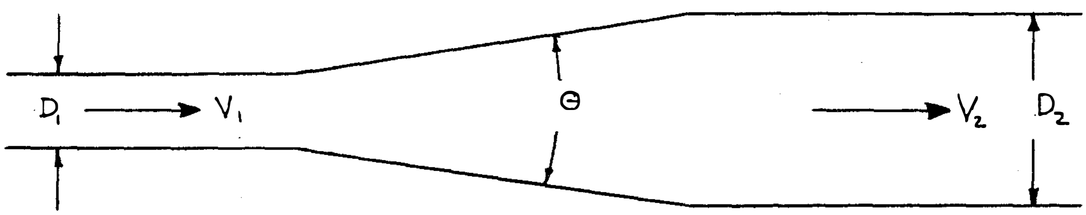


Figure 5-5 - Gradual Conical Expansion

The equation is:

$$H_L = K \frac{(V_1 - V_2)^2}{2g} \qquad (65)(D.E.)$$

where K is shown below

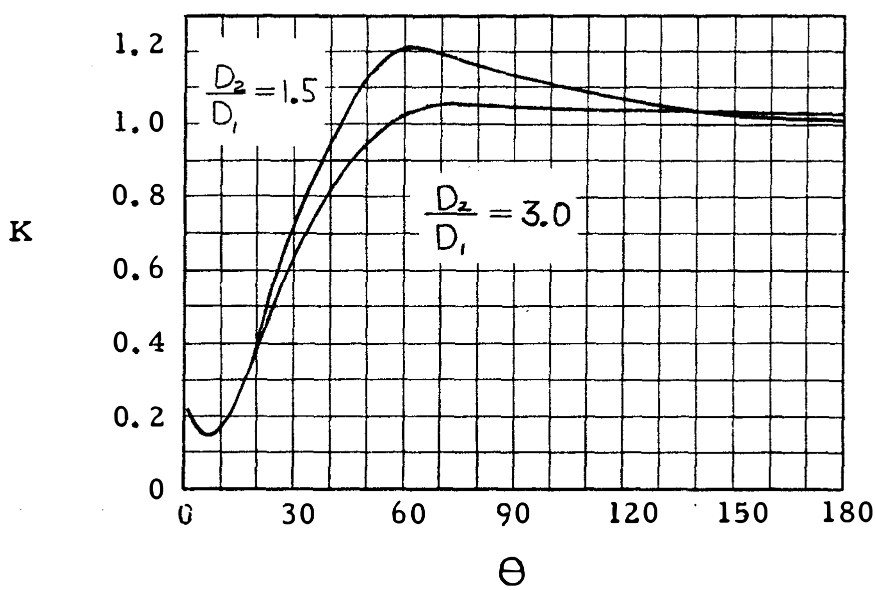


Figure 5-6 - Loss Coefficients for Gradual Conical Expansions

3. Ducts in Series

Two or more ducts of different sizes and roughnesses are in series, when they are connected so that the flow goes through one and then the other, etc. For this situation it is necessary to determine the losses due to (1) the length of duct one,, (2) the expansion or contraction, (3) the length of duct two, etc. A summation of the individual losses gives the total head loss.

4. Ducts in Parallel

If two or more ducts are connected as shown in Fig. 5-7 the system is classified as a parallel duct system.

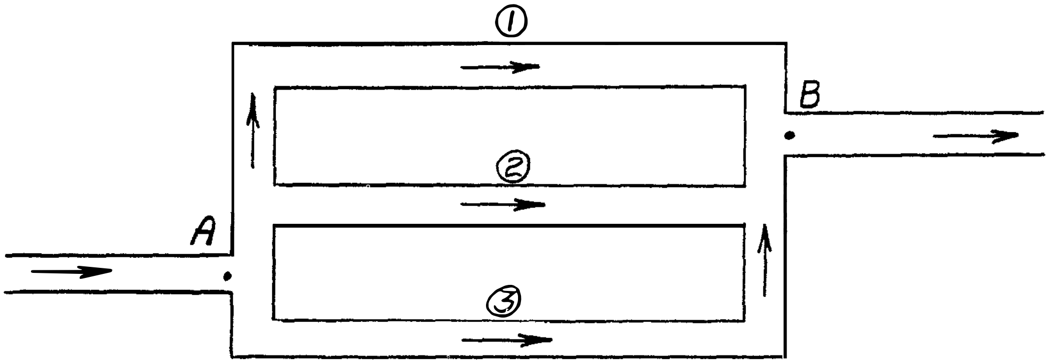


Figure 5-7 - Ducts in Parallel

At equilibrium, i.e. the steady state, the following equations are valid:

$$Q_{total} = Q_1 + Q_2 + Q_3 \tag{66}$$

$$H_{AB_1} = H_{AB_2} = H_{AB_3} \tag{67}$$

Or the total flow rate is the sum of the individual flow rates and the pressure drop or head loss is the same, regardless of the branch, between junctions.

C. Hydraulic Radius (non-circular cross-sections)

In applications where it is necessary to calculate the flow through non-circular and annular shapes, a special parameter has been devised. This term is defined as the hydraulic radius.

$$HR = \frac{\text{Cross-sectional area}}{\text{Wetted perimeter}} \tag{68}$$

For a circular conduit flowing full, i. e., with only one fluid occupying the total cross-section, the hydraulic radius becomes:

$$HR = \frac{\text{Diameter}}{4} \qquad (69)(D.E.)$$

If 4(HR) is substituted for D, the Reynolds number becomes

$$Re = V \frac{(4HR)}{\mu} \rho \qquad (70)$$

and the head loss equation can be rewritten as

$$H_L = f \frac{L}{(4HR)} \frac{V^2}{2g} \qquad (71)$$

Use of the hydraulic radius provides acceptably accurate results for square, oval, rectangular, triangular and similar type ducts.

Note:
Of the several references consulted, only one suggested that the use of the hydraulic radius for laminar flow might lead to erroneous results.

D. Compressible Flow

When a gas flows in a duct, head loss may be calculated by using the "incompressible flow approximation", provided that the loss is less than 10% of the initial absolute static pressure. This procedure is based on the assumption that the density of the fluid remains constant. As a result, the velocity of the fluid is constant and pressure forces on the body of fluid in the duct will be used only in overcoming friction forces.

Generally, the initial density of the gas is used (i. e. conditions as of the upstream tap). Then the Darcy Weisbach Equation becomes:

$$H_L = 144 \frac{(P_1 - P_2)}{\rho} = f \frac{L}{D} \frac{V^2}{2g} \qquad (72)(D.E.)$$

where:

p_1, p_2 are the initial and final pressures, psia

H_L is the head loss, feet of fluid flowing

ρ is the density of fluid, lbs/ft^3

L is the length of duct, feet

D is the diameter of duct, feet

V is the velocity of flow, ft/sec

g is the acceleration due to gravity, 32.2 ft/sec^2

f is the friction factor

If the pressure drop exceeds 10% of the upstream absolute static pressure, then true compressible flow equations must be used. These are beyond the scope of this Manual, but can be found in most Fluid Mechanics texts or Mechanical Engineering handbooks (see References).

E. Basic Types of Problems

1. Problem Types

There are three basic types of problems that are encountered in head loss calculations. They are:

I. Given the flow rate, length, diameter, viscosity and roughness, find the head loss. This is generally the most common type and is particularly applicable to electronic cooling systems where the flow rate is determined by the amount of cooling required.

II. Given head loss, length, diameter, viscosity and roughness, find the flow rate. This usually would not be the case in an electronic cooling system, since the flow rate is determined by the amount of heat to be dissipated.

III. Given head loss, flow rate, length, viscosity and roughness, find diameter. This type of problem is probable inasmuch as a given blower might be available, and the duct size may depend on the permissible pressure drop (since pressure drop is inversely proportional to diameter).

2. Sample Problem II

Atmospheric air at $90^{o}F$ is to flow at the rate of 300 cfm, thru a straight run of galvanized circular duct, 130 feet long. Find the diameter of the duct if the permissible head loss is 0. 35 inches water.

Solution:

The head loss equation states

$$H_L = f \frac{L}{D} \frac{V^2}{2g} \qquad (73)$$

but V is not obtainable.

However, since

$$Q = VA = \frac{V \pi D^2}{4} \qquad (74)$$

$$V = \frac{4Q}{\pi D^2} \qquad (75)$$

and

$$H_L = f \frac{L}{D} \left(\frac{4Q}{\pi D^2} \right)^2 \frac{1}{2g}$$

$$= f \frac{L}{D} \frac{16Q^2}{\pi^2 D^4 2g}$$

$$= \frac{8fLQ^2}{\pi^2 D^5 g}$$

or

$$D^5 = \frac{8LQ^2}{\pi^2 H_L g} (f) = C_1 f \qquad (76)$$

Where C_1 is constant for any given configuration and

$$C_1 = \frac{8LQ^2}{\pi^2 H_L g}$$

C_1 can now be calculated, but first change H_L = 0.35 inches water to equivalent feet of air

$$\rho_{air} = \frac{P}{RT} = \frac{14.7 \text{ lbs}}{\text{in}^2} \times \frac{144 \text{ in}^2}{\text{ft}^2} \times \frac{{}^{o}\text{R}}{53.3 \text{ ft}} \times \frac{1}{550^{o}\text{R}}$$

$$= \frac{14.7 \times 144}{53.3 \times 550} \frac{\text{lbs}}{\text{ft}^3}$$

$$= 0.0723 \frac{\text{lbs}}{\text{ft}^3}$$

$$H_L = 0.35'' \text{ water} \times \frac{\text{ft}}{12 \text{ in}} \times \frac{62.4 \text{ lbs}}{\text{ft}^3 \text{ water}} \times \frac{\text{ft}^3 \text{air}}{0.0723 \text{ lbs}}$$

$$= \frac{0.35 \times 62.4}{12 \times 0.0723}$$

$$= 25.2 \text{ feet air}$$

Now calculate C_1

$$C_1 = \frac{8LQ^2}{\pi^2 H_L g}$$

$$= 8 \times 130 \text{ ft.} \times \left(\frac{5 \text{ ft}^3}{\text{sec}}\right)^2 \times \frac{1}{\pi^2} \times \frac{1}{25.2 \text{ ft air}} \times \frac{\text{sec}^2}{32.2 \text{ ft}}$$

$$= \frac{8 \times 130 \times 25}{9.9 \times 25.2 \times 32.2}$$

$$= 3.24 \text{ ft}^5$$

then

$$\underline{\underline{D^5 = 3.24 \text{ f}}}$$

$$Re = \frac{VD\rho}{\mu} \qquad \text{but} \quad V = \frac{4Q}{\pi D^2}$$

then

$$Re = \frac{D\rho}{\mu} \times \frac{4Q}{\pi D^2} = \frac{4\rho Q}{\mu \pi D} = \frac{C_2}{D} \tag{77}$$

Where C_2 is a constant for a given configuration and

$$C_2 = \frac{4\rho Q}{\mu \pi}$$

Find C_2

$$C_2 = \frac{4\rho Q}{\mu \pi}$$

$$\mu = 4 \times 10^{-7} \frac{\text{lb-sec}}{\text{ft}^2} \qquad \text{(from tables)}$$

then

$$C_2 = 4 \times 0.0723 \frac{\text{lb}}{\text{ft}^3} \times \frac{5\ \text{ft}^3}{\text{sec}} \times \frac{\text{ft}^2}{4 \times 10^{-7}\text{lb-sec}} \times \frac{\text{sec}^2}{32.2\ \text{ft}} \times \frac{1}{\pi}$$

$$= \frac{4 \times 0.0723 \times 5 \times 10^7}{4 \times 32.2 \times \pi}$$

$$= 0.00357 \times 10^7 = 35,700.$$

then

$$Re = \frac{C_2}{D} = \frac{35700}{D}$$

When the above constants have been determined the following procedure is to be followed:

(1) Assume a value for f by inspection of the Moody Diagram.

(2) Solve equation (76) for D.

(3) Solve equation (77) for Re.

(4) Find the relative roughness $\frac{\epsilon}{D}$.

(5) Using Re and $\frac{\epsilon}{D}$, find new f from Moody Diagram.

(6) Using new f, repeat steps 1 - 5.

(7) When the value of f does not change, all equations are satisfied and problem is solved.

Assume the value of f

i.e. $f = 0.020$

Solve for D

$$D^5 = C_1 f = 3.24f$$

$$= 3.24 \times 0.02 = 0.0648$$

$$D = \sqrt[5]{0.0648} = \sqrt[5]{6480 \times 10^{-5}}$$

$$= 5.8 \times 10^{-1} = 0.58 \text{ ft}$$

Now solve for Re

$$Re = \frac{C_2}{D} = \frac{35700}{0.58} = 61,600$$

$$\frac{\epsilon}{D} = \frac{0.0005}{0.58} = 0.00086$$

Then from Moody Diagram

$Re = 61,000$ $\quad\quad \frac{\epsilon}{D} \quad 0.00086$

$f = 0.023$

Calculate new D

$$D^5 = C_1 f = 3.24 \times 0.023$$

$$D = \sqrt[5]{0.0745 \text{ ft}^5} = \sqrt[5]{7450 \times 10^{-5}}$$

$$= 0.595 \text{ ft.}$$

Calculate new Re

$$Re = \frac{35700}{D} = \frac{35700}{0.595} = 60,000.$$

and

$$\frac{\epsilon}{D} = \frac{0.0005}{0.595} = 0.00084$$

Then from Moody Diagram

$$Re = 60,000 \qquad \frac{\epsilon}{D} \quad 0.00084$$

$$f = 0.023$$

Note that f does not change.

Note:

It is generally the case that the solution for f converges very rapidly.

Then

$$D = 0.595 \text{ ft} = 7.14 \text{ inches}$$

If this exact size is not available, the next larger size must be selected.

VI FORCED-AIR COOLING DESIGN

A. General Design Considerations

Forced-air cooling designs require careful consideration of the pertinent factors. Improper designs can lead to difficulties far worse than those which could be encountered with other methods of heat removal. Forced convection, at small flow rates, can readily provide thermal resistances of the order of half those obtained with free convection and radiation. In general, thermal resistances of the order of $10^{o}C/watt$ per square inch can be achieved with a reasonable coolant flow. Increased flow will, under certain conditions, lead only to smaller gains and the point of diminishing returns may be encountered.

From a thermal viewpoint, the ideal cooling design is one in which all heat sources are operated at their maximum allowable temperatures. Under such conditions shipboard equipments having heat concentrations as high as 10 kw per cubic foot could be cooled with forced convection. Unfortunately, such ideal situations are seldom encountered in practice. Further, reliability and life requirements frequently dictate permissible temperature levels which are considerably less than the maximum allowable temperature. Thus, it is seldom possible to properly cool heat concentrations greater than 4 kw per cubic foot in a typical shipboard equipment without excessive pressure losses and very high weight rates of flow. In general, heat concentrations in practical instances will usually be considerably less than 2 kw per cubic foot.

The design of forced-air cooling systems for electronic equipment should be based mainly upon the attainment of safe operating temperatures at the heat sources and minimization of the energy required to move the air through the cooling system, i.e., cooling power. Two other factors of importance are the volume and weight of the cooling systems.

Heat flow within and from heat-producing electronic parts is discussed in detail in NAVSHIPS 900,192 (CAL Report #HF-845-D-8), a companion manual. This information, which is also valid for forced-air cooling, is not repeated herein. In general, most electronic parts are designed for cooling by natural means and their external geometry is not ideally suited for forced-air cooling. Only a few parts, such as certain transmitting tubes and finned rectifiers, are specifically designed for cooling by forced air. However, most parts can readily be cooled by forced air because of their comparatively low unit heat dissipation. This does not mean that the electrical ratings should be correspondingly increased because of the reduced thermal resistances which may be achieved with forced convection.

Heat sources operating at high surface temperatures should always use radiation shields to decrease the effect of their radiation upon other electronic parts. The shields should, ideally, be liquid cooled. However, conduction to a suitable chassis or sink is usually adequate. Heat sources of similar power dissipation should be grouped together if possible, for economical usage of cooling air and cooling power economy. When two or more ducted chassis are placed in parallel, dampers or throttling plates may be required to assure the correct distribution of air flow to each chassis by changing the relative resistance of the ducts to the air flow.

Heat sources with low unit heat dissipations require much lower flow rates than those having high unit heat dissipations. Parallel flow is generally recommended for cooling of large heat sources having high unit heat dissipations. Crossflow, though slightly less efficient should generally be employed for small heat sources, since the duct work is simpler and less space is required. Under the same conditions, the surface temperatures will be slightly higher than for parallel flow, but if the power dissipated by the heat sources is low, the cooling air temperature rise will be small and the difference in cooling effectiveness for parallel and crossflow cooling will not be appreciable. In crossflow, a staggered arrangement increases turbulence, thus lowering the thermal resistance and improving cooling. Crossflow-in-line arrangement is simpler and cheaper, but does not cool as effectively as a staggered arrangement.

For equipments having a small power dissipation, the heat dissipated by the parts to the cooling air can be assumed to be equal to the total input power. For high-power equipment, the heat transferred to the cooling air will approximate 75% of the total input power. It should be emphasized that, when the total input power is considered dissipated by a given equipment, the results will include a safety factor.

In the design of an equipment for forced-air cooling, the following are to be considered:

a) Determination of inlet cooling-air pressure and temperature.

b) Selection of part arrangements and spacings based on electronic and space requirements, as well as cooling power considerations.

c) Determination of the allowable temperature level of each part.

d) Determination of the required Reynolds number by use of several methods outlined later.

e) Calculation of the weight rates of flow based upon physical dimensions of the systems and the prescribed Reynolds number.

f) Determination of the overall pressure losses (see Chapter V) and required cooling power of the system.

B. Simplified Design Methods

1. General

There are two basic attitudes for cooling heat-producing electronic parts with forced air:

> Parallel flow, wherein the air flows parallel to the major axis of the part.
>
> Crossflow, wherein the air flows over the part normal to the major axis.

Electron tubes usually produce most of the heat in electronic equipment. Consequently, much of the design section of this manual is devoted to forced-air cooling techniques pertinent to tubes. Several alternate methods of designing forced-air cooling systems are presented herein.

Figure 6-1 shows the variation of required air temperature with flow rate for parallel flow cooling of an electron tube. Flow rate-of-change increases rapidly as the air temperature approaches the maximum part surface temperature. It is more economical to use low air temperatures and flow rates because the power to convey the air increases as the square of the weight rate of flow. The weight rate of flow required to maintain a specified temperature differential between part surface and cooling air is independent of air density. However, at fixed weight rates of flow, the cooling-air power requirement is inversely proportional to the air density (Ref. 7).

A significant feature of cooling by crossflow is the peripheral variation of point heat transfer on the surfaces of cylindrical heat sources. The hot spot locations are determined principally by the basic configuration and spacings, and slightly affected by the air flow rate. Figure 6-2 contains typical variations of relative point heat transfer coefficients for two different configurations. In the design of assemblies for forced-air cooling, the optimum positioning of hot spots in peripheral regions of probable maximum heat transfer can aid in maximizing cooling.

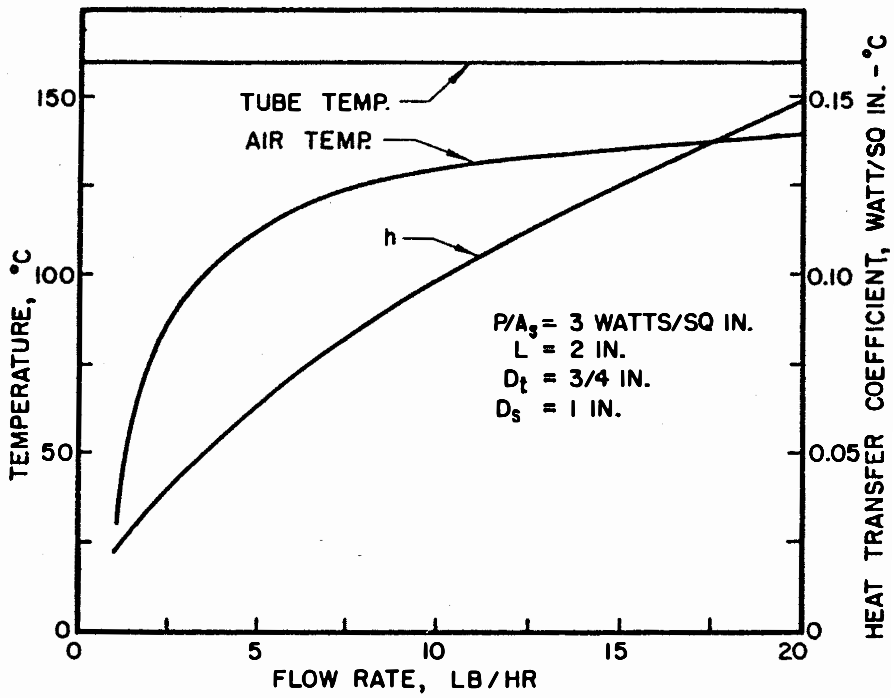


FIG. 6-1 VARIATION OF REQUIRED AIR TEMPERATURE WITH FLOW RATE FOR COOLING OF TUBE BY FORCED FLOW PARALLEL TO SURFACE

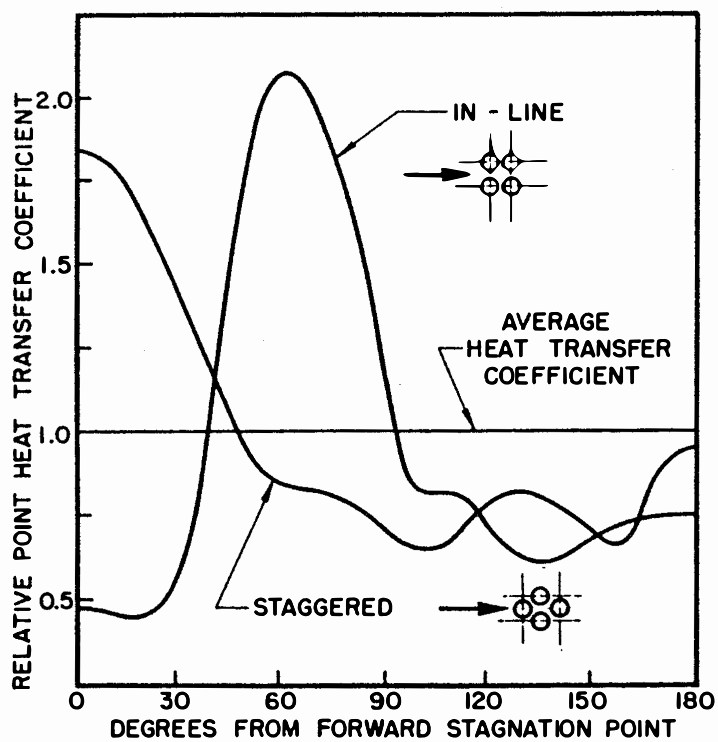


FIG. 6-2 TYPICAL VARIATIONS OF POINT HEAT TRANSFER COEFFICIENTS FOR IN-LINE AND STAGGERED CYLINDERS IN CROSSFLOW

2. Recommended Design Procedures

a. Electron Tubes

Forced-air cooling is one of the better methods of removing heat from electron tubes. While most tubes are not specifically designed for this mode of cooling, acceptably low thermal resistances can usually be achieved at relatively low flow rates. Plate temperature is an excellent index of the thermal condition of a tube cooled by forced air. Details pertaining to plate temperature measurement and a method of determining the point of diminishing returns of cooling electron tubes are outlined in CAL Reports HF-845-D-8 (NAVSHIPS 900,192) and HF-1053-D-3 and are not presented herein.

A general method of determining the air flow rate required to cool electron tubes in either parallel or crossflow configurations has been developed at Cornell Aeronautical Laboratory. The method involves the use of curves of the heat transfer coefficient, h, versus the Reynolds number, Re, for each type of tube configuration, along with the solution of a series of general equations represented by nomographs.

First step is to evaluate the heat transfer coefficient, h, for the particular type of tube from the relation:

$$h = \frac{q}{A_s(\Delta t)_s} \qquad (78)(D.E.)$$

where:

h is the total heat transfer coefficient - $\frac{\text{watts}}{\text{in.}^2\text{-}^\circ C}$

q is the power dissipated - watts

A_s is the surface area of the component - square inches

Δt_s is the difference in the inlet air bulk temperature and surface temperature, i.e. $(t_s - t_b)$, in $^\circ C$

Figure 6-3 is the nomograph of this equation covering the normal range of values and offers a quick method of solving for h. The surface area (glass) of the tube in contact with the moving air can be approximated as described in the illustrative problems. The heat to be dissipated, q, is known and Δt_s is the difference between the desired operating temperature of the envelope and the inlet air temperature.

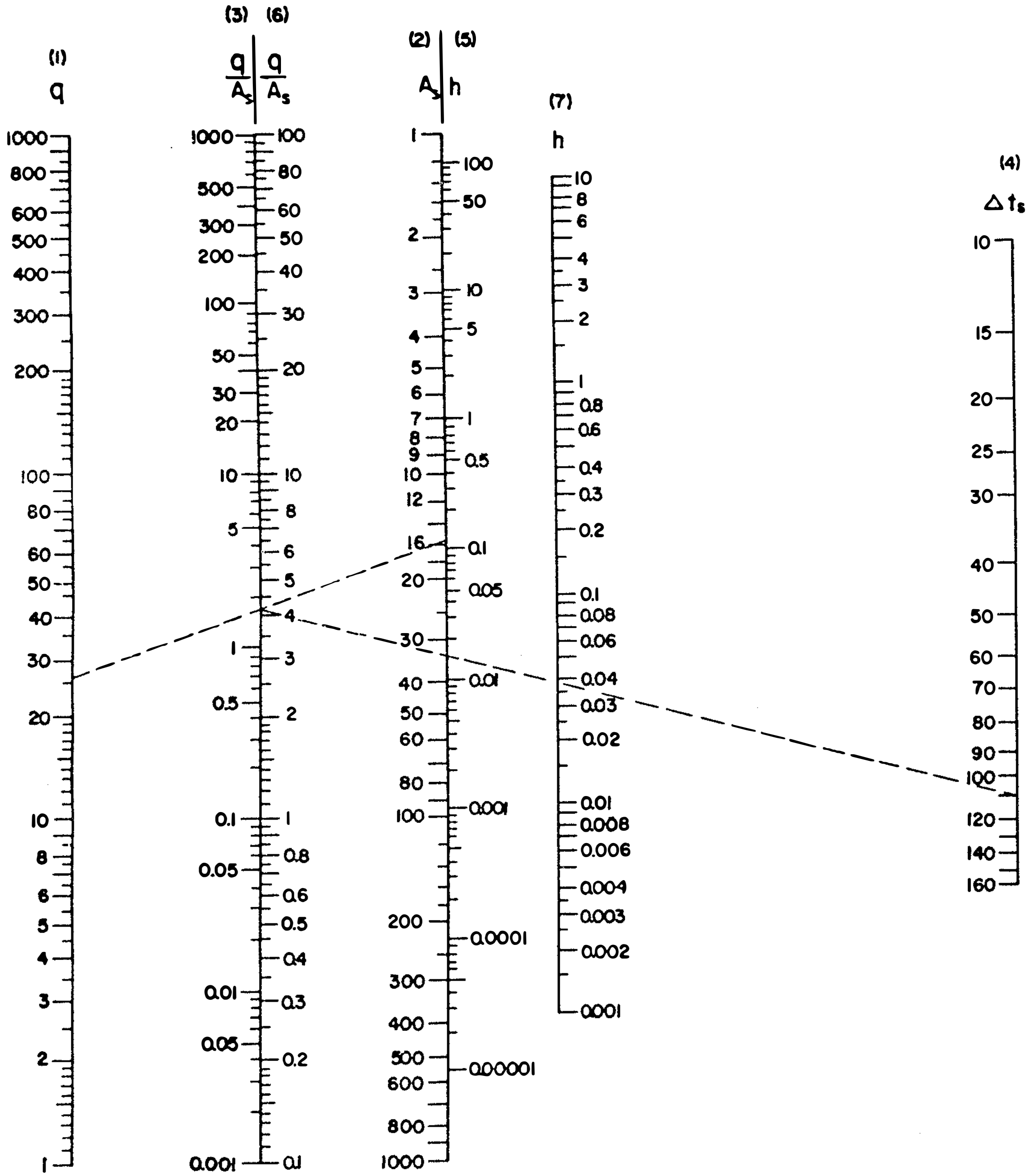


FIG. 6-3

DESIGN NOMOGRAPH FOR EQUATION (78)

$$h = \frac{q}{A_s \Delta t_s}$$

PROCEDURE:

(1) & (2) → (3)

(3) & (4) → (5)

(6) & (4) → (7) if $0.1 < \frac{q}{A_s} < 100$

BETTER RESOLUTION IS OBTAINED BY USING THE (6) & (7) SCALES

THE BROKEN LINES REFER TO THE SOLUTION FOR h IN SAMPLE PROBLEM III ON PAGE 84

DESIGNED BY J. B. COLEMAN
DRAWN BY R. B. SCOTT

The next step is to select the h vs. Re curve (part three of this section), which is derived from a tested setup, that most closely approaches the particular tube and configuration under consideration. The limits listed with each curve are those shown by the curve. These are the empirically proven ranges over which these curves are valid. However, the ranges can be extended by approximately 25% of the limit in either direction through extrapolation, with relatively little error.

Example (See Figure 6-15)

The proven limits are for Reynolds Numbers between 1200 and 12000. Applying the 25% criterion, the extrapolated limits become 900 and 15,000, respectively.

Figures 6-5 through 6-15 are the various h versus Re curves for several types of tubes based upon studies at this Laboratory. It should be realized that the heat transfer coefficient, h, referred to in these curves is the total heat transfer coefficient; that is, it includes all the modes of heat transfer. These coefficients are based on experimental data. The mathematical steps necessary to arrive at an h vs. Re curve are shown in Appendix H. Using the heat transfer coefficient, h, from step one, the Reynolds number may be selected from the appropriate curve.

The final step involves the use of another nomograph (Fig. 6-4) and the solving for m, the weight rate of flow in pounds per minute, from the relation:

$$m = \frac{(Re)A_c(\mu_f)}{D_e} \qquad (79)(D.E.)$$

where:

A_c is the minimum net cross sectional flow area - in.^{2}

μ_f is the air viscosity at the film temperature - lb./ft.-hr.

D_e is the equivalent diameter - in.

The equivalent diameter, D_e, is generally taken as the major diameter of a tube in crossflow or the average diameter, if a group of tubes in crossflow. In parallel flow, D_e is generally taken as the Hydraulic Radius of the configuration.

This general method gives accurate and quick solutions for determining the air flow requirements. Several alternate methods are discussed in Chapter III of this manual.

A thorough description of the values used for these terms for each tube and set of conditions is provided in Appendix H.

(1) h vs. Re Curves

This section presents design curves of total heat transfer coefficient versus Reynolds number. They have been developed to cover the majority of the configurations encountered in the forced-air cooling of vacuum tubes. Accompanying each curve is a description of the test configuration used as a basis for that particular curve. After the description for each of the h vs. Re curves are listed the approximations for computing the average surface temperature, t_s, the cross-sectional area, A_c, the surface area, A_s, and the equivalent diameter, D_e, pertinent to each configuration. The reader should use the same approximations when using the curves. For configurations not covered by the curves, equations and design nomographs have been devised. These are included in Section B-2 of this chapter.

(a) Octal-Base Tubes

Two types of configurations are available. They are -

i. Figure 6-5. h vs. Re_f for a single octal-base tube in crossflow. These data are based on a single instrumented 6L6GBY in crossflow.

The following are the physical parameters of the configuration:

t_s is the average surface temperature of the bulb.

A_c is the net cross-sectional area equal to the duct area minus the projected area of the tube.

A_s is the tube surface area approximated by a two cylinder configuration (see page 85).

D_e was assumed as the major diameter of the bulb.

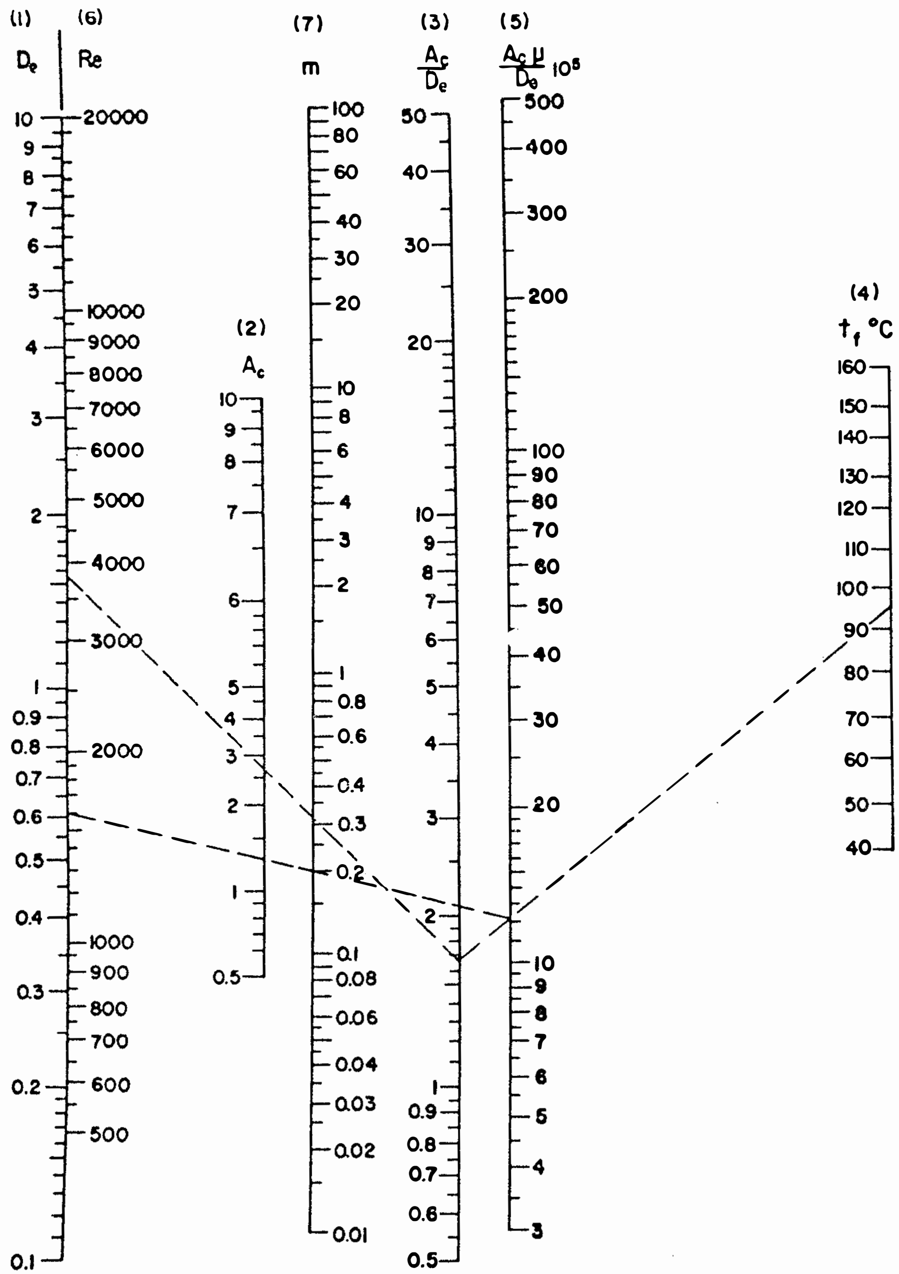


FIG. 6-4

DESIGN NOMOGRAPH FOR EQUATION (79)

$$m = \frac{Re\, A_c\, \mu_f}{D_e}$$

PROCEDURE:

(1) & (2) → (3)
(3) & (4) → (5)
(5) & (6) → (7)

THE BROKEN LINES REFER TO THE SOLUTION FOR m IN SAMPLE PROBLEM III ON PAGE 84

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DRAWN BY R.B. SCOTT

Note:

The tube used for this study was one made by the Tube Laboratory at Cornell University, Ithaca, New York. It has a thermocouple welded to its anode structure and is commonly known as an instrumented tube.

ii. Figure 6-6. h vs. Re_f for octal-base tube chassis crossflow-in-line. These data are based on a 6L6 chassis having 27 tubes in 3 longitudinal rows by 9 transverse rows. Note that two curves are presented. They are: (1) h-overall, based on the overall characteristics of the system, to be used when designing on an overall basis, and (2) h-last transverse row, to be used when designing on the basis of the last transverse row of tubes.

The following are the physical parameters of the configuration:

t_s is the average bulb surface temperature of six tubes; three in the first transverse row, and three in the last transverse row.

A_c is the net cross-sectional area equal to the duct area minus the projected area of three tubes (or one transverse row) averaged for nine transverse rows.

D_e is the approximate average diameter of the three types of tubes used.

A_s is the total surface area of 27 tubes.

(See Appendix H for shape approximations.)

(b) Miniature Tubes

The design curves for two tube types, the 6AQ5 and the 12BY7, in three configurations are presented:

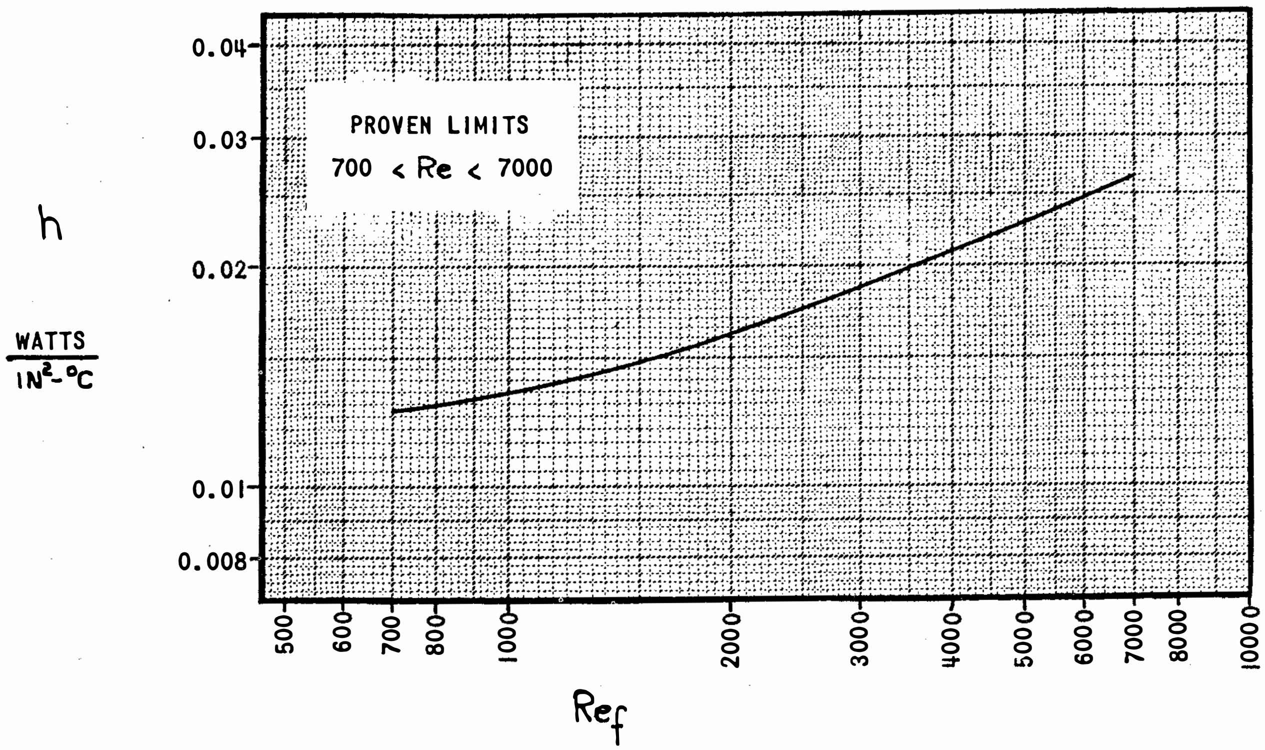


FIG. 6-5 h vs Re_f FOR SINGLE OCTAL BASE TUBE IN CROSSFLOW

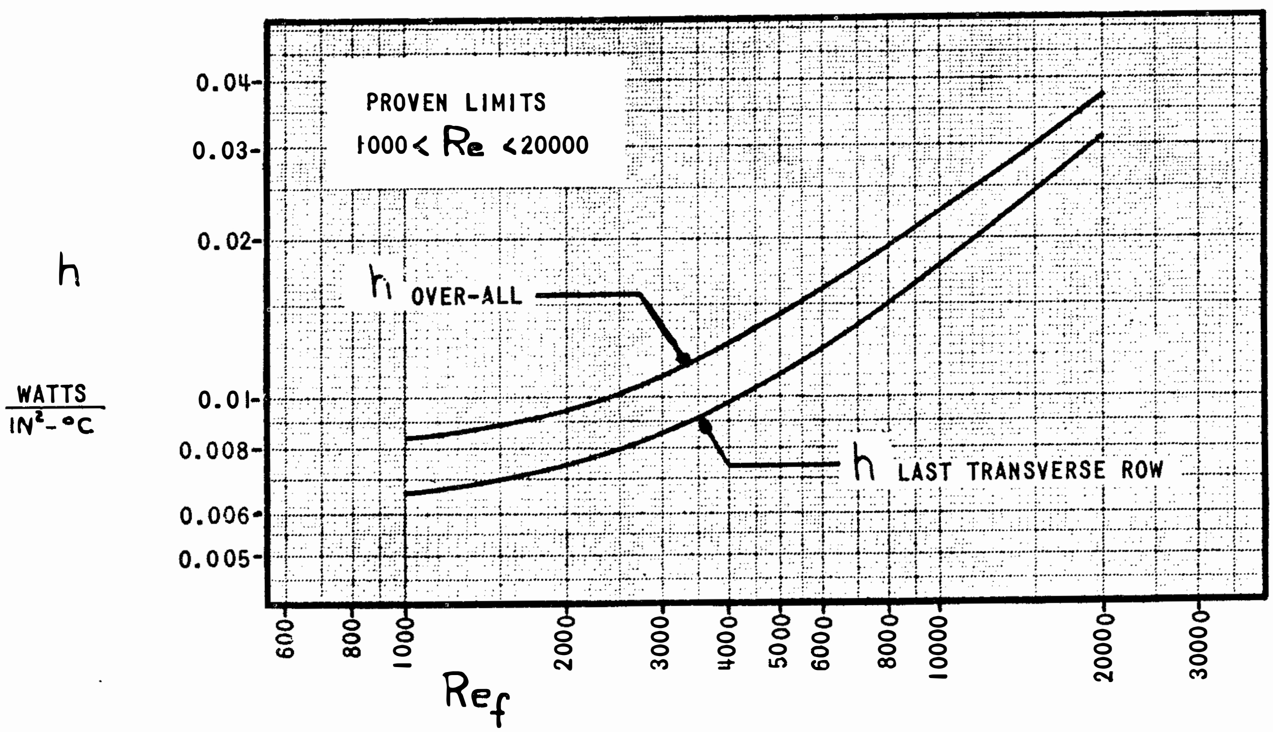


FIG. 6-6 h vs Re_f FOR OCTAL BASE TUBE CHASSIS CROSSFLOW-IN-LINE

i. Figure 6-7. h vs. Re_f for a single seven-pin miniature tube (6AQ5) in crossflow.

The following are the physical parameters of the configuration:

D_e is the tube bulb diameter.

t_s is the average surface temperature of the bulb.

A_c is the area of the duct minus the projected area of the tube.

A_s is the total surface area of the tube.

(See Appendix H for approximations.)

ii. Figure 6-8. h vs. Re_f for a single nine-pin miniature tube (12BY7) in crossflow.

The following are the physical parameters of the configuration:

D_e is the tube bulb diameter.

t_s is the average surface temperature of the bulb.

A_s is the total surface area of the tube.

A_c is the area of the duct minus the projected area of the tube.

(See Appendix H for approximations.)

iii. Figure 6-9. h vs. Re_f for seven-pin miniature tube chassis, crossflow-in-line. These data are based on a 6AQ5 chassis (crossflow-in-line) with 56 tubes; four longitudinal rows by 14 transverse rows.

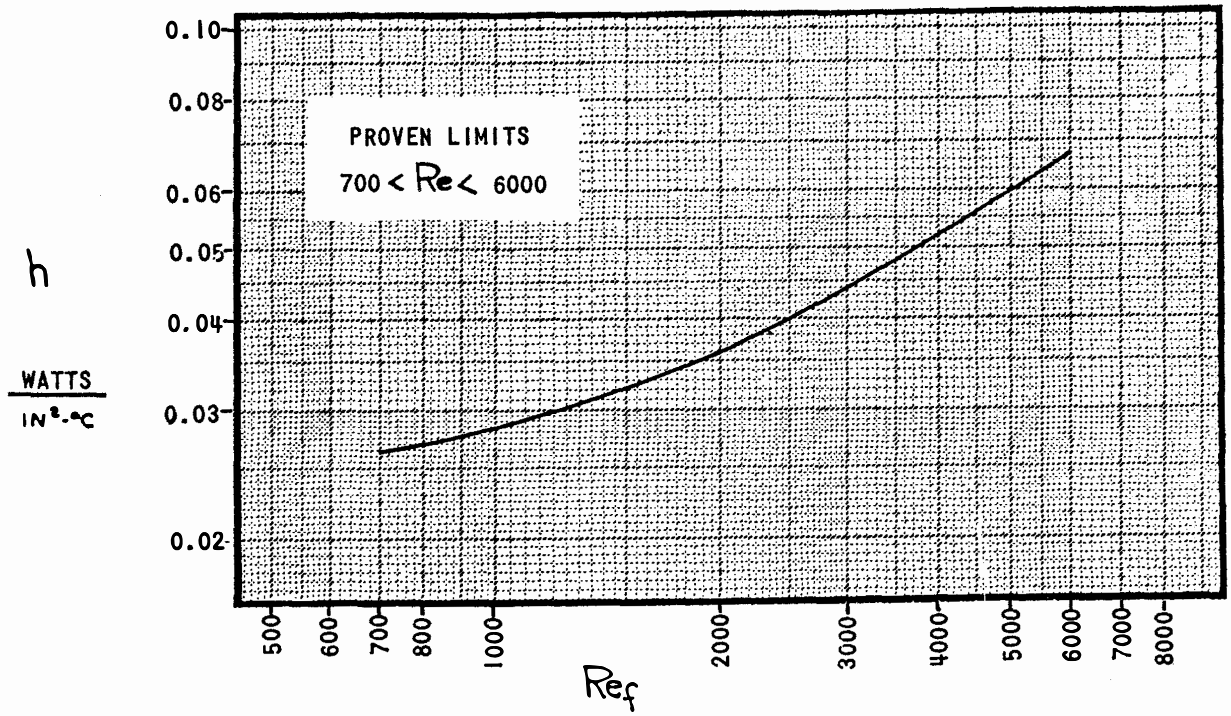


FIG. 6-7 h vs Re_f FOR SINGLE SEVEN PIN MINIATURE TUBE IN CROSSFLOW

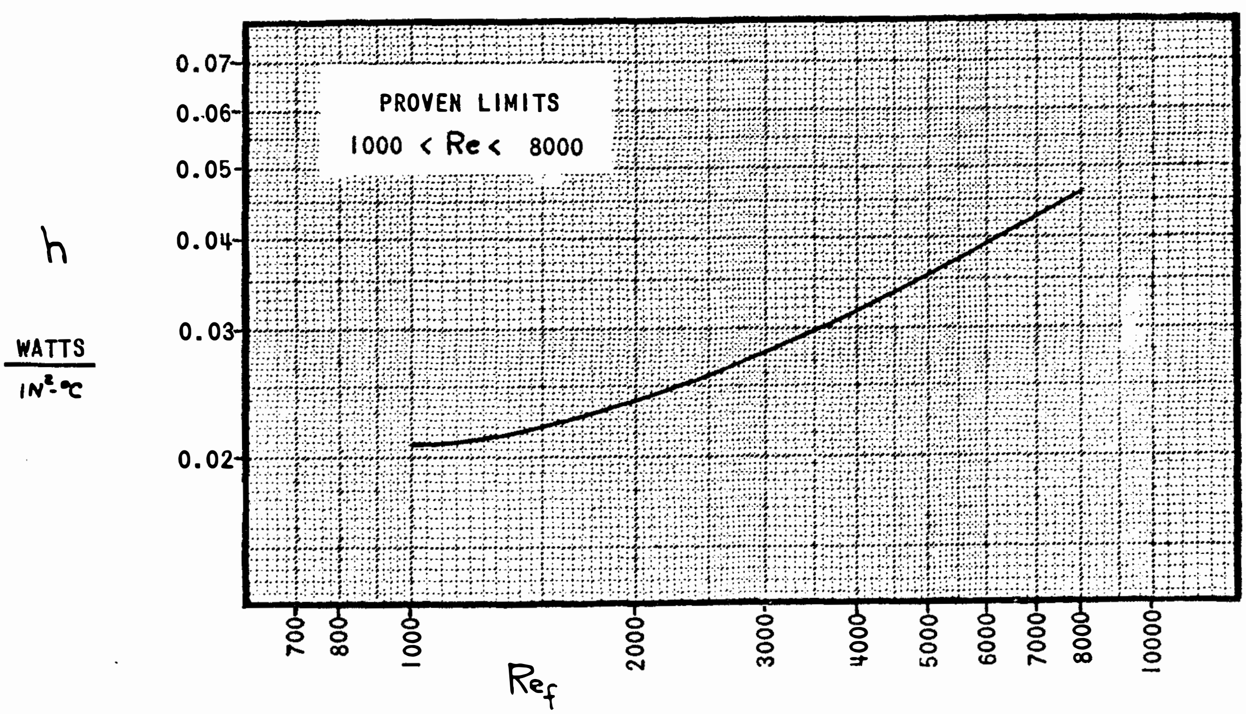


FIG. 6-8 h vs Re_f FOR SINGLE NINE PIN MINIATURE TUBE IN CROSSFLOW

This curve is for a blackened duct thermally bonded to the chassis.

