

HEAT EXCHANGE INSTITUTE, INC.

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STANDARDS for **DIRECT CONTACT BAROMETRIC AND LOW LEVEL CONDENSERS**

SIXTH EDITION

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Heat Exchange Institute
1300 Sumner Avenue
Cleveland, Ohio 44115-2851

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FOREWORD

The Sixth Edition of this Standard has been developed by the Vacuum Technology Section of the Heat Exchange Institute, Inc. The Standards are presented with the intent to be helpful to those engaged in the writing of specifications and, ultimately, the selection of equipment.

These Standards provide practical information on nomenclature, dimensions, construction, and testing of direct contact barometric fabricated condensers. The Standards for Direct Contact Barometric and Low Level Condensers are continually reviewed at scheduled meetings under the direction of the Vacuum

Technology Section. Suggestions for the improvement of this Standard are welcome and should be sent to:

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In preparation of this Standard, consideration has been given to the work of other organizations, such as The American National Standards Institute, The American Society of Mechanical Engineers, and others. Credit is hereby given to all those whose standards may have been helpful in this work.

1.0 INTRODUCTION

1.1 Purpose

This Standard contains a set of criteria for the specification, construction, and installation of Direct Contact Barometric and Low Level Condensers. It intends to provide a common understanding to which purchasers and manufacturers may make reference.

The guidelines described herein are meant to maximize the usefulness and benefits of Direct Contact Barometric and Low Level Condensers and work to eliminate any difficulties in their supply. The purpose is to provide practical methods by which a purchaser's requirements can be assured.

1.2 Scope

This Standard concentrates on those issues which can be standardized in an agreement between a purchaser and manufacturer of Direct Contact Barometric and Low Level Condensers. The issues are those which set the minimum requirements for construction and installation. No attempt is made to comprehensively describe the sizing, operational behavior, maintenance, or troubleshooting of direct contact condensers. These are issues left to individual manufacturer's determination and purchaser's choice.

2.0 FUNDAMENTALS

2.1 Nomenclature

With the intent of establishing standard terminology, four types of barometric condensers shown in Figs. 1, 2, 3, and 4 will be used. These drawings are merely illustrative for the purpose of indicating names of parts.

2.2 Operating Principles

Direct contact condensers are constructed to effect condensation of steam and other condensable vapors by means of direct contact with the condensing water. There are two principle types, namely, *counterflow* and *parallel flow*.

2.2.1 Counterflow condensers are so constructed that the steam or vapor and the condensing fluid flow in opposite directions.

2.2.2 Parallel flow condensers are so constructed that the steam or vapor and the condensing fluid flow in the same direction.

2.3 Arrangements for Water Removal

Counterflow and parallel flow condensers are further classified according to the method by which the mixture of condensing water and condensed vapor is removed, namely, barometric and low level.

2.3.1 A barometric condenser is elevated to a sufficient height to permit drainage through a tailpipe by gravity alone, without mechanical assistance.

2.3.2 A low level condenser requires a pump for removal of the condensing water/condensate mixture.

2.4 Definitions of Terms

Definitions of terms used in these Standards are given as follows:

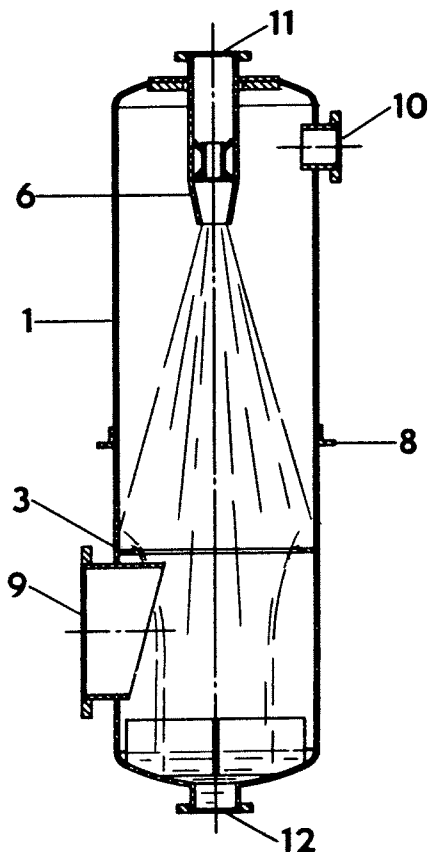
- a) Condenser Load consists of condensable vapors and non-condensable gases, expressed separately in pounds per hour.
- b) Condenser Duty is the total net heat absorbed by the condensing water, expressed in Btu per hour.
- c) Absolute Pressure is the pressure measured from absolute zero, i.e., from an absolute vacuum.