The following are the physical parameters of the configurations:

D_e is the tube diameter (one tube).

A_s is the total surface area of 56 tubes.

A_c is the duct area minus the projected area of four tubes.

t_s is the average surface temperature of 11 tubes across the length of the chassis (i.e., approximately one tube per transverse row was instrumented).

Similar curves were plotted for several other configurations. Among these were: (1) galvanized duct, insulated from chassis, (2) galvanized duct, bonded to chassis, (3) inclusion of radiation baffles between the rows of tubes, and (4) energized integral filament transformers, mounted on the chassis underside. All of these curves were parallel to the one shown and deviated by only a few percent.

(c) Subminiature Tubes (Type T-3)

Three flow configurations are presented for the same type of assembly.

i. Figure 6-10. h v.s. Re_b for subminiature tubes in parallel flow.

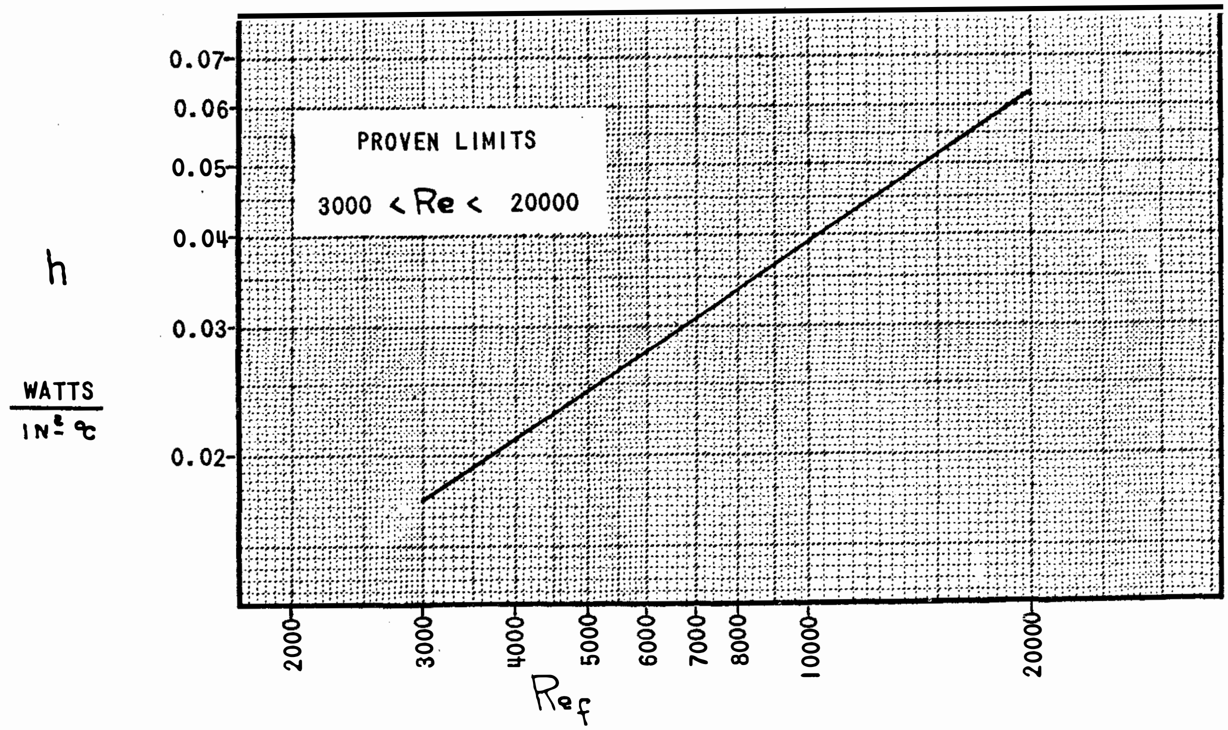


FIG. 6-9 h vs Re_f FOR SEVEN PIN MINIATURE TUBE CHASSIS, CROSSFLOW-IN-LINE

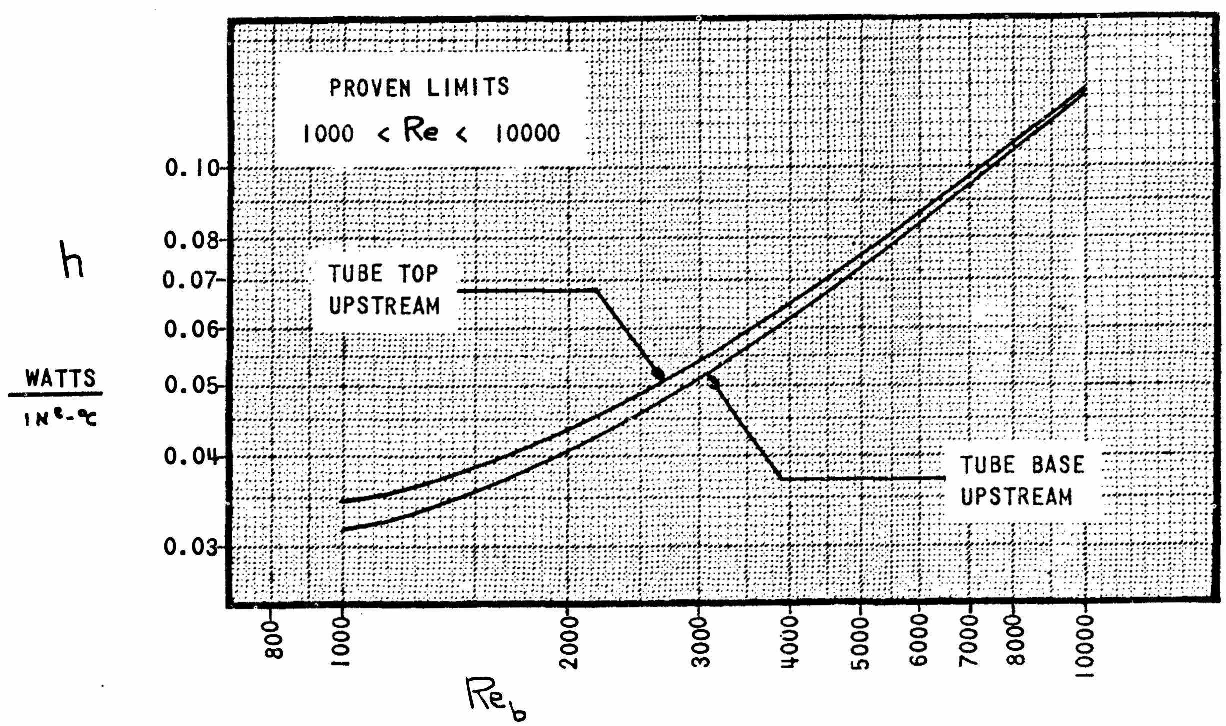


FIG. 6-10 h vs Re_b FOR SUB-MINIATURE TUBES IN PARALLEL FLOW

These data are based on three 5902's in a duct, mounted on five different panels.

The following are the physical parameters of the configuration:

t_s is the average surface temperature of the three tubes.

D_e is the hydraulic radius of the configuration in the duct.

A_s is the total surface area of three tubes.

A_c is the duct area minus the projected area of three tubes.

(The above are valid for all three 5902 cases.)

ii. Figure 6-11. h vs. Re_f for subminiature tubes in crossflow-in-line.

These data are based on 5902's (three tubes, one longitudinal row), mounted on five different panels. The results were very similar so that an average curve is presented.

iii. Figure 6-12. h vs. Re_f for subminiature tubes in crossflow. Data based on 5902's (three tubes, one transverse row).

(d) Transmitting Tubes

The data for two basic types of tubes are presented, i.e., the carbon anode "50 watt" transmitting type in which class fall the 203A, 211, 805, etc. and the radiation cooled type similar to the 4-250A wherein the anode operates in the visible portion of the spectrum (cherry red).

i. Figure 6-13. h vs. Re_f for "50 watt" type transmitting tubes in crossflow-in-line.

Data based on type 805 tube chassis, (four tubes, one longitudinal row).

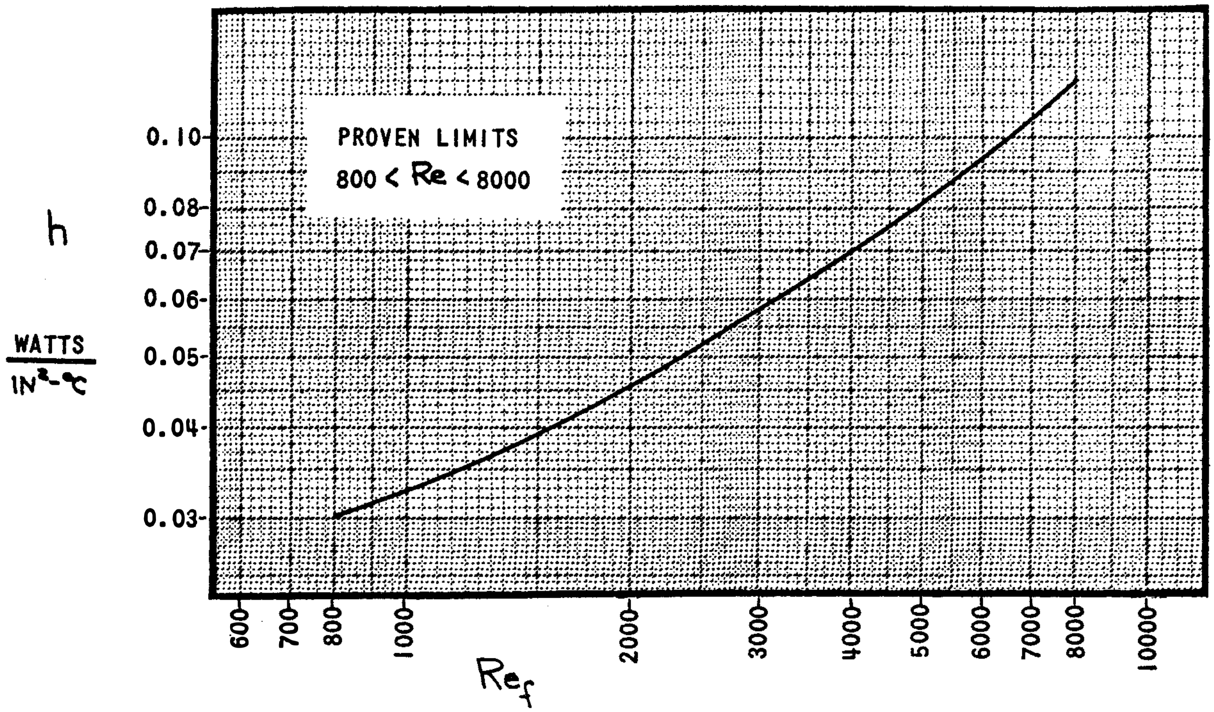


FIG. 6-11 h vs Re_f FOR SUB-MINIATURE TUBES IN CROSSFLOW-IN-LINE

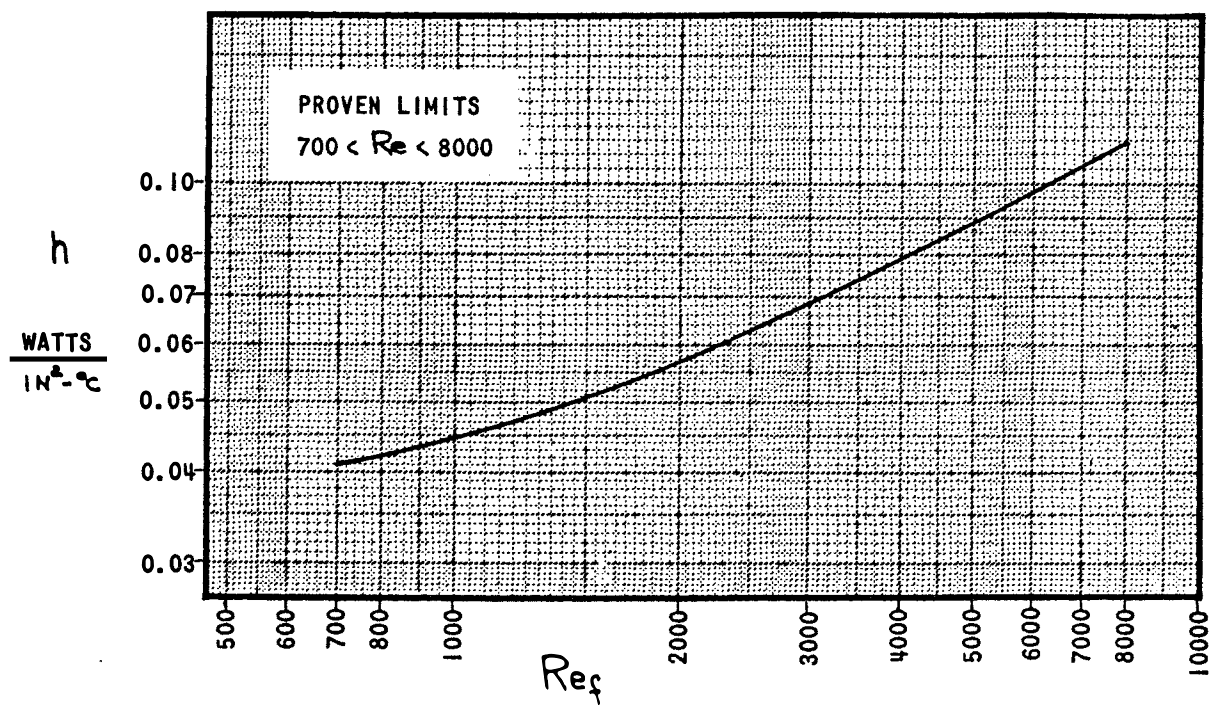


FIG. 6-12 h vs Re_f FOR SUB-MINIATURE TUBES IN CROSSFLOW

The following are the physical parameters of the configuration:

D_e is the diameter of the tube.

A_s is the total area of four tubes.

t_s is the average surface temperature of the four tubes.

A_c is the duct area minus the projected area of one tube.

(See Appendix H for approximations.)

ii. Figure 6-14. h vs. Re_f for 4-250A type transmitting tetrode in parallel flow (with shield).

Data are based on the 4-250A chassis, three tubes in parallel flow.

A cylindrical shield is used to direct the air flow parallel to the major axis of the tube.

The following are the physical parameters of the configuration:

D_e is the hydraulic radius of the annular section formed by the major diameter of the tube and the shield.

A_s is the surface area of one tube.

t_s is the average surface temperature of the tube.

A_c is the minimum annular area between the tube and the shield.

iii. Figure 6-15. h vs. Re_f for 4-250A type transmitting tetrode in parallel flow (without shield).

This case covers the situation wherein there is no shield to confine the air flow. It is essentially "free flow" for this type of configuration.

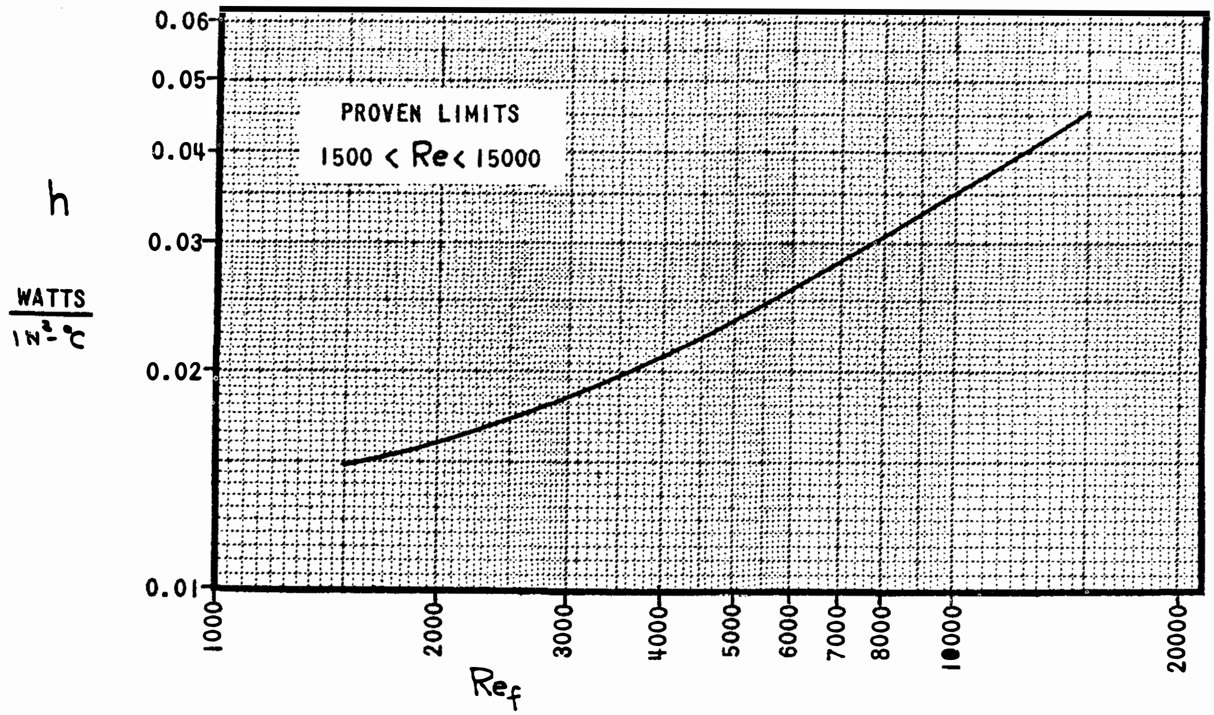


FIG. 6-13 h vs Re_f FOR "50 WATT" TYPE TRANSMITTING TUBES, CROSSFLOW-IN-LINE

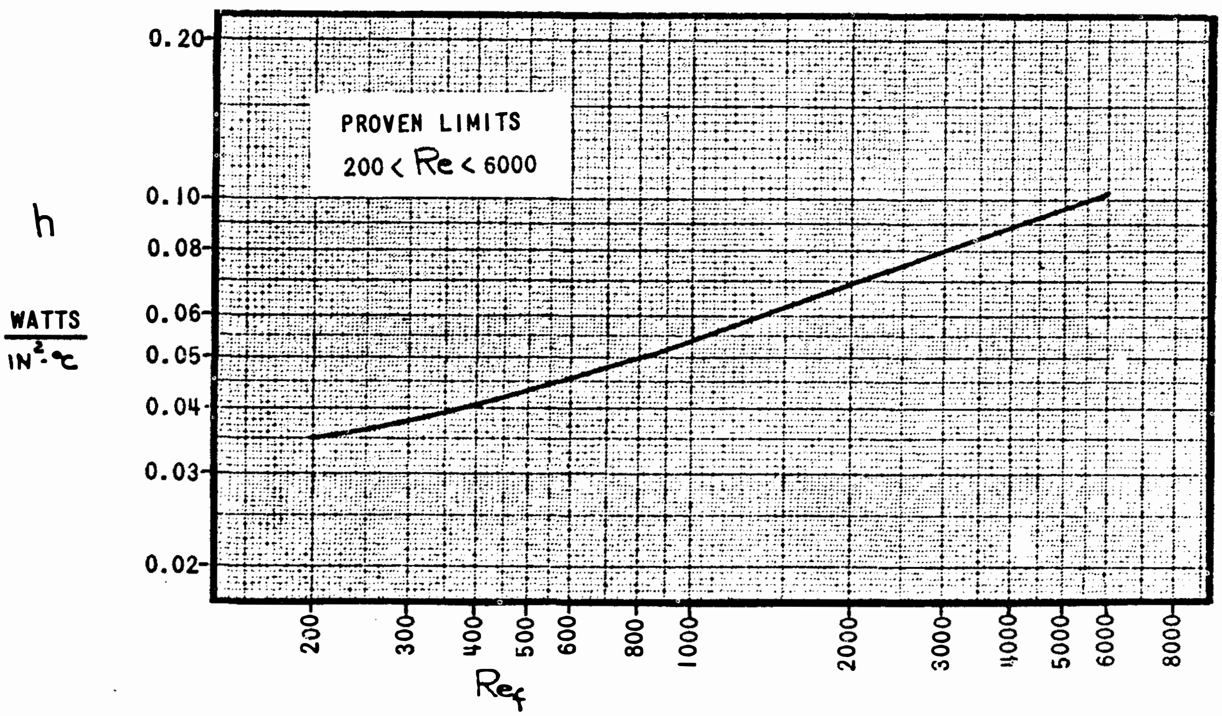


FIG. 6-14 h vs Re_f FOR 4-250A TYPE TRANSMITTING TETRODE, PARALLEL FLOW (WITH SHIELD)

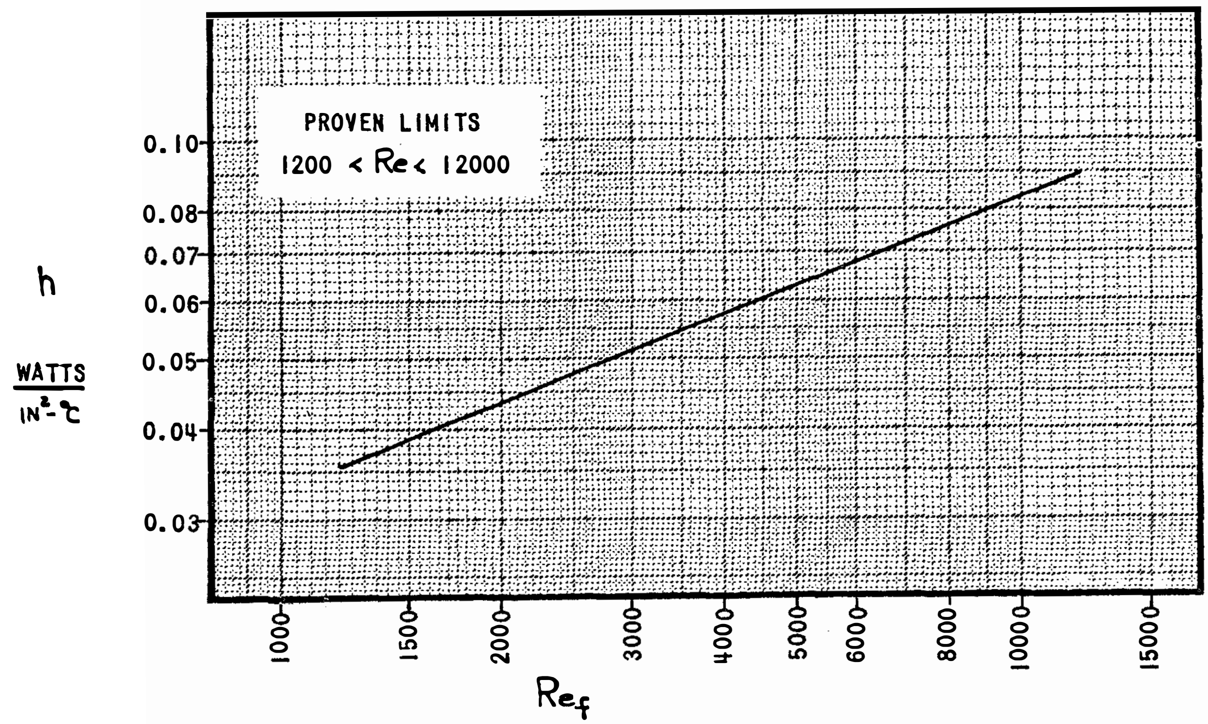


FIG. 6-15 h vs Re_f FOR 4-250A TYPE TRANSMITTING TETRODE, PARALLEL FLOW (WITHOUT SHIELD)

NOTE

The design methods presented in this chapter are based on clean parts, i.e., parts that are free from dust and oil. Oil films and dust can combine to form an insulating film on a heat source surface. This high thermal resistance film then becomes part of the heat path. Unless parts and their fins are kept clean, they will run much hotter than predicted.

The following are the physical parameters of the configuration:

D_e is the hydraulic radius of the base section of the tube.

A_s is the surface area of the tube.

t_s is the average surface temperature of the tube.

A_c is the area of the five holes in bottom of the tube base.

(2) Design Examples

The following are illustrative examples which demonstrate a solution by the recommended method.

(a) Sample Problem III

Find the required air flow to cool a single 6L6GBY tube in crossflow for the following conditions:

heat dissipated = 26 watts

maximum tube temperature = 150°C

inlet air temperature = 40°C

duct size = 2 in. wide by 3.5 in. high

Figure 6-16 shows the tube in the duct.

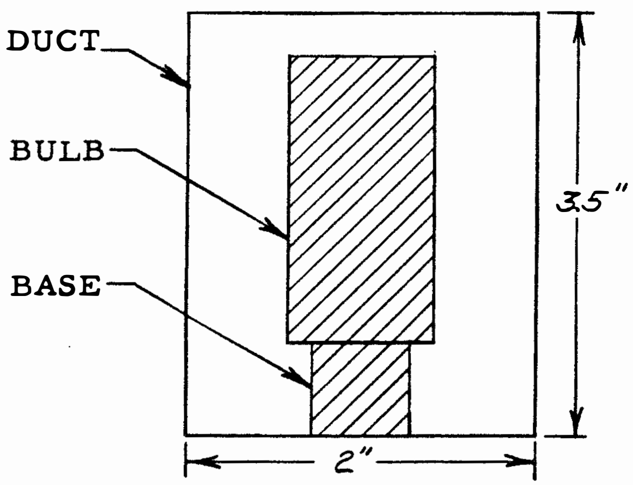


Figure 6-16 - Single Tube in a Duct.

To approximate the tube surface area, assume that it is composed of two cylinders, 2-1/8 inches by 1-9/16 inches in diameter and 3/4 inches by 1-3/8 inches in diameter (see Figure 6-17).

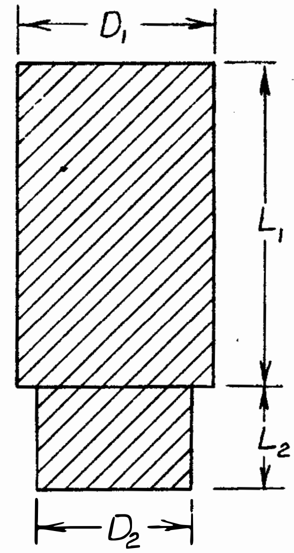


Figure 6-17 - Cylindrical Approximation of a 6L6GBY.

where in Figure 6-17:

D_1 = 1.56 inches, major diameter of tube envelope.

L_1 = 2.12 inches, length of cylindrical approximation of glass envelope.

D_2 = 2.12 inches, diameter of tube base.

L_2 = 0.75 inches, length of tube base.

Then the surface area is:

$$A_s = \pi \left(D_1 L_1 + D_2 L_2 + \frac{(D_1)^2}{4}\right)$$

$$A_s = \pi \left[1.56(2.12) + .75(1.37) + \frac{(1.56)^2}{4}\right]$$

$$A_s = 15.6 \text{ in.}^2$$

The temperature rise, equal to the difference between average tube bulb temperature and the inlet air bulk temperature $(t_s - t_b) = 150 - 40 = 110^\circ C$.

Using the nomograph (Fig. 6-3) expressing the relation

$$h = \frac{q}{A_s \; \Delta \; t_s}$$

the value of h is found to be 0.015 watts/in.^{2}-$^\circ$C.

The second step is to find the correct Reynolds number Re_f from Fig. 6-5. The curve gives a value of $Re_f = 1600$.

The film temperature, t_f, upon which the Reynolds number is based, is equal to one half the sum of the average allowable tube bulb temperature and the inlet air bulk temperature.

$$t_f = \frac{t_s + t_b}{2} = \frac{150 + 40}{2} = 95^\circ C$$

The viscosity for air based upon this film temperature is $\mu_f = .052$ (Appendix).

For the third step, use the nomograph of Fig. 6-4, based on the relation:

$$m = \frac{Re(A_c) \; \mu_f}{D_e}$$

A_c is the net cross-sectional air flow area equal to the difference between the duct area and the projected area.

$$A_c = 2(3.5) - \left[1.56(2.12) + .75(1.37)\right]$$

$$A_c = 7.0 - (3.31 + 1.04)$$

$$A_c = 2.65 \text{ in.}^2$$

D_e is equal to the major diameter of the tube in cross-flow.

$$D_e = 1.56 \text{ in.}$$

Using the nomograph of Fig. 6-4, the required weight rate of flow of air is found to be:

$$m = 0.20 \text{ lb./min.}$$

The results for the preceding calculations are listed below for comparison with actual measured conditions using a different size duct.

Design Example			Test		
A_c	=	2.65 in.^{2}	A_c	=	5.79 in.^{2}
t_s	=	150°C	t_s	=	152°C
q	=	26 watts	q	=	26 watts
t_b	=	40°C	t_b	=	40°C
m	=	0.20 lb/min.	m	=	0.26 lb/min.

Comparison with test data for a slightly different configuration shows close agreement. Decreasing the net cross-sectional flow area below a minimum does not have a significant effect on the air flow requirement.

(b) Sample Problem IV

This problem outlines the procedure for the thermal design of a forced-air cooling system for a miniature tube chassis in crossflow, with tubes-in-line. This configuration represents a major cooling problem. Such a setup, with each tube dissipating considerable wattage, and being in close proximity to its neighbors, would seriously impair reliability and long life, unless preventive measures were taken. A group, or several groups, of such tubes might easily be found in, for example, a computer. The use of forced-air cooling, as shown in the following problem, would considerably improve the reliability of the equipment.

The chassis will utilize 56 type 6AQ5 tubes mounted in four longitudinal rows by 14 transverse rows. Figure 6-18 shows the arrangement of the tubes and duct dimensions.

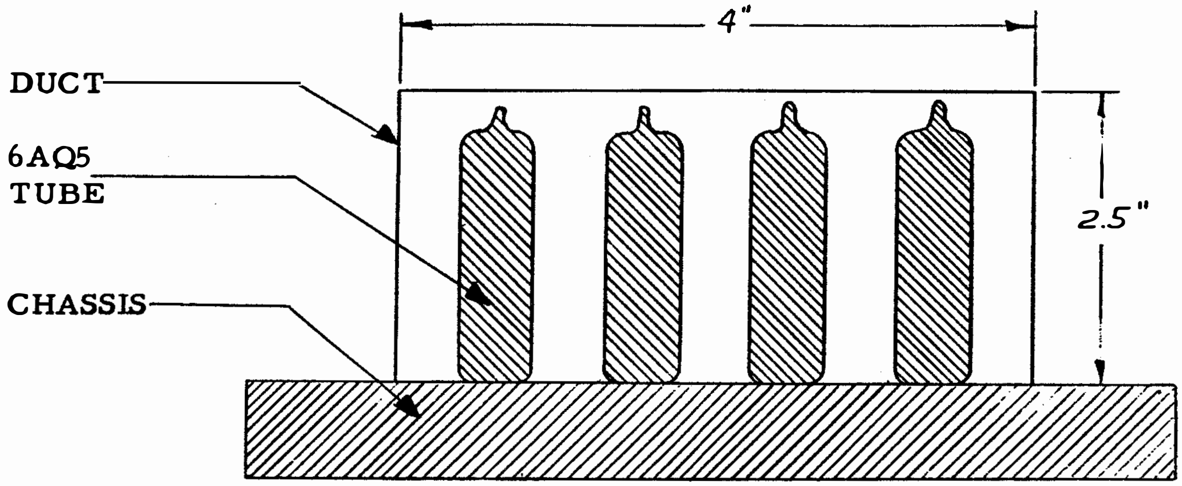


Figure 6-18 - End View of 6AQ5 Chassis

The temperature of the tubes in the last row will be critical, since the air will be heated before reaching them. The conditions for the problem are as follows:

t_s, maximum allowable tube surface temperature $= 150^{\circ}C$.

inlet air bulk temperature, $t_b = 40^{\circ}C$.

total heat dissipated, $q = 700$ watts

duct size = 2.5 in. high by 4.0 in. wide

The duct should be bonded to the chassis to enhance conduction, and should be blackened inside.

The 6AQ5 tube may be assumed to be a cylinder 0.71 inches in diameter and 2.12 inches high. The surface area for 56 tubes is:

$$A_s = 56 \left[\pi (.71)(2.12) + \frac{(.71)^2}{4} \right]$$

$$A_s = 56 (4.72 + .40)$$

$$A_s = 287 \text{ in.}^2$$

$$\Delta t_s = (t_s - t_b) = 150 - 40 = 110^{\circ}C$$

Using the nomograph of Fig. 6-3 expressing the relation;

$$h = \frac{q}{A_s \; \Delta t_s}$$

The value of h is equal to 0.021 watt/in.^{2}-^{o}C.

The second step is to find the Reynolds Number, Re_f. From the h vs. Re_f curve Fig. 6-9, Re_f = 4000.

The film temperature upon which the Re_f is based is:

$$t_f = \frac{t_s + t_b}{2} = \frac{190}{2} = 95^oC$$

viscosity of air, μ_f = .052 (Appendix).

Using Fig. 6-4, the nomograph based on the relation:

$$m = \frac{Re\, A_c\, \mu_f}{D_e}$$

net cross sectional area A_c = 2.5(4) - 4(2.12)(.71)

= (10 - 6.0) = 4.0 in.^{2}

The equivalent diameter D_e = 0.71"

therefore:

m = 1.55 lb/min.

The conditions and results of the above problem are listed below along with actual measured data obtained with a different size duct for general comparison.

Design Problem	Test
A_c = 4.0 in.^{2}	A_c = 5.23 in.^{2}
t_s = 150^{o}C	t_s = 163^{o}C
t_b = 40^{o}C	t_b = 40^{o}C
q = 700 watts	q = 670 watts
m = 1.55 lb/min.	m = 2.25 lb/min.

(c) Sample Problem V

To illustrate the use of the general method of forced-air cooling, a chassis design using parallel flow with three 5902 tubes mounted on a panel with clips as shown in Fig. 6-19 will be presented:

Figure 6-19 (a & b) are end and top views respectively of the configuration.

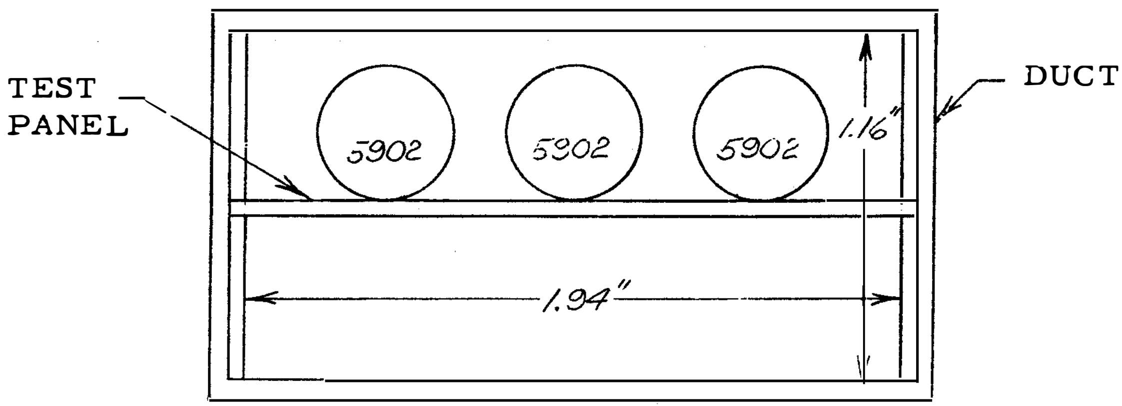


Figure 6-19-a - End View

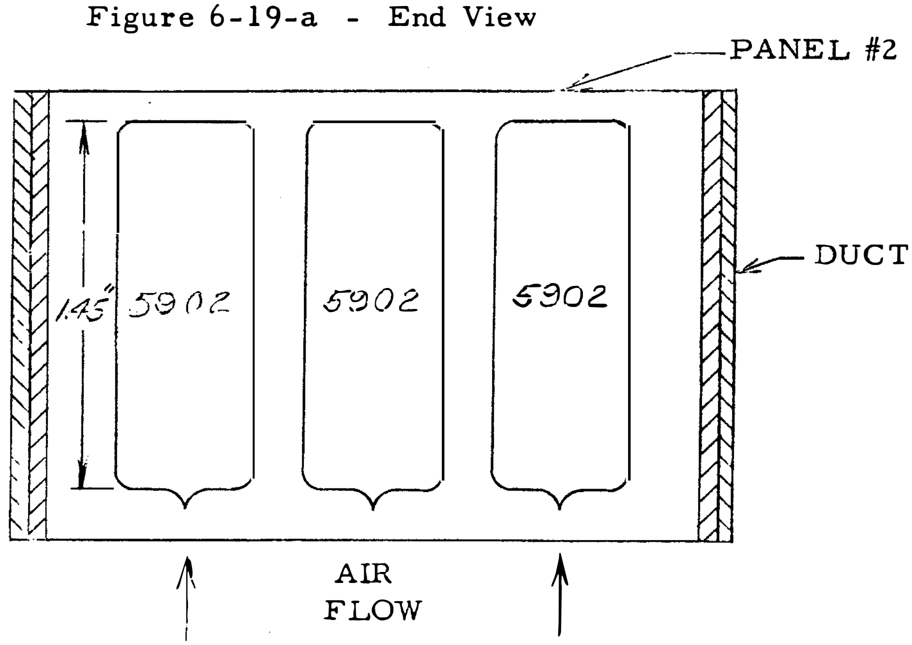


Figure 6-19-b - Top View

Views of the 5902 Configuration
(Tube Top Upstream)

The design conditions are:

maximum tube surface temperature $= 150^{\circ}C$

inlet air temperature $= 40^{\circ}C$

power to be dissipated = 18 watts

duct dimensions = 1.0 inches high by 2.0 inches wide

panel #2 = 0.05 inches thick

$$A_s\text{, surface area,} = 3\pi\left[(.375)(1.45) + \frac{(.375)^2(2)}{4}\right]$$

$$A_s = 9.42\ (.615) = 5.80\ \text{in.}^2$$

The difference between the tube surface temperature and the inlet air bulk temperature is:

$$\Delta t_s = 150 - 40 = 110^{\circ}C$$

Using the nomograph of Fig. 6-3, the value of the heat transfer coefficient, h, is 0.028 watt/in.^{2}-$^{\circ}$C. Referring to the proper h vs. Re curve, Fig. 6-10, read Re = 500, based upon a film temperature of:

$$t_f = \frac{t_s + t_b}{2} = \frac{190}{2} = 95^{\circ}C$$

Then:

$$\mu_f = .052\ \text{lb/ft-hr} \quad \text{(Appendix)}$$

The last step, using nomograph of Fig. 6-4, is:

$$\text{net area } A_c = 1.0(2) - \left[\frac{3(\pi)(.375)^2}{4} + .05(2)\right]$$

$$A_c = (2 - 0.43) = 1.57\ \text{in.}^2$$

The equivalent diameter D_e is equal to four times the hydraulic radius, or:

$$D_e = \frac{4A_c}{\text{Per.}}$$

where:

A_c is the net cross-sectional area

Per. is the wetted perimeter

$$D_e = \frac{4\ (1.57)}{7 + 3\ (\pi).375 + 4.1}$$

$$D_e = \frac{6.28}{14.63} = .43 \text{ in.}$$

The weight rate of flow under these conditions will be

$$m = .15 \text{ lb/min.}$$

Following is a tabulated comparison using actual measured data obtained with a different size duct.

Design Problem	Test
$A_c = 1.57 \text{ in.}^2$	$A_c = 1.81 \text{ in.}^2$
$t_s = 150^\circ C$	$t_s = 121^\circ C$
$t_b = 40^\circ C$	$t_b = 40^\circ C$
$q = 18$ watts	$q = 17.7$ watts
$m = .15$ lb/min.	$m = .24$ lb/min.

This general method of designing for forced-air cooling of vacuum tubes provides close correlation to actual test conditions and is believed to be a fast, easy method for such design calculations.

b. Electron Tubes in Tube Shields

(1) Crossflow

An acceptable tube shield for crossflow air cooling must have slots or windows around the periphery to allow the air to come in contact with the glass tube surface, or the shield must be fitted with a metal liner or fingers to insure adequate glass-to-metal contact for the removal of heat by conduction to the air stream. In essence, such a shield provides a tube with metal fins.

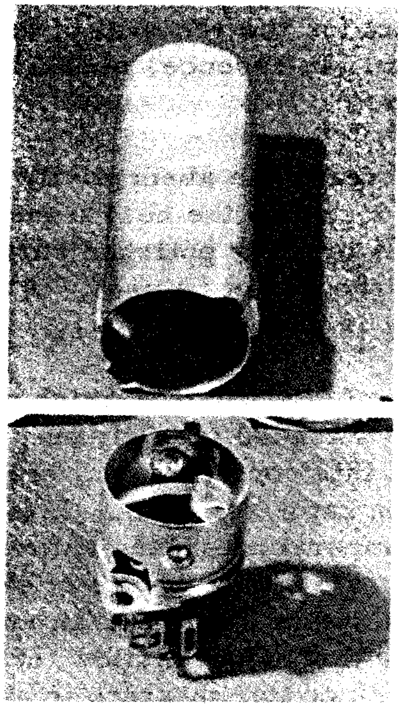
Further, it is necessary to increase the absorptivity of the inner surface of the shield to increase the heat transfer by radiation from the envelope. A brightly polished surface is a poor radiation absorber and should not be used. A dull, oxidized and blackened surface is preferred.

Figure 6-20 shows three types of tube shields which were evaluated in crossflow. It was intended to study only the effects of forced convection on this configuration, thus the shields were without conduction paths to a sink. Fig. 6-21 is a plot of the tube surface temperature on the bottom-downwind side versus air flow rate for each type of shield. Also plotted is the tube surface temperature for an unshielded tube. This illustrates that, in crossflow forced-air cooling, the lowest tube surface temperature is attained with a bare tube. Figure 6-22 is a plot of the downwind shield temperature versus air flow rate for each of the three shields. Again the conduction shield has the best thermal characteristics.

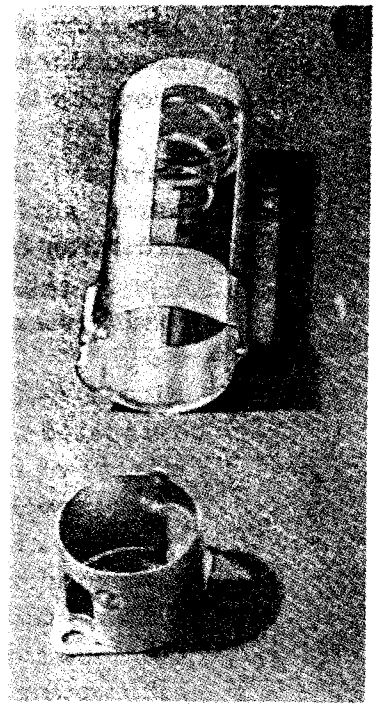
The JAN shield had very poor thermal characteristics in crossflow cooling because the air had no means of contacting the tube surface to provide heat removal by convection. A corrugated metal liner aided in the removal of heat by conduction to the external surface of the shield and thence to the air. Such an arrangement is excellent for parallel flow cooling when the air is forced through the metal liner (see parallel flow).

The radiation-convection shield had vertical slots around the periphery to allow air to contact the surface of the bulb. This shield did not have adequate metal-to-glass contact to provide for good conduction to its outer surface. The tube surface temperatures were, however, much lower than the JAN tube shield. The best shield for crossflow cooling was the conduction shield. This shield had three vertical slots equally spaced around the periphery and included a metal liner which contacted the tube surface to provide conduction cooling.

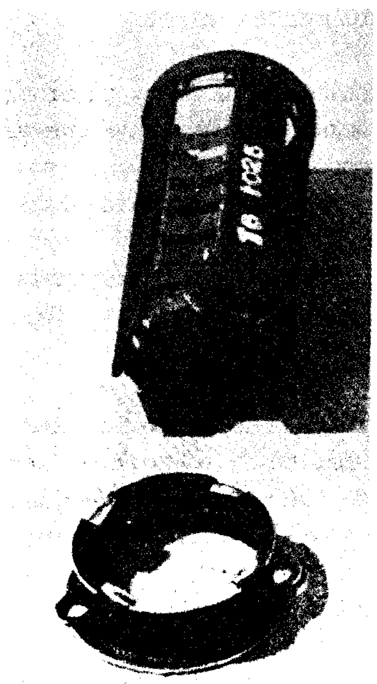
If temperature-sensitive parts and other tubes constitute the surrounding objects (in which case radiant heat transfer to these parts is not desired), then radiation shields should be used in crossflow cooling.



(a) JAN SHIELD



(b) RADIATION-CONVECTION SHIELD



(c) CONDUCTION SHIELD

FIG. 6-20 THREE TYPES OF SHIELDS USED IN CROSSFLOW COOLING

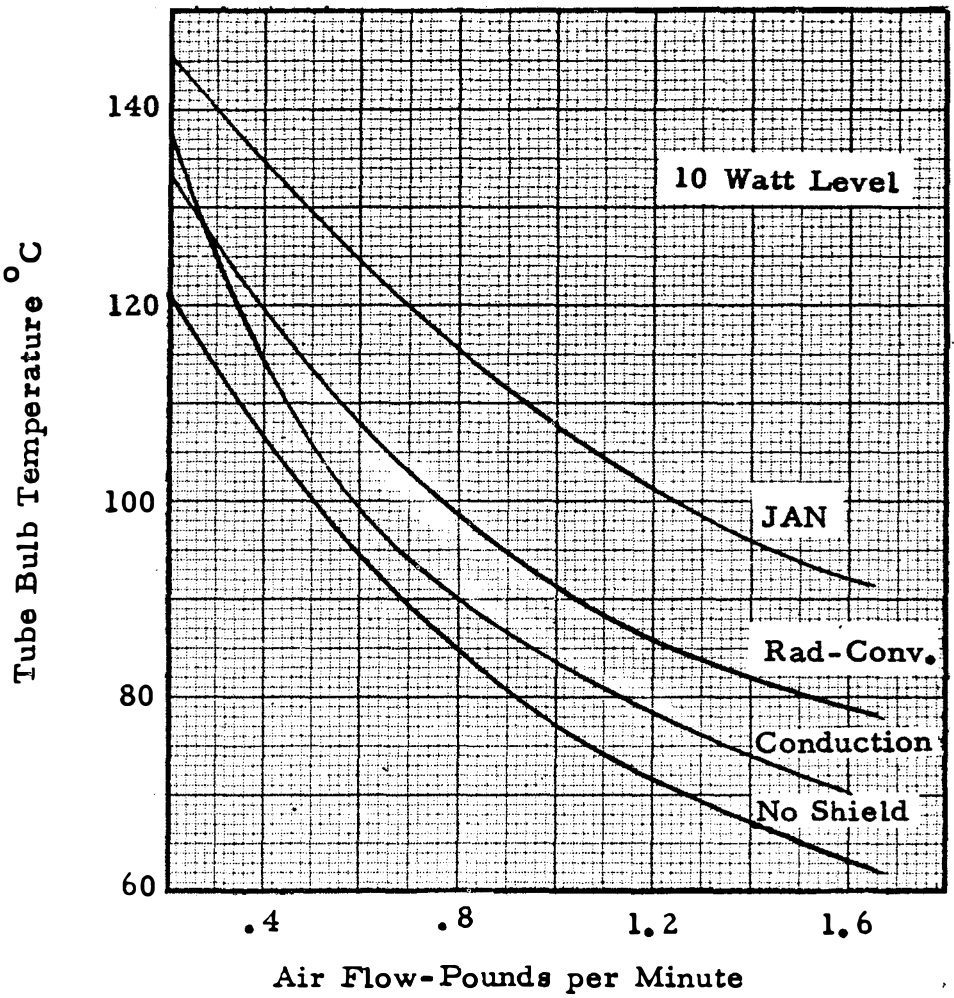


Fig. 6-21 Downwind Bottom Bulb Temperature for Seven Pin Miniature Tubes in Crossflow

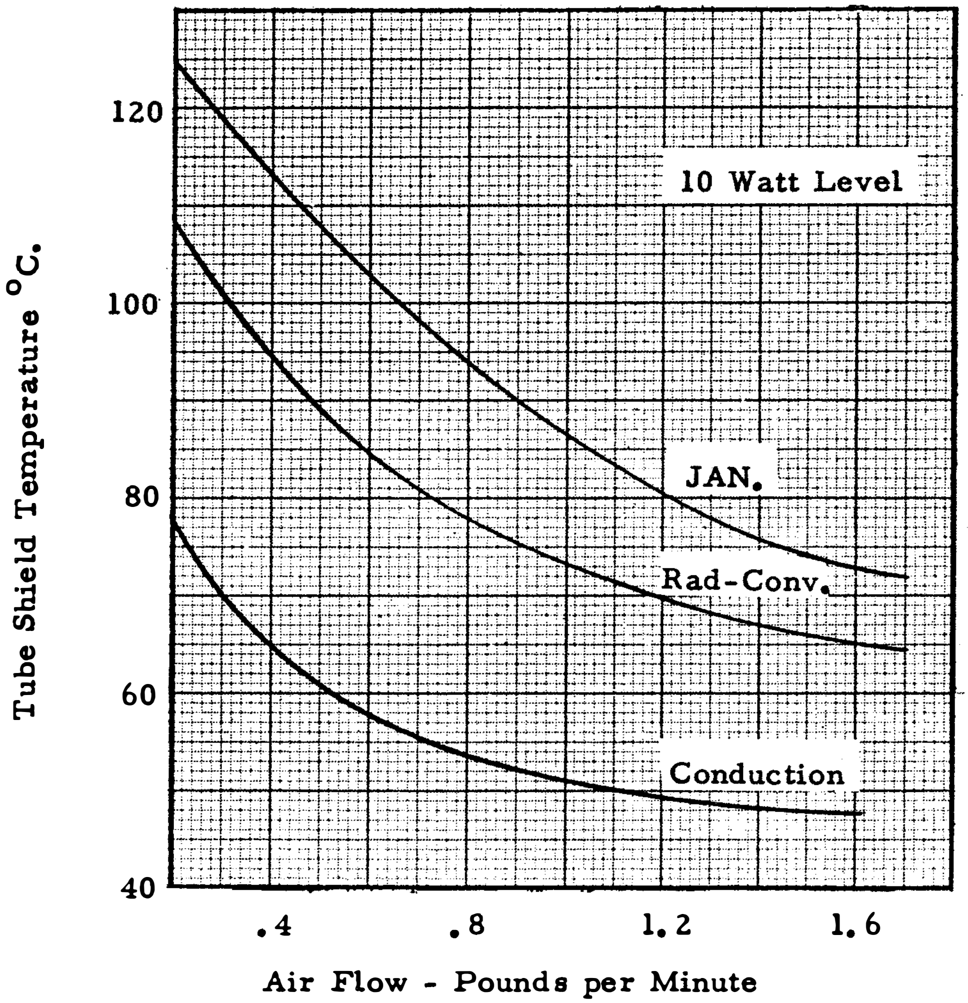


Fig. 6-22 Downwind Shield Temperature for Seven Pin Miniature Tube Shields in Crossflow

(2) Parallel Flow

Shields are effective in parallel flow because they guide the air stream and increase turbulence. Figure 6-23 illustrates an application of special miniature tube sockets which allow a controlled flow of air inside the tube shield. This is an effective approach and provides for the economical usage of air. The chassis must be closed and relatively air-tight in order to serve as an effective plenum chamber as shown in Fig. 6-23. The plastic insert in the socket has slots around its periphery which act as nozzles for the cooling air.

Figure 6-24 shows bulb hot spot temperature vs. air flow rate for flow in both directions, i.e., upwards and downwards. (Upward flow is accomplished by pressurizing the plenum chamber, and downward flow, by partially evacuating it.) Hot spot temperatures are about the same for each direction of flow. However, the chassis is cooler and the hot spot higher on the tube for the upward flow condition. With downward flow the chassis tends to become hotter, which may result in undesired heat conduction to other components.

Liners in the tube shields caused a lowering of the hot spot temperature. The air required for a given degree of cooling for shields without liners is several times that of shields with liners.

The heat transfer coefficient is defined as:
(See Fig. 6-25)

$$h = \frac{\text{watts dissipated in the tube}}{\text{surface area of bulb x bulb hot-spot temp. rise}} \qquad (80)$$

With the special shield and corrugated liner the equation is:

$$h = 0.026 \ m^{0.459} \qquad (81)(\text{D. E.})$$

where:

m is the weight rate of flow - lb/hr.

Without the liner, the equation is:

$$h = 0.0154 \ m^{0.459} \qquad (82)(\text{D. E.})$$

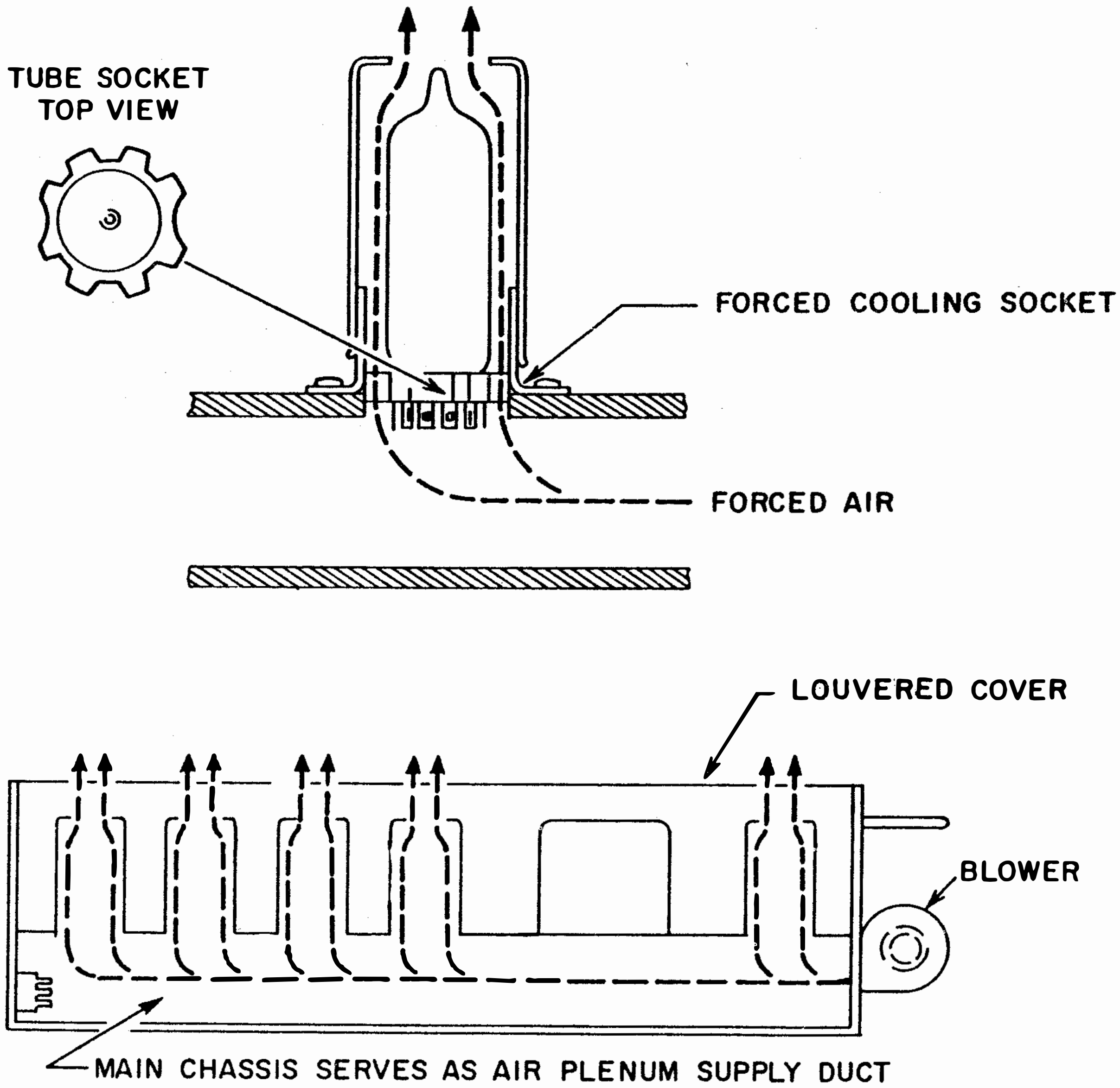
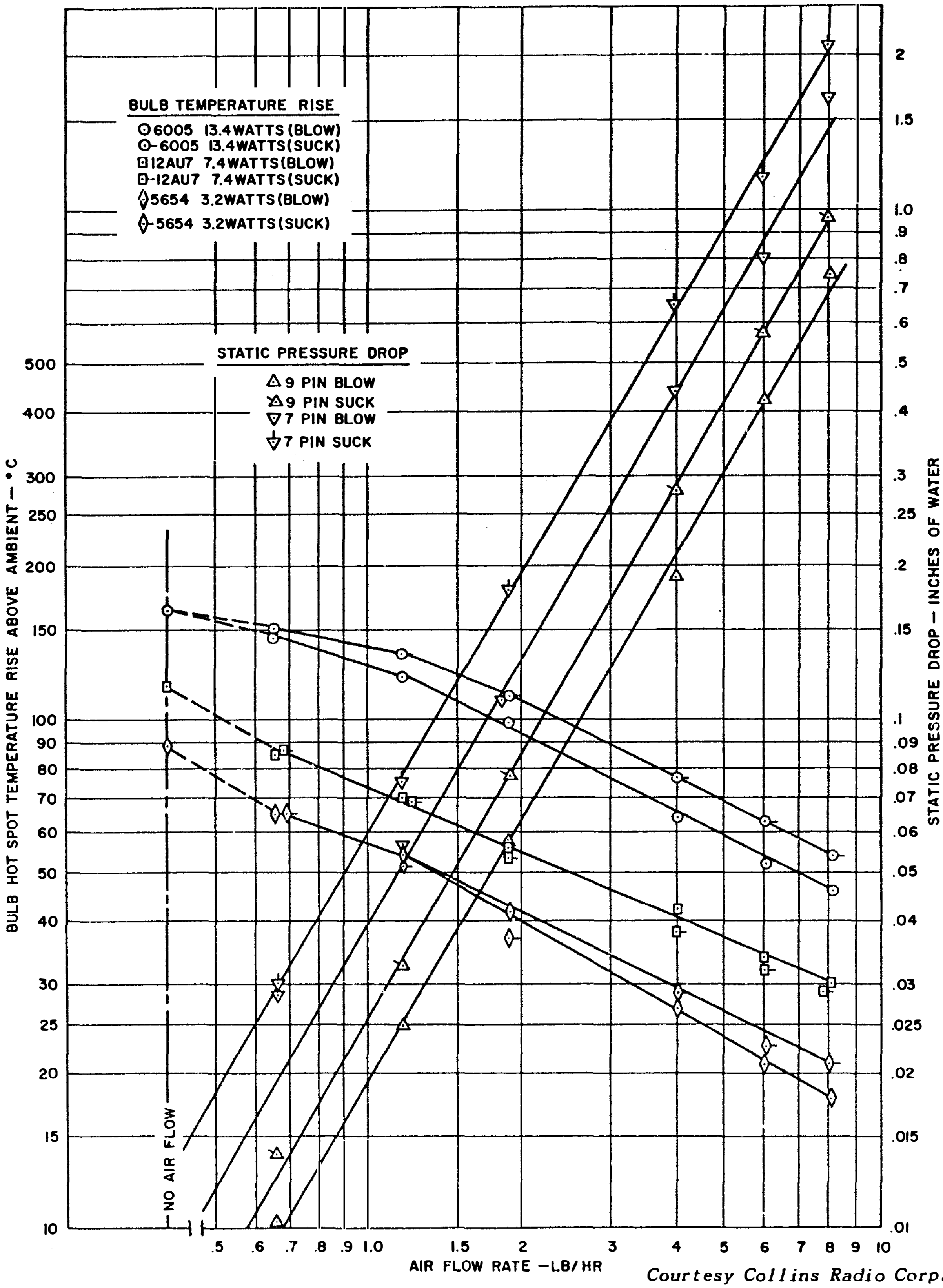


FIG. 6-23 FORCED-AIR COOLING SOCKET ON CONVENTIONAL PAN CHASSIS TYPE EQUIPMENT

Courtesy Collins Radio Corp.



Courtesy Collins Radio Corp.

FIG. 6-24 BULB HOT SPOT TEMPERATURE RISE AND STATIC PRESSURE DROP VERSUS AIR FLOW RATE IN POUNDS PER HOUR FOR A 12AU7, 5654 AND 6005

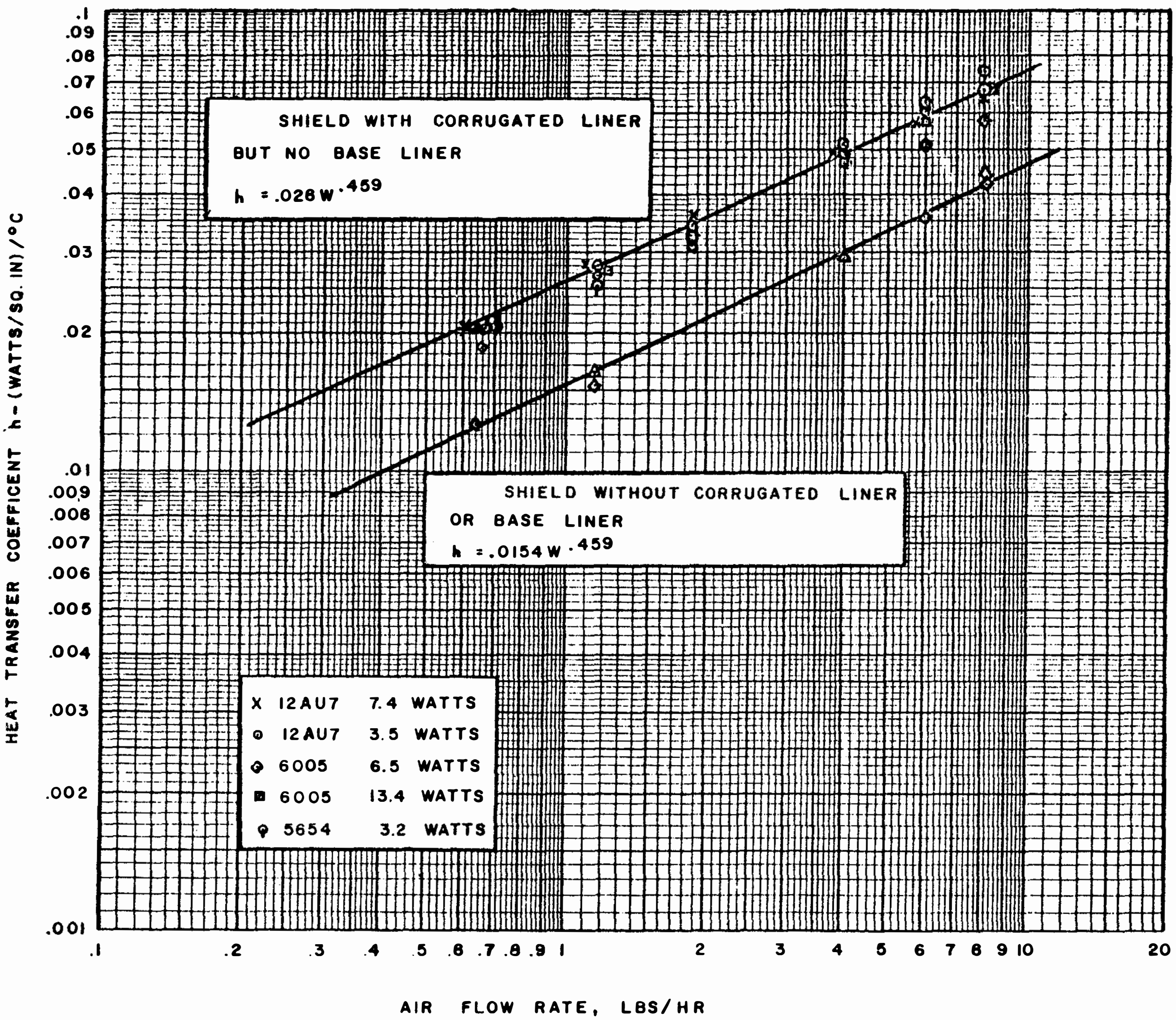


FIG. 6-25 HEAT TRANSFER COEFFICIENT VERSUS AIR FLOW RATE FOR SHIELDED TUBE WITH AND WITHOUT CORRUGATED LINER

Courtesy Collins Radio Corp.

The slopes of the two curves are identical, but the coefficient with a liner is considerably higher than when no liner is used. This means that, to attain the same bulb temperature, less air is required when a liner is used, compared to that when the liner is excluded.

Sample Problem VI

A 12AU7 dissipating 7.4 watts, is to be cooled, with an allowable bulb temperature rise of 45°C, in order to have a 100°C bulb temperature in a 55°C environment (ambient). Without forced-air cooling, the bulb temperature will be about 116°C.

First, solve for h

$$h = \frac{7.4 \text{ watts}}{3.81 \text{ in.}^2 \times 45^\circ\text{C}} = 0.0432 \text{ watts/in.}^2\text{-}^\circ\text{C}.$$

Now using Fig. 6-25, find the amount of air needed to obtain a heat transfer coefficient, h, of 0.0432 for each case.

with liner - 3.1 lb/hr.

without liner - 8.6 lb/hr. (almost 3 times as much)

This example illustrates the quick method for approximating air flow requirements to obtain a given bulb temperature for a shielded miniature tube cooled with parallel flow. This method of cooling transfers the heat produced in the tube directly to the cooling air with a minimum of heating of the chassis and neighboring components.

Large vacuum tubes also can be cooled effectively by parallel flow inside of shields. Experimental data on type 4-250A (5D22) transmitting tubes were correlated by Fig. 6-14, a plot of heat transfer coefficient, h, vs. Reynolds number, Re. The required heat transfer coefficient is computed by Equation (83):

$$h = \frac{q}{A_s \, \Delta t_s} \tag{83}$$

The corresponding Reynolds number is read from Fig. 6-14. Since:

$$Re = \frac{D_e m}{A_c \mu_f} \qquad (84)$$

Where D_e is taken as four times the hydraulic radius of the annular air flow space,

$$D_e = 4HR = \frac{2\,A_c}{\pi(r_1 + r_2)} \qquad (85)$$

The required weight rate of flow, m, can be computed, using the nomograph of Fig. 6-4.

c. External Anode Tubes

Special high-frequency and high-powered transmitting tubes with finned external anode "coolers" intended for forced-air cooling require special design considerations. External anodes are in general thermally desirable. Such anodes eliminate the largest resistance to heat transfer which normally exists in a tube, that is, the thermal resistance offered by the vacuum separating the anode from the glass bulb. External anodes can in some instances be cooled by metallic conduction or liquid cooling, thus permitting other effective modes of heat transfer. Temperatures of external anodes will be considerably lower than those of the anodes in conventional tubes, thus the internal control elements can be cooler or can operate at higher unit heat dissipations.

External anode tubes are generally more sensitive than conventional tubes to temperature gradients on their outer surfaces. This limitation is primarily due to the small dimensions of the glass or ceramic seals. The thermal, electrical, and mechanical stresses in the seals can be severe unless uniform cooling techniques are used. Tubes such as the 4X150A and 4X250A must be forced-air cooled even when only the heaters are energized. In general, the base seals are the most sensitive to temperature, with the plate-screen or plate-control grid seals running a close second. Seal temperatures on tubes such as the 4X150A are limited to 150°C maximum. However, it has been found during a companion phase of this contract that longer and more reliable service can be obtained when the seals are maintained in the vicinity of 100°C, with 125°C being the maximum reliable seal temperature. The limitation of the anode temperature is frequently the melting point of the solder used to attach the fins and cooler to the anode. Most of these solders melt in the vicinity of 220°C. Recently these temperature limitations have been eliminated by brazing or silver soldering the metal parts of the anode.

The tube manufacturers' ratings for air flow over external anode tubes have been found to be somewhat marginal, especially at the seals. It is generally recommended that the nominal air flow ratings be exceeded by 20 per cent for reliable operation. Further, some manufacturers rate tube air flow in CFM. This preferably should be a weight rating of flow rather than a volumetric rating. The pressure drop data for air flow over or through external anodes have been found to be reasonably accurate, the largest variations being at the special sockets for cooling these tubes by forced air.

Parallel flow should be used with tubes such as the 4X150A, whereas crossflow should be used with tubes such as the 2C39. With parallel flow it is recommended that the air first be directed to the base and the seals of the tube, thence flowing over the anode. Where possible, "chimneys" should be used at plate-screen seals to maximize turbulence at the seal surface. This flow direction will produce the lowest seal temperatures which can be obtained in a given situation. If the air is first passed over the anode, the seals are in effect cooled with "second hand" heated air. The air temperature rise through the anode will be high since most of the heat produced is dissipated at the anode. The seals should not be cooled with discharge temperature air.

d. Resistors

(1) Stacked Resistor Assemblies

Automatically produced assemblies of resistors as shown in Fig. 6-26 usually need more than natural convection to provide adequate cooling. The printed wiring or plastic decks upon which the resistors are mounted form barriers against natural convection, and conduction through the plastic is negligible.

Forced-air cooling is the most economical means of attaining a reliable operating temperature for composition carbon and similar resistors under average operating conditions. For severe applications, conduction through metal offers the lowest thermal resistance. (See CAL Report No. HF-1053-D-7.)

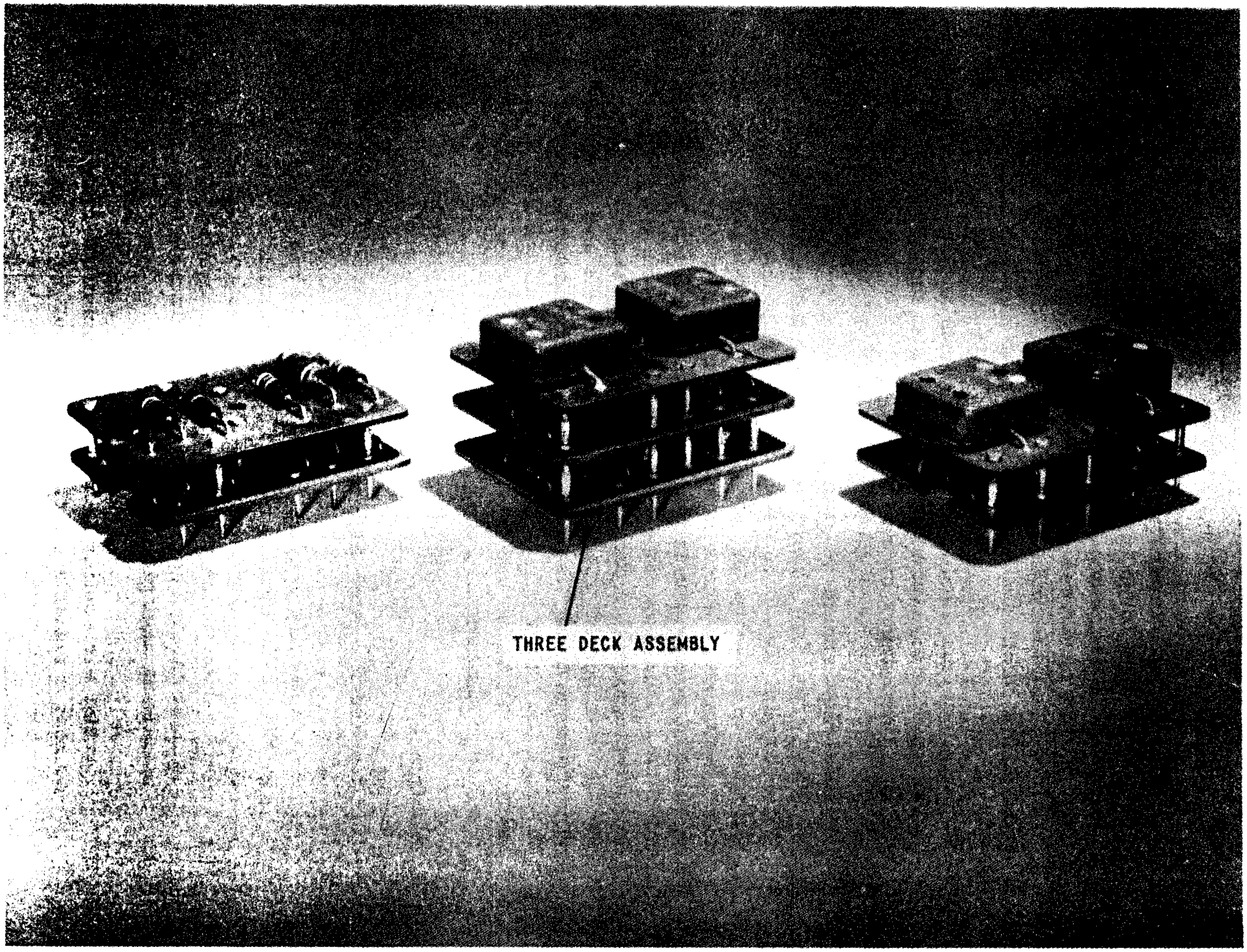


FIG. 6-26 TYPICAL STACKED RESISTOR ASSEMBLIES

Design curves based upon data obtained at Cornell Aeronautical Laboratory are presented in Figs. 6-27 and 6-28. These data are based upon the assemblies being located in a close-fitting duct, but isolated from the duct to prevent conduction. The proper air flow, with the assembly in a given attitude and power level, may be selected from these curves. The curves are based upon the hottest resistor body temperature of the assembly. Attitude D has the axes of the resistors parallel to the air flow and attitude E has the axes of the resistors perpendicular to the air flow. Flow rates above 0.15 lb/min. show no appreciable decrease in resistor body temperature. Air flow rates of the order of .075 lb/min. are adequate for cooling the assemblies.

(2) Banks of Resistors

In the design of electronic units, it is often necessary or convenient to locate several resistors in close proximity to each other. When it is necessary to group heat generating elements such as resistors, the spacing between them should be as great as possible. If resistors are to be mounted on a vertical deck or chassis, as is commonly done to conserve space, the resistors' axes should be vertical. If the resistors are not of convenient length, then they may be mounted with their axes horizontal. However, with the axes horizontal, the resistor bodies should be staggered so that they are not directly above one another.

Typical data and comparison curves are presented herein which are based upon the following resistor arrangements and spacings:

elements in horizontal position; $S/D = 2.0$

elements in vertical position; $S/D = 2.0$

elements in vertical position; $S/D = 1.5$

where:

S is the distance between resistors

D is the resistor diameter.

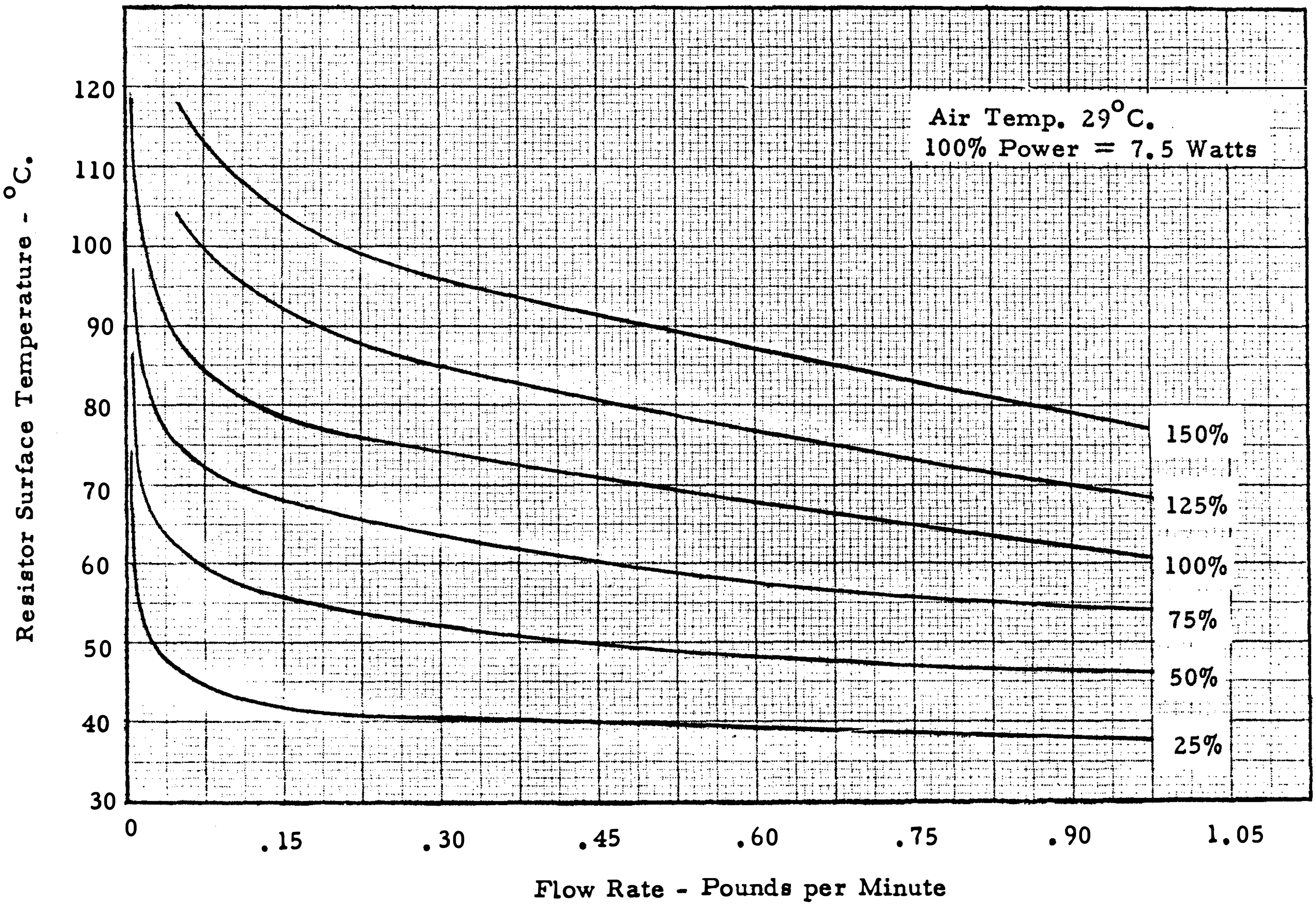


Fig. 6-27 Resistor Surface Temperature vs. Air Flow Rate For the Hottest Resistor in Attitude D.

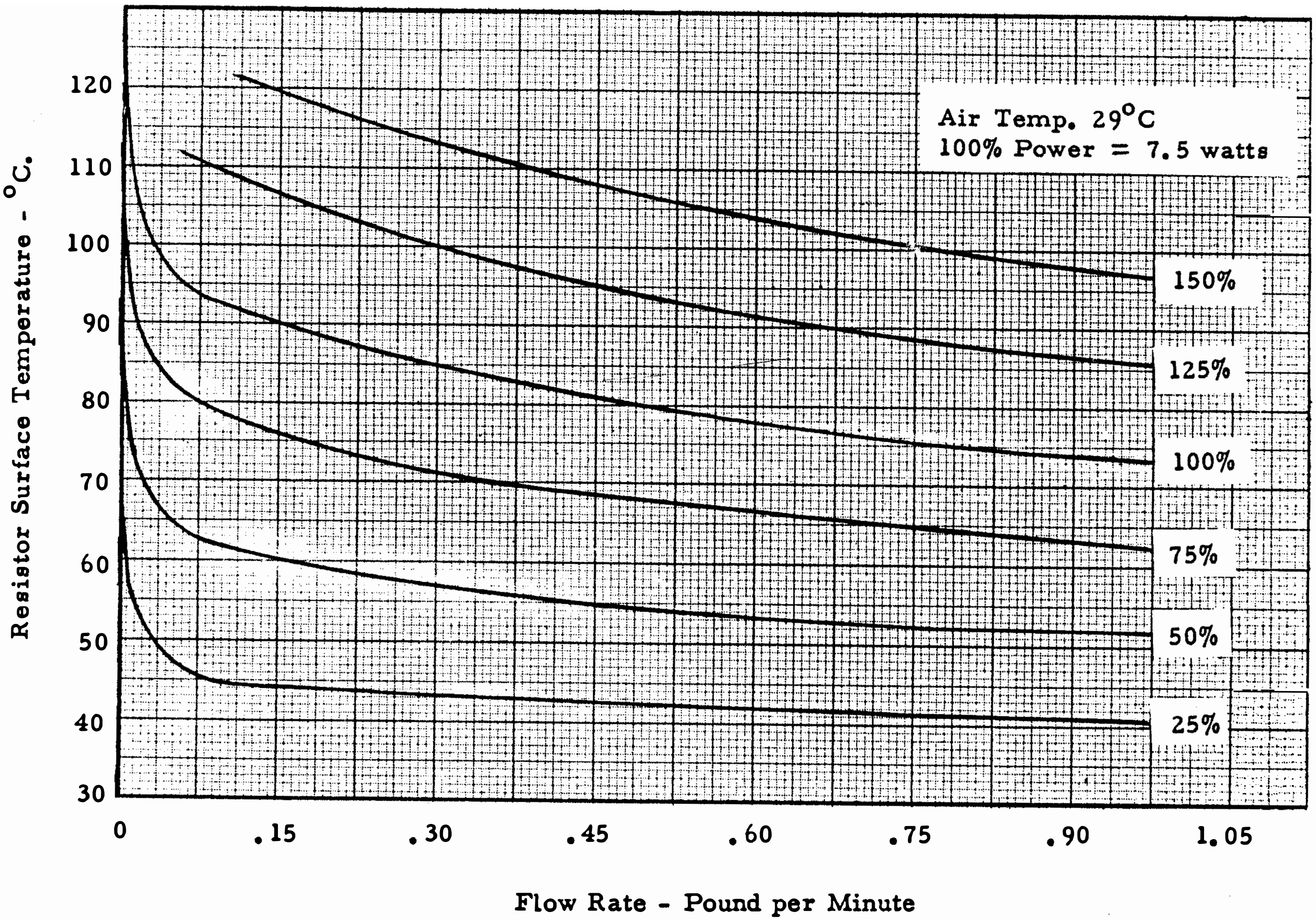


Fig. 6-28 Resistor Surface Temperature vs. Air Flow Rate For the Hottest Resistor in Attitude E.

Generally, data from such configurations are correlated by means of Equation (86):

$$\frac{hD}{k_f} = \phi\left(\frac{DV\rho}{\mu_f}\right) \tag{86}$$

This equation was found to be valid within experimental limits between Reynolds Numbers of 0.2 to 250,000.

Figure 6-29 shows the data for single cylinders. The experimental points lie generally above the predicted curve, indicating better heat transfer in the bank than would be expected from single cylinders. This improvement in heat transfer may be due to improved turbulence obtained in a bank of staggered resistors.

Other configurations of resistors can be treated in the same manner as electron tubes (cylinders) as outlined in Section C of this Chapter.

e. Semiconductors

(1) General

Semiconductor devices are extremely temperature sensitive. The heat produced in a semiconductor device is generated at the junction, and the temperature rise of the junction governs the successful application of the part. The thermal limitations of semiconductor rectifiers are described in detail in NAVSHIPS 900,192 (CAL Report No. HF-845-D-8) a companion manual together with internal thermal resistance data. Internal temperature gradients are not considered in this manual.

(2) Rectifiers

Typical temperature rises for various cooling conditions with finned rectifiers, including the rise between fin and ambient air, are presented in Fig. 6-30. Radiation from the fins, when the rectifier is not cooled by forced air causes the curves for free convection conditions to curve slightly from the linear form.

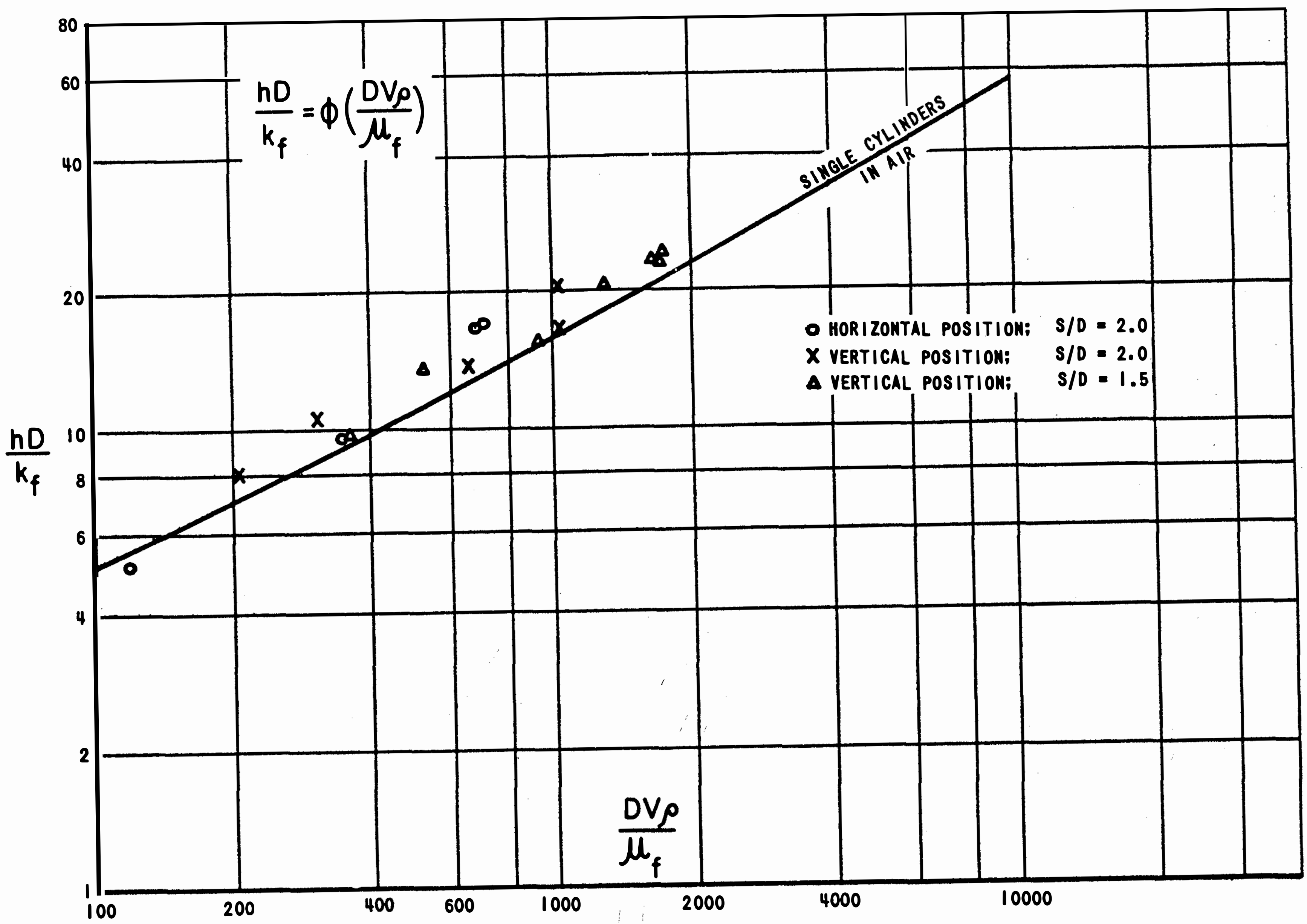


FIG. 6-29 CORRELATION OF HEAT TRANSFER BY FORCED CONVECTION FROM A BANK OF RESISTORS

For commonly encountered applications, the device manufacturers usually publish ratings determined in the foregoing manner for various ambient temperatures and velocities of cooling air. A set of such curves for a typical device is shown in Fig. 6-31. Operation at any point on one of these curves develops the same junction temperature.

(3) Transistors

Transistors are of course even more temperature sensitive than rectifiers or diodes. Methods of thermally rating transistors are given in NAVSHIPS 900,192 (CAL Report No. HF-845-D-8) and are not presented herein. A common method for the forced-air cooling of small heat sources, such as power transistors having high heat concentrations and high unit heat dissipations, is to mount the device at the center of a plate, or fin, which serves to increase the surface area and reduce the unit heat dissipation.

(a) Parallel Flow

When air is forced across such a plate parallel to the plate's surface, a local Reynolds number of the order of 300,000 may be required for fully developed turbulent flow. Such high values will generally not be attained in electronic equipment, so parallel-flow cooling of these small plates must be in the laminar region. For a plate of length L, in the direction of the air flow, and width W, with free stream velocity V_o, Jakob gives the following equation:

$$q = 0.664\, k_f\, (Re)^{1/2} (Pr)^{1/3}\, W \Delta t_s \qquad (87)(D.E.)$$

Using

$$Pr = 0.71 \text{ for air}$$

then

$$q = 0.593\, k_f\, (Re)^{1/2}\, W \Delta t_s \qquad (88)(D.E.)$$

where:

$$Re = \frac{\rho V_o L}{\mu_f}$$

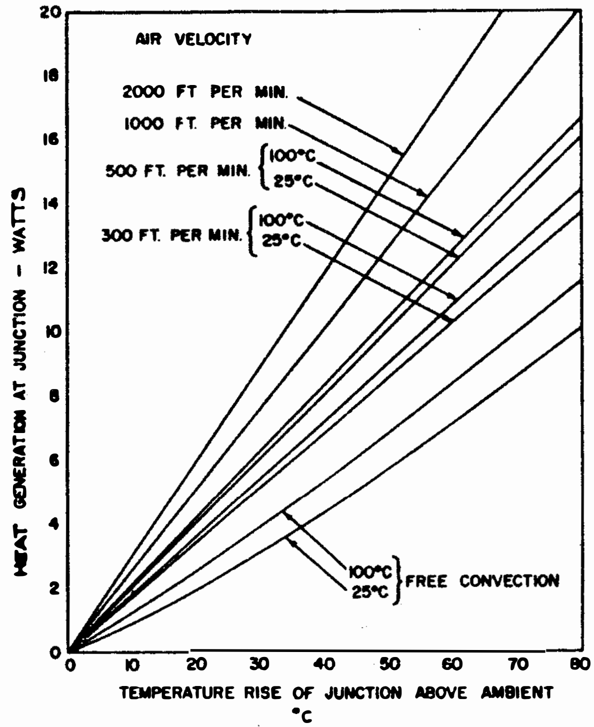


FIG. 6-30 HEAT DISSIPATION CHARACTERISTICS OF A SEMICONDUCTOR RECTIFIER

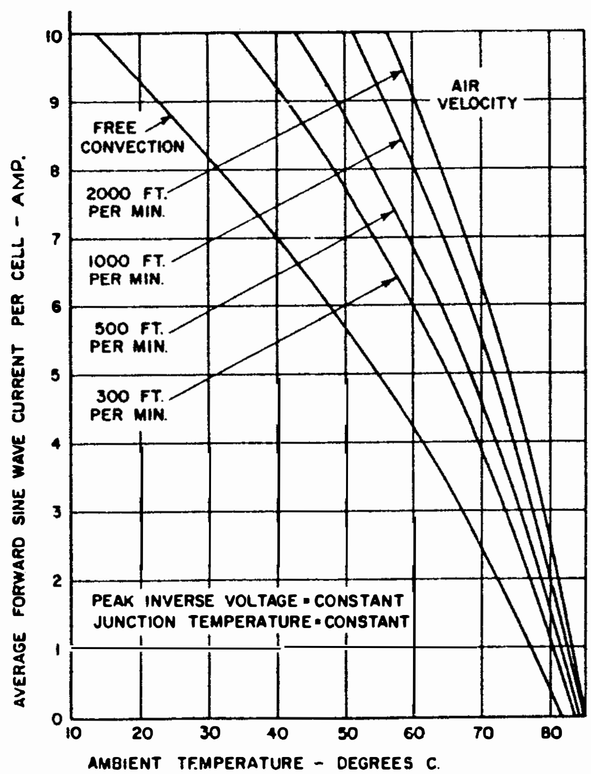


FIG. 6-31 CONSTANT JUNCTION TEMPERATURE RATINGS OF A GERMANIUM RECTIFIER CELL

V_o is the free stream velocity

L is the length of the plate in the direction of flow

W is the width of the plate

The Reynolds number can be computed using the free stream velocity V_o and the plate length L as the characteristic length. Equation (87) is applicable to small plates at uniform temperature for laminar flow at small Reynolds numbers.

A plate with a heat source mounted at its center will not be at a uniform temperature. The center will be hotter than the edges. In laminar flow, cooling is most effective at the leading edge where the boundary is thinnest. The temperature at the leading edge will be lowered, causing heat to flow more rapidly in that direction. With a low thermal resistance from the heat source to the plate, results should be somewhat better than predicted by Equation (87).

Sample Problem VII

A transistor dissipating 10 watts is mounted on a copper plate. The maximum allowable surface temperature is 80^oC and the cooling air is at 40^oC. Find a suitable size of plate which will satisfy these conditions at a reasonable flow velocity.

From Equation (87), with $t_f = 60^oC$, $\Delta t_s = 40^oC$, and $q = 10$ watts, the required Reynolds number is given by:

$$Re = \frac{330,000}{W^2} \qquad (89)$$

where:

W is the width of plate, inches

From Equation (88), the velocity may be computed for assumed values of length.

$$Re = 7.10\ V_oL \qquad (90)$$

where:

V_o is the free stream velocity, ft/min.

L is the length, inches

Table X gives computed values of Reynolds Number, Re, free stream velocity, V_o, and the heat transfer coefficient, h, for different assumed plate sizes.

TABLE X

COMPUTED VALUES OF Re, V_o AND h FOR SAMPLE PROBLEM VII

Item	Plate Size inches	Re	V_o $\frac{ft.}{min.}$	V_o $\frac{mi.}{hr.}$	h $\frac{watt}{in.^2 - {}^oC}$
1	2 by 2	82,400	5790	66.	0.0625
2	4 by 4	20,600	725	8.2	0.0156
3	6 by 6	9,170	215	2.4	0.00696
4	6 by 6	16,600	480	5.45	0.01038
5	6 by 6	16,600	480	5.45	0.01040

The heat transfer coefficient is calculated by:

$$h = \frac{q}{LW \Delta t_s} \tag{91}$$

Items 1, 2 and 3 of Table X give computed values of Re, V_o and h for three different plate sizes. A 6 by 6 inch plate gives a reasonably low Reynolds number and good heat transfer. For a 2 by 2 inch plate the air velocity is uneconomically large.

Items 4 and 5 of Table X give, respectively, computed and measured values for a plate operating at a film temperature of 100°C. The agreement between computed and measured heat transfer coefficients is very good.

Convective heat transfer can, of course, be increased by increasing turbulence, but for these small plates the turbulence must be fine-grained with a very short mixing length. An approximation of the degree of turbulence can generally be determined by the injection of smoke into the flow path. If the smoke dissipates rapidly, the flow is highly turbulent. However, if the smoke tends to linger, or if it can be traced for more than a few inches, the flow is not sufficiently turbulent. Generally, such turbulence cannot be attained by the use of the turbulators discussed in Section C-9 of this Chapter. It appears that the proper turbulence could be achieved by an array of small nozzles directing air obliquely onto the plate, but references to such a method have not been found in the literature. Because of the pressure required, the plates would have to be arranged in parallel rather than in cascade, with an air manifold to feed the nozzles.

Because of space limitations, cooling plates for transistors will normally be mounted in ducts. The boundary layer of air on the top surface of the plate is less than a fraction of an inch thick, so that for any reasonable duct height Equation (87) is applicable. The air flow requirement may be computed by:

$$m = \rho V_o A_c \tag{92}$$

where A_c is the duct cross-sectional area. The properties of air should be taken at the film temperature, $t_f = (t_s + t_b)/2$, as in most other calculations in this manual.

(b) Crossflow

Turbulence and heat transfer would be expected to increase in air impinged perpendicularly to the cooling plate surface, but no data have been found for this mode of cooling. The practical objections to this mode are the ducting requirements and the difficulty in arranging circuitry. It appears that parallel flow is the only practical way of cooling components of this type by forced air.

(c) Heat Exchanger Chassis

Special heat exchanger chassis (see Fig. 6-32) have been developed for cooling power transistors. These chassis have many small fins for high thermal efficiency and actually are miniature versions of forced-air cooled cold plates. (See Section D of this Chapter.)

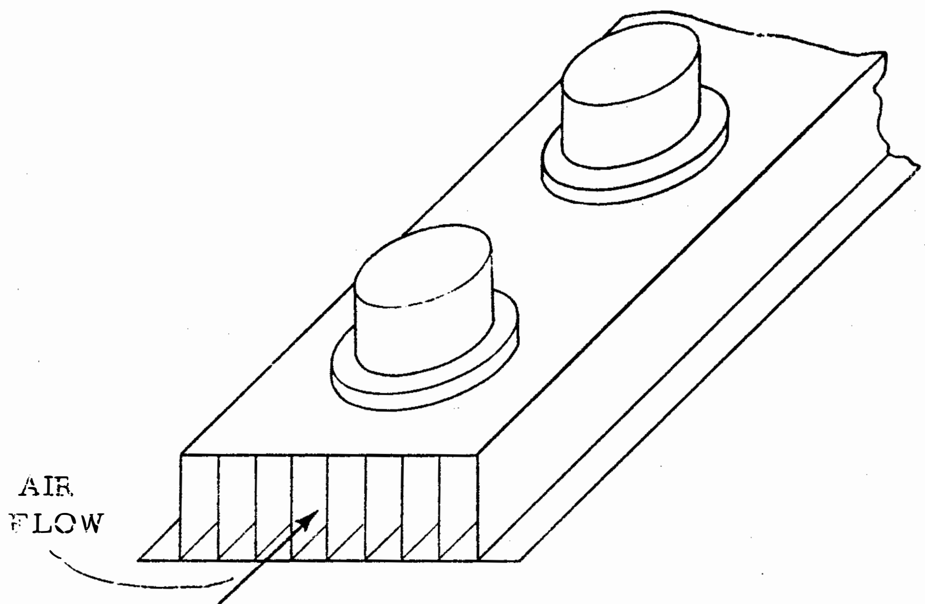


Figure 6-32 - Transistors Mounted on a Heat Exchanger Chassis.

C. Application of Classical Theory

The design procedures for cooling configurations not previously covered can be formulated by using modified classical methods. As explained in Chapter III, dimensional analysis of forced-air cooling yields the equation:

$$Nu = C\,Re^{n} \tag{93}$$

where C and n are undetermined constants.

This leads to Equation (94) for the air flow required in forced-convection cooling:

$$m = q^{\frac{1}{n}} \; \frac{1}{C}^{\frac{1}{n}} \left[\left(\frac{D_e}{A_s} \right)^{\frac{1}{n}-1} \; \frac{A_c}{A_s} \right] \left[\frac{\mu_f}{k_f^{\frac{1}{n}}} \cdot \frac{1}{(t_s - t_b)^{\frac{1}{n}}} \right] \tag{94}$$

where:

m is the weight rate of flow.

D_e is a characteristic linear dimension.

A_s is the area of hot surface at temperature t_s.

A_c is the net cross-sectional area.

The viscosity and thermal conductivity of air are taken at the film temperature $t_f = (t_s + t_b)/2$.

Equation (94) is generally applicable. In practical cases, the average surface temperature, the average incoming air temperature and the rate of heat transfer vary from point to point, and, strictly, this equation should be written in differential form and integrated over the entire configuration. This is obviously impractical. However, a configuration can be analyzed piece-by-piece (i.e. in stages) by successive applications of Equation (94).