- d) Static Pressure is the pressure measured in the vapor in such manner that no effect on the measurement is produced by the velocity of the vapor.
- e) Suction Pressure is the absolute static pressure prevailing at the vapor inlet of the condenser expressed in pounds per square inch, inches or millimeters of mercury.
- *f) Gas-Vapor Outlet Pressure is the absolute static pressure prevailing at the gas-vapor outlet of the condenser expressed in pounds per square inch, inches or millimeters of mercury.
- g) Inlet Vapor Temperature is the total temperature of the vapor at the condenser inlet.
- h) Inlet Water Temperature is the temperature of the condensing water at the condenser water inlet.
- i) Outlet Water Temperature is the temperature of the condensing water at the condenser water outlet.
- j) Outlet Temperature is the temperature of the gas-vapor mixture leaving the condenser or external air cooler.
- k) Temperature Rise is the difference between outlet and inlet water temperatures as defined in i) and h) above.
- l) Terminal Difference is the difference between the temperature corresponding to the partial absolute pressure of the condensing vapor at the condenser inlet and the outlet water temperature.
- m) Gas-Vapor Approach is the difference between the vapor outlet temperature and the condensing water inlet temperature.
- n) Condenser Pressure Drop is the difference between the suction pressure and the gas-vapor outlet pressure described in e) and f) above.
- o) Suction Lift is the vertical distance the condensing water can be elevated due to the reduced pressure prevailing in the condenser. This dimension is referred to the centerline of the water inlet if the connection is on the side of the condenser shell, or is referred to the face of the inlet if the connection is on the top of the shell.

- p) Effective Height of Tailpipe is the vertical distance between the face of the water outlet of the condenser and the maximum water level in the hotwell under maximum flow conditions.
- q) Tailpipe Seal is the vertical distance between the end of the tailpipe and the minimum water level in the hotwell.

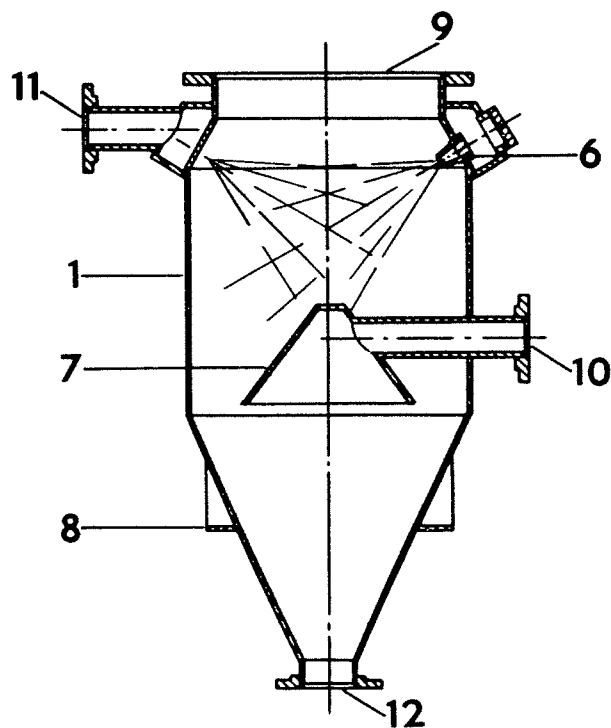
- r) Water Removal Pump Submergence is the vertical height of water column above the horizontal centerline of the pump impeller.
- *Where non-condensibles other than air are present, the total quantity of gases must be used.