1. Single Cylinder in a Duct in Crossflow

Most heat sources in electronic gear approximate cylinders (vacuum tubes, resistors) or prisms (transformers, chokes). The forced-air flow must have a Reynolds number high enough to insure good turbulence but not so high that the blower power is excessive. The heat source is placed in a duct of such size that the net area A_c (gross duct area minus source silhouette area) is small enough to obtain this condition. Experimental work indicates that Reynolds numbers between 2000 and 10,000 are desirable. For this range it was found that, in Equation (94):

$$\frac{1}{n} = 1.62$$

and

$$\frac{1}{C}^{\,1.62} = 17.0$$

Robinson (Ref. 8) found it necessary to add a factor X_{st} to Equation (94) in order to correlate measurements on vacuum tubes. Equation (94) then becomes:

$$m = 17\, q^{1.62} \left[\left(\frac{D_e}{A_s}\right)^{0.62} \bullet \frac{A_c}{A_s} \right] \left[\frac{\mu_f}{k_f^{1.62}} \bullet \frac{1}{\Delta t_s^{1.62}} \right] \left[1 + \frac{1}{\sqrt{S_T}} \right]^{1.62} \qquad (95)(D.\ E.)$$

For circular tubes or bulbs, D_e is the average diameter, and S_T is the duct width divided by the tube diameter. A prism such as a transformer should be so placed that its smaller surface faces the air stream. D_e is the smaller dimension of this surface. A_c is found by subtracting the part silhouette area from the gross duct area.

In order to simplify the solution of Equation (95), it may be written as:

$$m = C\, q^{1.62}\, X_d\, X_t\, X_{st} \qquad (96)(D.\ E.)$$

where:

m is the weight rate of flow

C is a constant

q is the heat dissipated, watts

$$X_d = \left[\left(\frac{D_e}{A_s}\right)^{0.62} \bullet \left(\frac{A_c}{A_s}\right) \right]$$

$$X_t = \left[\frac{\mu_f}{(k_f)^{1.62}} \bullet \frac{1}{(\Delta t_s)^{1.62}} \right]$$

$$X_{st} = \left[1 + \frac{1}{\sqrt{S_T}} \right]^{1.62}$$

D_e - inches

A_c and A_s - square inches

X_d and X_{st} are functions of the geometry of the configuration and X_t is a function of the temperatures involved. Equation (96) is solved by the use of two CAL nomographs (Fig. 6-33 and Fig. 6-34). These nomographs, which are described below, incorporated most of the quantities in their construction, reducing the calculations required to those concerned only with simple geometrical and temperature quantities. Equation (95) may be solved without the use of the nomographs, providing a consistent set of units is used.

Equation (96) is further simplified to:

$$m = \hat{G} \cdot \hat{H} \qquad (97)$$

where:

$\hat{G}$ is found from the nomograph of Fig. 6-33

$\hat{H}$ is found from the nomograph of Fig. 6-34

The nomograph of Fig. 6-33 solves the equation:

$$\hat{G} = C \; X_d \; X_{st} \qquad (98)$$

The values necessary (the entering values) for solution are D_e/A_s, S_T and C.

In Equation (95), C is given as 17.0, a value quoted by heat transfer authorities for round tubes. Experimental work at this Laboratory indicates lower values for certain (electron) tubes as shown on scale (6) of Fig. 6-33. The use of C = 17.0 will yield conservatively large air flow rates, but may result in uneconomically large Reynolds numbers.

An index number appears at the head of each scale in Fig. 6-33 and the ensuing nomographs to show the order of use. Values in columns (1) and (2) yield (3), values in (3) and (4) yield (5), etc. The proper sequence is shown on each chart in the following form:

$$(1) \;\&\; (2) \longrightarrow (3)$$

The nomograph of Fig. 6-34 solves the equation:

$$\hat{H} = q^{1.62} \; X_t \qquad (99)$$

The values necessary (the entering values) are q and Δt_s.

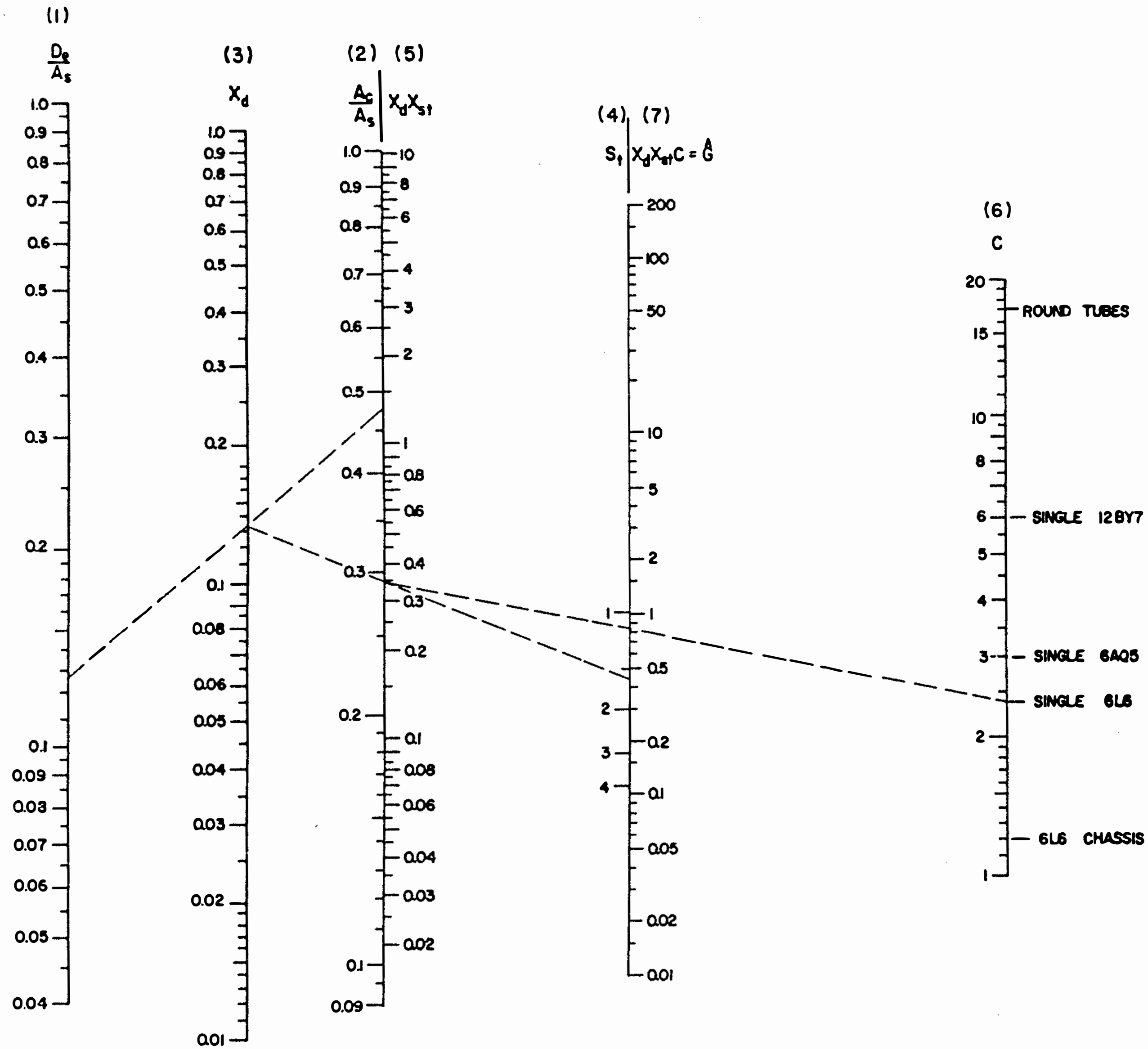


FIG. 6-33

DESIGN NOMOGRAPH NO. 1 FOR SINGLE TUBE IN CROSSFLOW

$$\frac{A}{G} = C X_d X_{st} \quad (98)$$

PROCEDURE:

(1) & (2) → (3)

(3) & (4) → (5)

(5) & (6) → (7)

THE BROKEN LINES REFERS TO THE SOLUTION FOR SAMPLE PROBLEM VIII ON PAGE 120

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DRAWN BY R.B. SCOTT

After the quantities $\hat{G}$ and $\hat{H}$ have been determined, the weight rate of flow, m, is found either by using scales (3), (4) and (5) of Fig. 6-34, or by separate multiplication of $\hat{G}$ and $\hat{H}$, if better resolution is desired.

Since scales become successively smaller in proceeding through a series of factors on a nomograph, better results are obtained by breaking the computation into several steps.

These charts, like any nomograph, may be worked backward or forward. Symbolically, if

$$(1) \;\; \& \;\; (2) \longrightarrow (3)$$

then

$$(3) \;\; \& \;\; (2) \longrightarrow (1)$$

The symbol "&" signifies laying a straightedge on the two scales indicated and the symbol $\longrightarrow$ signifies reading the result on the third scale.

These charts are prepared for t_b, average inlet air temperature, of $40^{\circ}C$ and sea level atmospheric pressure. However, the results are somewhat insensitive to t_b. A $15^{\circ}C$ change in t_b results in about 4% change in the final result. The viscosity and thermal conductivity of gases are independent of pressure, except for the low pressures approaching a vacuum (i. e., where the mean free paths of the molecules are long). These nomographs are thus quite generally applicable.

It is important to note that t_b is the average or "cup-mixing" temperature of the air at the tube location, and will therefore increase by steps when several heat sources are arranged in a row.

When m has been calculated it is necessary to make sure that the Reynolds Number, Re is of a suitable value (i. e., between 2000 and 10,000). If Re is too small, Equation (95) is not applicable because of low turbulence. If Re is too large the required blower power is excessive. The nomograph of Fig. 6-4 solves the equation:

$$Re = \frac{m\, D}{A_c \; \mu_f} \tag{100}$$

If Re is excessively large or small, the duct size should be changed accordingly. This, in turn, changes X_d and X_{st}.

Radiation can account for a substantial part of the total heat flow and therefore, calculations based on pure convection will yield conservatively large values of the weight rate of flow. In this Laboratory, tests of a single tube showed that as much as 40% of the heat was dissipated by radiation at a surface-temperature rise (t_s - t_b) of 100°C. For banks of tubes and assemblies of components, each heat source "sees" less absorbing surface than a single source and the radiation correction is less. A bank of 27-6L6 tubes showed a radiation correction of about 5%.

Sample Problem VIII

A single 6L6GBY is to be cooled in crossflow with forced air.

The following are known:

$$\text{duct height} = 4.0 \text{ inches}$$

$$\text{duct width} = 2.5 \text{ inches}$$

$$\left.\begin{array}{lcl} D_e & = & 1.562 \text{ inches} \\ A_s & = & 12.34 \text{ square inches} \\ q & = & 9.10 \text{ watts} \\ t_s & = & 63.5\ ^\circ C \end{array}\right\} \text{tube}$$

$$A_{c_{\text{duct}}} = 5.88 \text{ square inches}$$

$$t_b = 40.0^\circ C$$

Determine the weight rate of flow and the Reynolds number.

Solution

(1) Find the geometrical ratios

$$\frac{D_e}{A_s} = \frac{1.562}{12.34} = 0.127$$

$$\frac{A_c}{A_s} = \frac{5.88}{12.34} = 0.476$$

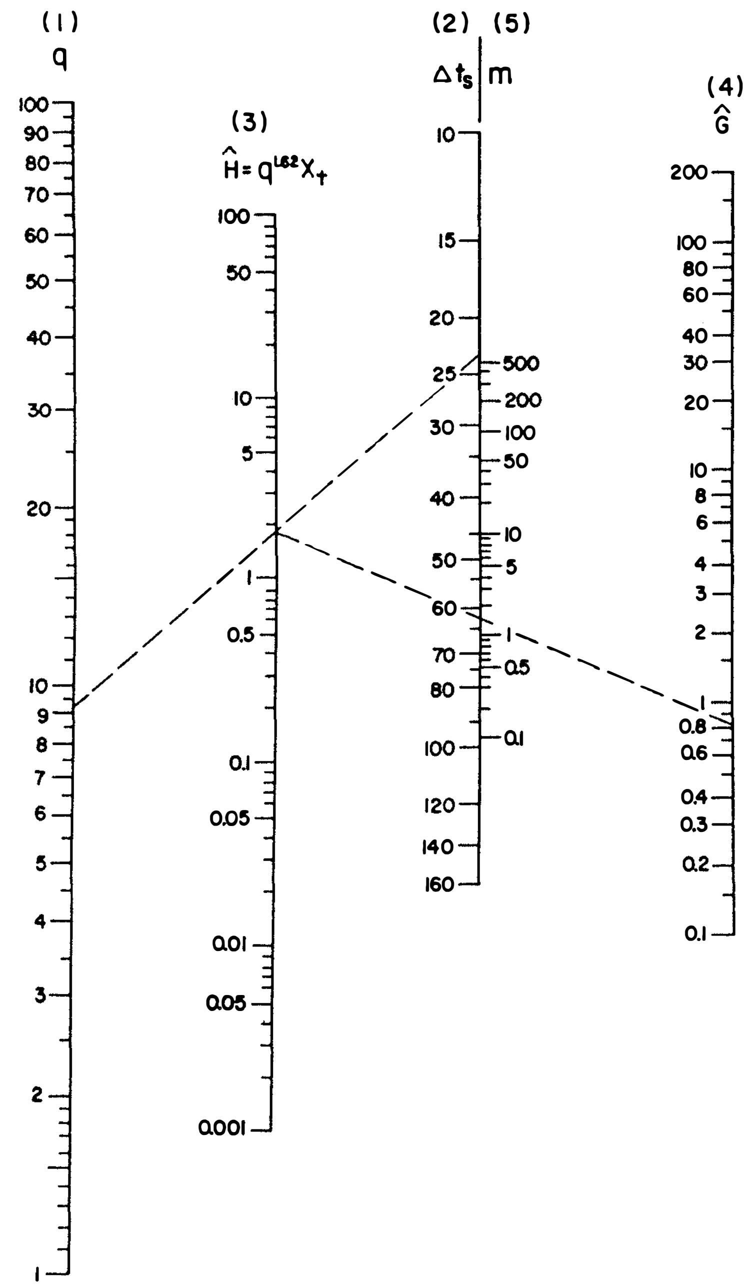

FIG. 6-34

DESIGN NOMOGRAPH NO. 2 FOR SINGLE TUBE IN CROSSFLOW

$$\hat{H} = q^{1.62} X_t \qquad (99)$$

WITH THE RESULT OF FIG. 6-33

$$m = \hat{G} \cdot \hat{H}$$

PROCEDURE:

(1) & (2) → (3)

(3) & (4) → (5)

THE BROKEN LINES REFER TO THE SOLUTION FOR $\hat{H}$ AND m IN SAMPLE PROBLEM VIII ON PAGE 120

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(2) From the nomograph of Fig. 6-33, read:

$$X_d = 0.132$$

(3) Find S_T

$$S_T = \frac{W}{D_e} = \frac{2.5}{1.562} = 1.60$$

(4) Again refer to Fig. 6-33, and read

$$X_d \ X_{st} = 0.34$$

(5) Apply C for the 6L6GBY, then

$$\hat{G} = 0.81$$

(6) Solve for $\triangle t_s$

$$\triangle t_s = t_s - t_b = 63.5 - 40.0 = 23.5^{\circ}C$$

(7) With q = 9.1 watts, enter the nomograph of Fig. 6-34 and read:

$$\hat{H} = 1.8$$

(8) then $m = \hat{G} \bullet \hat{H}$

$$= 0.81 \times 1.80$$

$$= 1.45 \ \frac{lbs}{min}$$

(9) Solve for t_f:

$$t_f = \frac{t_s + t_b}{2} = \frac{63.5 + 40.0}{2} = \frac{103.5}{2}$$

$$t_f = 51.75^{\circ}C$$

(10) From the nomograph of Fig. 6-4, reversing the order of the last step, read:

$$Re = 5700$$

(11) Actual measurements for this case from a laboratory experiment gave:

$$m = 1.41 \frac{\text{lbs}}{\text{min}}$$

and

$$Re = 5550.$$

<u>Note:</u>

For the small difference in temperature, i.e., $\Delta t_s = 23.5^oC$ at $t_s = 63.5^oC$, the amount of radiant heat transfer is negligible.

Sample Problem IX

A 6AQ5 dissipates 13.1 watts. An air flow of 0.5 lbs/min. at 40°C is available for cooling. Determine the duct size necessary to keep the envelope temperature, t_s, at 100°C.

From the tube dimensions, the following are available:

$$D_e = 0.71 \text{ inches}$$

$$A_s = 5.23 \text{ square inches}$$

tube silhouette area = 1.5 square inches

(1) From the nomograph of Fig. 6-34, with

$$q = 13.1 \text{ watts, and}$$

$$\Delta t_s = t_s - t_b = 100 - 40 = 60^oC$$

read:

$$\hat{H} = 0.67$$

(2) From the equation

$$m = \hat{G} \cdot \hat{H}$$

or

$$\hat{G} = \frac{m}{\hat{H}} = \frac{0.5}{0.67}$$

then

$$\hat{G} = 0.75$$

(3) Working backwards in Fig. 6-33, with C for the 6AQ5, read:

$$X_d \; X_{st} = 0.25$$

(4) Now assume an $S_T = 1.5$

(5) From Fig. 6-33, then

$$X_d = 0.094$$

(6) Assume an Re = 4000

with

$t_f = 70^\circ C$, find from Fig. 6-4 that

$A_c = 1.25$ square inches

(7) then

$$\frac{D_e}{A_s} = 0.136 \qquad \frac{A_c}{A_s} = 0.24$$

(8) From Fig. 6-33, knowing X_d from step (7) and $X_d \; X_{st}$ from step (3) read:

$$S_T < 1.0$$

This is an impossible result.

(9) Examination of Fig. 6-33 shows that for an $X_d \, X_{st} = 0.25$ then

$$0.08 < X_d < 0.130$$

(10) Now assume an $S_T = 2.5$

(11) With this, $X_d \doteq 0.113$, within the limits specified by step (9).

(12) Since $X_d = 0.113$ and $\frac{D_e}{A_s} = 0.136$ from Fig. 6-33 read:

$$\frac{A_c}{A_s} = 0.39$$

$$A_c = 0.39 \cdot A_s = 0.39 \times 5.23$$

$$A_c = 2.02 \text{ square inches}$$

(13) Based on this, Fig. 6-4 gives

$$Re = 2400$$

(14) Then

$$S_T = 2.5 = \frac{W}{D_e}$$

or

$$W = 2.5 \, D_e = 2.5 \times 0.71$$

$$= 1.77 \text{ inches}$$

(15) Total duct area

$$A = A_c + (\text{silhouette area})$$

$$= (2.02 + 1.5) \text{ square inches}$$

$$= 3.52 \text{ square inches}$$

(16) Then the duct height:

$$L = \frac{3.52}{W}$$

$$= 2.0 \text{ inches}$$

(17) Since the 6AQ5 is 2.12 inches high, the duct height and A_c must be increased somewhat. Therefore, use a duct which is 2.2 x 1.77 inches. This gives $A_c = 2.4$ square inches and results in $Re = 2100$.

In a laboratory test, this tube, in a duct 2.55 by 1.75 inches, dissipated 13.1 watts at $t_s = 114.3^\circ C$, $t_b = 39.5^\circ C$ with $m = 0.49$ lb/min. at $Re = 1685$. The somewhat smaller duct and larger Reynolds number determined above account for the lower envelope temperature. At the temperature differential of $60^\circ C$ between envelope and duct wall, there is an appreciable radiation loss so that the calculated result is conservative.

2. A Longitudinal Row of Tubes in Crossflow

The amount of heat absorbed by moving air is given by Equation (101):

$$q = 7.62 \quad m \,\Delta t_b \qquad (101)$$

where:

q is the amount of heat, watts

m is the weight rate of flow of air, lbs/min.

Δt_b is the temperature change in air, ^{o}C

Consequently, the air temperature increases at the n'th heat source by the amount:

$$(\Delta t_b)_n = \frac{q_n}{7.62m} \qquad (102)(D.E.)$$

and after n heat sources in a longitudinal row

$$(t_b)_n = (t_b)_{IN} + \frac{\sum q}{7.62m} \qquad (103)(D.E.)$$

When several heat sources are arranged in a longitudinal row, the same calculations must be made for each source, t_b being increased at each step according to Equation (102). If the heat sources in a longitudinal row are identical, the last one (at air discharge point) will be the hottest, and calculations should be made for it. The calculations are necessarily a trial and error process. A value of m may be assumed, (Δt_b) calculated from the known $\sum q$, and m calculated for the last tube using its desired t_s. This is repeated until the two values of m agree.

If several different heat sources are arranged in a longitudinal row, they should be generally arranged in increasing order of heat production, the source of highest power dissipation being last. Temperature sensitive sources should be first. Care should be taken to arrange heat sources and ducting so that the cross-sectional area is about the same at different stations, because sudden changes in Reynolds number cause additional losses, thereby increasing the power required. It is sometimes advantageous to introduce baffles (turbulators) to increase turbulence near the tubes. Throttling plates may also be required to equalize pressure drops in parallel ducts to obtain the required air stream division. These devices are discussed in Section C-9 of this Chapter.

3. A Bank of Tubes in Crossflow

Equipments generally have heat sources arranged in banks of longitudinal and transverse rows. The cooling design problem for several identical heat sources in a transverse row may be solved by treating them as a single source, i.e., multiplying q, A_c and D_e by the number of sources in the transverse row. It is not advisable to place heat sources of different geometries in a bank in a single duct, as the resultant Reynolds numbers may be so different that one or the other is not efficiently cooled. Better practice is to provide parallel ducts, so that each can be designed for efficient operation. When parallel ducts are used it is necessary to design them for the same total pressure drop, or to provide dampers or throttling plates to regulate the air flow.

With heat sources having complex shapes or unusual temperature distribution, the preliminary prototype should be so designed that adjustments, particularly of duct lay-out, are easily made. The thermal and electrical design must proceed hand in hand. Each working model can then be tested thermally as well as electrically and these two aspects of design can be refined together. There is no adequate mathematical substitute for measured values of temperatures, flow rates and pressures. Heat flow problems are more analogous to electric field problems than to circuit problems. Like all field problems the chief difficulty is that of defining the boundary conditions in a complicated geometry. Only trivially simple configurations can be solved explicitly. Therefore, it is necessary to resort to empirical methods based on extensive experimental work. As an illustration, a rather complicated problem, which was originally presented by Robinson in Ref. 9, will be solved by a method outlined previously.

Sample Problem X

A power supply is to occupy a volume not exceeding 750 cubic inches. Dimensions, wattage and allowable temperatures of the heat-producing parts are given in Table XI.

TABLE XI

COMPONENT DESCRIPTIONS FOR SAMPLE PROBLEM X

Designation	Component	q watts	Allowable t_s, °C	Dimensions inches
V_3, V_9	6AS7-G	41.5	160	2-1/16 x 4-3/4
V_1, V_2, V_7, V_8	5R4GY	26.	160	2-1/16 x 4-3/4
V_4, V_{10}	12AX7	2.	160	7/8 x 1-15/16
V_5, V_6, V_{11}, V_{12}	5651	3.	160	3/4 x 1-7/8
R_2, R_{12}	500 ohm resistor	40.	300	9/16 x 2
R_1, R_{11}	75 ohm resistor	25.	300	1-9/16 x 2-1/4
T_1, T_4	plate trans.	26.5	200	2-1/4 x 1-5/8 x 1-7/8
T_3	filament trans.	15.	200	1-17/32 x 1-5/8 x 1-7/8
T_2	filament trans.	10.	200	1-17/32 x 1-5/8 x 1-7/8
L_1, L_2	choke	16.	200	2-1/8 x 2-11/16 x 2-5/8
C_1, C_2	capacitor	0.	125	1 x 1-3/4 x 3-5/8

Air at 85°C and 30 inches of mercury is available for cooling. Devise a thermal design and compute the required air flow.

Data for the thermal design are given in Table XII.

TABLE XII

COMPONENT THERMAL DESIGN DATA FOR SAMPLE PROBLEM X

Designations	V_3, V_9	V_1, V_2, V_7, V_8	T_1, L_1, T_4, L_2
q watts	41.5	26.	42.5
D_e in.	1-7/8	1-7/8	3.75
A_c sq. in.	4.25	4.25	7.9
A_s sq. in.	34.	34.	50.75

It seems reasonable to place V_3 and V_9, the tubes of highest power dissipation, in the first bank, followed by the four other tubes of the same size but lower power, V_1, V_2, V_7 and V_8. Assume a duct size 5.5 by 5 inches, giving duct area of 27.5 sq. in. The silhouette area of one tube is 9.5 sq. in. and A_c/A_s for one tube is 0.125. From Figures 6-33 and 6-34, it is found that an air flow of 3.0 lb/min. is required per tube. From Fig. 6-4 the Reynolds Number is 17,000. This is high but probably feasible. Air temperature rise calculated from Equation (99) is 1.8°C.

Figure 6-34 is now used to calculate Δt_s for the next tube, $\hat{G}$ being the same. The result is $\Delta t_s = 49°C$. The temperature of the second tube is 86.8 + 49 = 135.8°C which is well under the 160°C specified. The temperature of the third tube, with essentially the same operating conditions, will be 88.6 + 49 = 137.6°C.

The air temperature after the third tube in the row is calculated from Equation (102).

$$85 + \frac{93.3}{7.62 \times 3} = 89.1°C$$

Transformers T_1 and T_4 will be placed next in line after the large tubes. Figure 6-34 gives $\Delta t_s = 150°C$, which is too large. The duct size might be reduced to throttle air flow and increase turbulence but the ductwork would be awkward. It is better to place the transformers in a separate duct. The arrangement of the components in the first duct can be completed by installing the two capacitors in the entrance where they have 85°C air and will not throttle the air flow, and by placing V_4 and V_{10} between the large tubes.

The second duct will contain the transformers, chokes, resistors and the four small tubes. A duct size of 5 by 3 inches will accommodate T_1 and L_1 side by side. Construction will be easier if the duct is the same height as the other. It will be 5.5 by 3 in. with the transformers mounted on a plate dividing the two ducts. Net duct area $A_c = 7.86$ in.^{2} For a rough estimate, the geometric parameters of T_1 and L_1 will be added since they are similar devices. Use $D_e = 3.75$, $A_s = 50.75$. $D_e/A_s = 0.074$. $A_c/A_s = 0.155$. $S_T = 1.47$. From Fig. 6-33, $\hat{G} = 2$. From Fig. 6-34, allowing the maximum Δt_s of 115^o, m = 1.4 lb/min. From Fig. 6-4, the Reynolds Number is 9000. The rise in air temperature in the first bank is, from Equation (102), 4^oC. Since the second bank is identical to the first, it is advisable to increase the air flow to 1.5 lb/min.

The small low-power tubes and transformers T_2 and T_3 are placed next in the rows, and finally, the four resistors, since they have the highest allowable surface temperatures. The assembly will occupy a space of 10 by 8.5 by 7 inches and have a volume of 595 cubic inches, allowing a chassis 1 in. deep. A total of 7.5 pounds of air per minute is required. The dimensions and arrangements are substantially the same as determined by Robinson (Ref. 9) who calculated an air flow of 7.07 lb/min. and a total volume of 557. in.^{3}.

4. Single Miniature and Subminiature Tubes in Concentric Shields in Parallel Flow

In cooling miniature and subminiature tubes with air flowing parallel to the major axes of the tubes, the air velocity near the envelope surfaces should be high in order to develop turbulent flow. If the air stream enters at the base of the tube through an orifice of some sort, too high a velocity may be developed, which in turn requires excessive blower power. A point can be found at which the cooling efficiency is maximum.

If a tube is held in a tightly-fitting metal shield designed for conduction cooling, heat removal is best accomplished by conduction to a sink and little is gained by forcing air over the surface. If conduction to a heat sink is impossible because of circuit or construction requirements, air, forced through a properly designed shield, will significantly reduce the hot spot temperature. A poorly designed shield will, however, raise the hot spot temperature. Clips and tightly fitting shields are not recommended for forced-air cooling. Rather, the shield should clear the bulb by at least 1/32 inch to permit the parallel flow of cooling air. Actually, such a shield forms a chimney, or duct, to guide the air. Corrugated metal inserts, installed between the shield and the tube, will provide increased cooling, because the effective surface area of the tube is increased. However, the inserts should not restrict the flow of air and only those inserts having corrugations parallel to the direction of the air flow should be used.

The nomograph and equations developed for this method of cooling have been proven for subminiature tubes. There is nothing to indicate that the ranges cannot be extended to encompass miniature tubes. Hence, using Equation (104), curves to cover the miniature tube dimension range were formulated and included.

For efficient cooling it is necessary to develop a sheet of turbulent air hugging the tube envelope. This may be accomplished by forcing the air through orifices in the chassis, closely surrounding the tube base, and supplemented in some cases by a shield surrounding the tube, as shown in Fig. 6-35:

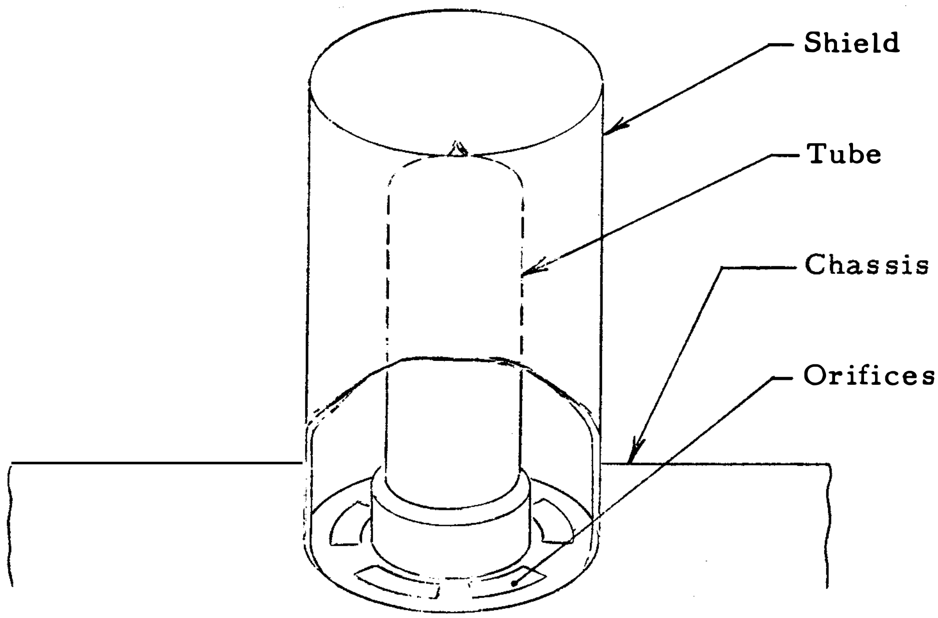


Figure 6-35 - Typical Shielded Configuration for Parallel Flow Cooling.

Air flow requirements for a single tube in concentric shield are given by the equation:

$$\frac{q}{\Delta t_s} = \frac{12.6 m^{0.57} L_t^{0.314}}{b^{0.28} (D_t + b)^{0.57}} \qquad (104)(D.E.)$$

where:

q is the amount of heat dissipated, watts.

Δt_s is the allowable surface temperature rise, $^{\circ}C$.

m is the weight rate of flow - $\frac{lbs}{min}$

L_t is the length of the tube, inches

b is the width of the annular space, inches

D_t is the diameter of the tube, inches

The dimensions are shown in Fig. 6-36:

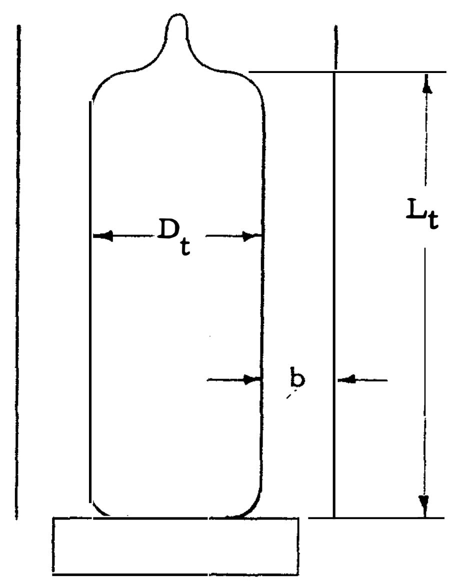


Figure 6-36 - Dimensions for Use in Equation (104)

The nomograph of Fig. 6-37, based on Equation (104) is to be used in the design of configurations of this nature. The following sample problem is designed to illustrate the use of Fig. 6-37.

Sample Problem XI

A tube 1.0 inch long and 0.37 inch in diameter dissipates 5.0 watts. The allowable temperature rise is 150°C. The available air flow is 0.008 lb/min. Find the annular width, b.

The following procedure is used:

(1) The value of m is known.

(2) A straight line from m through $\Delta t_s/q$ is projected to the left vertical scale of the plot.

(3) From this point move horizontally to the line corresponding to L_t.

(4) From this point move vertically to the line corresponding to D_t.

(5) From this point move horizontally to the required value of b on the right vertical scale.

Solution

(a) Locate $m = 0.008 \frac{\text{lbs}}{\text{min}}$ on the m-scale

(b) Solve for $\frac{\Delta t_s}{q}$

$$\frac{\Delta t_s}{q} = \frac{150}{5.0} = 30.0$$

(c) Draw line (1) (Refer to Fig. 6-37) from m = 0.008 through $\Delta t_s/q = 30$ to the left vertical scale of the plot.

(d) Draw line (2) horizontally to the curve, $L_t = 1.0$.

(e) Draw line (3) vertically to the curve, $D_t = 0.37$.

(f) Finally, draw line (4) horizontally to the b-scale, where the answer is read:

$$\underline{b = 0.031 \text{ inches}}$$

A more general type of problem is one where L_t, D_t, q, and Δt_s are given, and both m and b must be determined. The procedure outlined above must be revised in the following manner:

For step (1), assume a value of m; follow steps (2) through (5) as listed above; add a final step, step (6) which is to determine the Reynolds number in order to see if it is in the recommended range.

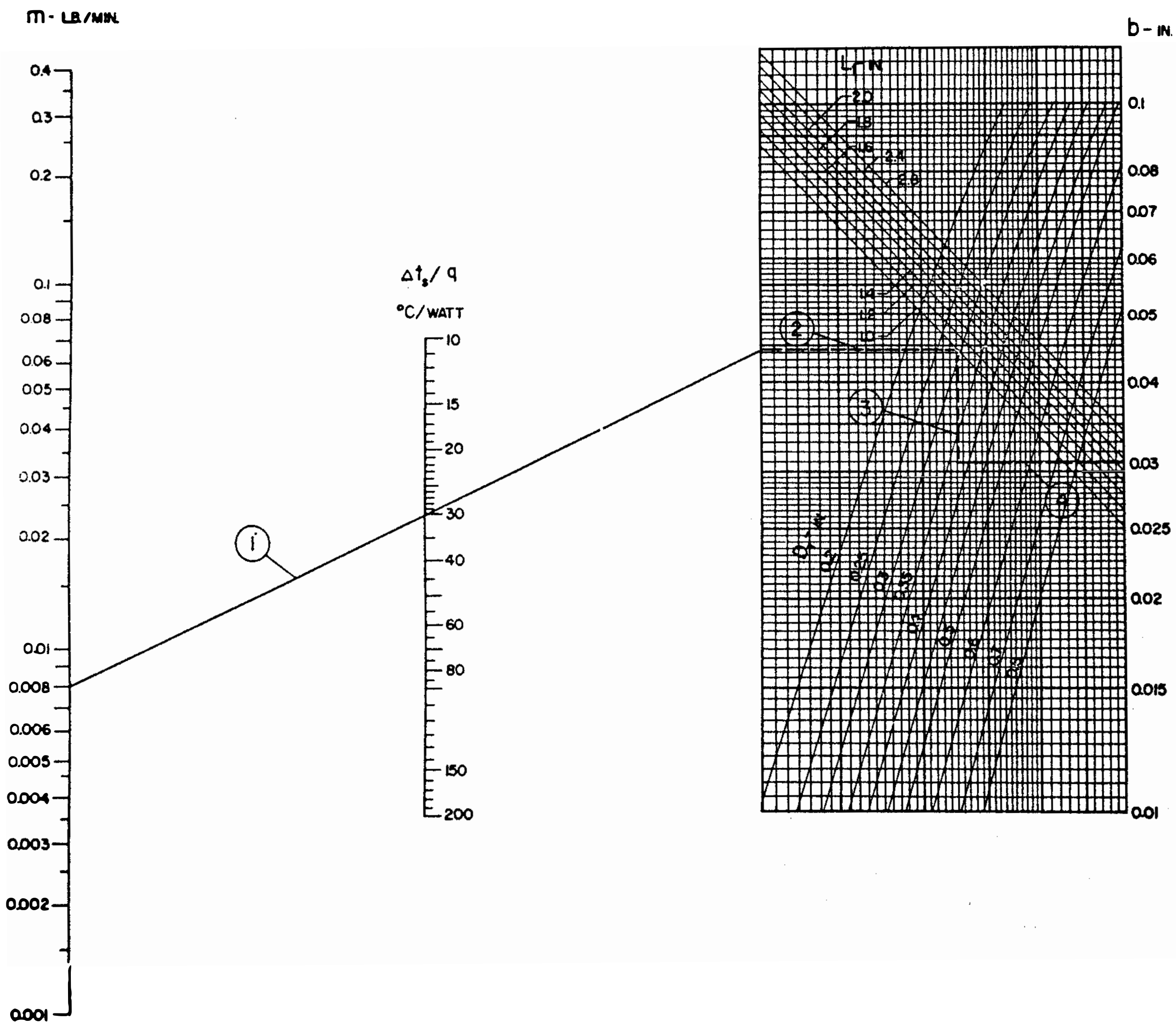


FIG. 6-37

DESIGN NOMOGRAPH FOR MINIATURE AND SUB-MINIATURE TUBES IN CONCENTRIC SHIELDS

FOR PROCEDURE, SEE PAGES 131-132

THE BROKEN LINES REFER TO THE SOLUTION FOR b IN SAMPLE PROBLEM III ON PAGE 131

BASED ON CHART FROM O.S.U. REPORT NO. 50

Generally, it is more convenient to determine several values of both m and b, then to solve for the Reynolds number, using the tabular form shown in the following example. The nomograph of Fig. 6-4 can be used. In this nomograph, A_c is the annular area between the tube and the shield, and D is the tube diameter.

The values of A_c and m obtained for subminiature tubes may be below the lowest scale values in Fig. 6-4, but since Equation (100) is linear, A_c and m can each be multiplied by 10 or 100 without changing the Reynolds number scale. The procedure outlined above is illustrated in the following sample problem.

Sample Problem XII

A 5643 thyratron dissipates 10 watts. The desired hot spot envelope temperature is 150°C. Cooling air at 40°C is available. Find the spacing, b, and the corresponding weight rate of flow, m.

Known are the following:

$$D_t = 0.40 \text{ inches}$$

$$L_t = 1.20 \text{ inches}$$

$$\Delta t_s = 110^\circ C$$

$$\frac{\Delta t_s}{q} = \frac{110}{10} = 11$$

$$t_f = 95^\circ C$$

(1) Assume a weight rate of flow

i.e. $m = 0.060 \frac{\text{lbs}}{\text{min}}$

(2) Locate m on the m-scale of the nomograph of Fig. 6-37.

(3) On the $\frac{\Delta t_s}{q}$ scale, locate $\frac{\Delta t_s}{q} = 11.0$

(4) Project the line between steps (2) and (3) to the left vertical scale of the plot.

(5) Proceed horizontally until $L_t = 1.20$ inches is reached.

(6) From here, proceed vertically until $D_t = 0.40$ inches is reached.

(7) From here, proceed horizontally to the b-scale and read b = 0.050 inches.

(8) Calculate A_c

$$A_c = \frac{\pi}{4} \left[(D_t + 2b)^2 - (D_t)^2 \right]$$

$$= \frac{\pi}{4} \left[(0.40 + 2(0.05))^2 - (0.40)^2 \right]$$

$$= 0.0628 \text{ square inches}$$

(9) From Fig. 6-4, using A_c, D_t, t_f and m read Re = 5000.

(10) Repeat steps (1-9) for assumed values of

$$m = 0.048,\ 0.036,\ 0.024\ \frac{\text{lbs}}{\text{min}}$$

TABLE XIII

SUMMARY OF CALCULATIONS FOR SAMPLE PROBLEM XII

m (lb/min.)	0.060	0.048	0.036	0.024
b (inch)	0.050	0.036	0.022	0.011
A_c (in.^{2})	0.0628	0.0452	0.0276	0.0139
A_c/D_e	0.157	0.113	0.069	0.035
Re	5,000	5,600	6,900	9,500

The solution of the problem results in four alternate answers, anyone of which will do what is required. The selection of one of these solutions is dependent on the pressure requirements of the system. If the pressure drop is to be minimized, the solution with m = 0.060 should be selected. However, if the amount of cooling air is limited, then select the minimum flow rate solution.

The final design should, in any case, give a value of Reynolds number between 2000 and 10,000, depending on the amount of throttling caused by the shape of the air entry ports. The arrangement of these ports does not affect cooling efficiency but does affect the overall pressure drop, and depends partly on the construction methods used. A full annulus can be provided by locating an orifice deck above the chassis. If orifices are to be punched in the chassis, round or segmental holes can be used. The selection of the entrance ports is dependent on both the desired pressure characteristics and the chassis or equipment under consideration.

Small holes exert a throttling effect and require lower values of Reynolds number in the annular passage between tube and shield. Thus, in this example, a shield 0.444 inch in diameter would be satisfactory if a full annular entrance is used, but a design using several small nozzles might require a shield diameter of 0.50 inch or more, with a correspondingly larger air flow.

"Since the maximum air velocity and turbulence occurs at the discharge from the orifice, inlet orifices should be oriented as closely as practicable to the component's heat-dissipating surface, in order to make the most effective use of the available air supply." (Ref. 14) When several tubes are cooled by this method, the orifices and shields must be designed to distribute the air among them properly. That is, the total pressure drop across each tube, when each receives the required amount of air, should be the same. It may be found, however, that the pressure necessary to force the required air flow through the shield is excessive. In such cases the shield may be shortened, flared, or omitted entirely. Figures 6-38, 6-39 and 6-40 illustrate several satisfactory arrangements of shields and orifices.

It is emphasized again that best results can be achieved by introducing a high degree of turbulence initially. This is accomplished by means of a nozzle or a concentric array of small holes at the base end of the tube. The air, already turbulent, then maintains a satisfactorily high Reynolds number over the tube surface, requiring about 50% as much air as does a laminar type of flow for the same degree of cooling. Obviously, it is necessary to design the entrance nozzle to minimize the required pressure drop. Nozzles of reasonably efficient design can easily be formed in metal plates by punch press tools.

Reference 21 reports that square rather than round shields are more effective in forced-air cooling. The optimum spacing between a shield and a hot cylinder was found to be 1/8 inch. It was also claimed that aluminum paint on the shield provides temperatures "20% lower than flat black". If the shield had a high emissivity on

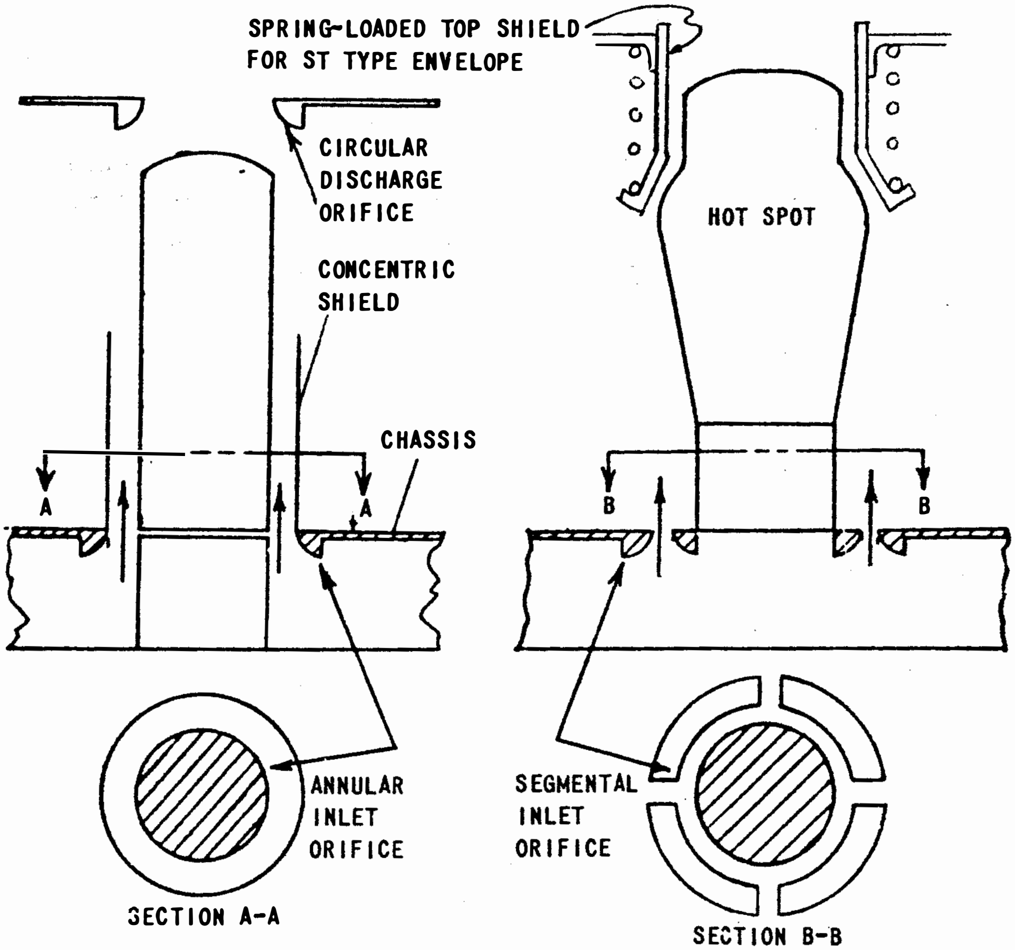


FIG. 6-38 ORIFICES AND SHIELDS

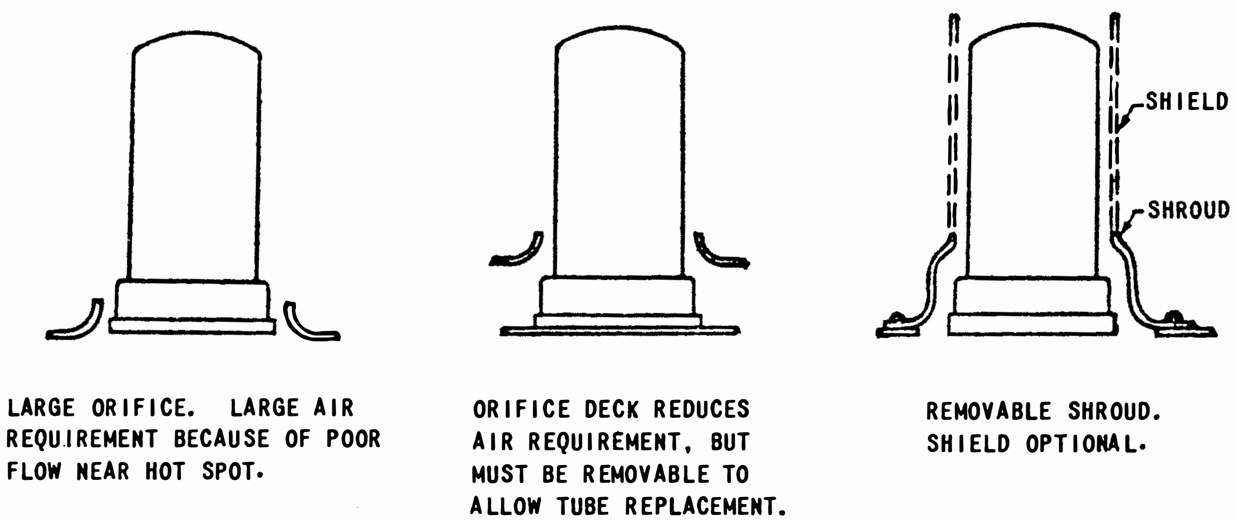
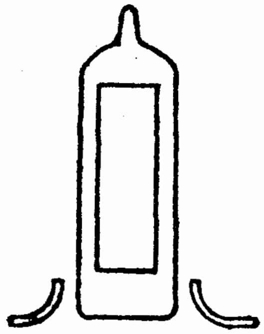
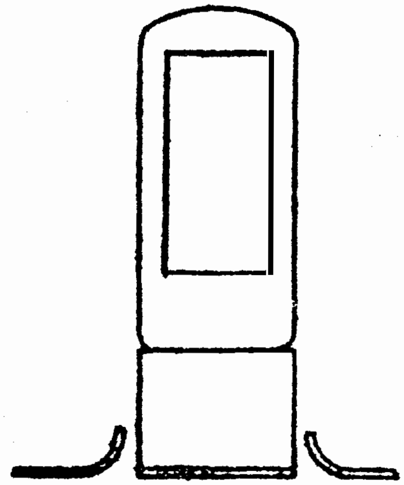


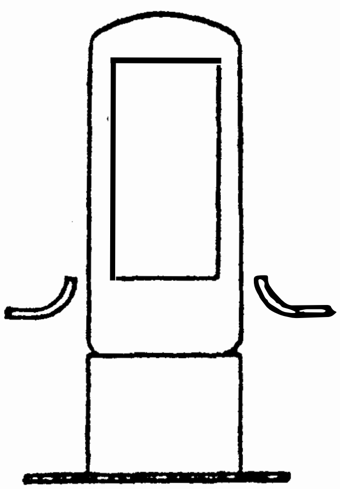
FIG. 6-39 SHIELDS AND ORIFICES FOR METAL ENVELOPE TUBES



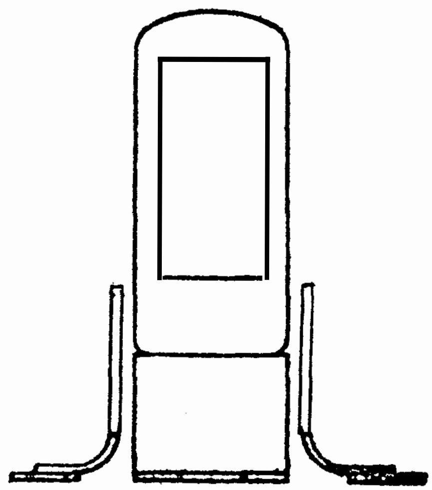
MINIATURE BASELESS TUBE WITH SIMPLE ORIFICE. SATISFACTORY AIR FLOW.



TUBE WITH BASE. AIR NOT WELL DIRECTED TO HOT SPOT.



ORIFICE DECK DIRECTS AIR TO HOT SPOT.



SHIELD DIRECTS AIR TO HOT SPOT.

FIG. 6-40 METHODS OF DIRECTING AIR FLOW

both surfaces, it would form an additional heat path, but at the expense of radiating some of the heat to the neighboring components. Square shields were not investigated at CAL.

5. Single Tube in a Rectangular Duct in Parallel Flow

For parallel turbulent flow in the range of Reynolds numbers between 4000 and 40,000, the numerical constants in Equation (94) are generally accepted as:

$$\frac{1}{n} \doteq 1.67$$

and

$$\frac{1}{C}^{1.67} \doteq 1.738$$

The geometrical factor X_{st} does not appear and Equation (94) becomes

$$m = 1.738 \left(\frac{D_e}{A_s}\right)^{0.67} \cdot \frac{A_c}{A_s} \cdot \frac{\mu_b}{(k_b \Delta t_s)^{1.67}} q^{1.67} \qquad (105)(\text{D. E.})$$

Equation (105) is solved by the nomograph of Fig. 6-41, the application of which is defined in terms of the symbols in parentheses at the top of each scale.

(1) & (2) $\longrightarrow$ (a)

(3) & (4) $\longrightarrow$ (b)

(a) & (b) $\longrightarrow$ (c)

(5) & (c) $\longrightarrow$ (6) the answer.

When m has been calculated it is necessary to make sure that Re is of a suitable value, i.e., between 2000 and 10,000 to assure sufficient turbulence and reasonable pressure drop. This is accomplished by the use of Fig. 6-4 as explained above in Section VI. B. 2. a.

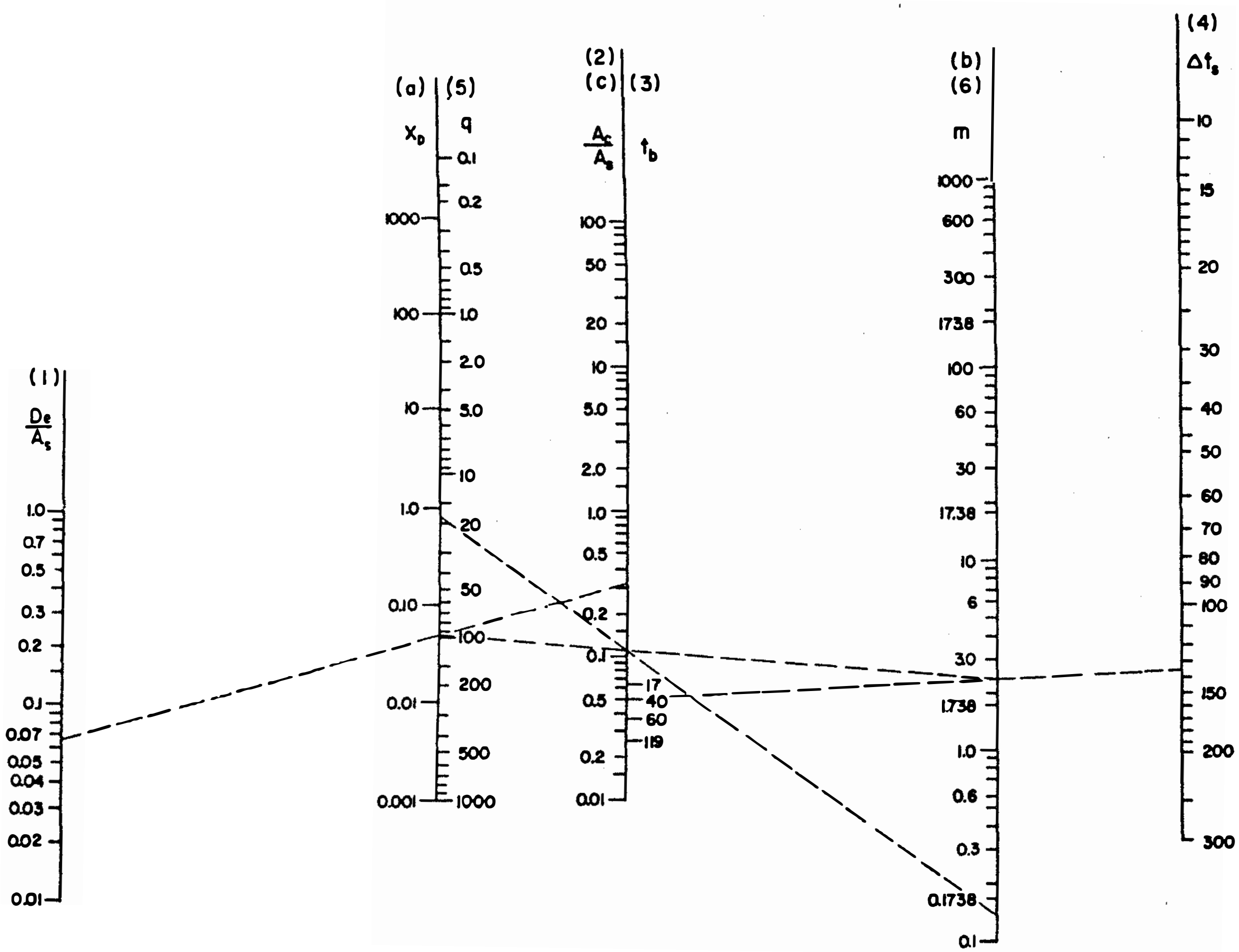


FIG. 6-41

DESIGN NOMOGRAPH FOR PARALLEL FLOW COOLING OF ELECTRON TUBES

$$m = 1.738\left[\frac{q}{k_b\,\Delta t_s\,A_s}\right]^{1.67} D_e^{0.67} A_c \mu_b$$

PROCEDURE:

(1) & (2) → (a)
(3) & (4) → (b)
(b) & (a) → (c)
(5) & (c) → (6)

THE BROKEN LINES REFER TO THE SOLUTION FOR m IN SAMPLE PROBLEM XIII ON PAGE 139

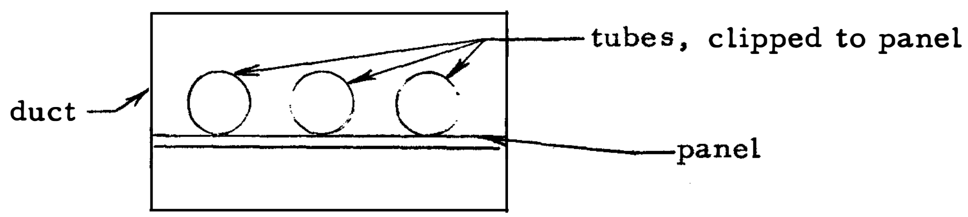
DESIGNED BY J. B. COLEMAN
DRAWN BY R. B. SCOTT

6. A Transverse Row of Tubes in Parallel Flow

The procedure is the same as for a single tube, except that A_s and the silhouette area of one tube must be multiplied by the number of tubes in the bank. The method of computation is illustrated by the following illustrative problem.

Sample Problem XIII

An electronic assembly is housed in an insulated duct. It consists of three 5902 subminiature type tubes mounted on a small thin plate, the tube axes being parallel to the plate as per sketch.

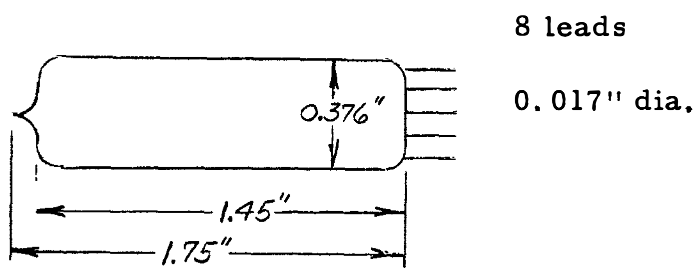


Air Flow into Duct
(i. e. into paper)

The following data are known:

duct dimensions - 1-15/16 inches wide by 1-5/32 inches high

tube dimensions -



entering air temperature, $t_b = 40^{\circ}C$.

desired tube envelope surface temperature, $t_s = 175^{\circ}C$ (to remain constant).

If each tube develops $q = 5.9$ watts of heat, what is the required weight rate of flow of air (m) necessary to keep $t_s = 175^{\circ}C$?

The following quantities are calculated -

A_c - [cross-sectional area of duct] minus [(cross-sectional area of tube) times three tubes].

A_s - surface area of tube envelope times three

D_e - diameter of tube envelope

Δt_s - temperature difference between the temperature of the incoming air and the temperature of the tube envelope.

Method of solution employing the nomograph of Fig. 6-41:

(1) The ratio $\frac{D_e}{A_s}$ is calculated and located on the $\frac{D_e}{A_s}$ scale.

(2) The ratio $\frac{A_c}{A_s}$ is calculated and located on the $\frac{A_c}{A_s}$ scale.

(3) A straight edge laid between the above two points, steps 1 and 2, gives reference point (a) on the q-scale.

(4) t_b is located on the t_b scale.

(5) Δt_s is located on the Δt_s scale.

(6) A straight edge laid between the two points, steps 4 & 5, gives reference point (b) on the m-scale.

(7) A straight edge laid between points (a) and (b) gives reference point (c), which is a pivot point.

(8) The value of q is then located on the q-scale.

(9) A straight edge laid from the value of q, through pivot point (reference point (c)) indicates an m on the m-scale. This m is read directly and is the quantity desired.

Solution

(1) Net cross-sectional area - A_c

A_c = (duct area) -3 (tube cross-sect. area)

$$= (1\text{-}15/16'' \times 1\text{-}5/32'') - \frac{3\pi}{4}(0.376'')^2$$

$$= 1.938 \times 1.156 - \frac{3\pi}{4}(0.142)$$

$$= 2.24 \text{ in.}^2 - 0.33 \text{ in.}^2$$

$$A_c = 1.81 \text{ in.}^2$$

(2) Surface area of tube - A_s (an approximation)

Considering the tube to be a cylinder of diameter = 0.376" and length = 1.45"

$$A_{s \text{ total 3 tubes}} = 3\left[\pi D_e L + \frac{\pi (D_e)^2}{4}(2)\right]$$

$$= 3\left[\pi (0.376)(1.45) + \frac{2\pi}{4}(0.376)^2\right]$$

$$= 3\left[1.71 + 0.223\right] = 3\ (1.933)$$

$$A_s = 5.80 \text{ in.}^2$$

(3) $\Delta t_s = t_s - t_b = 175°C - 40°C = 135°C$

(4) $\frac{D_e}{A_s} = \frac{0.376}{5.80} = 0.065 \frac{\text{in.}}{\text{in.}^2}$

Locate this on graph, scale (1)

(5) $\frac{A_c}{A_s} = \frac{1.81}{5.80} = 0.312$, scale (2)

(6) Now find reference point on scale (a).

(7) Locate t_b (scale (3)), Δt_s (scale (4)) and find reference point on scale (6).

$$t_b = 40^{\circ}C \qquad \Delta t_s = 135^{\circ}C$$

(8) Using reference points on scales (a) and (b), find pivot point on scale (c).

(9) Total q = 3(5.9) = 17.7 watts. Locate this on scale (5).

(10) Then, using scales (5) and (c), locate on scale (6);

$$m = 0.128 \frac{\text{lbs}}{\text{min}} \qquad \underline{\text{ANS.}}$$

(11) Same conditions as above problem except

$$q = 3 \times 7.9 = 23.7 \text{ watts}$$

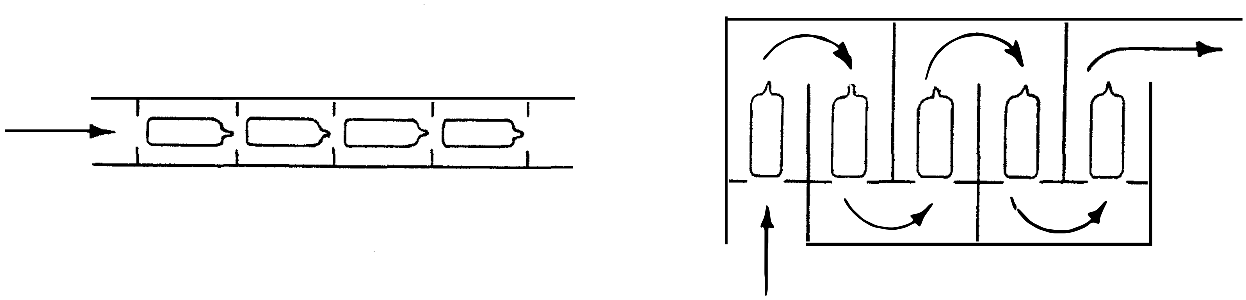
then

$$m = 0.22 \frac{\text{lbs}}{\text{min}}$$

7. A Longitudinal Row of Tubes in Parallel Flow

As in crossflow, the air temperature increases at each transverse row of tubes, because of the heat absorbed by the air at the previous row. This is computed as explained above in Section VI. B. 2. b.

It will generally be found more efficient to confine the air flow by means of a shield, as shown in Section VI. C. 4., or in a configuration as shown in Fig. 6-42.



(a) Straight Row (b) Broken Row

Figure 6-42 - Longitudinal Row of Tubes in Parallel Flow

This results in simplifying the calculations required.

8. Prismatic Component Parts

Heat transfer data for heat sources of prismatic shape are not found in the general literature, but extensive experimental work at Ohio State University (Ref. 12) indicates that an equation similar to Equation (93) correlates the data. This equation is:

$$Nu = 0.446 \ Re^{0.57} \left[\frac{6 \ A_c}{A_o + 5 \ A_c} \right]^{0.57} \tag{106}$$

This leads to an equation for the required weight rate of flow:

$$m = 0.637 \ q^{1.75} \left[\left(\frac{D_e}{A_s} \right)^{0.75} \cdot \left(\frac{A_c}{A_s} \right) \right] \left[\frac{\mu_f}{k_f^{1.75}} \cdot \frac{1}{(\Delta t_s)^{1.75}} \right] \left[1 + \frac{5 \ A_c}{A_o} \right] \tag{107)(D.E.}$$

Equation (107) can be reduced to the following form (Equation (108)) which is then solved by means of the nomograph of Fig. 6-43.

$$m = 0.637 \ q^{1.75} \ X_d \ X_t \ X_a \tag{108}$$

where:

m is the weight rate of flow, lbs/min.

q is the amount of heat dissipated, watts

$$X_d = \left[\left(\frac{D_e}{A_s} \right)^{0.75} \cdot \left(\frac{A_c}{A_s} \right) \right]$$

where:

D_e, A_s, and A_c are as explained previously.

$$X_t = \left[\frac{\mu_f}{k_f^{1.75}} \cdot \frac{1}{(\Delta t_s)^{1.75}} \right]$$

$$X_a = \left[1 + \frac{5\,A_c}{A_o}\right]$$

where:

A_c and A_o are as defined previously.

Note:
Equation (107) may be solved, using any consistent set of units, if it is desired not to use the nomograph.

The nomograph of Fig. 6-43 may be used to solve Equation (108) in the following manner:

(a) A line from D_e/A_s on scale (1) to A_c/A_s on scale (2) gives X_d on scale (3).

(b) A line from X_d to A_c/A_o on scale (4) gives $\hat{G}$ on scale (5).

$\hat{G}$ is a geometrical factor, where

$$\hat{G} = X_d \; X_a$$

(c) A line from q on scale (a) to $\triangle t_s$ on scale (b) gives $\hat{H}$ on scale (c).

$\hat{H}$ is a temperature factor, where

$$\hat{H} = 0.637 \; q^{1.75} \; X_t$$

(d) m is then found by multiplying $\hat{G}$ and $\hat{H}$, i.e.:

$$m = \hat{G} \cdot \hat{H}$$

$\hat{G}$ and $\hat{H}$ are actually intermediate steps in the solution of Equation (108). They serve to group the quantities used so that better visualization of the method may be attained. Since the range of values is large, a nomograph scale for multiplying $\hat{G}$ and $\hat{H}$ would give very poor resolution. The nomograph of Fig. 6-43 is designed for $t_b = 40^\circ C$ but the temperature factor is relatively insensitive to t_b so that good results are obtained over a considerable range of air temperature.

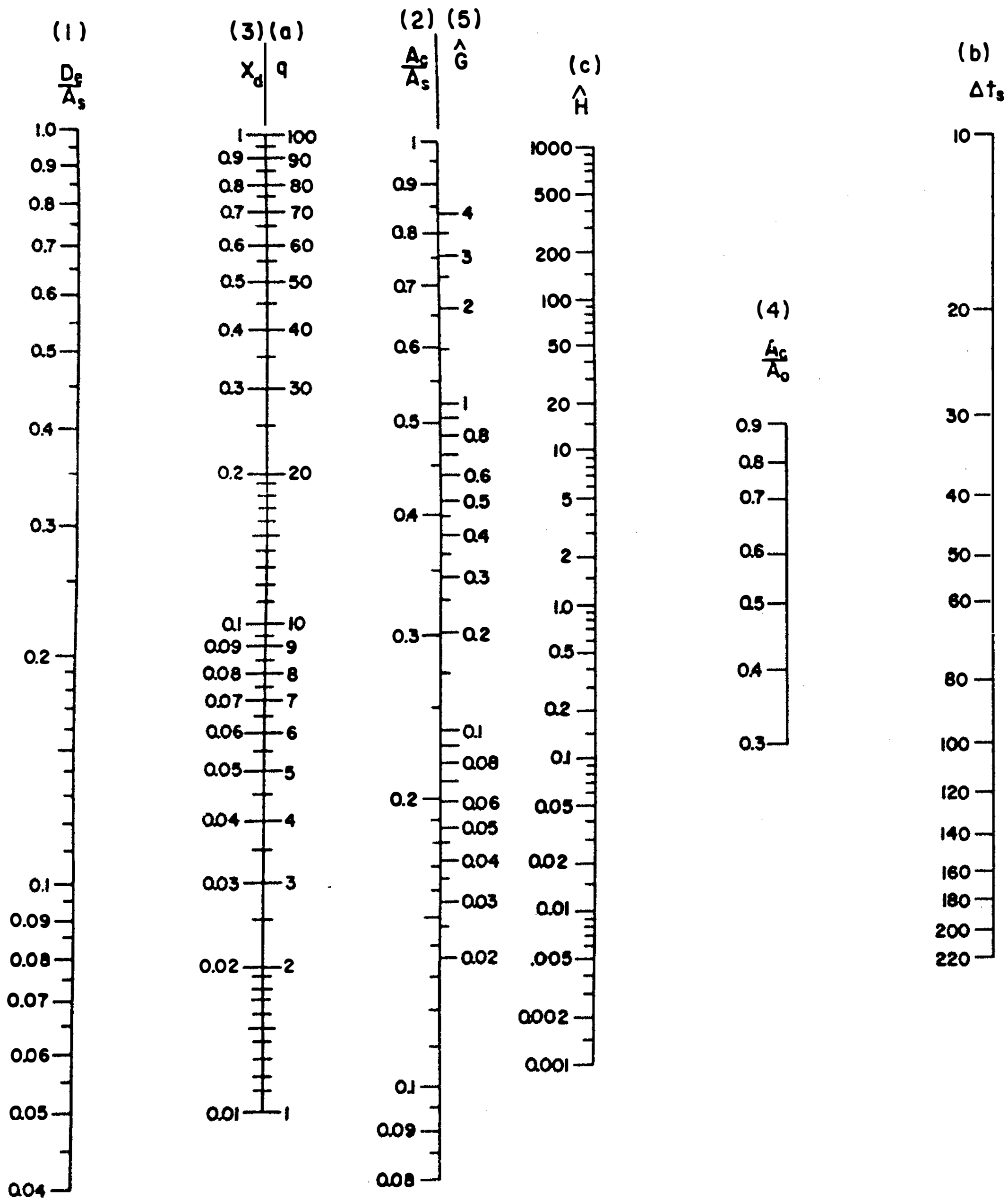


FIG. 6-43

DESIGN NOMOGRAPH FOR PRISMATIC COMPONENT PARTS

PROCEDURE:

(1) & (2) → (3)

(3) & (4) → (5) = $\hat{G}$

(a) & (b) → (c) = $\hat{H}$

THEN, $m = \hat{G} \cdot \hat{H}$

DESIGNED BY J. B. COLEMAN
DRAWN BY R. B. SCOTT

If several prismatic component parts are placed in the same duct, the calculation is repeated successively, allowance being made for the rise in air temperature. Since prismatic shapes cause much more disturbance of the air flow than do cylindrical shapes, it is recommended that the methods given in Reference 12 be applied when the number of such parts exceeds three or four. While turbulence is desired, the flat faces of the prisms may cause stagnation, with consequent reduction in heat transfer. The spacing of the units is critical. Staggering and the use of turbulating devices is recommended and computation methods are given in detail in Reference 12.

9. Turbulators

It has been emphasized that turbulent rather than laminar air flow is desirable because turbulence results in a much thinner layer of air (boundary layer) through which the heat must flow by conduction. Designs of forced-air cooled equipment should always assure turbulence near the surfaces of heat-producing parts. This is generally accomplished by placing heat sources near each other or near duct walls so that the clearance spaces are small. A rough check on the design can be made by computing the Reynolds number. It should be larger than 4000, although a Reynolds number down to 2000 is acceptable. However, between 2000 and 4000, there exists a possibility of incomplete turbulence, possibly resulting in marginal design. Therefore, it may be better to use a Reynolds number of 4000 as the minimum design quantity. The upper limit of Reynolds number is dictated by pressure drop and blower power required.

Air flow patterns can be observed by injecting smoke at several points along the flow path. If, at any point, low turbulence exists, it will be evident by a concentration of smoke occurring at that point. Corrective steps, described below, may then be taken. If components of different size and shape are placed in series in a duct, the turbulence will vary and may become undesirably low in some places. When such spots are found, the turbulence may be increased by introducing "turbulators" into the air stream. A sharp-edged plate set normal to the stream is an effective turbulator, the action of which is sketched in Fig. 6-44. Both upstream and downstream turbulators are shown. It is evident that the turbulator throttles the flow and increases the required blower power by increasing the number of obstacles in the flow path.

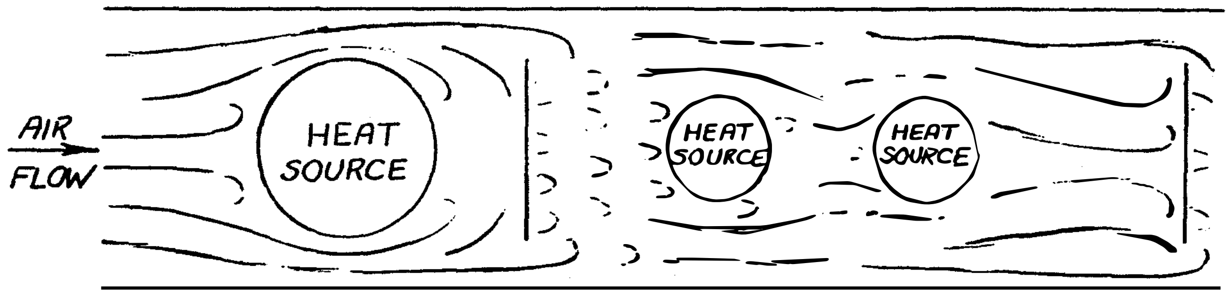


Figure 6-44 - Sharp-Edged Plate Turbulator

Another useful device for increasing turbulence is a transverse row of small rods placed in front of a heat source as shown in Fig. 6-45. The laminar flow is broken up, thus increasing the heat-transfer coefficient.

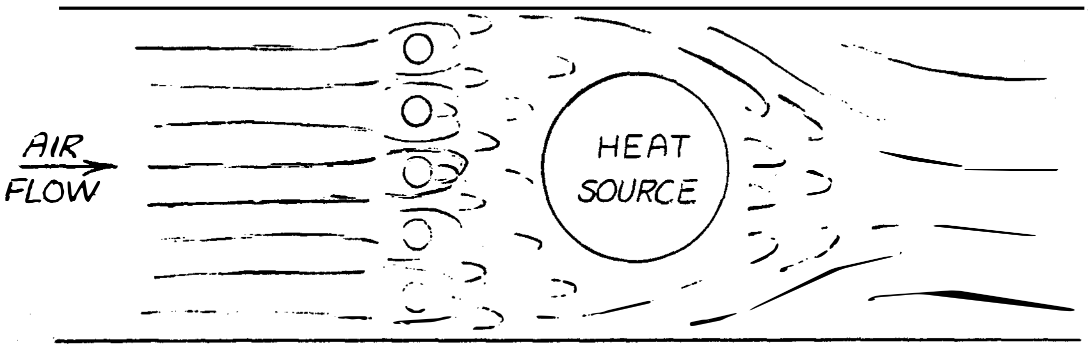


Figure 6-45 - Rod Type Turbulator

No definite design rules are given for turbulators. If hot spots are discovered and a study of the air stream indicates low turbulence, various types and locations of turbulators should be tried until an effective arrangement has been found. Sketches of air flow, similar to Figures 6-44 and 6-45 will frequently be helpful.

The nozzle arrangements described in the discussion of parallel flow cooling may also be thought of as turbulators. Nozzles and orifice plates may be found useful in eliminating local hot spots.

10. Finned Electron Tubes

For a discussion and examination of finned high-power electron tubes, i.e., of the order of 25 kw., see Reference 37.

D. Cooling of Enclosures and Duct Design

1. General

Air ducts impose resistances to air flow which must be overcome by pressure differences resulting from the expenditure of energy in maintaining the flow. A reasonably precise estimate of the flow resistances offered by the system is essential for satisfactory duct design. The theoretical resistance can be computed from the methods and data given in Chapter V of this manual.

The drop in pressure in air-transmission systems is due to friction losses and dynamic losses. Pressure increases and decreases may also be caused by changes in duct areas. The friction losses for turbulent flow are due to the friction of air against the side of the duct, and to internal friction between the air molecules. The dynamic losses are caused by changes in the direction or in the velocity of air flow, and may be caused by changes in size and shape of the cross-section of the duct, by elbows, and by obstructions to flow offered by dampers. Turbulent air flow is not desirable in ducts because pressure loss due to friction is directly proportional to the degree of turbulence.

2. Basic Ground Rules

The following factors must be considered when designing an air-duct system:

a. The air should be conveyed as directly as possible at permissible velocities.

b. Avoid sharp elbows and bends in ducts. Splitters and turning vanes should be used to minimize pressure losses.

c. Avoid both sudden enlargements and abrupt contractions. The angle of divergence of enlargements should not exceed 20 degrees. The angle of convergence in contractions should not be greater than 60 degrees.

d. For the greatest air-carrying capacity, rectangular ducts should be made as close to square as possible. In rectangular ducts, aspect ratios, that is, the ratio of the larger duct cross-section dimension to the smaller, greater than 6 to 1 should be avoided.

e. Make the ducts as air tight as possible. All laps should be in the direction of flow. Avoid raw edges on splitters and vanes.

f. Ducts should be made from smooth materials to minimize friction losses.

g. Provide dampers in all duct branches for final balancing.

3. Design Procedures

The steps necessary in the design of an air-duct system are:

a. Lay out the most convenient, direct system to provide for proper air distribution.

b. Lay out the return system.

c. Decide on the temperature of the air that is to be supplied and determine the air requirement for each chassis to be cooled, based upon the dissipated power and predicted temperature rise.

d. Size the ducts, using the methods described.

e. Check the calculations by determining the friction loss from the fan to the discharge outlet at the end of each branch.

f. Determine the total pressure loss from the fan to the farthest outlet. Add the pressure losses for all equipment, outlet loss, and loss in the return-duct system to determine the pressure that must be produced by the fan to insure adequate flow through the system.

4. Design Methods

Sample problem I in Chapter V illustrates the use of the basic fluid flow equations to solve for friction loss in a straight duct. The fittings, such as elbows and branch takeoffs, which cause dynamic pressure losses are converted into Equivalent Lengths of Straight Duct (ELSD) by obtaining the ELSD from Table XIV. The equivalent lengths are added to the length of straight duct before calculating friction loss. For any fluid other than air, these calculations are necessary, but for standard air with a density of 0.075 lb./ft.^{3} and a temperature between 50 to 90°F flowing through a round galvanized duct, a friction chart has been designed to eliminate calculations. Figures 6-46a and 6-46b are the air friction charts, divided into two parts, covering air flow of 10 to 1000 cfm in part a. and 1000 to 100,000 cfm in part b. Note that these charts have four variables: air flow (cfm), air velocity (fpm), duct diameter (in.), and pressure loss (in. of H_2O per 100 ft. of straight duct). Determining any two of these four variables immediately fixes the values of the other two.

Note that the size of the ducts are given as diameters. All air ducts are first sized as round ducts; then, if rectangular ducts are to be used, their dimensions can be selected to provide equivalent air-carrying capacity. Figure 6-47 speeds the conversion from round to rectangular ducts.

In sizing a duct system, the equal friction method is recommended. This method allows the entire air-duct network to be designed for an equal friction drop for each foot of duct length, preventing widely varying resistances between sections. The method is fast and relatively simple to use. Little balancing is required for symmetrical layouts in which all runs have about the same resistance.

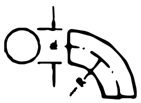
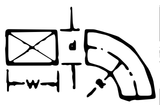
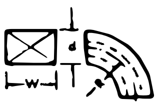
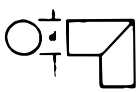
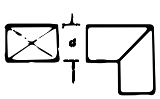
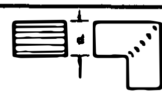
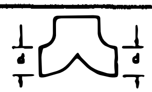
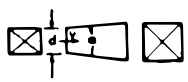
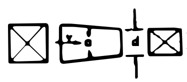
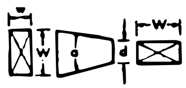
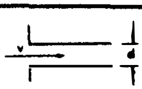
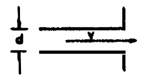
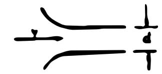
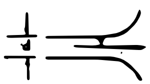
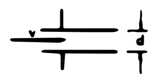
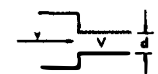
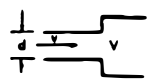
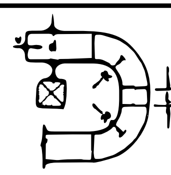
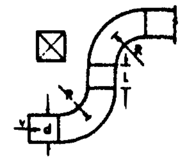
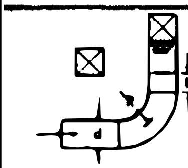
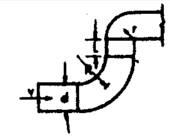
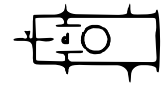
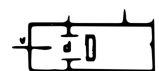
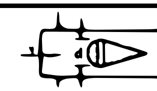
The step-by-step procedure of the equal friction method follows:

1. Determine the required air flow needed at the inlet temperature desired.

2. Determine the main duct velocity (maximum 2000 ft./min.).

3. From Fig. 6-46. determine the main-duct diameter for the total air flow (cfm) and velocity (fpm) just selected. At the same time read the pressure loss per 100 ft. of straight duct. This figure will be used in sizing the remaining ducts.

TABLE XIV EQUIVALENT LENGTHS AND STANDARD DUCT FACTORS FOR COMMON FITTINGS

Element	Conditions	Elsd Factor
 1. Round Radius Elbow	R/D = 0.5 R/D = 0.75 R/D = 1.0 R/D = 1.5 R/D = 2.0	43D 23D 15D 10D 9D
 2. Rectangular Radius Elbow	W/D 0.5: R/D = 0.5, 0.75, 1.0, 1.5 W/D 1 to 3: R/D = 0.5, 0.75, 1.0, 1.5	79D, 29D, 17D, 11D 57D, 20D, 12D, 8D
 3. Rectangular Radius Elbow	No. Vanes 1: R/D = 0.5, 0.75, 1.0, 1.5 No. Vanes 2: R/D = 0.5, 0.75, 1.0, 1.5	42D, 10D, 8D, 7D 27D, 7D, 6D, 9D
 4. Round Section Mitre Elbow		53D
 5. Rectangular Mitre Elbow		76D
 6. Rectangular Square Elbow	Single Vane Thickness. Double Vane Thickness. (Use Manufacturer's Data)	21D
 7. Rectangular Tee	Consider equal to an elbow. Base loss in each elbow on duct dimension indicated.	
 8. Radius Tee	Consider equal to an elbow. Base loss in each elbow on duct dimension indicated.	
 9. 45° Elbow	Either rectangular or round. Vaned or Unvaned.	½ of value for similar 90° elbow.
 10. Expansion	a = 5° a = 10° a = 20° a = 30° a = 40°	10D 17D 27D 36D 43D
 11. Contraction	a = 30° a = 45° a = 60°	1D 2D 4D
 12. Transition Piece		9D
 13. Abrupt Entrance		30D
 14. Abrupt Exit		60D
 15. Bell Mouth Entrance		2D
 16. Bell Mouth Exit		60D
 17. Re-entrant Entrance		51D
 18. Abrupt Contraction	v/V = 0.00 0.25 0.50 0.75	30D 27D 19D 11D
 19. Abrupt Expansion	V/v = 0.00 0.20 0.40 0.60 0.80	60D 39D 22D 9D 2D
 20. Double Elbows	Loss for both elbows L = 0 L = D Elbows Vaned	 26D 19D 9D
 21. Double Elbows	L = 0 L = D Elbows Vaned	38D 40D 12D
 22. Double Elbows	L = 0 L = D Elbows Vaned	24D 28D 12D
 23. Double Elbows	Direction of arrow Reverse direction	70D 62D
 24. Pipe Running Through Duct	E/D = 0.10 0.25 0.50	12D 33D 120D
 25. Bar Running Through Duct	E/D = 0.10 0.25 0.50	40D 80D 240D
 26. Streamlined Covering Over Obstruction	E/D = 0.10 0.25 0.50	4D 14D 54D

4. Along the pressure-loss line, at the air flow rate carried by each section of duct, read from Fig. 6-46 the corresponding duct diameters and air velocities. Record all information from Fig. 6-46 in a table for ready reference.

5. Enter the duct lengths in the work table. From Table XIV obtain the ELSD factors for all fittings in the system and enter them in the work table. Calculate the ELSD of all fittings. Add the total friction lengths of all ducts.

6. Determine the actual pressure drop in inches of water for each section of duct.

7. Add the pressure drops from the fan to the end of each duct run. These pressure drops should be almost equal. If not, re-size the ducts. Do this by assuming a higher pressure loss per 100 ft. of duct in those cases in which the pressure drop must be increased. Use a lower pressure loss per 100 ft. if the drop must be decreased. Read the new duct diameters from Fig. 6-46.

8. Recalculate the ELSD of all fittings for the new diameters.

9. Recalculate the actual pressure drop in inches of water for each section of duct. Remember to use the new pressure losses per 100 ft. that were assumed in step (7) to re-size the ducts.

10. Pressure drops through all runs should now be equal and the system should balance.

11. From Fig. 6-47 select suitable rectangular duct equivalents for the round duct diameters.

12. The fan selected must be able to overcome the static pressure of such accessory equipment as air cleaners or filters. A small factor of safety should also be included for leakage from the system

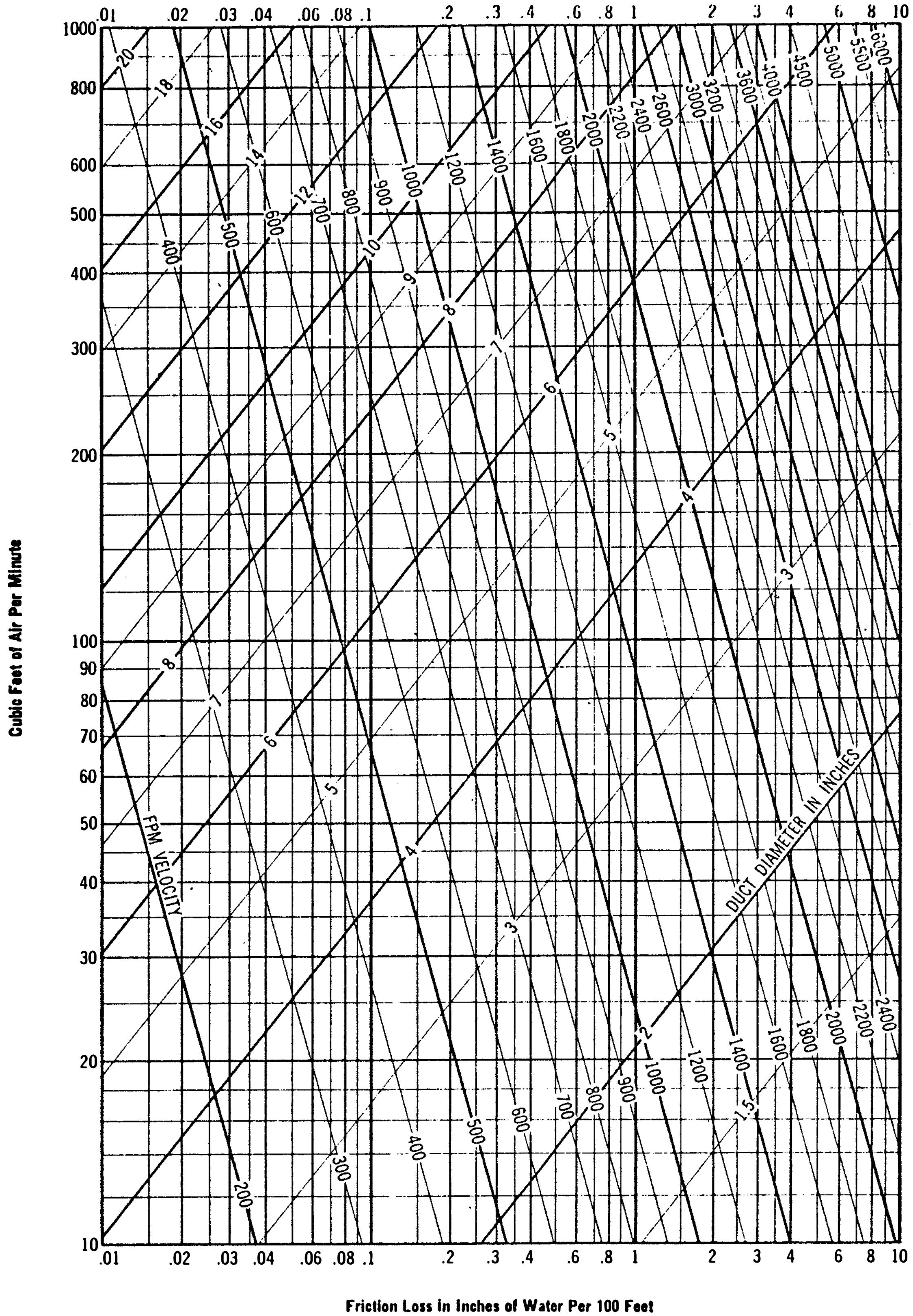


FIG. 6-46a AIR FRICTION CHART - PART a

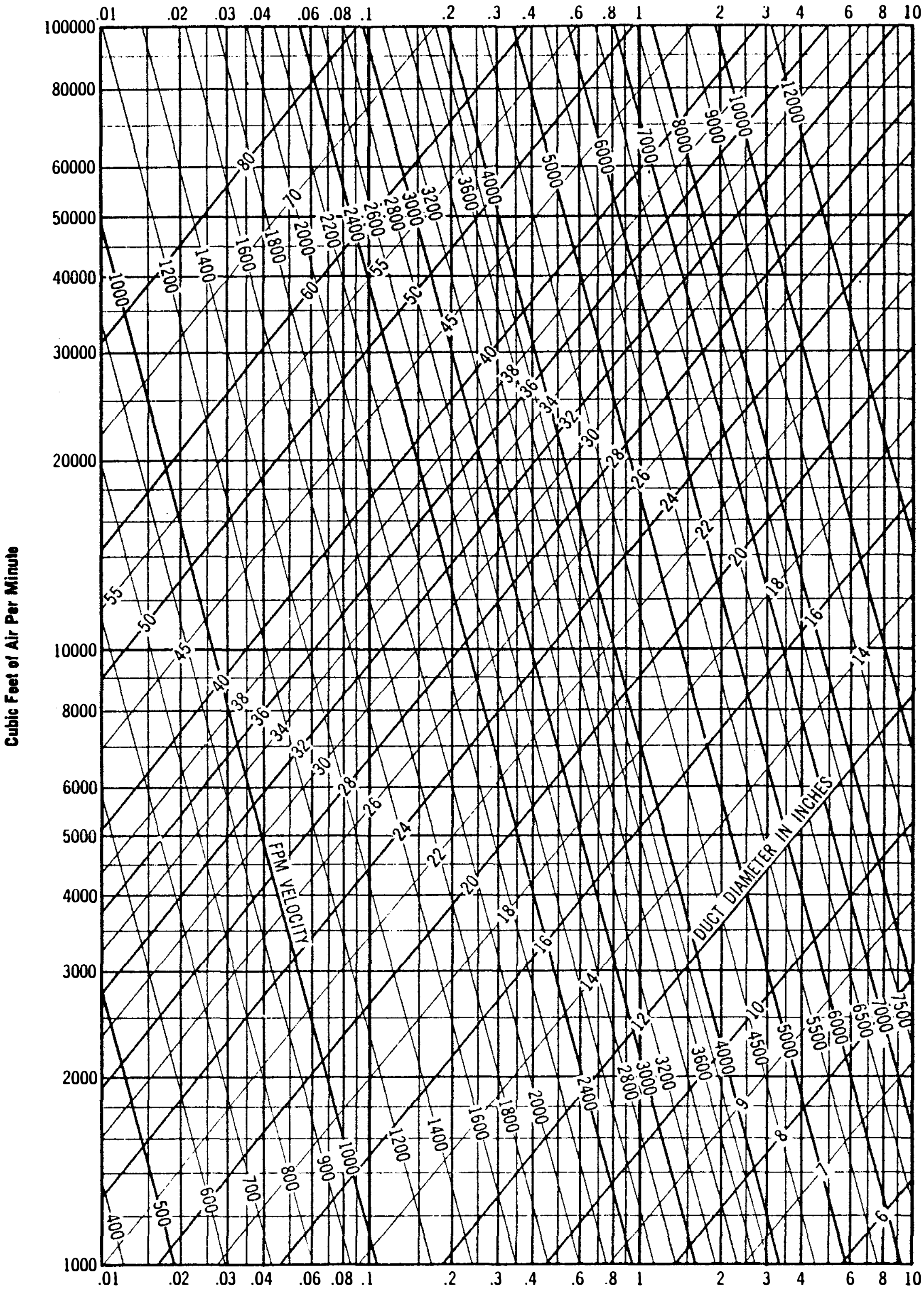


FIG. 6-46b AIR FRICTION CHART - PART b

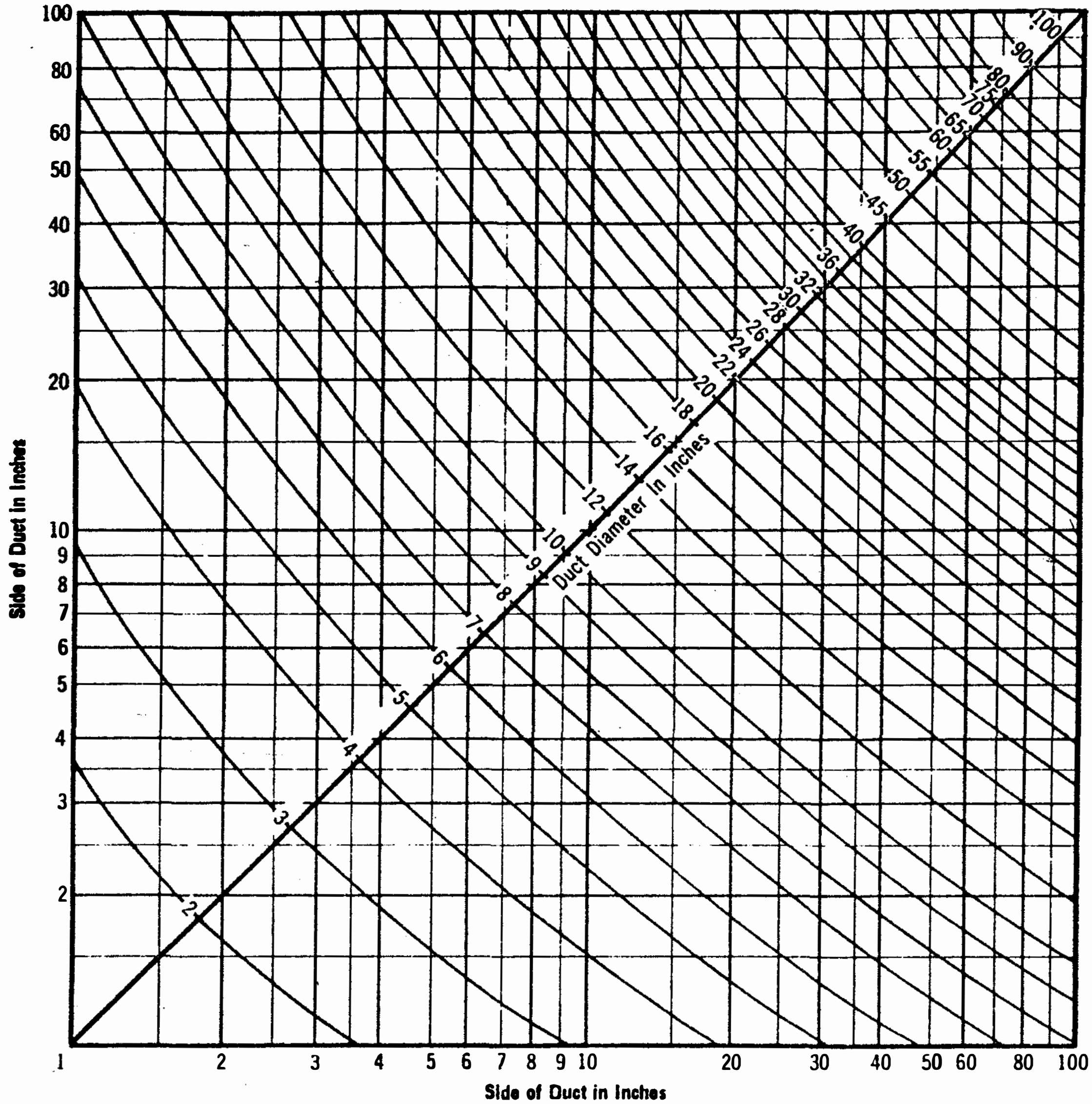


FIG. 6-47 CONVERSION CHART FOR CONVERTING SQUARE OR RECTANGULAR DUCTS TO ROUND DUCTS

5. Construction

Aluminum is the most commonly used metal for duct work. It is resistant to corrosion, light in weight, easy to fabricate, and, because it is thicker than steel for the same weight, it tends to have greater rigidity. The construction of fittings and changes of shape cannot be definitely outlined, because of the various conditions peculiar to each installation, but in general, long radius elbows and graded changes in shape tend to maintain uniform velocities accompanied by decreased turbulence and lower resistance. Examples of some generally accepted duct fittings are shown in Fig. 6-48.