Fig. 1
SPRAY TYPE COUNTERFLOW



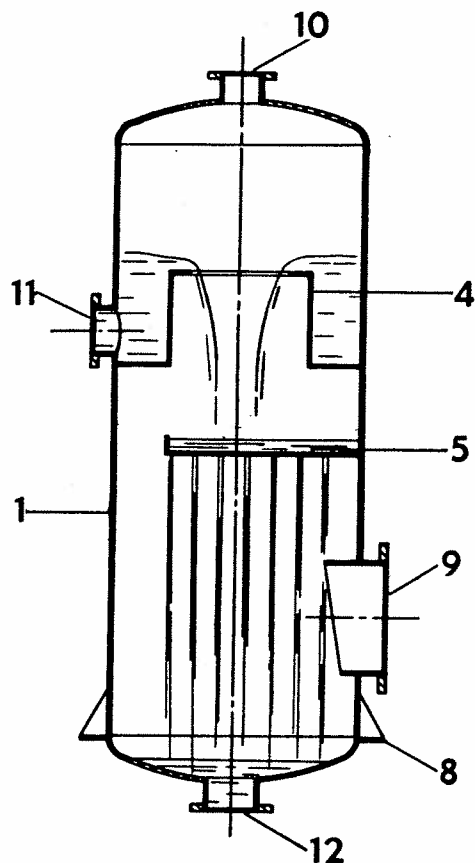
1. Shell
2. Diverting Disc
3. Diverting Ring
4. Distribution Weir
5. Distribution Tray
6. Spray Nozzle(s)

Fig. 2
SPRAY TYPE PARALLEL FLOW



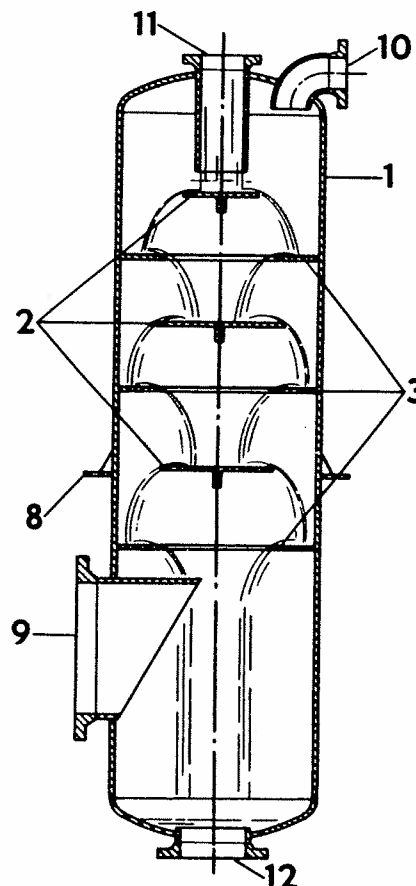
7. Water Hood
8. Supporting Feet
9. Vapor Inlet
10. Gas-Vapor Outlet
11. Water Inlet
12. Water Outlet

Fig. 3
WEIR AND TRAY TYPE COUNTERFLOW



1. Shell
2. Diverting Disc
3. Diverting Ring
4. Distribution Weir
5. Distribution Tray
6. Spray Nozzle(s)

Fig. 4
RING AND DISC COUNTERFLOW



7. Water Hood
8. Supporting Feet
9. Vapor Inlet
10. Gas-Vapor Outlet
11. Water Inlet
12. Water Outlet

2.5 Notation

2.5.1 Symbols

The following symbols are used in formulae in this Standard:

- C_p Specific heat at constant pressure, Btu per pound per °F (0.24 for air, 1.0 for fresh water).
 d Inside diameter, inches.
 d_n nominal pipe size, inches
 D Inside diameter, feet.
 f Friction Factor, dimensionless.
 F Pressure drop in pipe, inches of water per 100 feet of pipe.
 g Acceleration due to gravity, 32.17 feet per second per second.
 h Head, feet of water.
 H Total required pumping head, feet of water.
 h_f Enthalpy of liquid, Btu per pound.
 h_g Enthalpy of vapor, Btu per pound.
 M_n Molecular weight, noncondensable gas.
 M_v Molecular weight of water vapor.