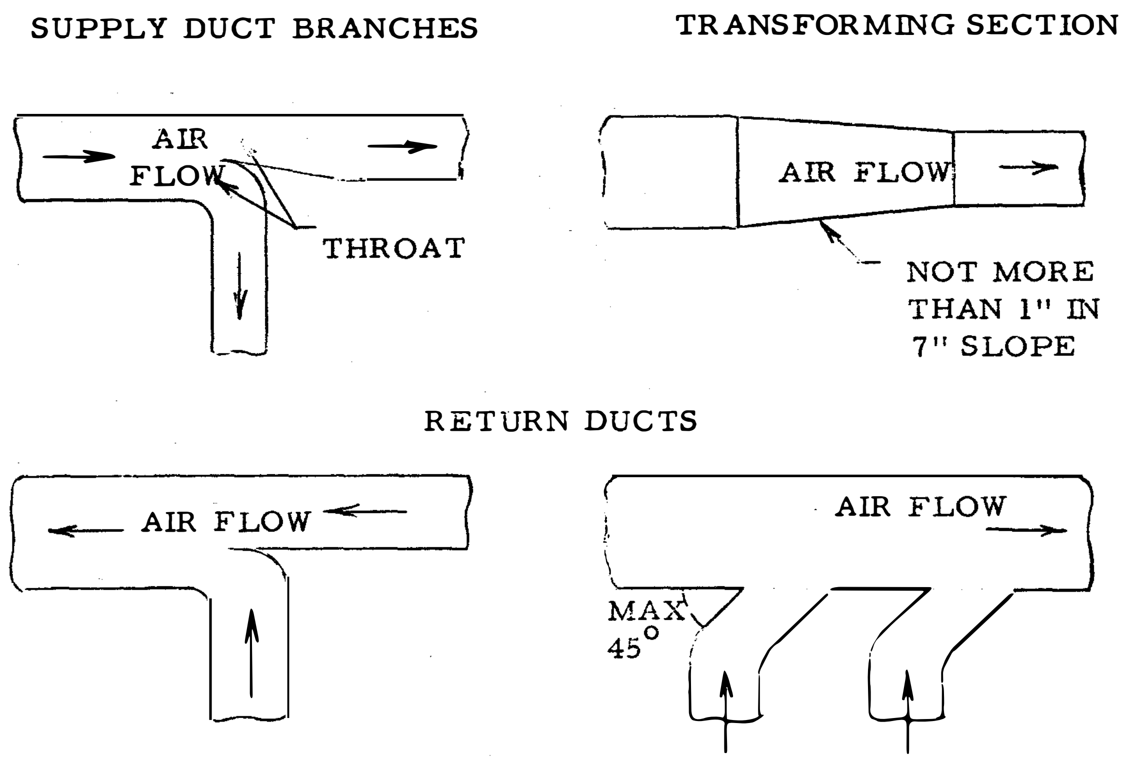


Figure 6-48 - Some Fittings for Ducts

6. Air Cleaners

In the closed forced-air cooling systems used in electronic enclosures, air cleanliness is not of the greatest importance. If outside air is to be introduced into the equipment, heavy dust particles affected by gravity might settle somewhere in the equipment. The finer particles remain suspended in the circulating air and are ejected from the equipment when the vertical velocity exceeds the critical velocity of the dust particles.

a. Dry filters strain the air through orifices smaller than the dust particles. The screens are made of cellulose, cloth felt, or similar material. They depend upon the fineness of mesh to screen out the impurities. "The effectiveness of dry filters is a function of the denseness and thickness of the filtering medium." Dry filtering has a high resistance to air flow. NOT RECOMMENDED for military electronic equipment.

b. Wet filters function through impact against viscous-coated barriers having interstices larger than the dust particles. Viscous filters usually use oil as a means of dust separation and can be very effective. However, once the dust is in contact with the oil it is difficult to release. This feature, while advantageous in the collection of dust, is somewhat of an inconvenience in its ultimate disposal. NOT RECOMMENDED for military electronic equipment.

c. There has been developed a dry filter which operates on the principle of electrostatic precipitation without the associated high voltage power supply, charged wire electrodes, and electronic equipment. A plastic material is given an electrical charge which reorients the molecules to maintain a permanent electrostatic charge (electret). The charged plastic is then shredded and packed into a conventional filter unit. Dust particles are attracted to the charged plastic filter by the electric field. When dirty, the filters can be removed and washed with a fine spray of cool water which neutralizes the static charge and releases the dirt. When dry, the filters resume their static charge and may be used again. This type of filter is RECOMMENDED for military electronic equipment. The shredded plastic material must be firmly held in place in the filter by crossties to prevent settling and dislodgement of the filter material during washing or transportation. Any significant displacement of the filter material will render the filter ineffective.

It is necessary to keep all filters clean and free from excessive oil in order that dust and oil be kept from the parts to be cooled. As explained previously, the resulting films can hinder the transfer of heat from the parts.

Filters should be readily accessible for periodic cleaning. When dirty, filters may present a fire hazard inasmuch as the dirt they accumulate may be flammable. The duct and connections to filters should be designed to allow uniform flow over the entire filter area.

7. Physical Example

As an example, the internal ducting of a rack-type cabinet for forced-air cooling of electronic equipment will be designed. The cabinet is to contain five chassis to be cooled by forced air. One chassis will be the power supply and the other four chassis will incorporate electron tubes as primary heat sources. In general, the tubes will be operated near their maximum ratings and tightly packaged to obtain a severe cooling situation. Several chassis of this type were designed, constructed, and tested to determine their individual cooling air requirements. Figure 6-49 shows the arrangement of the chassis within the cabinet. The following are brief descriptions of each chassis:

(1) Power supply chassis - 558 watts diss.

(2) 6L6 chassis - 27 tubes - 558 watts diss.

(3) 6AQ5 chassis - 53 tubes and 3 thermatrons - 900 watts dissipation.

(4) 805 chassis - 4 tubes - 630 watts diss.

(5) 4-250A chassis - 3 tubes - 1000 watts diss.

The total dissipation was in the neighborhood of 3800 watts including the power supply.

Standard rack chassis were used for each unit and were made considerably larger than the space occupied by the tubes in order to accommodate filament transformers, bias supply, and plate current monitoring circuits. Each chassis had a close-fitting cover over the tubes to form a duct for crossflow forced-air cooling. The 4-250A, the only exception, required parallel flow. Initial calculations indicated that no more than 150 watts could be readily removed from the 4-250A's by forced convection alone without high flow rates. This further supported the belief that much of the dissipated energy from such tubes is rejected by direct radiation from the plate because the rated gross dissipation is of the order of 300 watts per tube. Therefore, provisions were made for radiation and forced convection. Each tube was enclosed in a blackened shield welded to a coil of copper tubing through which

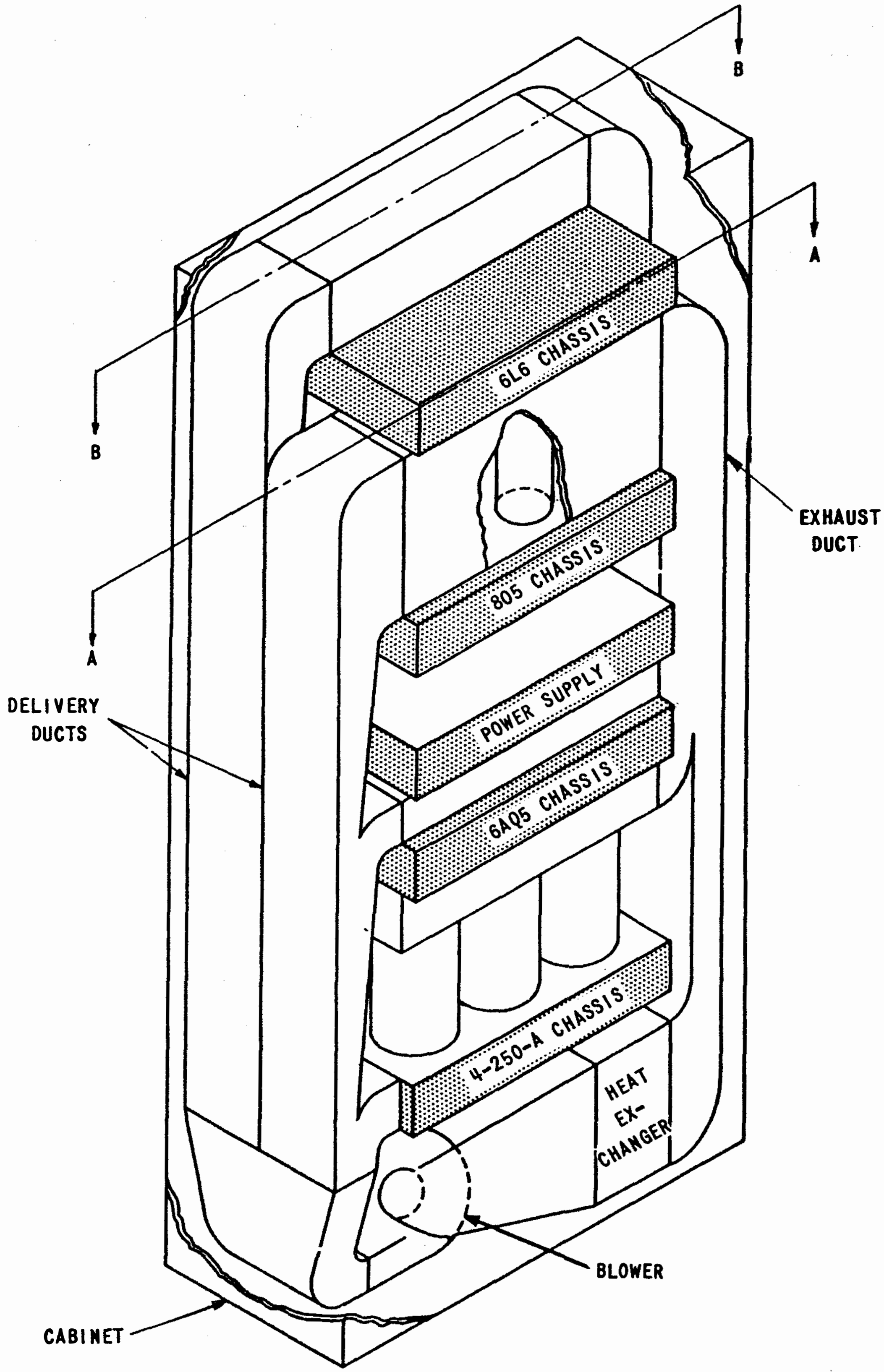


FIG. 6-49 RACK-TYPE FORCED-AIR COOLED CABINET

water was passed. Air entered the bottom of the chassis, passed through the socket and jacket and was exhausted. The bulb-base temperatures of the tube could not exceed 160°C. A duct system was to be designed within the cabinet to deliver the required amount of air to each chassis.

The cooling requirements for each chassis are:

CHASSIS	FLOW RATE	PRESSURE DROP
805	7.5 lb./min.	0.22" H_2O
6L6	3.0 lb./min.	0.22" H_2O
6AQ5	3.0 lb./min.	0.22" H_2O
4-250A	4.0 lb./min.	1.15" H_2O
Power Supply	5.0 lb./min.	0.50" H_2O
Total	22.5 lb./min.	

Note:

With the exception of the power supply, each of the chassis listed above was fabricated and tested at CAL during the Forced-Air Cooling Studies preceding this manual. The required flow rate and pressure drop are based on optimum cooling conditions observed during the tests. The power supply was assumed to have the flow rate and pressure drop listed, based on the above mentioned tests. This was necessary since no power supplies, as such, were tested.

The total air flow necessary to cool these chassis was 26.0 lb/min. A factor of 15% was added to the air flow to allow for the heat dissipated by the blower and any leakage from the system. The duct system within the cabinet was much too short for using the equal friction method of duct sizing because the ducts had negligible resistance to air flow compared to the chassis and heat exchangers. Only the delivery duct had to be sized and this was a constant size throughout. Air was forced through the system by the brute-force method and adjustable baffles were located at the entrance to each chassis. The baffles insured proper air distribution since the chassis had unequal flow resistance. The size of the delivery duct was determined in the following manner:

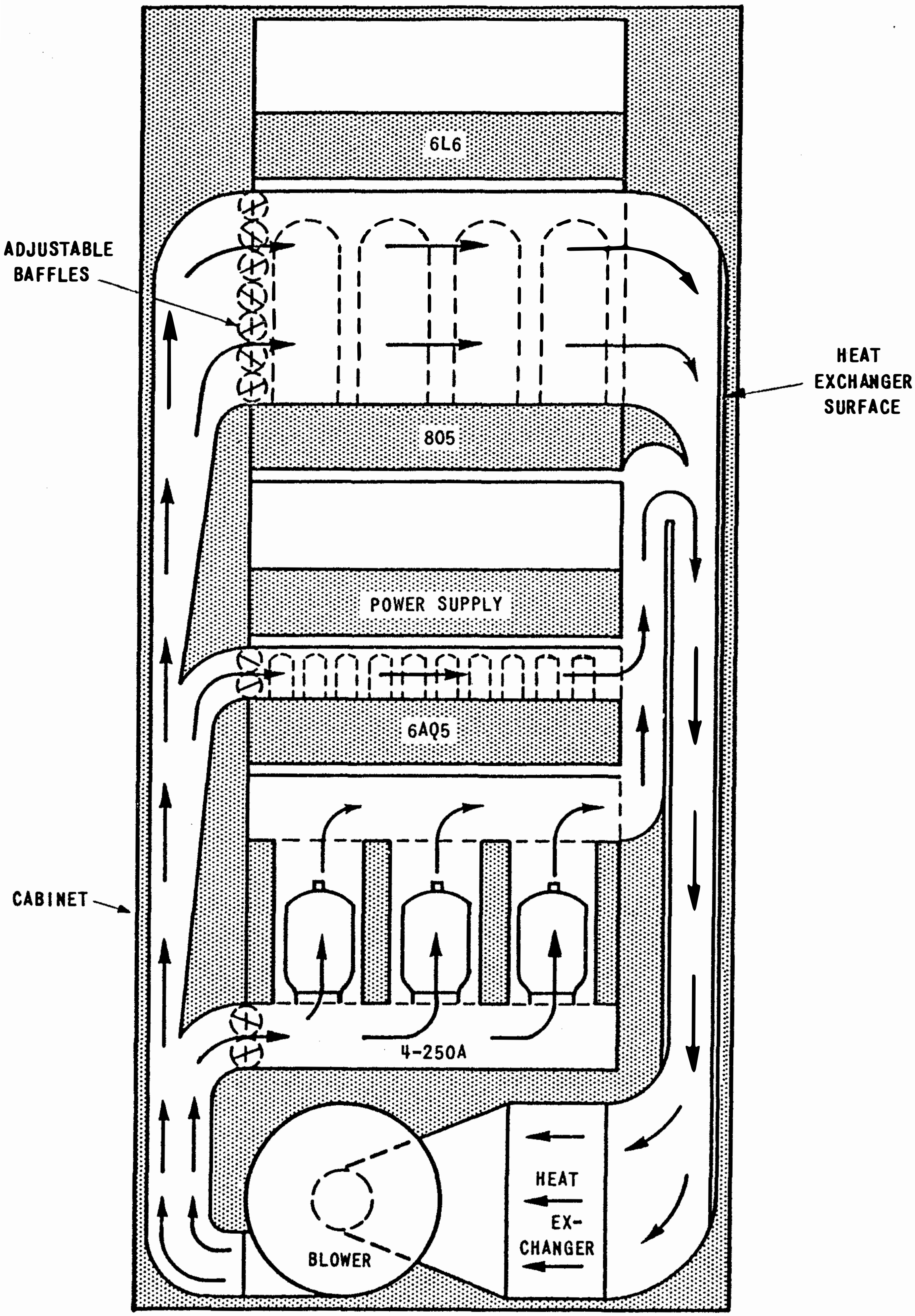


FIG. 6-50 SECTION A-A OF FIG. 6-49

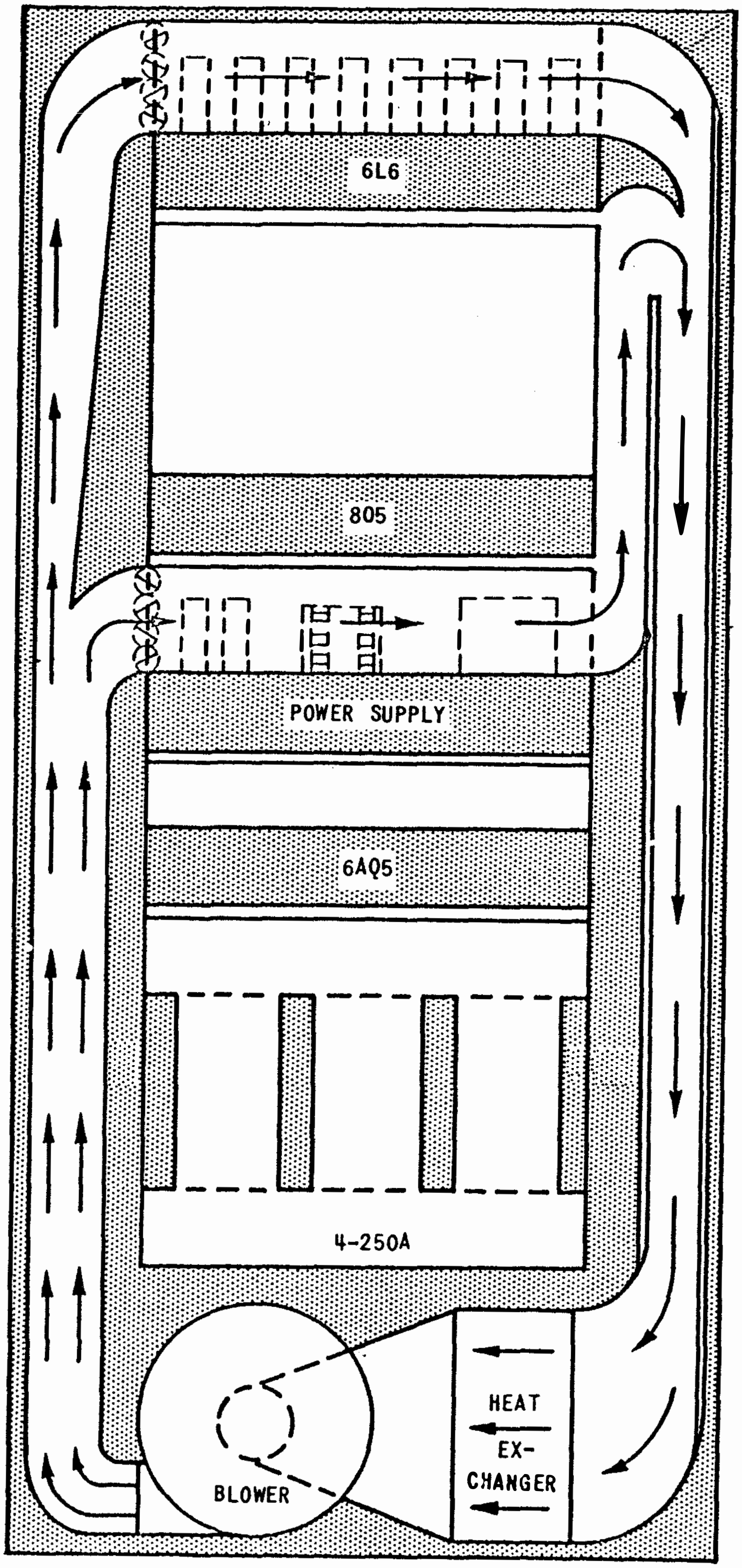


FIG. 6-51 SECTION B-B OF FIGURE 6-49

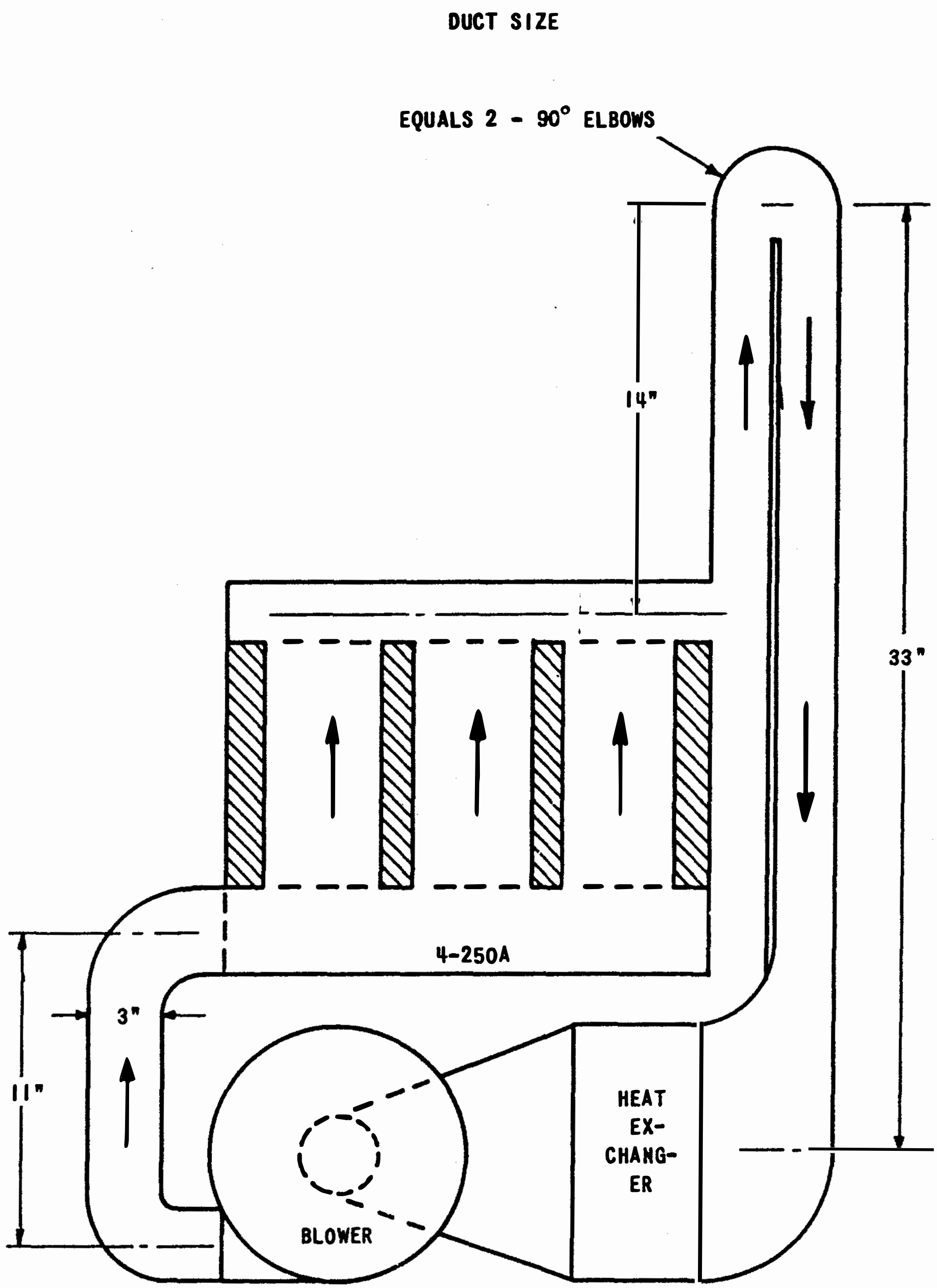


FIG. 6-52 SECTION VIEW OF 4-250A CHASSIS AND BLOWER FOR CABINET OF FIG. 6-49

density of standard air $= 0.075$ lb/ft.^{3}

volume of air required $= \dfrac{26}{.075} = 347$ cfm.

maximum velocity $= 2000$ fpm.

The duct diameter from Fig. 6-46a. is 5.7 inches, and is equivalent, in capacity, to a 3 inches by 10 inches rectangular duct, having 1.0 inches H_2O pressure loss per 100 feet of straight duct. The conversion of round ducts to equivalent rectangular ducts is accomplished by means of Fig. 6-47. As seen in Fig. 6-49, the blower is located at the bottom of the cabinet and delivers the air into a 10 inch by 3 inch duct. The duct splits into two equally sized ducts 5 inches by 3 inches , each delivering air to their respective chassis as shown in Figures 6-50 and 6-51.

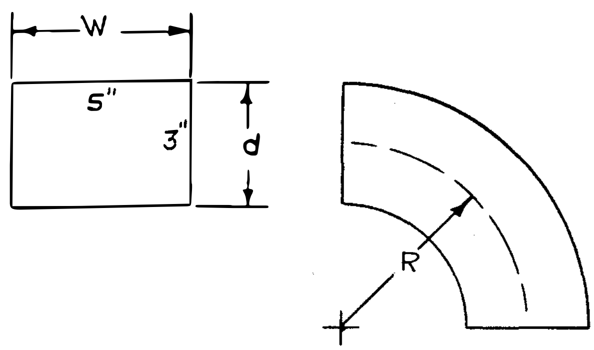
In order to select the proper blower, the duct run having the largest pressure loss was calculated. In this case it was the 4-250A chassis which was calculated as follows: (See Fig. 6-52)

Length of straight duct = 58 inches

Diameter of round duct equivalent to
5" x 3" $\doteq$ 4.3"

5 elbows of the type shown in Fig. 6-53.

There remains to be calculated the losses incurred by the elbows.



$$\frac{R}{D} = \frac{3}{4.3} = 0.7$$

$$\frac{W}{D} = \frac{5}{4.3} = 1.16$$

Figure 6-53 - Elbows Used in Cabinet of Figure 6-49.

Where in Fig. 6-53:

W is the width of rectangular duct

d is the depth of rectangular duct

R is the radius of the elbow

D is the diameter of the equivalent round duct.

For one elbow:

$$\frac{W}{D} = \frac{5}{4.3} = 1.16$$

$$\frac{R}{D} = \frac{3}{4.3} = 0.7$$

From Table XIV, interpolation gives:

ELSD Factor = 27D

Then equivalent length of straight duct is:

$$\text{ELSD} = 5 \text{ elbows} \times \frac{27 \times 4.3}{\text{elbow}}$$

$$= 590 \text{ inches}$$

Total length of duct

$$L = 590 + 58 = \text{approx. } 54 \text{ ft.}$$

$$\text{friction loss} = \frac{54}{100} \times 1.0 = 0.54 \text{ in. } H_2O$$

This, together with the pressure drop through the chassis totals 1.15 + 0.54 = 1.69 in. H_2O. The pressure drop through the heat exchanger must be added before the blower size can be selected. It is assumed to be .25 in. of H_2O. Thus the total pressure delivered by the fan is 1.94 in. H_2O.

The water-cooled heat exchangers should be capable of changing the air temperature 20°C and delivering the air to the chassis at 40°C. Heat exchanger design is discussed in the Liquid Cooling Manual, C. A. L. Report No. HF-845-D-9.

E. Forced-Air Cooled Chassis

"Cold-plate" chassis are basically heat exchangers on which are mounted the heat-producing parts or assemblies. The heat flows by metallic conduction from the heat sources to the exchanger fins thence into the moving air. The design of conductive metal heat-flow paths is discussed in detail in NAVSHIPS 900,192 (CAL Report No. HF-845-D-8) Design Manual of Natural Methods of Cooling Electronic Equipment. The cold-plate chassis may be cooled by forced air flowing through tubes which are in intimate contact with one side of plate, or the chassis itself may consist of a finned heat exchanger through which the air is forced.

The design of heat exchangers is discussed in a companion Manual, CAL Report No. HF-845-D-9, Design Manual of Methods of Liquid Cooling Electronic Equipment. The thermal requirements of cold chassis are such that conductive heat-flow paths having low thermal resistances from the heat sources to the air cooled surfaces are mandatory. This necessitates careful and perhaps expensive construction. It is necessary that the mating surfaces of the chassis and assemblies through which heat must flow be smooth with reasonably high contact pressures. Thus, the cold chassis must be rigid so that no mechanical distortion can occur when heat sources are mounted thereon. Further, the method of attachment should permit maintenance accessibility for component replacement with a minimum of effort.

The air within the heat exchanger passages should be turbulent. Therefore, relatively high air-pressure drop usually results. Practical cold chassis exchangers have air-pressure drops ranging from 2 to 20 in. H_2O at full capacity. Cold chassis have been built with 300 to 800 watts per sq. ft. of effective plate area.

Forced-air cooled heat exchanger chassis for power transistors are essentially miniaturized cold plates. In general, such exchangers must be efficient to minimize the temperature rise between the coolant air and the temperature sensitive transistors. The air velocity and the effective fin area must be relatively large.

Forced-air cooled modules or assemblies similar to those described by MIL-E-19600 (AER) permit the air to contact the heat sources more directly. This does not necessarily provide a lower thermal resistance to the air than can be achieved with cold chassis because the surface areas of some heat sources are marginally small. CAL Report No. HF-845-D-7 presents information pertinent to forced-air vs. conduction cooling of resistors.

VII METHODS OF MEASURING PRESSURE AND FLOW RATE*

A. Pressure Drop

Pressure drop is the difference between two absolute pressures. It is analogous to the potential difference (i.e., voltage difference) encountered in electrostatics. Since an absolute quantity is more difficult to measure than the difference between two absolute quantities, difference measurement is preferred. In the case of pressure, the reference is usually the atmosphere. However, when considering the flow of a fluid through a duct, pressure drop refers to that difference between an upstream pressure and a downstream pressure. An example of this is the pressure drop across an obstruction to the flow of fluid. Except in special cases, pressure drop measurements should be made only between two taps on the same cross-section, such as two points on the same duct.

After the rate of flow of coolant has been determined by the cooling needs of the equipment under consideration, the total pressure drop through the system for the prescribed flow rate must be found. Since fans, etc., are rated on the basis of rate of flow and pressure drop, the knowledge of both these quantities facilitates the selection of the most economical system. Knowledge of the pressure drop across a given type of obstruction or restriction to the flow permits the velocity to be calculated and, therefore, the flow rate, since the rate of flow is proportional to the velocity of the flow.

1. Units of Pressure Measurement

There are a number of dimensional systems for pressure intensity. The following illustrate the more common units at standard atmospheric pressure:

(1.) 14.7 pounds per square inch absolute (lbs/in.^{2}, psia).

(2.) 2116. pounds per square foot absolute (lbs/ft.^{2}, psfa).

(3.) 1.00 atmospheres (atm.)

(4.) 29.92 inches of mercury (in. Hg.) (30.0" Hg. for calculations).

(5.) 33.93 feet of water (ft. H_2O) (34. ft. H_2O for calculations).

*Note: The theory portions of this section are applicable to any fluid flowing in a conduit.

(6.) 407.1 inches of water (in. H_2O)
(407.0" H_2O for calculations).

Note:
It is common practice to use the approximate values in the parentheses in calculations.

The pressure drop of air flowing in a duct is generally quite small, and it is customary to express it as an equivalent height of water, i.e., inches of water.

2. Pressure Measuring Techniques

When a fluid flows in a duct, three types of pressures can be measured: (1) static pressure, (2) total pressure, otherwise known as stagnation pressure, and (3) velocity pressure, known also as dynamic or impact pressure.

In any given system, only two of the foregoing pressures need to be measured, since total pressure equals the sum of static and velocity pressures.

$$\text{(total)} = \text{(static)} + \text{(velocity)} \qquad (109)$$

$$p_t = p_s + p_v$$

3. Principles of Manometers

In general, a manometer is an instrument which displays a pressure differential based upon the relative pressures and the mass of the indicating fluid. A typical U-tube manometer is illustrated in Fig. 7-1.

The following generalized procedure may be followed in solving all manometer problems.

Step a. Start at one end (or any meniscus if the circuit is continuous), and determine the pressure intensity at that point in inches of water or by an appropriate indication if it is unknown.

Step b. Add to this the change in pressure intensity, in inches of water, from one meniscus to the next (plus if the meniscus is lower, minus if higher). This is the product of the difference in elevation in inches and the specific gravity of the fluid used in the manometer tube at that point.

Step c. Continue until the other end of the gage is reached (or the starting meniscus), and equate the expression to the pressure at that point, known or unknown.

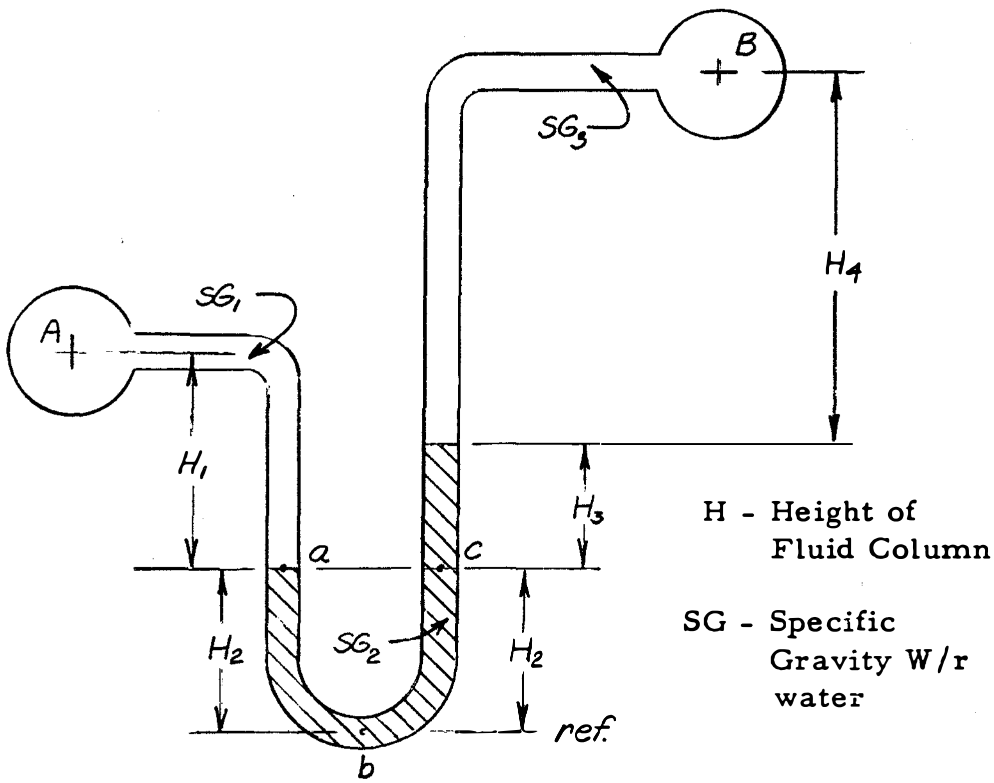


Figure 7-1 - Typical U-Tube Manometer

a. Sample Problem XIV

In Fig. 7-1, the pressure at point A, H_A, is known, as are heights H_1, H_2, H_3, and H_4 and specific gravities SG_1, SG_2, and SG_3. Find the pressure at point B (H_B) and the pressure difference (H_A - H_B) in inches of water.

Note:

When using a pressure unit such as the equivalent height of a column of fluid, i.e., inches of water, it is common practice to use the symbol, "H", which denotes the height of the column. "H", is also used to represent "pressure head".

Step 1. Start at point A. The pressure here is H_A.

Step 2. The next pressure to consider is that due to the height, H_1, of fluid of specific gravity SG_1. The equivalent pressure in inches of water is $(+ H_1 \; SG_1)$.

Step 3. The pressures at point (a) and (c) are identical. This is so because the points are both in the same fluid and are at equal heights above a horizontal reference. The terms would appear as:

$$+ H_2\, SG_2 \; - \; H_2\, SG_2 \; = 0.$$

Step 4. The equivalent pressure in inches of water due to the column of fluid H_3 inches high and of specific gravity SG_2 is $(- H_3\, SG_2)$.

Step 5. The equivalent pressure of the column of fluid H_4 inches high and of specific gravity SG_3 is $(- H_4\, SG_3)$.

Step 6. All the pressures in the system are known with the exception of H_B. The equation may be written as follows:

$$H_A + H_1\, SG_1 + H_2\, SG_2 - H_2\, SG_2 - H_3\, SG_2 - H_4\, SG_3 = H_B \quad (110)$$

and H_B may be found.

By rearranging the equation, the pressure difference may be found.

$$H_A - H_B = - H_1\, SG_1 + H_3\, SG_2 + H_4\, SG_3 \quad (111)$$

Almost any kind of manometer system may be solved in a like manner.

b. Static Pressure Measurement

Static pressure is defined as the compressive pressure existing in a fluid and is a measure of its potential energy. It may exist in a fluid at rest or in motion and is virtually the means of providing flow and maintaining it against resistance. Some static pressure measuring devices are shown below.

(1) Piezometer Opening

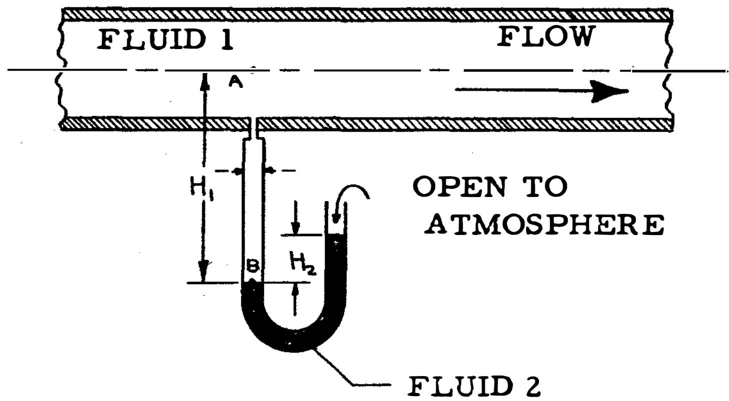


Figure 7-2 - Piezometer Opening

In Fig. 7-2, A is the point at which the static pressure is to be measured. The pressure that is actually indicated by H_2, the reading, is that at point B. If the ratio of the specific gravities of the two fluids, i.e. the manometer fluid (fluid 2) to the fluid flowing (fluid 1), is large, then the pressure at point B will be essentially that at point A. If the ratio is small, then the reading, H_2, must be corrected to include the effect of column H_1. A sample problem follows illustrating the effect of the correction when standard air ($\rho = 0.0765$ lbs/ft.^{3}) is flowing in the duct and water ($\rho = 62.4$ lbs/ft.^{3}) is the manometer fluid.

Sample Problem XV

Find the effect on the accuracy of the pressure reading in Fig. 7-2, when the correction is and is not applied.

$$H_1 = 4'', \; H_2 = 1''.$$

Solution:

The pressure at point A is 1 in. H_2O if the correction is not applied.

$$1 \text{ in. } H_2O \times 62.4 \frac{\text{lbs}}{\text{ft}^3} \times \frac{1 \text{ ft}}{12 \text{ in.}} = 5.2 \frac{\text{lbs}}{\text{ft}^2}$$

correcting for the column of air between B and A,

$$4'' \text{ air } \times \frac{1 \text{ ft}}{12 \text{ in.}} \times 0.0765 \frac{\text{lbs}}{\text{ft}^3} = \frac{0.0765}{3} = 0.0255 \frac{\text{lbs}}{\text{ft}^2}$$

$$\text{true static pressure at A} = 5.2 - 0.0255 = 5.1755 \frac{\text{lbs}}{\text{ft}^2}$$

$$\% \text{ error } = \frac{5.2 - 5.1755}{5.1755} = \frac{0.0255}{5.1755} \approx \frac{0.026}{5.2}$$

$$= 0.5\%$$

From the above, it is apparent that when measuring static pressure in this fashion, no corrections are necessary since the error in reading the pressure difference will in all probability be much greater than that due to neglecting the correction.

Some precautions that should be heeded when using this type of pressure tap and similar types, are as follows:

1. The tap should be inserted only where the duct has a smooth inner surface.

2. The tap diameter in the duct should be small with respect to the size of the duct.

3. The tap opening in the duct should not have burrs on its inner surface, i.e., it should be polished.

4. If a thin-walled air duct is to be tapped, provisions should be made so that the hole will have a length of at least one-eighth inch. This can be accomplished by "building up" the thickness of the duct in the vicinity of the tap opening.

(2) Static Tube (Piezometer Opening)

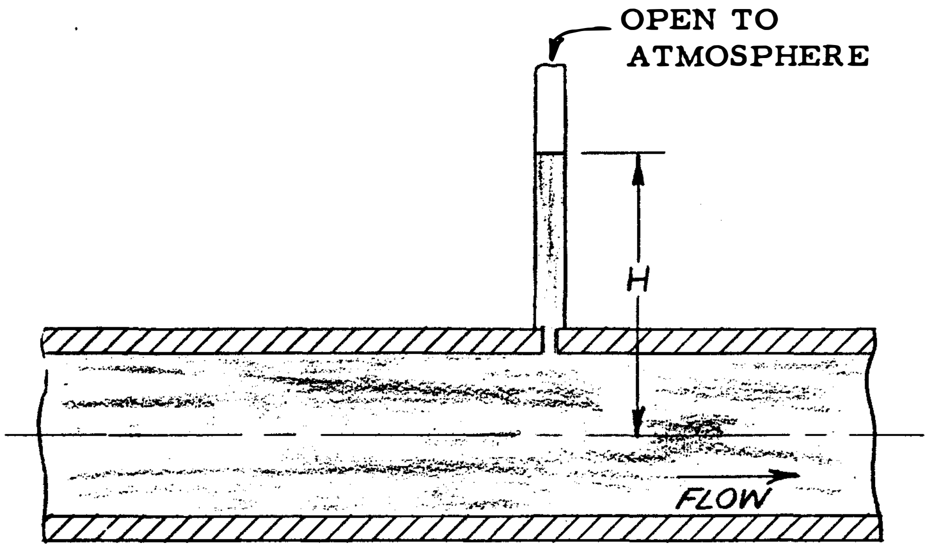


Figure 7-3 - Static Tube (Piezometer Opening)

In this method, the static pressure of the fluid causes it to rise to given level, H, above the center line of the duct so that the height, H, is the average static pressure reading in inches of fluid flowing in the duct, referenced directly to atmospheric pressure.

The limitations of this method are: (a) only a liquid of low volatility should be used, and (b) the static pressure cannot be less than atmospheric, otherwise air will be introduced into the system.

(3) Static Tube

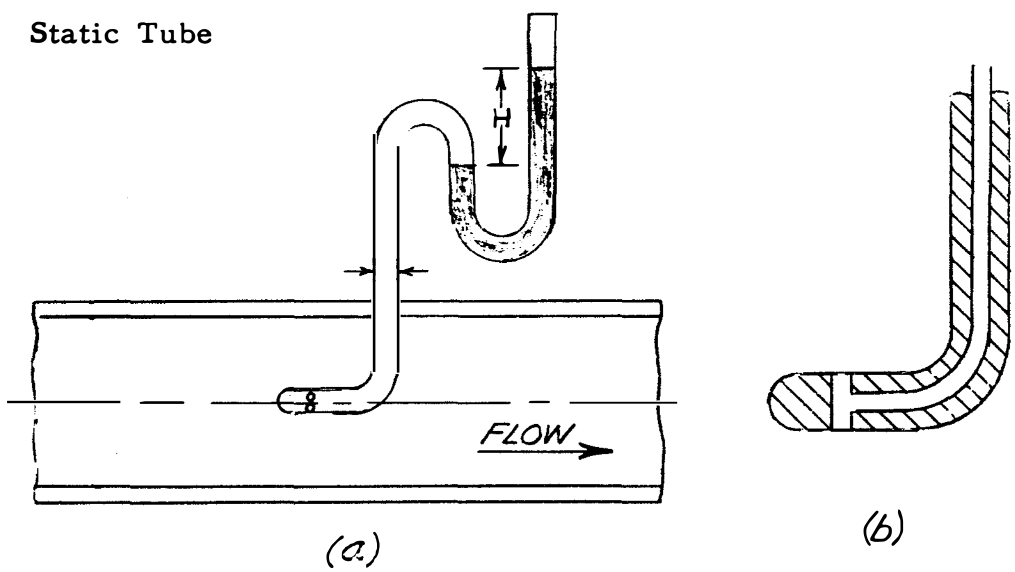


Figure 7-4 - Static Tube

The static tube (Fig. 7-4) is an instrument which is independent of the smoothness condition of the inner surface of the duct. It is especially useful in measuring static pressure in a rough-walled duct (e.g. a corrugated steel duct).

The static tube of Fig. 7-4a, is a tube directed upstream with its end, or nose, closed and having radial holes in the cylindrical portion downstream from the nose. It is imperative that the axes of both the duct and the static tube be coincidental in order that accurate readings be obtained. The pressure indicated is "H" ft. of manometer fluid. A cross-sectional view appears in Fig. 7-4b.

A disadvantage of this instrument is that the indicated pressure reading may not be the true pressure reading. This is possible because of the disturbance of the flow by both the cylindrical portion and its perpendicular leg. It is recommended therefore, that the static tube be calibrated against some accurate device and a calibration coefficient obtained.

(4) Piezometer Ring

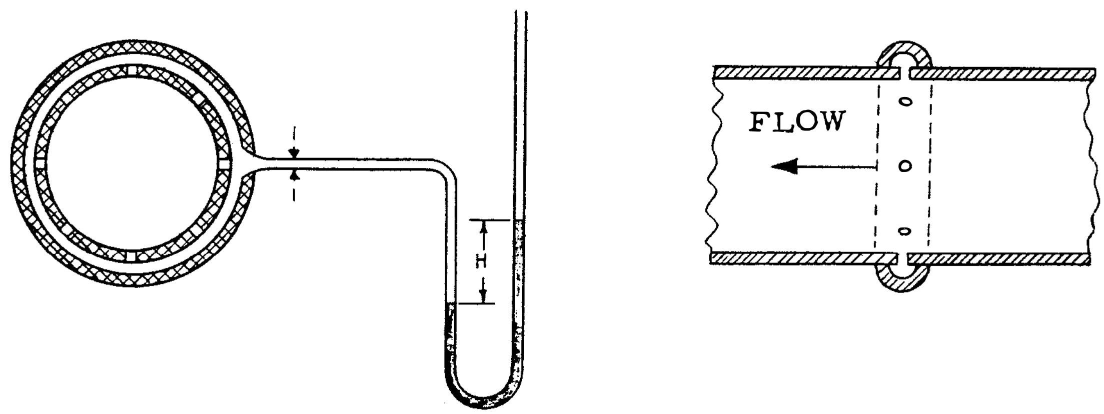


Figure 7-5 - Piezometer Ring

This instrument utilizes several piezometer openings connected in parallel (see Fig. 7-5). It is recommended for measuring the static pressure intensity of gases flowing in ducts. The flow of a gas generally does not assume the uniform pattern of liquid flow. Gases have a tendency to spiral through a duct, therefore a measurement of average static pressure is necessary. This is most easily accomplished with the piezometer ring.

c. Velocity and Total Pressures

Velocity pressure is the pressure corresponding to the velocity of the flow and is a measure of the kinetic energy of the fluid.

Total pressure is the sum of static and velocity pressure and is a measure of the total energy of the fluid. Total pressure is generally measured with a simple pitot tube. In Fig. 7-6, the measurement of total pressure in open channel flow of liquid is illustrated.

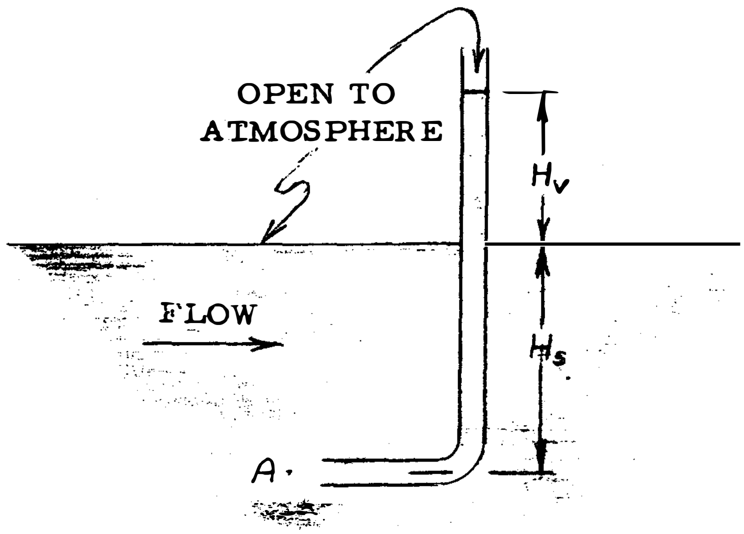


Figure 7-6 - Simple Pitot Tube for Open Channel Flow

In Fig. 7-6:

H_v is the velocity pressure at A, inches of fluid

H_s is the static pressure at A, inches of fluid

$H_s + H_v = H_t$, the total pressure at A

When a fluid flows in a duct, the system of Fig. 7-7 is commonly used to measure total pressure. This technique is utilized often when the fluid is a gas (e. g. air).

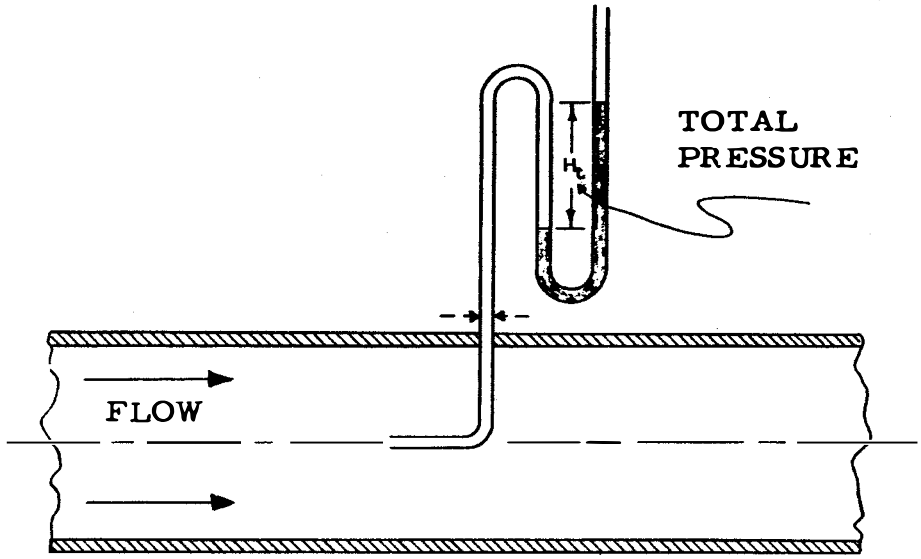


Figure 7-7 - Simple Pitot Tube in a Duct

When a measurement of velocity pressure alone is desired, either of two techniques may be used: the first is a combination of the pitot tube and a piezometer opening (see Fig. 7-8), the second is the pitot-static tube (see Fig. 7-9). Measurement of velocity pressure is fundamental in the determination of the velocity of the flow.

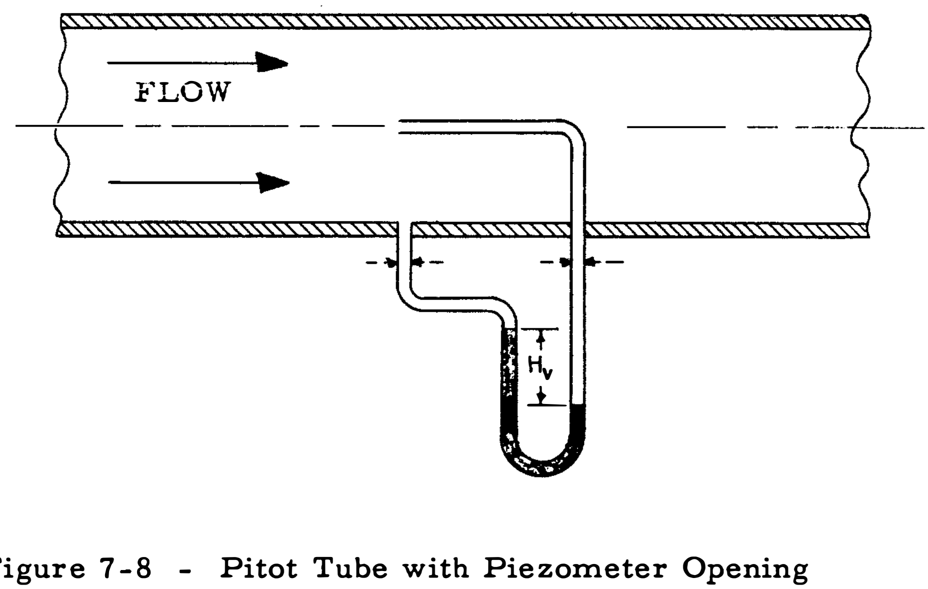


Figure 7-8 - Pitot Tube with Piezometer Opening for Measuring Velocity Pressure (H_v)

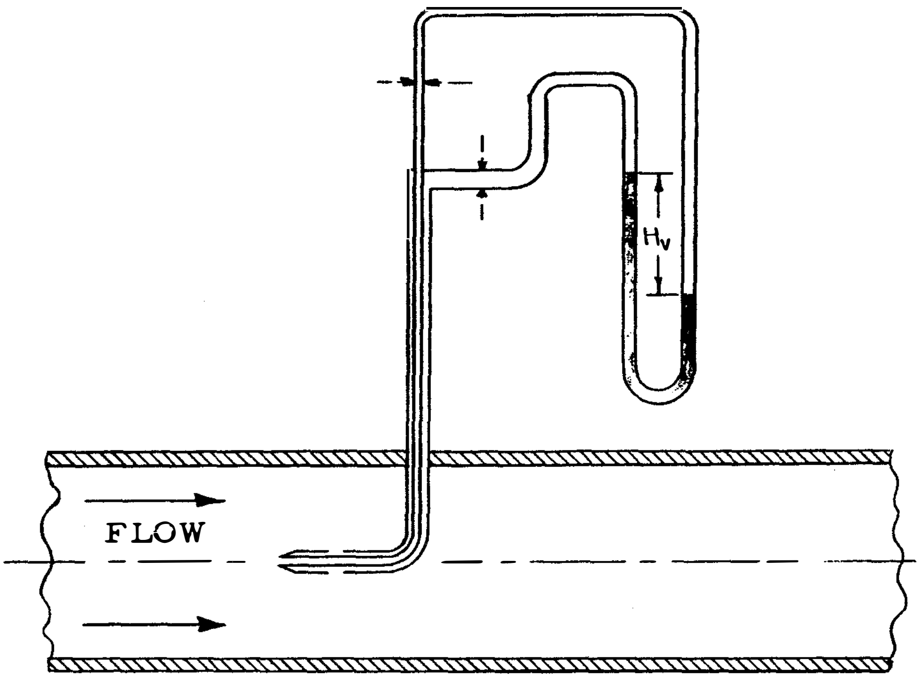


Figure 7-9 - Pitot-Static Tube for Measuring Velocity Pressure (H_V).

The above instruments should be calibrated, with the exception of the Prandtl tube. It is not necessary to calibrate the Prandtl tube because it is a form of pitot-static tube, in which the disturbances due to the nose and leg cancel each other.

In the previous sketches of pressure measuring devices, the U-tube manometer was used to indicate the pressure. However, at the points indicated by the two arrows, (Fig. 7-10), tubing, either rubber hose, copper or something similar, may be attached and then connected to a different type of manometer.

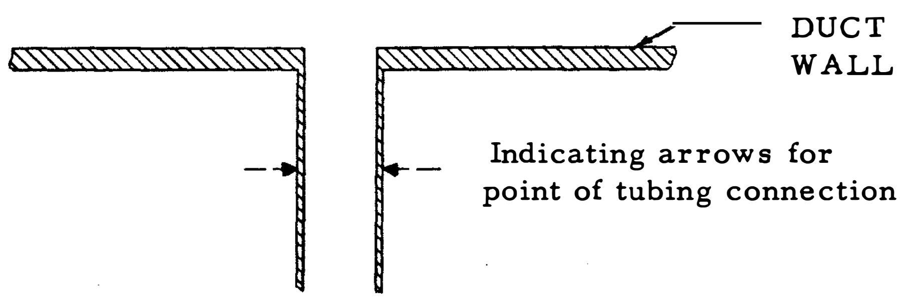
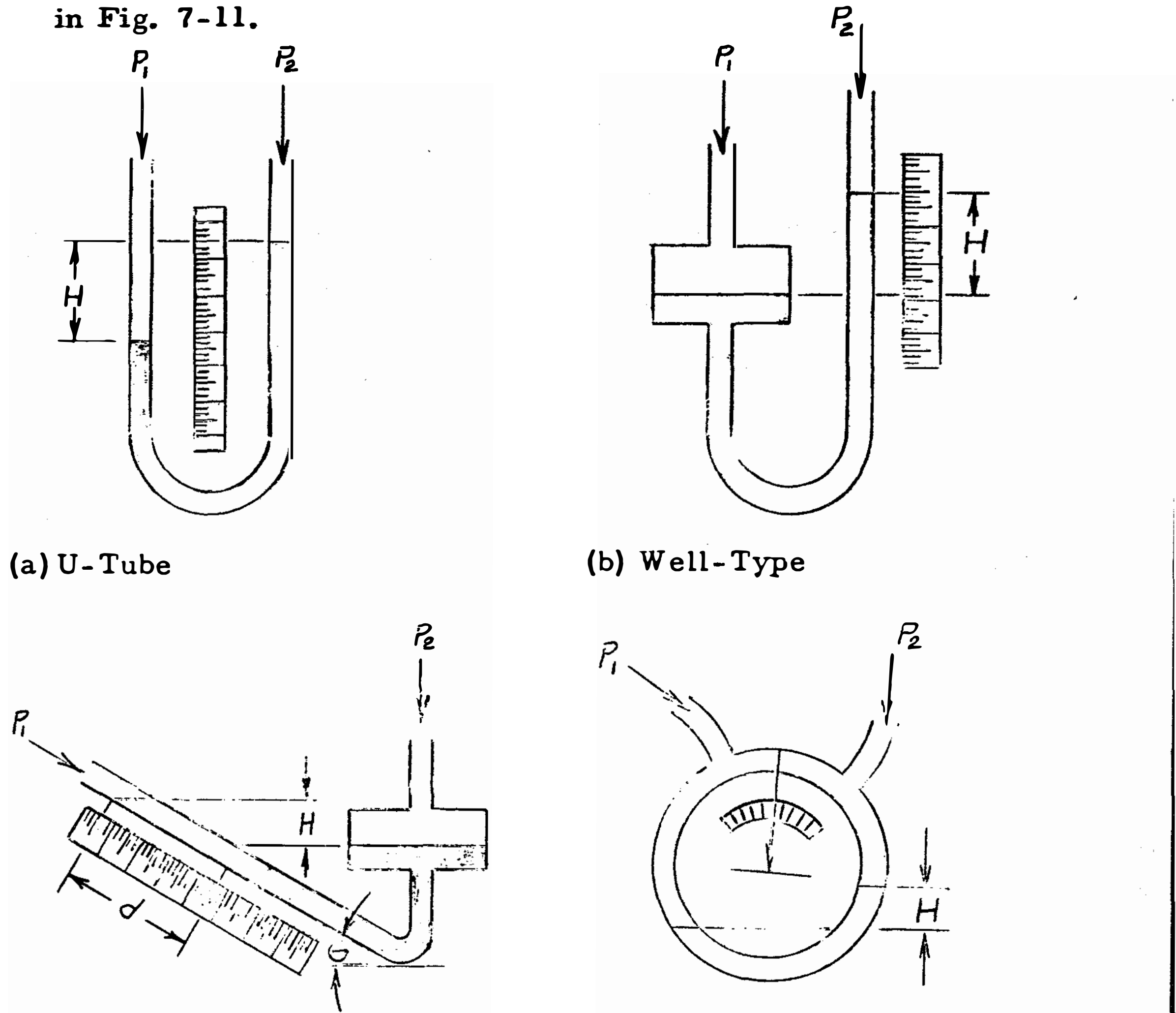


Figure 7-10 - Tubing Connection Symbol

Some of the more common types of manometers are shown in Fig. 7-11.



(a) U-Tube

(b) Well-Type

(c) Inclined tube

(d) Ring type

Note:

p_1 is the pressure to be measured

p_2 is a reference pressure where $p_1 > p_2$

$H = p_1 - p_2 =$ pressure in height of manometer fluid

Figure 7-11 - Basic Liquid Manometers

The manometers in Fig. 7-11 exhibit the essential principles of the U-tube. However, each type has advantages compared to the simple U-tube.

The well or cistern manometer is usually set up so that the level of the liquid in the well or reservoir is on the same horizontal plane as the point at which the pressure measurement is desired. This reduces the possible error due to columns of the fluid flowing (see Sample Problem XV, page 170). It also offers the advantage of having a permanently fixed scale with zero at the level of the fluid in the well. The error introduced by assuming this fixed zero point, i. e., assuming that the level of fluid in the well is constant, is small if the ratio of tube area to well area is small. For instance, if the cross-sectional area of the indicating tube is 1% that of the well, then the error introduced is 1%.

The inclined tube manometer is similar to the well type but differs with respect to the indicating tube. This tube is located at an angle, θ, with respect to the horizontal plane and hence has the ability to magnify the reading according to Equation (112).

$$H = d \sin \theta \tag{112}$$

where

H is the pressure reading in inches of fluid

d is the scale measurement of difference

θ is the angle between inclined tube axis and the horizontal plane

This "magnification" results in more accurate readings.

The operation of the ring manometer (Fig. 7-11d) depends on a rotation of the ring. After a pressure differential is applied, the fluid in the circular tube shifts. After the shift, the ring will rotate until it finds a new equilibrium point. The equilibrium point is determined by the center of gravity of the manometer fluid. This system can be very accurate if properly constructed eliminate friction in the bearings and resistance to the moving of the connecting tubes. The manometer can be used over a large range if an arm and counterweight system is attached. However, the accuracy of the readings decreases as the range increases.

The two most commonly used manometer fluids are mercury and water. Some of the other fluids include kerosene, alcohol, and various oils.

The inclined manometer is recommended for measuring pressures in a duct through which air is flowing, using water as the manometer fluid. It is recommended because the pressure differences, either between two points in the same duct, or, in some cases, with reference to the atmosphere, are generally small and quite easily within the range of most inclined gages.

B. Velocity and Flow Rate

1. Calculation of the Velocity of the Flow

Velocity can be calculated using the measured velocity pressure.

From Bernoulli's equation, the following expression can be derived (for incompressible flow) (see Ref. 3).

$$V = C_c \sqrt{2\ gH \left[\frac{SG_o}{SG} - 1\right]} \qquad (113)(D.E.)$$

where:

V is the velocity of flow, feet per second

C_c is the calibration coefficient $= \left[\frac{\text{actual velocity}}{\text{theoretical velocity}}\right]$

g is the gravity constant $= 32.2$ ft./sec.^{2}

H is the reading of manometer in feet of fluid (i.e., velocity pressure)

$\frac{SG_o}{SG}$ is the ratio of specific gravity of manometer fluid to specific gravity of fluid flowing in the duct.

Equation (113) is applicable to either of the systems illustrated in Figures 7-8 and 7-9.

It has been shown (see Ref. 3, p. 176) that air may be considered to be incompressible in the following ranges for the purpose of these measurements:

static pressure	-	0 to 2.0 psig
	or	0 to 55.5 inches of water
velocity	-	0 to 500 feet per second

This conclusion was arrived at by plotting the equations for velocity pressure head for both compressible flow and incompressible flow, using air at standard conditions, i.e.:

$$p_{static} = 2116. \text{ pounds per square foot}$$

$$\rho = 0.0765 \text{ pounds per cubic foot}$$

$$k_c = 1.40$$

The curves for both of the equations were coincidental over the ranges previously stated.

Generally, in an electronic cooling system, the limits would not be exceeded, and hence, air flowing in a duct may be assumed to be incompressible.

2. Flow Rate Measurement

The rate of flow of a fluid (particularly gases) in a duct is most easily and accurately measured by head meters. A head meter consists of two units: (1) a primary device producing a differential head or pressure difference which varies as the square of the rate of flow, and (2) a secondary device, usually a manometer of some kind, for measuring the differential head. The common head meters are: (1) thin plate, sharp-edged orifice; (2) Venturi tube; (3) flow nozzle or rounded entrance nozzle; and (4) pitot tube.

Note:

The principles of measuring flow rate with a pitot tube are different than those of the head meters. Hence, in discussing flow rate, the pitot tube will be mentioned separately.

Pressure loss caused by the pitot tube is zero; for the other three devices, a loss of 10 to 90 percent of the differential pressure will occur. The accuracy of all four types of meters will be within two percent if they are properly installed and operated.

The meter location is very important in the case of head meters, as upstream disturbances affect the flow. It is a good general practice to allow 10 diameters of straight duct upstream and 5 diameters downstream, if possible, and to use straighteners (egg crate or nest of several tubes) several diameters upstream from the meter.

The generalized flow equation for any head meter measuring a liquid, or measuring a compressible fluid with a differential head less than 40 percent of the upstream absolute pressure, is:

$$Q = Y M C_d A_c \sqrt{2gH} \qquad (114)$$

where:

Q is the volumetric flow rate in cubic per second

Y is the compressibility factor

M is the velocity of approach factor or "meter constant"

C_d is the coefficient of discharge for the particular type of meter

A_c is the measured area of minimum section of the throat, tube or orifice in square feet

g is the acceleration due to gravity (32.2 ft/sec^2)

H is the differential head in feet of fluid flowing (conditions as of the upstream tap)

Equation (114) can be further generalized as follows:

$$Q = B_f A V \qquad (115)$$

where:

Q is the flow rate

B_f is the flow factor (i.e., dependent upon coefficient of discharge, velocity and compressibility factors)

A is the Area

V is the velocity (i.e. $\sqrt{2gH}$) through above area

Essentially, then, the measurement of the differential head permits the velocity to be calculated, and hence, by the continuity or simple flow equation, the flow rate (see Equation (116) for incompressible flow), is:

$$Q = V_1 A_1 = V_2 A_2 \qquad (116)(D. E.)$$

The continuity equation states that for a given system the product of velocity and area is always constant, i.e., the flow rate.

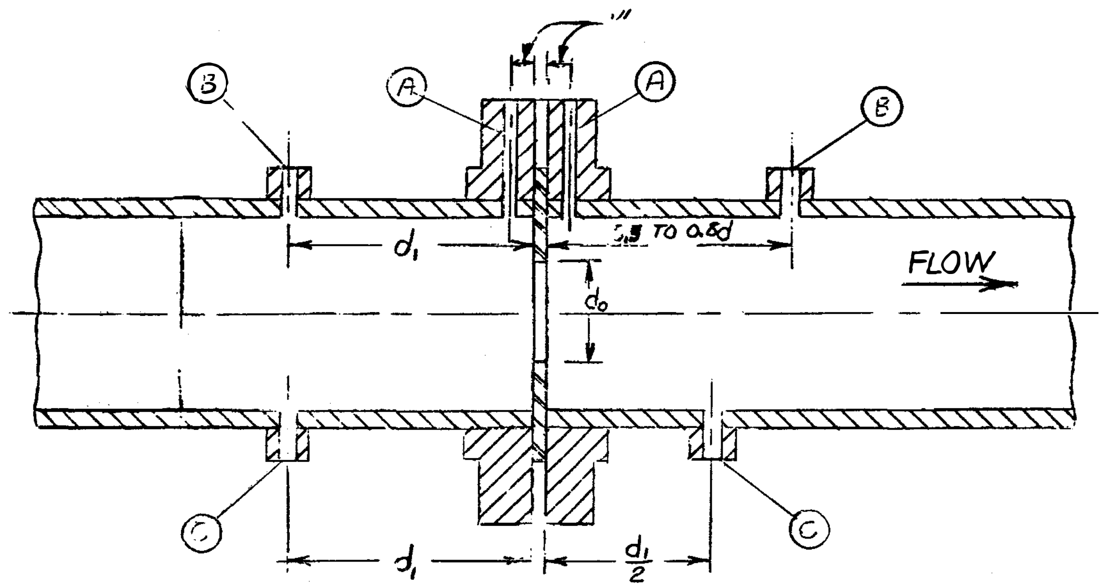
Each of the four types of head meters is discussed below.

a. Thin-Plate, Sharp-Edged Orifice

The orifice meter is the most common flow-metering device. It is relatively inexpensive, easy to construct, and easy to install. A thin-plate concentric orifice is a flat diaphragm with a circular hole in the center. It is usually clamped concentrically between the flanges in a duct.

Three sets of pressure taps are approved by the ASME for the measurement of the differential head: (a) flange taps (b) vena contracta taps, and (c) radius taps.

The specifications for each type of tap are shown in Fig. 7-12.



A. Flange Taps

B. Vena Contracta Taps

C. Radius Taps

Figure 7-12 - Thin-Plate, Sharp-Edged Orifice

b. **Venturi Meter**

The venturi meter is somewhat different from the other head meters in that its construction gives rise to only a minimum amount of turbulence. Generally having a smooth inner surface, it consists of the following: an upstream section the same size as the duct, which has pressure taps for measuring the upstream static pressure intensity; a converging conical section; a cylindrical throat, which also has pressure taps; and a gradually diverging conical section leading to a cylindrical section the same size as the duct. A differential manometer is usually connected between the pressure taps.

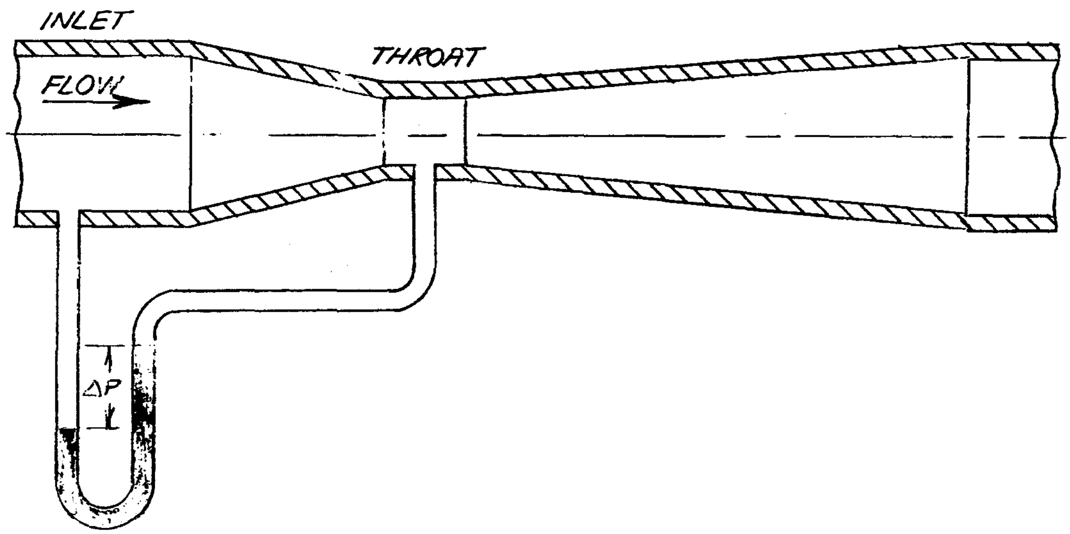


Figure 7-13 - Venturi Meter

The size of a venturi meter is specified by the duct and throat diameters, e.g. a 6" x 4" venturi meter fits a 6-inch diameter duct and has a 4-inch diameter throat. Venturi meter design requires the diverging after-section. This after-section helps to restore some of the static pressure that is lost when the flow is constricted. The venturi has greater pressure recovery than either the orifice or the flow nozzle. In this respect it is much better than the other two types of heat meters.

The venturi converging section has an included angle of between 25° and 30°. The diverging section should never exceed $7\text{-}1/2^{\circ}$ included angle.

c. Flow Nozzles

Flow nozzles are generally designed to be clamped between the flanges of a duct and usually have rather abrupt curvatures of the converging surfaces. The curvature of the inner surface may be either circular or elliptical, depending on the type of nozzle under consideration. The flow nozzle terminates in a short cylindrical section, which exhausts into the duct for the continuance of the flow. A typical flow nozzle with the conventional pressure taps is shown in Fig. 7-14.

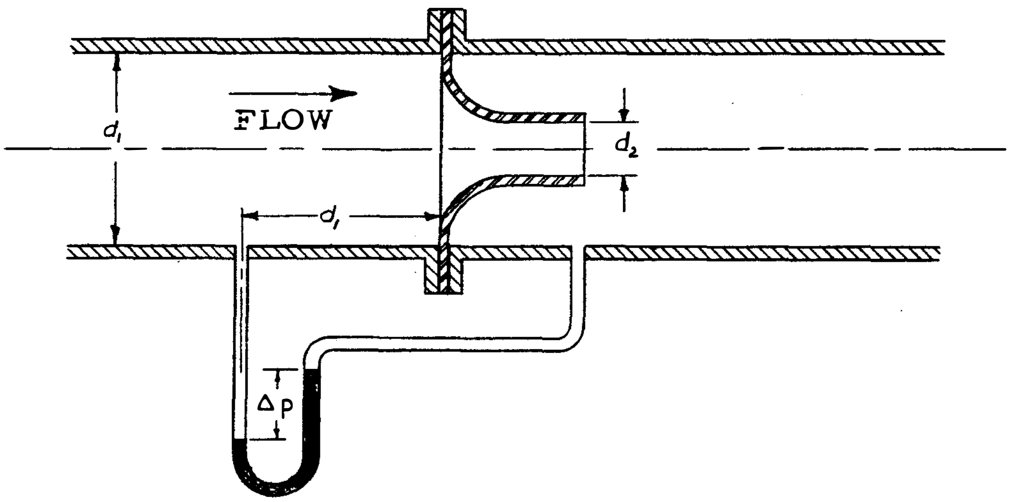


Figure 7-14 - Flow Nozzle

d. Recommendation

The orifice meter is recommended for measurement of air flow primarily because of its simple construction and easy installation. Once calibrated, the orifice is more than sufficiently accurate for the systems encountered in electronic cooling.

3. Theoretical Flow Rate Calculation

For the calculation of the flow rate of a fluid through a duct, the following are required: (a) the average velocity of the fluid and (b) the cross-sectional area of the conduit at the point where the velocity is measured. The fundamental operating principles of the orifice, venturi, and nozzle are the changing (reducing) of the flow path cross-section and the measurement of the corresponding change in static pressure. From the continuity equation:

$$Q = V_1A_1 = V_2A_2 \tag{117}$$

or:

$$\frac{V_1}{V_2} = \frac{A_2}{A_1} \tag{118}$$

It can be seen that velocity changes inversely with area. If, in a given system, a constriction of the flow occurs because of the presence of one of these meters, the velocity increases at that section of reduced area and the static pressure decreases as a result. Some of the potential energy, or static pressure of the flow, is converted to kinetic energy or velocity pressure, in the transition from the duct to the section of reduced area. Measurement of the change in static pressure permits the velocity at the constriction to be calculated. This velocity, multiplied by the area of the reduced section results in the volumetric flow rate.

$$\text{(Vol. flow rate)} = \text{(velocity of flow)} \times \text{(area)} \qquad \text{(119)(D. E.)}$$

$$Q = V \times A$$

This is the fundamental flow rate equation.

If the weight rate of flow is desired, the following form is utilized.

$$\text{(wt. rate of flow)} = \text{(density)} \times \text{(vol. flow rate)} \qquad \text{(120)(D. E.)}$$

$$m = \rho \times Q$$

The equation for the theoretical flow rate for orifice, venturi, and nozzle as a function of the static pressure drop measurement is:

$$Q = A_r \sqrt{2gH} = A_r \sqrt{2g\left(\frac{p_1 - p_2}{\rho}\right)} \qquad \text{(121)(D. E.)}$$

or:

$$m = \rho A_r \sqrt{2g\left(\frac{p_1 - p_2}{\rho}\right)} = A_r \sqrt{2g\,(p_1 - p_2)\rho} \qquad \text{(122)(D. E.)}$$

where:

Q is the volumetric flow rate, cubic feet per second

A_r is the area of reduced section, square feet

g = 32.2 feet per second per second

$(p_1), (p_2)$ are the absolute static pressures, upstream and downstream, respectively, in pounds per square foot.

ρ is the density of flowing fluid upstream tap conditions, pounds per cubic foot.

m is the weight rate of flow, pounds per second

A sample problem will be presented later to illustrate how simply these equations can be manipulated to fit any particular system.

a. Coefficient of Discharge

The coefficient of discharge is defined as:

$$C_d = \frac{\text{actual flow rate}}{\text{theoretical flow rate}} \qquad (123)(D.E.)$$

It is a function of the area ratio of the constriction to the duct, the velocity of the flow (more exactly, the Reynolds number of the flow), and the type of meter.

(1) Orifice Meters

As the flow passes through the orifice opening, the flow cross-section contracts further and approaches a minimum cross-section, approximately 0.62 that of the opening. This phenomenon is called "vena contracta". The approximate C_d for orifices is 0.60.

(2) Venturi

In the venturi, there is no flow contraction as in the orifice. This is due to the gradual transition between the duct and the throat. C_d for venturis range from 0.95 to 0.99.

(3) Flow Nozzles

The curved entrance and short cylindrical section of the nozzle prevent contraction. Generally for a smooth-surfaced nozzle, C_d varies from 0.97 to 0.99.

The coefficient of discharge is usually determined experimentally, although it can be calculated from design data. Most Mechanical Engineering handbooks and Fluid Mechanics textbooks contain sufficient data for these calculations.

Note:

The coefficients of discharge listed above are merely representative values and are not to be used. The C_d for a particular meter can usually be obtained from the manufacturer.

b. Velocity of Approach Factor

In the three types of head meters mentioned, the velocity of approach factor is dependent on the ratio of the constriction area to the duct area.

$$M = \frac{1}{1 - \left(\frac{A_2}{A_1}\right)^2} = \frac{1}{1 - \left(\frac{d_2}{d_1}\right)^4} \tag{124}$$

where:

M is the velocity of approach factor

A_2, d_2 are the area or diameter, respectively, of the constriction.

A_1, d_1 are the area or diameter, respectively, of the ducts.

Dimensional units should be such that the ratios are pure numbers. More often than not, the factor is left in the form of the area ratio in most flow equations. i.e.

$$M = \frac{1}{1 - \left(\frac{A_2}{A_1}\right)^2}$$

c. Compressibility Factor and Compressible Flow

A normally low-density gas can, of course, be compressed or expanded to a different density along the length of a duct. Therefore, a compressibility factor must be introduced into the flow equations when measuring rate of flow of gas in a conduit. Generally, at the low flow rates present in an electronic cooling system, air, the most common coolant, can be considered to be incompressible within the limits and for the conditions previously stated. However, a development of the equations necessary to account for possible compressibility follows.

The equation for ideal flow of an incompressible fluid through either the venturi, flow nozzle or orifice is:

$$Q_{ideal} = \frac{A_2}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{2g\left(\frac{p_1 - p_2}{\rho}\right)} \qquad (125)$$

where:

Q is the volumetric flow rate in cubic feet. per second

A_2 is the area of the constriction (i. e. throat of venturi, hole in orifice, hole in nozzle) in square feet.

A_1 is the area of the duct in square feet.

$g = 32.2$ ft/sec^2

p_1, p_2 are the upstream and downstream absolute static pressures respectively in pounds per sq. ft.

ρ is the density of the fluid flowing lbs/ft^3

To find the actual flow rate for an incompressible fluid, multiply the ideal Q by the coefficient of discharge, C_d.

When the actual flow rate of a compressible fluid, such as air, is desired, the equation becomes:

$$Q_{actual} = \frac{C_d A_2 Y}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{\frac{2g (p_1 - p_2)}{\rho_1}} \qquad (126)(D. E.)$$

Where Y is a function of the area ratio, A_2/A_1, the pressure ratio, p_2/p_1, and the specific heat ratio, k_c.

$$k_c = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}} = \frac{c_p}{c_v}$$

and is equal to:

$$Y = \left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} \left(\frac{k}{k-1}\right) \left\{ \frac{1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}}{1-\left(\frac{p_2}{p_1}\right)} \right\} \left\{ \frac{1-\left(\frac{A_2}{A_1}\right)^2}{1-\left(\frac{A_2}{A_1}\right)^2 \left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}} \right\} \right]^{1/2} \quad (127)$$

The compressibility factor, Y, is given as a function of the pressure, diameter, and specific heat ratios for air in Fig. 7-15.

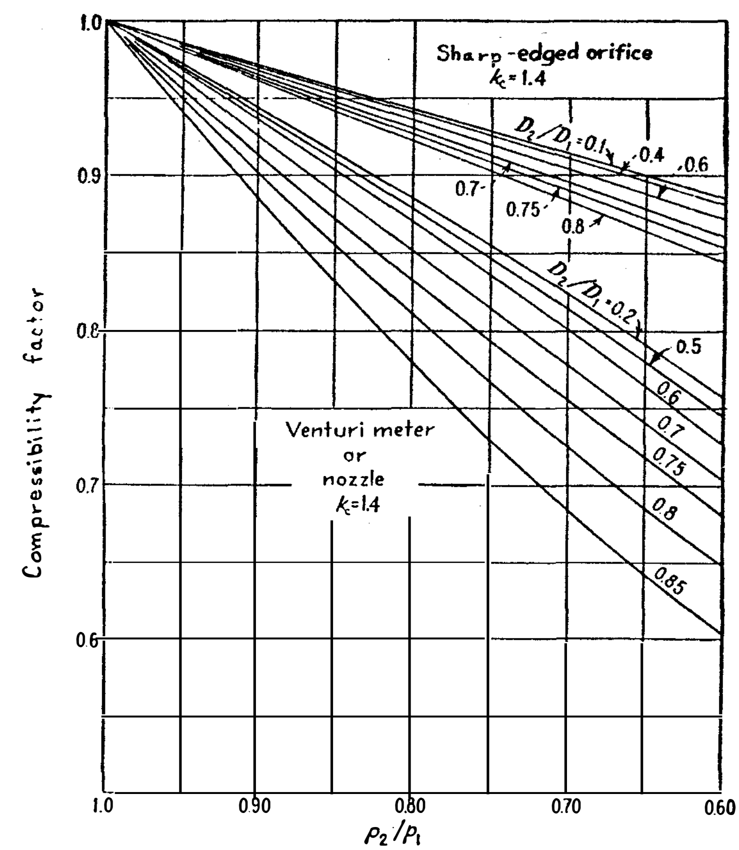


Figure 7-15 - Compressibility Factor

In a given system, with a measuring device that is permanently installed, and whose coefficient of discharge is known, the flow equation, Equation (126) can be reduced to the following form:

$$Q_{act} = C_s Y \sqrt{\frac{p_1 - p_2}{\rho_1}} \qquad (128)$$

where:

C_s is a constant for the system

$$C_s = \frac{C_d A_2 \sqrt{2g}}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}}$$

ρ_1 is the density of fluid flowing, (lb/ft^3), based on the conditions at the upstream pressure tap (see Equation (130)).

For determining the weight rate of flow, m, (lbs/sec.), Equation (128) is multiplied by the density of the fluid at the upstream tap, ρ_1:

$$m = C_s Y \rho_1 \sqrt{\frac{p_1 - p_2}{\rho_1}} = C_s Y \sqrt{\rho_1 (p_1 - p_2)} \qquad (129)(D. E.)$$

The density of a gas may be determined by the following procedure:

1. The following must be known

   a. p_1 - absolute static pressure at upstream tap (lbs/ft^2).

   b. T_1 - absolute temperature (^oR)

   c. R - gas constant (R = 53.3 ft/^oR for air)

2. Use these data in the following equation

$$\rho_1 = \frac{p_1}{RT_1} \qquad (130)$$

This will give ρ_1 in lbs/ft^3

4. Pitot Tube as a Head Meter

The pitot tube differs from the three head meters previously discussed. The pitot tube measures the velocity pressure at a point in a duct rather than a differential static pressure across a constriction. The velocity pressure is directly equal to the kinetic energy of the flow.

$$p_v = \frac{V^2}{2g} \tag{131}$$

or

$$V = \sqrt{2gp_v} \qquad \text{(132)(D. E.)}$$

where:

p_v is the velocity pressure in feet of fluid flowing

$g = 32.2 \text{ ft/sec.}^2$

V is the velocity of flow in feet per second

The pitot tube traverse is especially useful in determining the average velocity of flow of air in a duct. The average velocity must be obtained because of the variations of velocity diametrically across the duct. This is especially true for gases. They tend to spiral down a duct when under moderate or low pressures.

The traverse is determined by measuring several values of velocity pressure and averaging the result. Reference to most handbooks will give the procedure to be followed.

See the NAFM Bulletin #110, pages 11 to 27 inclusive, for a presentation of the accepted procedure.

Once the average velocity has been obtained the following equations may be used.

$$Q = V_{av} A \qquad \text{(133)(D. E.)}$$

$$m = \rho_1 V_{av} A \qquad \text{(134)(D. E.)}$$

5. Anemometers

Anemometers are used for measuring the velocity of flow of a gas. Since it measures only velocity, flow rate must then be calculated. These instruments, which are generally portable, are especially useful in measuring air velocity exiting from a duct system.

There are two types of anemometers that are in common usage. These are (1) the vane anemometer and (2) the hot-wire anemometer.

The vane anemometer consists of a series of vanes mounted on an axis which is coupled through a gear train or an electrical generator to some calibrated indicating device. The force of the gas flow causes the vanes to rotate at some speed proportional to the velocity of the flow.

The hot-wire anemometer consists of a short length of fine platinum wire which is heated by an electric current. The resistance to the flow of current through the wire is a function of its temperature. Flow of a gas around the hot wire cools it and thus changes its resistance. By holding either the voltage across the wire, or the current through the wire constant, the change in current or voltage, respectively, becomes a function of the velocity of the gas flow across the hot wire.

The hot-wire anemometer, when calibrated, has a reasonable degree of accuracy. However, the vane anemometer, due to its fundamentally mechanical operation, can have a considerable error, because the flow tends to evaporate bearing lubricants, thus changing the friction in the system.

It is recommended that anemometers be calibrated often, since their relatively complex structure is likely to lead to a change in calibration after a short period of usage.

When using an anemometer, it is advisable to make several measurements of the velocity and average the results to calculate flow rate. The following equations should be used.

a. Volumetric Flow Rate

$$Q = V_{av} A \quad \text{cfm} \tag{135}$$

b. Weight Rate of Flow

$$m = \rho V_{av} A = \rho Q. \qquad \frac{\text{lbs}}{\text{min}} \tag{136}$$

where:

Q is the volumetric flow rate, cubic feet per minute

V_{av} is the average velocity (measured) in feet per minute

A is the cross-sectional area of conduit in square feet

ρ is the density of fluid (upstream conditions) pounds per cubic foot

m is the weight rate of flow, pounds per minute

6. Sample Problem XVI

Air (k_c=1.4) flows through an 8-inch diameter duct. The flow rates, volumetric and weight, are to be found using a 4-inch diameter thin-plate orifice whose coefficient of discharge, $C_d = 0.67$. Manometer readings are: $H_1 = 5.33''$ H_2O, $H_2 = -1.4''$ H_2O with respect to atmosphere. The temperature at the upstream tap is 77.5°F and barometric pressure was measured at 29.42" Hg. (Gas constant for air, R = 53.3 ft/°R). Find the volumetric and weight rates of flow.

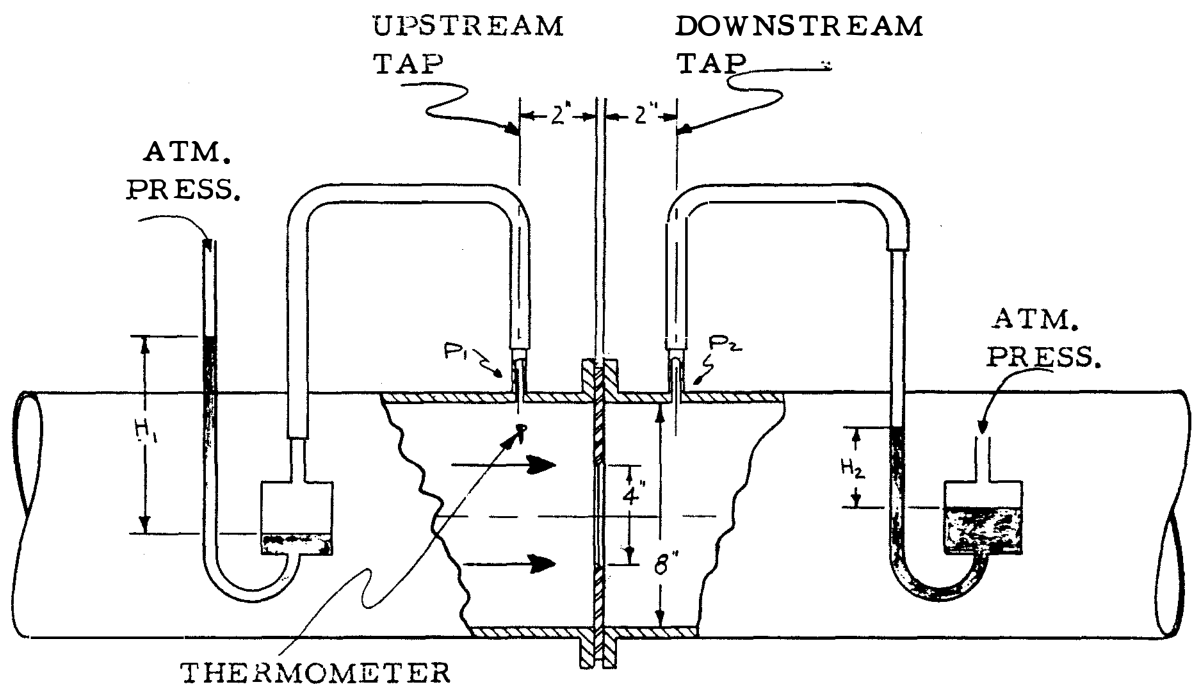


Figure 7-16 - Typical Installation for Measuring Flow of Air in a Duct With an Orifice.

Solution:

a. atmospheric pressure, p_A

$$p_A = 29.42 \text{ in Hg} \times \frac{2116 \text{ psf}}{30.0 \text{ in Hg.}} = \underline{\underline{2075 \text{ psfa}}}$$

b. upstream absolute static pressure, p_1

$$p_1 = 5.33 \text{ in } H_2O \times \frac{1 \text{ ft}}{12 \text{ in}} \times \frac{2116 \text{ psf}}{34 \text{ ft. } H_2O}$$

$$= 27.6 \text{ psf}$$

$$p_1 = 2075. + 27.6 \approx \underline{\underline{2102 \text{ psfa}}}$$

c. downstream absolute static pressure, p_2

$$p_2 = -1.4 \text{ in } H_2O \times \frac{1 \text{ ft}}{12 \text{ in}} \times \frac{2116 \text{ psf}}{34 \text{ ft. } H_2O}$$

$$= -7.25 \text{ psf}$$

$$p_2 = 2075. - 7.25 \approx \underline{\underline{2068 \text{ psfa}}}$$

d. pressure head, $(p_1 - p_2)$

$$(p_1 - p_2) = 27.6 - (-7.25) = 27.6 + 7.25$$

$$= \underline{\underline{34.85 \text{ psf}}}$$

e. upstream temperature, T_1

$$t_1 = 77.5^\circ F$$

$$T_1 \text{ (absolute)} = 460 + 77.5 = \underline{\underline{537.5^\circ R}}$$

f. upstream density of air, ρ_1

$$\rho_1 = \frac{p_1}{RT_1} = 2102.6 \frac{\text{lbs}}{\text{ft}^2} \times \frac{1\ ^\circ R}{53.3 \text{ ft.}} \times \frac{1}{537.5^\circ R}$$

$$\rho_1 = \frac{2106.2}{53.3 \times 537.5} = 0.0736 \ \frac{\text{lbs}}{\text{ft}^3}$$

g. Area of duct, A_1

$$A_1 = \frac{\pi (d_1)^2}{4} = \frac{\pi}{4} \left(8 \text{ in} \times \frac{\text{ft}}{12 \text{ in}}\right)^2 = \frac{\pi}{4} \times \frac{4}{9}$$

$$= \underline{\underline{0.349 \text{ ft}^2}}$$

h. Area of orifice opening, A_2

$$A_2 = \frac{\pi (d_2)^2}{4} = \frac{\pi}{4} \left(4 \text{ in} \times \frac{1 \text{ ft}}{12 \text{ in}}\right)^2 = \frac{\pi}{4} \times \frac{1}{9}$$

$$= \underline{\underline{0.0873 \text{ ft}^2}}$$

i. Ratio of absolute pressures, $\frac{p_2}{p_1}$

$$\frac{p_2}{p_1} = \frac{2067.75}{2102.6} = \underline{\underline{0.983}}$$

j. Compressibility factor, Y

$$Y = (\text{from Fig. 7-15})$$

$$= 0.990$$

k. System Constant, C_s (see page 190)

$$C_s = \frac{C_d \ A_2 \ \sqrt{2g}}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}}$$

$$= \frac{0.67 \times 0.0873 \text{ ft}^2 \times \sqrt{64.4 \ \frac{\text{ft}}{\text{sec}^2}}}{\sqrt{1 - \left(\frac{0.0873}{0.349}\right)^2}}$$

$$= \frac{0.0591 \times 8.04}{\sqrt{1 - (0.25)^2}}$$

$$= \frac{0.475}{\sqrt{1-0.0625}} = \frac{0.475}{\sqrt{0.9375}} = \frac{0.475}{0.97}$$

$$= \underline{\underline{0.49}} \; \frac{(ft)^{5/2}}{sec}$$

l. Volumetric Flow Rate, Q

$$Q_{act} = C_s \; Y \sqrt{\frac{P_1 - P_2}{\rho_1}} \qquad (128)$$

$$= 0.49 \; \frac{ft^{5/2}}{sec} \times 0.99 \sqrt{34.85 \; \frac{lbs}{ft^2} \times \frac{ft^3}{0.0736 \; lbs}}$$

$$= 0.49 \times 0.99 \times \sqrt{474} \; \frac{ft^3}{sec}$$

$$= 0.485 \times 21.8 = 10.6 \; \frac{ft^3}{sec}$$

$$= 10.6 \; \frac{ft^3}{sec} \times \frac{60 \; sec}{min} = 636 \; \frac{ft^3}{min}$$

m. Weight Rate of Flow, m

$$m = \rho_1 \; Q = 0.0736 \frac{lbs}{ft^3} \times 10.6 \; \frac{ft^3}{sec}$$

$$= 0.78 \; \frac{lbs}{sec}$$

$$= 0.78 \frac{lbs}{sec} \times 60 \; \frac{sec}{min} = \underline{\underline{46.8 \; \frac{lbs}{min}}}$$

APPENDIX A

SYMBOLS AND NOMENCLATURE

Symbol	Definition	Typical Units	Reference
A	Area - - - - - - - - - - - - - - -	in.^{2}, ft.^{2}	———
A_c	Cross-sectional area - - - - - -	in.^{2}, ft.^{2}	———
A_n	Net duct flow area - - - - - - - -	in.^{2}, ft.^{2}	p. 25
A_o	Gross duct flow area- - - - - - -	in.^{2}, ft.^{2}	p. 25
A_r	Area of reduced section - - - - -	ft.^{2}	p. 185
A_s	Surface area - - - - - - - - - -	in.^{2}, ft.^{2}	———
B	Impeller width - - - - - - - - - -	in.	p. 40
B_f	Flow factor - - - - - - - - - - - -	———	p. 181
b	width of annular space - - - - - -	in.	p. 130-131
C_c	Calibration coefficient- - - - - -	———	p. 179
C_d	Coefficient of discharge - - - - -	———	p. 186
C_s	System constant - - - - - - - - -	———	p. 190
c_p	Specific heat at constant pressure-	$\frac{\text{watt-min.}}{\text{lb. -}^{o}\text{C}}$	———
c_v	Specific heat at constant volume -	$\frac{\text{watt-min.}}{\text{lb. -}^{o}\text{C}}$	———
D, d	Diameter (distance) - - - - - - -	in., ft.	———
D_e	Equivalent diameter - - - - - - - -	in., ft.	p. 71
ELSD	Equivalent length of straight duct -	———	p. 149
e	efficiency - - - - - - - - - - - - -	———	———
F	Configuration factor, crossflow cooling - - - - - - - - - - - - - -	———	p. 21-22
F_a	Configuration factor, radiation - -	———	p. 6

Symbol	Definition	Typical Units	Reference
F_e	Emissivity factor, radiation - - - -	——	p. 6
f	Friction factor- - - - - - - - - - - -	——	p. 45-48
G	Mass velocity of flow $= V\rho$ - - -	$\frac{\text{lbs.}}{\text{min.-ft.}^2}$	——
$\hat{G}$	Intermediate step in nomographs of Fig. 6-33 and Fig. 6-43. - - - -	——	p. 118-119, 144
Gr	Grashof number $= \frac{g\beta\,\Delta t\,L^3\rho^2}{\mu^2}$	——	p. 10
g	Acceleration due to gravity $= 32.2 \frac{\text{ft.}}{\text{sec.}^2}$	——	——
$\hat{H}$	Intermediate step in nomographs of Fig. 6-34 and Fig. 6-43. - - - - -	——	p. 118-119, 144
H	Head (pressure head) (height of a column of fluid) - - - - - - - - - -	ft. of fluid	——
H_s	Static head - - - - - - - - - - - -	ft. of fluid	——
H_v	Velocity head - - - - - - - - - - -	ft. of fluid	——
H_t	Total head $H_t = H_s + H_v$ - - - -	ft. of fluid	——
H_L	Head loss - - - - - - - - - - - -	ft. of fluid	p. 45-46
H_c	Head loss due to sudden contraction	ft. of fluid	p. 55
H_e	Head loss due to sudden expansion	ft. of fluid	p. 55-56
HP	Horsepower - - - - - - - - - - -	——	——
HR	Hydraulic Radius - - - - - - - - -	ft., in.	p. 58-59
h	Overall coefficient of heat transfer	$\frac{\text{watt}}{\text{in.}^2\text{-}^\circ\text{C}}$	——
h_c	Coefficient of convective heat transfer - - - - - - - - - - - - -	$\frac{\text{watt}}{\text{in.}^2\text{-}^\circ\text{C}}$	——

APPENDIX A (Contd.)

Symbol	Definition	Typical Units	Reference
h_r	Coefficient of radiant heat transfer -	$\frac{\text{watts}}{\text{in.}^2\text{-}^oC}$	———
K	Loss coefficient - - - - - - - - - -	———	p. 53-54
k	Thermal conductivity - - - - - - -	$\frac{\text{watts-in.}}{\text{in.}^2\text{-}^oC}$	———
k_c	Specific heat ratio $= \frac{c_p}{c_v}$ - - - - - - -	———	p. 188
L	Length, (characteristic length) - - -	in., ft.	———
L_e	Equivalent length - - - - - - - - - -	in., ft.	p. 53-54
L/D	Length-to-diameter ratio- - - - - - -	———	p. 13
l_e	Equivalent length - - - - - - - - - -	in., ft.	p. 27
M	Velocity of approach factor - - - - - -	———	p. 181, 187
m	Weight rate of flow- - - - - - - - - -	$\frac{\text{lbs.}}{\text{min.}}$	———
N	Rotational speed- - - - - - - - - - -	rpm.	———
N_s	Specific speed - - - - - - - - - - - -	———	p. 42
Nu	Nusselt number $= \frac{h_c L}{k} = \frac{h_c D}{k}$ - -	———	p. 10-11
P	Power - - - - - - - - - - - - - - - - -	HP.	———
Pr	Prandtl number $= \frac{c_p \mu}{k}$ - - - - - -	———	p. 10-11
p	Pressure - - - - - - - - - - - - - -	$\frac{\text{lbs.}}{\text{ft.}^2}$	———
p_s	Static pressure - - - - - - - - - - -	in. H_2O	p. 35, 167
p_v	Velocity pressure - - - - - - - - - -	in. H_2O	p. 35, 167
p_t	Total pressure - - - - - - - - - - - -	in. H_2O	p. 36, 167
Q	Volumetric flowrate - - - - - - - - -	$\frac{\text{ft.}^3}{\text{min.}}$	———
q	Rate of heat transfer, amount of heat dissipated - - - - - - - - - -	watts	———

APPENDIX A (Contd.)

Symbol	Definition	Typical Units	Reference
q_c	Rate of convective heat transfer - - -	watts	———
q_r	Rate of radiant heat transfer - - - -	watts	———
R	Gas constant (air, $R = 53.3 \frac{ft.}{^oR}$) - - -	$\frac{ft.}{^oR}$	p. 190
Re	Reynolds number $= \frac{VD\rho}{\mu} = \frac{VD}{\nu} = \frac{DG}{\mu}$	———	p. 10-11
Re_o	Reynolds number based on gross duct flow area and prism side dimension -	———	p. 25
r	Radius - - - - - - - - - - - - - - -	ft., in.	———
S_T	Space parameter, crossflow cooling $= \frac{X_T}{D}$	———	p. 20
S_L	Space parameter, crossflow cooling $= \frac{X_L}{D}$	———	p. 20
SG	Specific gravity - - - - - - - - - - -	———	p. 167-169
T	Absolute temperature - - - - - - - - -	^oK, ^oR	———
t	Temperature - - - - - - - - - - - - -	^oC, ^oF	———
t_b	Bulk temperature (of a fluid) - - - - -	^oC	———
t_f	Film temperature $= \frac{t_s + t_b}{2}$ - - - - -	^oC	———
t_s	Surface temperature - - - - - - - - -	^oC	———
t_w	Wall temperature - - - - - - - - - - -	^oC	p. 12
V	Velocity - - - - - - - - - - - - - - -	$\frac{ft.}{min.}$	———
V_{av}	Average velocity - - - - - - - - - - -	$\frac{ft.}{min.}$	———
V_o	Free stream velocity - - - - - - - - -	$\frac{ft.}{min.}$	p. 112
VD"	Velocity diameter product used with Moody Diagram, Fig. 5-1 - - - - - - -	———	p. 47, 52
v	Specific volume - - - - - - - - - - -	$\frac{ft.^3}{lb.}$	p. 51

Symbol	Definition	Typical Units	Reference
W	Width - - - - - - - - - - - - - - -	in., ft.	———
X_a	Geometrical factor of the form $= \left(1 + \frac{5A_c}{A_o}\right)$	———	p. 144
X_D	Intermediate step in nomograph of Fig. 6-41 - - - - - - - - - -	———	p. 138-A
X_d	Geometrical factor of the form $= \left[\left(\frac{D_e}{A_s}\right)^{n-1} \cdot \left(\frac{A_c}{A_s}\right)\right]$	———	p. 117, 143
X_L	Longitudinal distance between cylinders in crossflow - - - - -	in.	p. 19-20
X_{st}	Geometrical factor of the form $= \left[1 + \frac{1}{\sqrt{S_T}}\right]^n$	———	p. 117
X_T	Transverse distance between cylinders in crossflow - - - - - - -	in.	p. 19-20
X_t	Temperature factor of the form $= \left[\frac{\mu_f}{(k_f)^n} \cdot \frac{1}{(\Delta\ t_s)^n}\right]$	———	p. 117, 143
Y	Compressibility factor - - - - -	———	p. 181,188,189
Z	Heat dissipation factor - - - - -	———	p. 26-27
z	Height (above a datum line) - - -	ft.	p. 37
β	Volumetric expansion coefficient	$1/^{o}C$	p. 9
Δ	Difference, (e.g. temperature difference - Δ t) - - - - - - - - -	———	———
ϵ	Absolute roughness - - - - - - -	ft.	p. 46, 49
ϵ/D	Relative roughness - - - - - - -	———	p. 46, 49
σ_s	Stefan-Boltzman Constant $= 0.0037 \times 10^{-8} \frac{\text{watt}}{\text{in.}^2 - {}^{o}K^4}$ - -	———	p. 6

APPENDIX A (Contd.)

Symbol	Definition	Typical Units	Reference
θ	Angle - - - - - - - - - - - - - -	degrees	———
ρ	Density - - - - - - - - - - - - - -	$\frac{\text{lbs.}}{\text{ft.}^3}$	———
ϕ	Function of. . . . - - - - - - - - -	———	———
μ	Absolute or dynamic viscosity - -	$\frac{\text{lb.}}{\text{ft.-hr.}}$	———
Σ	Summation - - - - - - - - - - -	———	———
ν	Kinematic viscosity - - - - - - -	$\frac{\text{ft.}^2}{\text{hr.}}$	———
Δt_s	$\Delta t_s = t_s - t_b$ - - - -	^oC	———
Δt_b	$\Delta t_b = t_{b1} - t_{b2}$ - - - -	^oC	———

Note: - Subscripts "f" and "b" on a property refer to that property being evaluated at the film or bulk temperature, respectively, of a fluid.

APPENDIX B

LIST OF CONVERSION FACTORS

k - thermal conductivity

english system	cgs. system	hybrid system
$1.0 \frac{\text{Btu-ft.}}{\text{hr.-ft.}^2\text{-}^\circ\text{F}}$ =	$0.004134 \frac{\text{cal.-cm.}}{\text{sec.-cm.}^2\text{-}^\circ\text{C}}$ =	$0.044 \frac{\text{watt-in.}}{\text{in.}^2\text{-}^\circ\text{C}}$

c_p - specific heat

$$1.0 \frac{\text{Btu.}}{\text{lb.-}^\circ\text{F}} = 1.012 \frac{\text{cal.}}{\text{gm-}^\circ\text{C}} = 31.6 \frac{\text{watt-min.}}{\text{lb.-}^\circ\text{C}}$$

μ - viscosity

$$1.0 \text{ centipoises} = 2.42 \frac{\text{lb.}}{\text{hr.-ft.}}$$

$$1.0 \frac{\text{Btu.}}{\text{hr.}} = 0.293 \text{ watts}$$

$$1.0 \frac{\text{Btu.}}{\text{min.}} = 17.58 \text{ watts}$$

$$1.0 \frac{\text{calories}}{\text{hr.}} = 0.001162 \text{ watts}$$

$$1.0 \frac{\text{ergs}}{\text{hr.}} = 2.78 \times 10^{-11} \text{ watts}$$

$$1.0 \frac{\text{joules}}{\text{hr.}} = 2.78 \times 10^{-4} \text{ watts}$$

APPENDIX C

LIST OF ASSOCIATED CORNELL AERONAUTICAL LABORATORY REPORTS

Description	Report Number	Date of Issue	BuShips Contract No.
Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment	HF-710-D-10 NAVSHIPS 900, 189	3 March 1952	NObsr-49228
Manual of Standard Temperature Measuring Techniques, Units, and Terminology for Miniaturized Electronic Equipment	HF-845-D-2 NAVSHIPS 900, 187	1 June 1953	NObsr-63043
A Guide Manual of Cooling Methods for Electronic Equipment	HF-710-D-16 NAVSHIPS 900, 190	April 1954	NObsr-49228
Design Manual of Natural Methods of Cooling Electronic Equipment	HF-845-D-8 NAVSHIPS 900, 192	Nov. 1956	NObsr-63043
Design Manual of Methods of Liquid Cooling Electronic Equipment	HF-845-D-9	Scheduled for June, 1958	NObsr-63043
The Thermatron	HF-1053-D-3	May 1957	NObsr-72531

APPENDIX D

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APPENDIX E

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Messrs. James Brush and Rodney Hall, Bureau of Ships, initiated and directed this program.

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Mr. Robert Scott, Asst. Res. Mech. Engr., prepared sections of the design and duct work chapters and correlated the test data.

Mr. Richard Wrobel, Asst. Res. Elec. Engr., prepared the pressure losses and measurements chapters, sections of the design chapter, correlated test data, edited, proofread, and dimensionally analyzed this Manual.

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Mr. Oren Landis, Elec. Engr. Asst., constructed apparatus, instrumented and performed the majority of the experiments mentioned herein.

APPENDIX F

TABLE XV

PROPERTIES OF AIR*

Temp. °F.	Temp. °C.	** c_p Specific Heat $\frac{Btu}{lb.-^oF}$	*** ρ Density $\frac{lb.}{cu. ft.}$	μ Viscosity $\frac{lb.}{ft.-hr.}$	k Thermal Conduct. $\frac{Btu}{hr.-ft.-^oF}$	$\frac{c_p \mu}{k}$ Prandtl No.	β Coeff. of Vol. Expan. $\frac{1}{^oR}$	*** (a)10^{-6} Free Conv. Modulus $\frac{1}{cu. ft.-^oF}$
-50	-46.0	0.239	0.0968	0.036	0.0116	0.74	0.00244	5.46
0	-17.8	0.239	0.0863	0.040	0.0132	0.72	0.00217	3.00
50	10.0	0.240	0.0779	0.043	0.0145	0.71	0.00196	1.81
100	37.8	0.240	0.0708	0.046	0.0158	0.70	0.00179	1.20
150	65.6	0.241	0.0651	0.049	0.0170	0.70	0.00164	0.82
200	93.3	0.241	0.0601	0.052	0.0182	0.69	0.00152	0.58
250	121.1	0.242	0.0559	0.055	0.0192	0.68	0.00141	0.41
300	148.9	0.242	0.0522	0.058	0.0204	0.68	0.00132	0.31
350	176.7	0.243	0.0490	0.060	0.0216	0.68	0.00123	0.23
400	204.4	0.245	0.0461	0.062	0.0227	0.67	0.00116	0.18
450	232.2	0.246	0.0436	0.065	0.0239	0.67	0.00110	0.14
500	260.0	0.247	0.0413	0.067	0.0250	0.66	0.00104	0.11
550	288.0	0.249	0.0393	0.070	0.0264	0.66	0.00099	0.086
600	315.6	0.250	0.0374	0.072	0.0271	0.66	0.00094	0.069
650	343.3	0.252	0.0358	0.074	0.0282	0.66	0.00090	0.055
700	371.1	0.253	0.0342	0.076	0.0291	0.66	0.00086	0.044

* Table derived mainly from Ref. 18

** Specific heat at constant pressure

*** Density and convection modulus for atmospheric pressure (29.92 in. Hg)

APPENDIX F

TABLE XVI

TOTAL NORMAL EMISSIVITY OF MATERIALS

Metals	Condition	Temperature Range °C	Emissivity
Aluminum	Highly polished	230 - 580	0.039 - 0.057
Aluminum	Commercial sheet	100	0.09
Aluminum	Rough polish	100	0.18
Brass	Polished	100	0.06
Brass	Dull plate	50 - 300	0.22
Copper	Polished	100	0.052
Copper	Oxidized	100	0.70
Molybdenum	Polished	100	0.071
Monel	Oxidized	200	0.41
Monel	Polished	230	0.35
Nickel	Polished	20	0.045
Nickel - silver	Polished	100	0.135
Platinum	Wire	27 - 1200	0.036 - 0.192
Silver	Polished	100	0.052
Stainless Steel	Polished	100	0.074
Alleghany #4	Polished	100	0.13
Alleghany #66	Polished	100	0.11
Type 304(8-18)	Rough, oxidized	215	0.44
Steel Sheet	Oxidized	24	0.80
Tin	Commercial plate	100	0.08
Tungsten	Polished coat	100	0.066
Zinc	Galvanized sheet	100	0.21
Glasses			
Fused quartz	1.96 mm. thick	100 - 500	0.77 - 0.67
Covex D	3.40 mm. thick	100 - 500	0.83 - 0.91
Nonex	1.57 mm. thick	100 - 500	0.84 - 0.82
Pyrex		260 - 540	0.95 - 0.85
Porcelain	Glazed	23	0.92
Paints			
Aluminum	Silicone vehicle	260	0.29
Aluminum	Lacquer	100	0.39 - 0.52
Radiator, white	Clean	100	0.79
Radiator, cream	Clean	100	0.77
Radiator, black	Clean	100	0.84
Radiator, bronze	Clean	100	0.51
Lacquer on iron	any color	25	0.87 - 0.91
Oil paints	any color	100	0.92 - 0.96

(Reference No. 20)

APPENDIX F

TABLE XVII

TEMPERATURE CONVERSIONS

°C	°F
−200	−328.0
−201	−329.8
−202	−331.6
−203	−333.4
−204	−335.2
−205	−337.0
−206	−338.8
−207	−340.6
−208	−342.4
−209	−344.2
−210	−346.0
−211	−347.8
−212	−349.6
−213	−351.4
−214	−353.2
−215	−355.0
−216	−356.8
−217	−358.6
−218	−360.4
−219	−362.2
−220	−364.0
−221	−365.8
−222	−367.6
−223	−369.4
−224	−371.2
−225	−373.0
−226	−374.8
−227	−376.6
−228	−378.4
−229	−380.2
−230	−382.0
−231	−383.8
−232	−385.6
−233	−387.4
−234	−389.2
−235	−391.0
−236	−392.8
−237	−394.6
−238	−396.4
−239	−398.2
−240	−400.0
−241	−401.8
−242	−403.6
−243	−405.4
−244	−407.2
−245	−409.0
−246	−410.8
−247	−412.6
−248	−414.4
−249	−416.2
−250	−418.0
−251	−419.8
−252	−421.6
−253	−423.4
−254	−425.2
−255	−427.0
−256	−428.8
−257	−430.6
−258	−432.4
−259	−434.2
−260	−436.0
−261	−437.8
−262	−439.6
−263	−441.4
−264	−443.2
−265	−445.0
−266	−446.8
−267	−448.6
−268	−450.4
−269	−452.2
−270	−454.0
−271	−455.8
−272	−457.6
−273	−459.4
−273.2	**−459.7**

Absolute Zero

Formulas:

$C = \frac{5}{9}(F - 32)$

$F = \frac{9}{5}C + 32$

°C	°F
−100	−148.0
−101	−149.8
−102	−151.6
−103	−153.4
−104	−155.2
−105	−157.0
−106	−158.8
−107	−160.6
−108	−162.4
−109	−164.2
−110	−166.0
−111	−167.8
−112	−169.6
−113	−171.4
−114	−173.2
−115	−175.0
−116	−176.8
−117	−178.6
−118	−180.4
−119	−182.2
−120	−184.0
−121	−185.8
−122	−187.6
−123	−189.4
−124	−191.2
−125	−193.0
−126	−194.8
−127	−196.6
−128	−198.4
−129	−200.2
−130	−202.0
−131	−203.8
−132	−205.6
−133	−207.4
−134	−209.2
−135	−211.0
−136	−212.8
−137	−214.6
−138	−216.4
−139	−218.2
−140	−220.0
−141	−221.8
−142	−223.6
−143	−225.4
−144	−227.2
−145	−229.0
−146	−230.8
−147	−232.6
−148	−234.4
−149	−236.2
−150	−238.0
−151	−239.8
−152	−241.6
−153	−243.4
−154	−245.2
−155	−247.0
−156	−248.8
−157	−250.6
−158	−252.4
−159	−254.2
−160	−256.0
−161	−257.8
−162	−259.6
−163	−261.4
−164	−263.2
−165	−265.0
−166	−266.8
−167	−268.6
−168	−270.4
−169	−272.2
−170	−274.0
−171	−275.8
−172	−277.6
−173	−279.4
−174	−281.2
−175	−283.0
−176	−284.8
−177	−286.6
−178	−288.4
−179	−290.2
−180	−292.0
−181	−293.8
−182	−295.6
−183	−297.4
−184	−299.2
−185	−301.0
−186	−302.8
−187	−304.6
−188	−306.4
−189	−308.2
−190	−310.0
−191	−311.8
−192	−313.6
−193	−315.4
−194	−317.2
−195	−319.0
−196	−320.8
−197	−322.6
−198	−324.4
−199	−326.2

°C	°F
0	32.0
−1	30.2
−2	28.4
−3	26.6
−4	24.8
−5	23.0
−6	21.2
−7	19.4
−8	17.6
−9	15.8
−10	14.0
−11	12.2
−12	10.4
−13	8.6
−14	6.8
−15	5.0
−16	3.2
−17	1.4
−18	−0.4
−19	−2.2
−20	−4.0
−21	−5.8
−22	−7.6
−23	−9.4
−24	−11.2
−25	−13.0
−26	−14.8
−27	−16.6
−28	−18.4
−29	−20.2
−30	−22.0
−31	−23.8
−32	−25.6
−33	−27.4
−34	−29.2
−35	−31.0
−36	−32.8
−37	−34.6
−38	−36.4
−39	−38.2
−40	−40.0
−41	−41.8
−42	−43.6
−43	−45.4
−44	−47.2
−45	−49.0
−46	−50.8
−47	−52.6
−48	−54.4
−49	−56.2
−50	−58.0
−51	−59.8
−52	−61.6
−53	−63.4
−54	−65.2
−55	−67.0
−56	−68.8
−57	−70.6
−58	−72.4
−59	−74.2
−60	−76.0
−61	−77.8
−62	−79.6
−63	−81.4
−64	−83.2
−65	−85.0
−66	−86.8
−67	−88.6
−68	−90.4
−69	−92.2
−70	−94.0
−71	−95.8
−72	−97.6
−73	−99.4
−74	−101.2
−75	−103.0
−76	−104.8
−77	−106.6
−78	−108.4
−79	−110.2
−80	−112.0
−81	−113.8
−82	−115.6
−83	−117.4
−84	−119.2
−85	−121.0
−86	−122.8
−87	−124.6
−88	−126.4
−89	−128.2
−90	−130.0
−91	−131.8
−92	−133.6
−93	−135.4
−94	−137.2
−95	−139.0
−96	−140.8
−97	−142.6
−98	−144.4
−99	−146.2

°C	°F
0	32.0
1	33.8
2	35.6
3	37.4
4	39.2
5	41.0
6	42.8
7	44.6
8	46.4
9	48.2
10	50.0
11	51.8
12	53.6
13	55.4
14	57.2
15	59.0
16	60.8
17	62.6
18	64.4
19	66.2
20	68.0
21	69.8
22	71.6
23	73.4
24	75.2
25	77.0
26	78.8
27	80.6
28	82.4
29	84.2
30	86.0
31	87.8
32	89.6
33	91.4
34	93.2
35	95.0
36	96.8
37	98.6
38	100.4
39	102.2
40	104.0
41	105.8
42	107.6
43	109.4
44	111.2
45	113.0
46	114.8
47	116.6
48	118.4
49	120.2
50	122.0
51	123.8
52	125.6
53	127.4
54	129.2
55	131.0
56	132.8
57	134.6
58	136.4
59	138.2
60	140.0
61	141.8
62	143.6
63	145.4
64	147.2
65	149.0
66	150.8
67	152.6
68	154.4
69	156.2
70	158.0
71	159.8
72	161.6
73	163.4
74	165.2
75	167.0
76	168.8
77	170.6
78	172.4
79	174.2
80	176.0
81	177.8
82	179.6
83	181.4
84	183.2
85	185.0
86	186.8
87	188.6
88	190.4
89	192.2
90	194.0
91	195.8
92	197.6
93	199.4
94	201.2
95	203.0
96	204.8
97	206.6
98	208.4
99	210.2

°C	°F
100	212.0
101	213.8
102	215.6
103	217.4
104	219.2
105	221.0
106	222.8
107	224.6
108	226.4
109	228.2
110	230.0
111	231.8
112	233.6
113	235.4
114	237.2
115	239.0
116	240.8
117	242.6
118	244.4
119	246.2
120	248.0
121	249.8
122	251.6
123	253.4
124	255.2
125	257.0
126	258.8
127	260.6
128	262.4
129	264.2
130	266.0
131	267.8
132	269.6
133	271.4
134	273.2
135	275.0
136	276.8
137	278.6
138	280.4
139	282.2
140	284.0
141	285.8
142	287.6
143	289.4
144	291.2
145	293.0
146	294.8
147	296.6
148	298.4
149	300.2
150	302.0
151	303.8
152	305.6
153	307.4
154	309.2
155	311.0
156	312.8
157	314.6
158	316.4
159	318.2
160	320.0
161	321.8
162	323.6
163	325.4
164	327.2
165	329.0
166	330.8
167	332.6
168	334.4
169	336.2
170	338.0
171	339.8
172	341.6
173	343.4
174	345.2
175	347.0
176	348.8
177	350.6
178	352.4
179	354.2
180	356.0
181	357.8
182	359.6
183	361.4
184	363.2
185	365.0
186	366.8
187	368.6
188	370.4
189	372.2
190	374.0
191	375.8
192	377.6
193	379.4
194	381.2
195	383.0
196	384.8
197	386.6
198	388.4
199	390.2

°C	°F
200	392.0
201	393.8
202	395.6
203	397.4
204	399.2
205	401.0
206	402.8
207	404.6
208	406.4
209	408.2
210	410.0
211	411.8
212	413.6
213	415.4
214	417.2
215	419.0
216	420.8
217	422.6
218	424.4
219	426.2
220	428.0
221	429.8
222	431.6
223	433.4
224	435.2
225	437.0
226	438.8
227	440.6
228	442.4
229	444.2
230	446.0
231	447.8
232	449.6
233	451.4
234	453.2
235	455.0
236	456.8
237	458.6
238	460.4
239	462.2
240	464.0
241	465.8
242	467.6
243	469.4
244	471.2
245	473.0
246	474.8
247	476.6
248	478.4
249	480.2
250	482.0
251	483.8
252	485.6
253	487.4
254	489.2
255	491.0
256	492.8
257	494.6
258	496.4
259	498.2
260	500.0
261	501.8
262	503.6
263	505.4
264	507.2
265	509.0
266	510.8
267	512.6
268	514.4
269	516.2
270	518.0
271	519.8
272	521.6
273	523.4
274	525.2
275	527.0
276	528.8
277	530.6
278	532.4
279	534.2
280	536.0
281	537.8
282	539.6
283	541.4
284	543.2
285	545.0
286	546.8
287	548.6
288	550.4
289	552.2
290	554.0
291	555.8
292	557.6
293	559.4
294	561.2
295	563.0
296	564.8
297	566.6
298	568.4
299	570.2

°C	°F
300	572.0
301	573.8
302	575.6
303	577.4
304	579.2
305	581.0
306	582.8
307	584.6
308	586.4
309	588.2
310	590.0
311	591.8
312	593.6
313	595.4
314	597.2
315	599.0
316	600.8
317	602.6
318	604.4
319	606.2
320	608.0
321	609.8
322	611.6
323	613.4
324	615.2
325	617.0
326	618.8
327	620.6
328	622.4
329	624.2
330	626.0
331	627.8
332	629.6
333	631.4
334	633.2
335	635.0
336	636.8
337	638.6
338	640.4
339	642.2
340	644.0
341	645.8
342	647.6
343	649.4
344	651.2
345	653.0
346	654.8
347	656.6
348	658.4
349	660.2
350	662.0
351	663.8
352	665.6
353	667.4
354	669.2
355	671.0
356	672.8
357	674.6
358	676.4
359	678.2
360	680.0
361	681.8
362	683.6
363	685.4
364	687.2
365	689.0
366	690.8
367	692.6
368	694.4
369	696.2
370	698.0
371	699.8
372	701.6
373	703.4
374	705.2
375	707.0
376	708.8
377	710.6
378	712.4
379	714.2
380	716.0
381	717.8
382	719.6
383	721.4
384	723.2
385	725.0
386	726.8
387	728.6
388	730.4
389	732.2
390	734.0
391	735.8
392	737.6
393	739.4
394	741.2
395	743.0
396	744.8
397	746.6
398	748.4
399	750.2

°C	°F
400	752.0
401	753.8
402	755.6
403	757.4
404	759.2
405	761.0
406	762.8
407	764.6
408	766.4
409	768.2
410	770.0
411	771.8
412	773.6
413	775.4
414	777.2
415	779.0
416	780.8
417	782.6
418	784.4
419	786.2
420	788.0
421	789.8
422	791.6
423	793.4
424	795.2
425	797.0
426	798.8
427	800.6
428	802.4
429	804.2
430	806.0
431	807.8
432	809.6
433	811.4
434	813.2
435	815.0
436	816.8
437	818.6
438	820.4
439	822.2
440	824.0
441	825.8
442	827.6
443	829.4
444	831.2
445	833.0
446	834.8
447	836.6
448	838.4
449	840.2
450	842.0
451	843.8
452	845.6
453	847.4
454	849.2
455	851.0
456	852.8
457	854.6
458	856.4
459	858.2
460	860.0
461	861.8
462	863.6
463	865.4
464	867.2
465	869.0
466	870.8
467	872.6
468	874.4
469	876.2
470	878.0
471	879.8
472	881.6
473	883.4
474	885.2
475	887.0
476	888.8
477	890.6
478	892.4
479	894.2
480	896.0
481	897.8
482	899.6
483	901.4
484	903.2
485	905.0
486	906.8
487	908.6
488	910.4
489	912.2
490	914.0
491	916.8
492	917.6
493	919.4
494	921.2
495	923.0
496	924.8
497	926.6
498	928.4
499	930.2

°C	°F
500	932.0
501	933.8
502	935.6
503	937.4
504	939.2
505	941.0
506	942.8
507	944.6
508	946.4
509	948.2
510	950.0
511	951.8
512	953.6
513	955.4
514	957.2
515	959.0
516	960.8
517	962.6
518	964.4
519	966.2
520	968.0
521	969.8
522	971.6
523	973.4
524	975.2
525	977.0
526	978.8
527	980.6
528	982.4
529	984.2
530	986.0
531	987.8
532	989.6
533	991.4
534	993.2
535	995.0
536	996.8
537	998.6
538	1,000.4
539	1,002.2
540	1,004.0
541	1,005.8
542	1,007.6
543	1,009.4
544	1,011.2
545	1,013.0
546	1,014.8
547	1,016.6
548	1,018.4
549	1,020.2
550	1,022.0
551	1,023.8
552	1,025.6
553	1,027.4
554	1,029.2
555	1,031.0
556	1,032.8
557	1,034.6
558	1,036.4
559	1,038.2
560	1,040.0
561	1,041.8
562	1,043.6
563	1,045.4
564	1,047.2
565	1,049.0
566	1,050.8
567	1,052.6
568	1,054.4
569	1,056.2
570	1,058.0
571	1,059.8
572	1,061.6
573	1,063.4
574	1,065.2
575	1,067.0
576	1,068.8
577	1,070.6
578	1,072.4
579	1,074.2
580	1,076.0
581	1,077.8
582	1,079.6
583	1,081.4
584	1,083.2
585	1,085.0
586	1,086.8
587	1,088.6
588	1,090.4
589	1,092.2
590	1,094.0
591	1,095.8
592	1,097.6
593	1,099.4
594	1,101.2
595	1,103.0
596	1,104.8
597	1,106.6
598	1,108.4
599	1,110.2

°C	°F
600	1,112.0
601	1,113.8
602	1,115.6
603	1,117.4
604	1,119.2
605	1,121.0
606	1,122.8
607	1,124.6
608	1,126.4
609	1,128.2
610	1,130.0
611	1,131.8
612	1,133.6
613	1,135.4
614	1,137.2
615	1,139.0
616	1,140.8
617	1,142.6
618	1,144.4
619	1,146.2
620	1,148.0
621	1,149.8
622	1,151.6
623	1,153.4
624	1,155.2
625	1,157.0
626	1,158.8
627	1,160.6
628	1,162.4
629	1,164.2
630	1,166.0
631	1,167.8
632	1,169.6
633	1,171.4
634	1,173.2
635	1,175.0
636	1,176.8
637	1,178.6
638	1,180.4
639	1,182.2
640	1,184.0
641	1,185.8
642	1,187.6
643	1,189.4
644	1,191.2
645	1,193.0
646	1,194.8
647	1,196.6
648	1,198.4
649	1,200.2
650	1,202.0
651	1,203.8
652	1,205.6
653	1,207.4
654	1,209.2
655	1,211.0
656	1,212.8
657	1,214.6
658	1,216.4
659	1,218.2
660	1,220.0
661	1,221.8
662	1,223.6
663	1,225.4
664	1,227.2
665	1,229.0
666	1,230.8
667	1,232.6
668	1,234.4
669	1,236.2
670	1,238.0
671	1,239.8
672	1,241.6
673	1,243.4
674	1,245.2
675	1,247.0
676	1,248.8
677	1,250.6
678	1,252.4
679	1,254.2
680	1,256.0
681	1,257.8
682	1,259.6
683	1,261.4
684	1,263.2
685	1,265.0
686	1,266.8
687	1,268.6
688	1,270.4
689	1,272.2
690	1,274.0
691	1,275.8
692	1,277.6
693	1,279.4
694	1,281.2
695	1,283.0
696	1,284.8
697	1,286.6
698	1,288.4
699	1,290.2

°C	°F
700	1,292.0
701	1,293.8
702	1,295.6
703	1,297.4
704	1,299.2
705	1,301.0
706	1,302.8
707	1,304.6
708	1,306.4
709	1,308.2
710	1,310.0
711	1,311.8
712	1,313.6
713	1,315.4
714	1,317.2
715	1,319.0
716	1,320.8
717	1,322.6
718	1,324.4
719	1,326.2
720	1,328.0
721	1,329.8
722	1,331.6
723	1,333.4
724	1,335.2
725	1,337.0
726	1,338.8
727	1,340.6
728	1,342.4
729	1,344.2
730	1,346.0
731	1,347.8
732	1,349.6
733	1,351.4
734	1,353.2
735	1,355.0
736	1,356.8
737	1,358.6
738	1,360.4
739	1,362.2
740	1,364.0
741	1,365.8
742	1,367.6
743	1,369.4
744	1,371.2
745	1,373.0
746	1,374.8
747	1,376.6
748	1,378.4
749	1,380.2
750	1,382.0
751	1,383.8
752	1,385.6
753	1,387.4
754	1,389.2
755	1,391.0
756	1,392.8
757	1,394.6
758	1,396.4
759	1,398.2
760	1,400.0
761	1,401.8
762	1,403.6
763	1,405.4
764	1,407.2
765	1,409.0
766	1,410.8
767	1,412.6
768	1,414.4
769	1,416.2
770	1,418.0
771	1,419.8
772	1,421.6
773	1,423.4
774	1,425.2
775	1,427.0
776	1,428.8
777	1,430.6
778	1,432.4
779	1,434.2
780	1,436.0
781	1,437.8
782	1,439.6
783	1,441.4
784	1,443.2
785	1,445.0
786	1,446.8
787	1,448.6
788	1,450.4
789	1,452.2
790	1,454.0
791	1,455.8
792	1,457.6
793	1,459.4
794	1,461.2
795	1,463.0
796	1,464.8
797	1,466.6
798	1,468.4
799	1,470.2

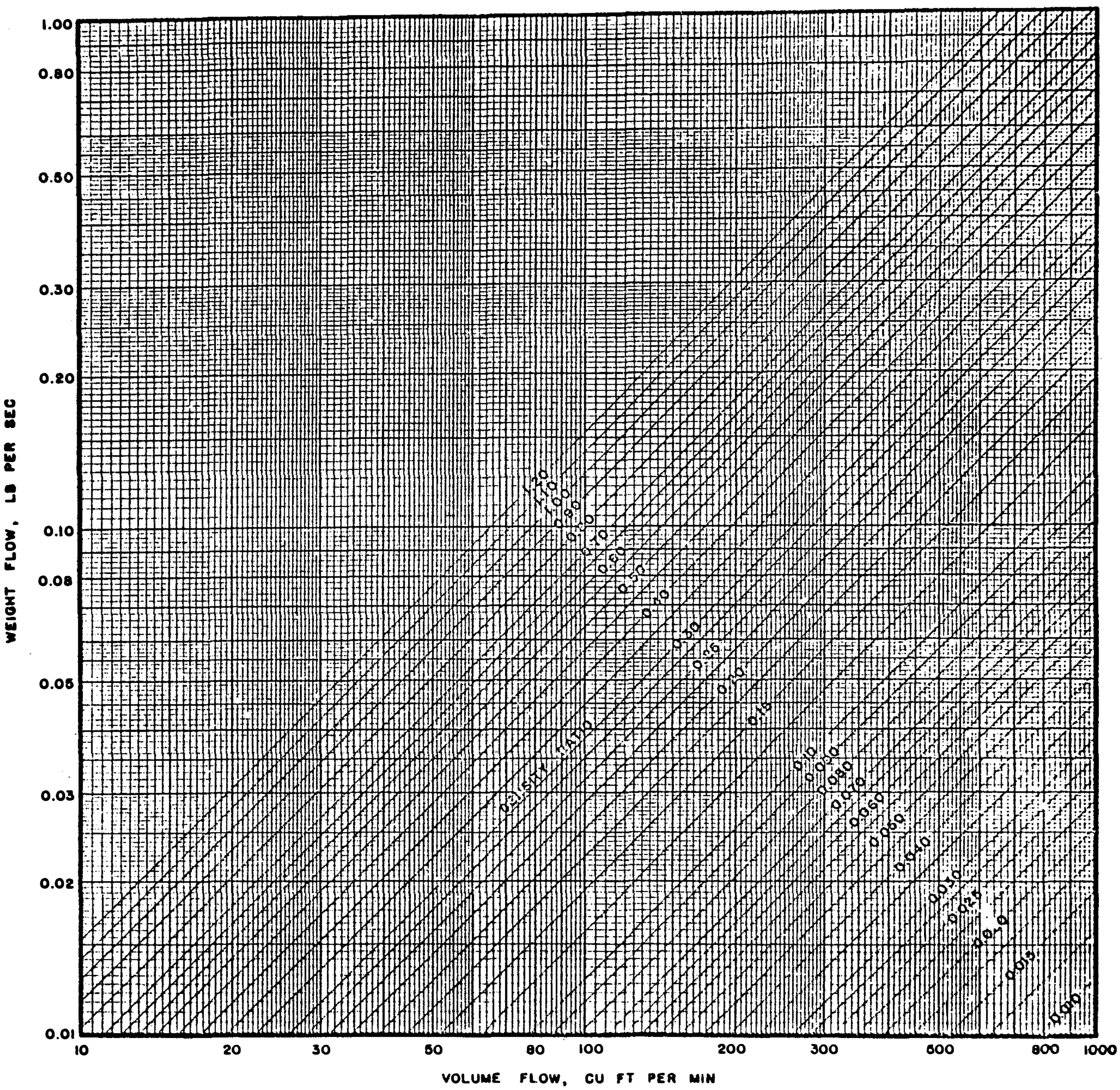


FIG. A-I VOLUME-WEIGHT FLOW CONVERSION CHART FOR AIR AT VARIOUS DENSITY RATIOS

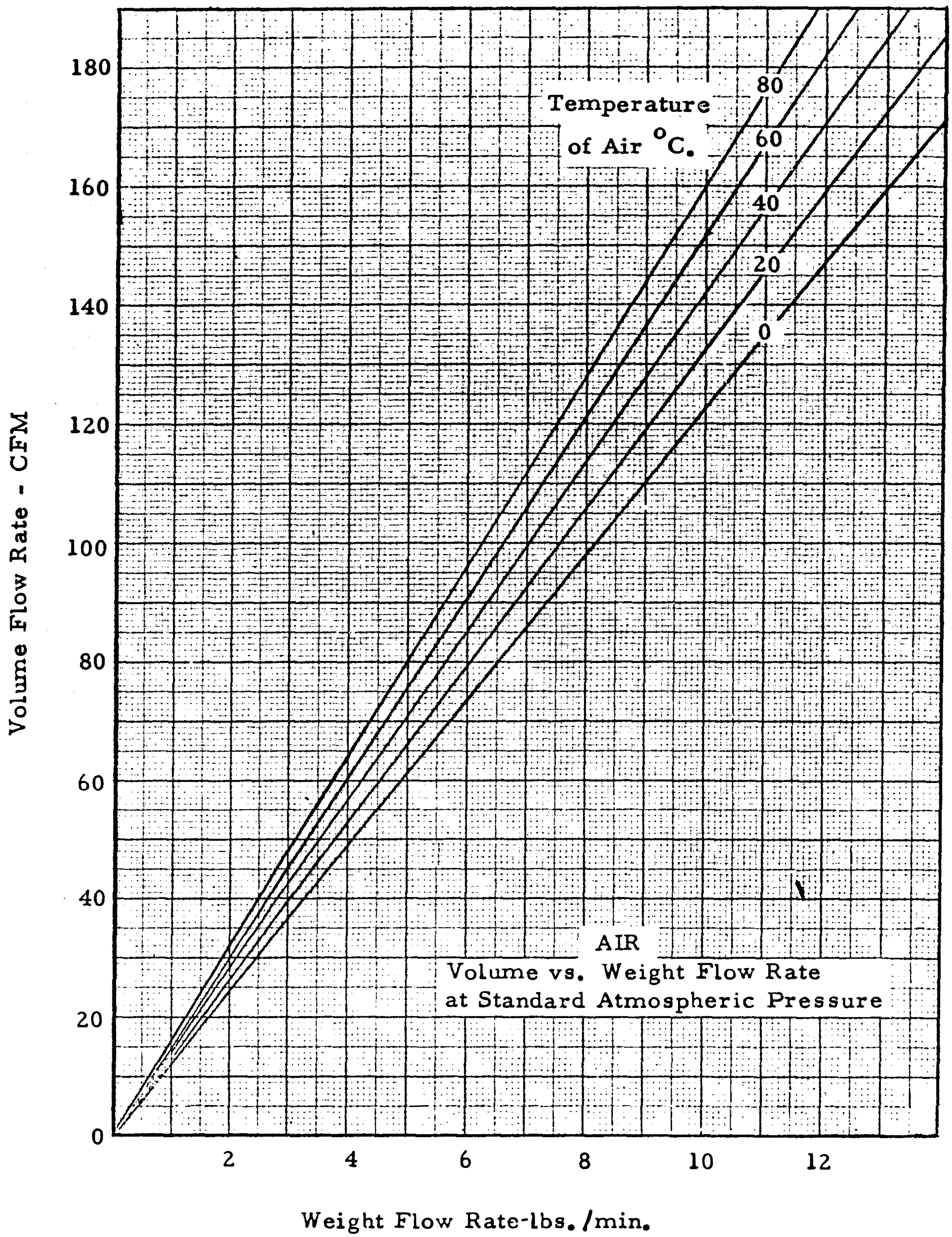


Fig. A-2 Volume-Weight Flow Conversion Chart for Air at Standard Atmospheric Pressure for Various Temperatures.

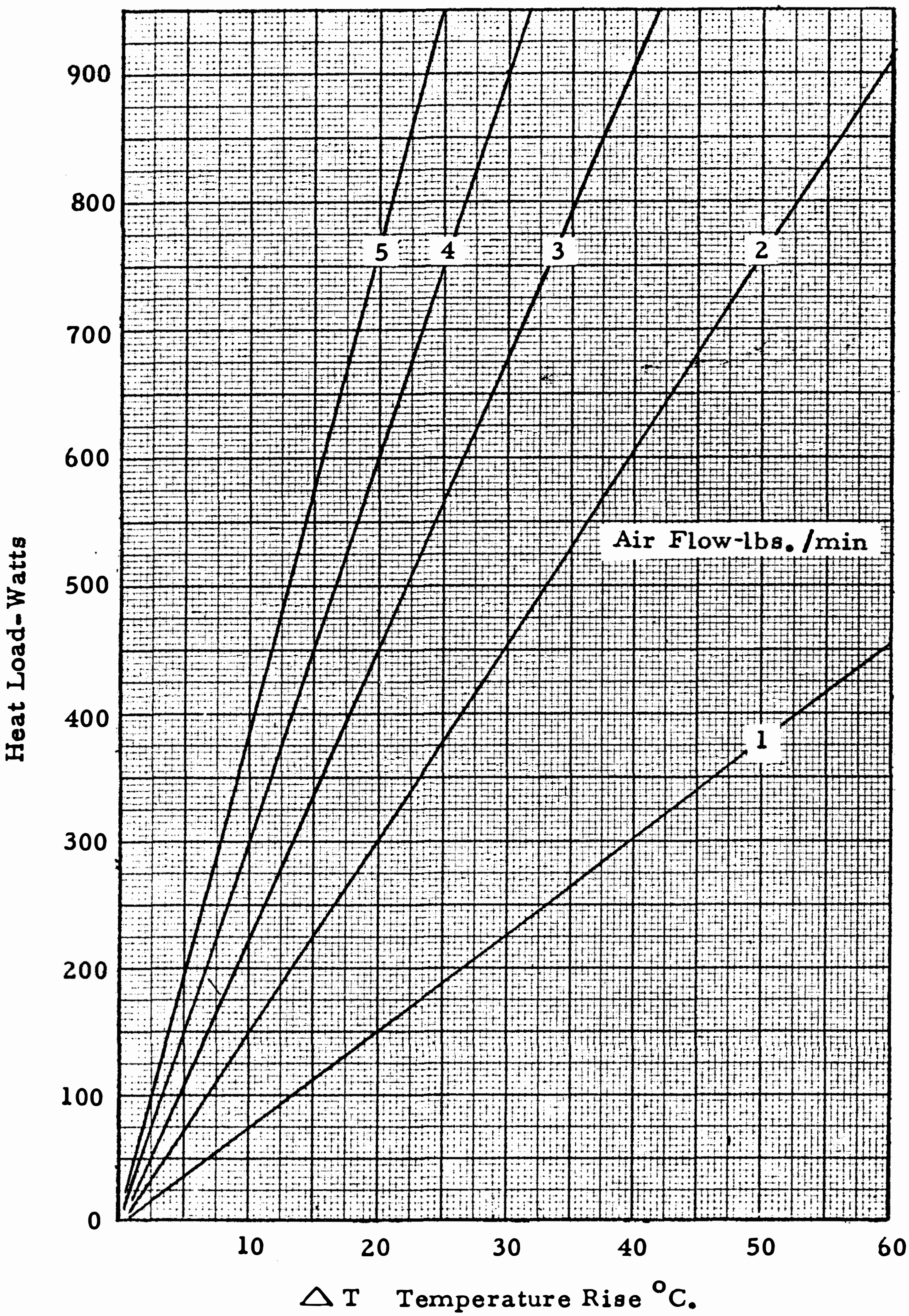


Fig. A-3 Permissible Air Temperature Rises for Various Heat Loads at Given Air Flow Rates.

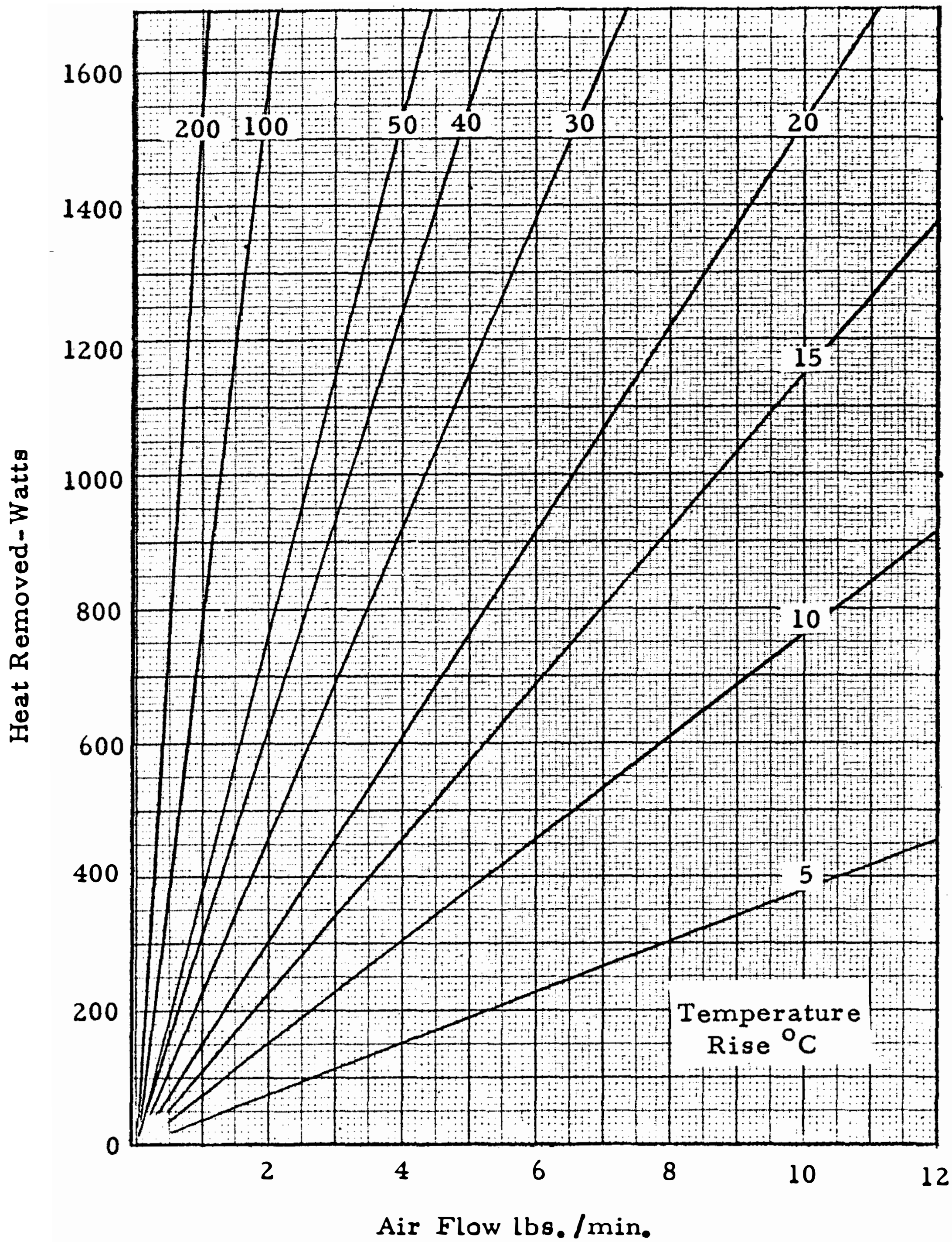


Fig. A-4 Air Flow Required to Remove Various Wattages for Given Air Temperature Rises.

APPENDIX H

SUMMARY OF PHYSICAL INVESTIGATIONS AT C. A. L.

A. Background for h vs Re Curves

In order to arrive at an h vs Re curve, one must of course determine the heat transfer coefficient, h, and the Reynolds Number, Re. h was calculated using the following equation:

$$q = h A_s \triangle t_s$$

or:

$$h = \frac{q}{A_s \triangle t_s}$$

where:

h = heat transfer coefficient, $\frac{\text{watts}}{\text{in.}^2\text{-}^oC}$

q = heat dissipated, watts

A_s = surface area of the part, in.^{2}

$\triangle t_s = t_s - t_b$, surface temperature rise, ^oC.

Each of the quantities listed above was determined in the following manner:

q = total amount of electrical watts converted to heat. (In all of the cases investigated, there was 100% conversion.)

A_s was calculated for each tube, based on a geometrical approximation of said tube; e.g. a miniature tube may be approximated by a cylinder of the same diameter and essentially the same length.

$\triangle t_s$ = the difference between the average incoming air temperature and the average tube envelope temperature. (These two were obtained, generally, by averaging the readings of several thermocouples.)

APPENDIX H (Contd.)

To calculate the Reynolds Number, the following equation was used:

$$Re = \frac{GD_e}{\mu} = \frac{mD_e}{A_c \mu}$$

where:

G = mass velocity, $\frac{\text{lbs.}}{\text{in.}^2\text{-min.}}$

D_e = effective diameter, inches

μ = viscosity, $\frac{\text{lbs.}}{\text{in.-min.}}$

m = weight rate of flow, $\frac{\text{lbs.}}{\text{min.}}$

A_c = net cross-sectional area, in.^2

Each of these quantities was determined in the following manner:

$$G = \frac{m}{A_c}$$

m - measured

A_c - measured, a function of the configuration, i.e. gross duct cross-sectional area minus projected area of tube or tubes.

μ - as found in tables, at film or bulk temperature, depending on configuration.

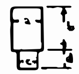
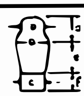
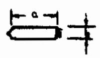
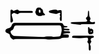
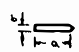
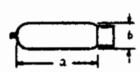
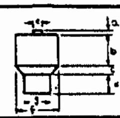
D_e - generally the major diameter of the part (tube) in crossflow. In parallel flow the Hydraulic Radius (HR) was used.

B. Summary of Physical Data

The following table, Table XVIII, presents a brief description of the investigations performed at Cornell Aeronautical Laboratory, for the Forced-Air Cooling Program.

TABLE XVIII

SUMMARY OF PHYSICAL INVESTIGATIONS

h vs. Re Curve	Configuration	Tube Type	D_e inches	Geometrical Approximation	Approximation Dimensions - inches	A_s Square-Inches	Duct Size inches	A_c Square-Inches
Fig. 6-5	Single tube in crossflow	6L6-GBY	1.5		a = 1.5 c = 1.38 b = 2.12 d = 0.75	15.3	4" high by 2.5" wide	5.79
Fig. 6-6	Bank of 27 tubes in cross-flow. 3 longitudinal rows by 9 transverse rows.	4 - 6L6's	average D_e = 1.65	assumed a cylinder	dia. = 1-9/32", ht. = 3-3/4"		4" high by 6.25" wide	average A_c = 10.35
		3 - 6L6-GBY's		same as for Fig. 6-5	a = 1.56 c = 1.38 b = 2.12 d = 0.75	46.8 (3 tubes)		
		20 - 6L6-GAY's			a = 1-3/4 d = 1-1/2 b = 1-3/8 e = 1-7/16 c = 1-5/16 f = 1-1/16	387.6 (20 tubes)		
Fig. 6-7	single tube in crossflow	6AQ5	0.71		a = 2.12 b = 0.71	5.14	2.55" high by 1.75" wide	2.96
Fig. 6-8	single tube in crossflow	12BY7	0.81		a = 2.25 b = 0.81	6.25	2.55" high by 1.75" wide	2.64
Fig. 6-9	Bank of 56 tubes in cross-flow. 4 longitudinal rows by 14 transverse rows (duct, blackened and bonded to chassis.)	6AQ5	0.71	same as for Fig. 6-7	same as for Fig. 6-7	287.	2.5" high by 4.5" wide	5.23
Fig. 6-10	3 tubes in a transverse row, in parallel flow, tube top upstream and tube base upstream	5902	HR = 0.528		a = 1.45 b = 0.376	5.80 (3 tubes)	1-5/32" high by 1-15/16" wide	1.81
Fig. 6-11	3 tubes, crossflow-in-line, (one longitudinal row)	5902	HR = 0.574	same as for Fig. 6-10	same as for Fig. 6 - 10	same as for Fig. 6 - 10	15/16" high by 2-11/16" wide	1.92
Fig. 6-12	3 tubes in crossflow, one transverse row	5902	HR = 0.213	same as for Fig. 6-10	same as for Fig. 6 - 10	same as for Fig. 6 - 10	essentially 1-5/16" high by 1-5/8" wide	0.85
Fig. 6-13	4 tubes, crossflow-in-line (one longitudinal row)	805	2.31		a = 6.0 b = 2.31	191." for four tubes	9.75" high by 3.62" wide	15.9
Fig. 6-14	3 tubes, parallel flow, tubes in shields	4-250A	HR = 0.81		a = 0.5 b = 2.0 c = 0.7 d = 1.125 e = 0.75 f = 3.56 g = 2.5	50.0 per tube	shield diameter = 4-3/8"	5.05 per tube
Fig. 6-15	3 tubes, parallel flow, no flow-confining shield (free flow)	4-250A	HR = 0.32	same as for Fig. 6-14	same as for Fig. 6 - 14	same as for Fig. 6 - 14	the tube has feed-through holes around its base, 5 holes at 5/16" dia.	1.56 per tube

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