- P Pressure (inches Hg, psia, etc.).
 P_n Partial pressure of noncondensable gas, in HgA.
 P_t Total pressure of mixture, in HgA.
 P_v Partial pressure of condensable vapor, in HgA.
 Q Condenser duty, Btu per hour.
 r Radius, inches.
 R_m Radius, inches.
 R_D Reynolds number.
 S Stress, psi.
 S_G Specific gravity of condensing water.
 t Temperature, °F.
 T Shell thickness, inches.
 TH Total head, feet of water.
 v Velocity, feet per second.
 w Flow rate, pounds per hour.
 W Flow rate, pounds per hour.
 μ Absolute viscosity, pounds per foot-second.
 ρ Vapor density, pounds per cubic foot.
 σ Stress, psi.

3.0 CONDENSER PERFORMANCE

3.1 Design Point

It is recognized that the performance of a condenser cannot be exactly predicted under each one of a number of possible operating conditions and, consequently, curves or tabulations of condenser performance data are only approximate, except for one specific condition termed the "Design Point".

3.2 Basic Formula

The quantity of condensing water required is obtained from the following basic formula:

$$\text{GPM} = \frac{\text{Condenser Duty}}{500 \times C_p \times S_G \times \text{Temperature Rise } (^{\circ}\text{F})}$$

3.3 Condenser Duty

3.3.1 Turbine Service

The heat released to the condensing water is assumed to average 950 Btu per pound for turbine service, unless otherwise specified. The condenser duty is obtained by multiplying this average Btu per pound by the pounds per hour of steam entering the condenser.

3.3.2 Process Service

The heat released to the condensing water is obtained by calculation of a heat balance in accordance with the following formula:

$$Q = \{(W_{S9} \times h_{g9}) - (W_{S10} \times h_{g10})\} - \{(W_{S9} - W_{S10}) \times h_{f12}\} + \{W_a C_p (t_9 - t_{10})\}$$

where:

Q = Condenser duty, Btu per hour

W_{S9} = Condensable vapor flow rate entering condenser, pounds per hour

W_{S10} = Vapor saturation flow rate at gas-vapor outlet, pounds per hour

h_{g9} = Total enthalpy of condensable vapor at vapor inlet, Btu per pound

h_{g10} = Total enthalpy of vapor at gas-vapor outlet, Btu per pound

h_{f12} = Total enthalpy of outlet water, Btu per pound

W_a = Total non-condensible flow rate, pounds per hour

C_p = Specific heat of non-condensible at constant pressure, Btu per pound per degree F (for air, use 0.24)

t_9 = Mixture temperature at vapor inlet, $^{\circ}\text{F}$

t_{10} = Mixture temperature at gas-vapor outlet, $^{\circ}\text{F}$

(Numerical subscripts refer to connections shown in Figs. 1, 2, 3, and 4.)

3.4 Terminal Difference

The terminal difference is governed by the condenser design, operating conditions or a combination of both. A minimum terminal difference of 5°F is recommended for normal operation conditions. When the condenser load contains a negligible amount of non-condensable gases, a minimum terminal differ-

ence of 3°F may be used. When the condenser load contains a large amount of non-condensable gases, a terminal difference greater than 5°F is recommended for a practical condenser design.

3.5 Outlet Water Temperature

Outlet water temperature is determined by subtracting the terminal difference from the temperature corresponding to the partial absolute pressure of the condensing vapor at the condenser inlet. The partial vapor pressure is calculated by assuming that the ratio of the partial vapor pressure to the total pressure of the mixture is equal to the ratio of mols of steam (or condensable vapor) to the total mols of the mixture. For example:

Condenser Load;

Quantity of air	100 lbs. per hr.
Quantity of steam	1,000 lbs. per hr.
Suction pressure	2.00 in. HgA
Molecular weight of air	28.97
Molecular weight of steam	18.0
Mols of air in mixture	$\frac{100}{28.97} = 3.45$

$$\text{Mols of steam in mixture} = \frac{1000}{18} = 55.55$$

$$\text{Total mols of mixture} = 3.45 + 55.55 = 59.00$$

Let P_v = partial pressure of condensable vapor, inches mercury absolute

P_t = total pressure of mixture, inches mercury absolute

$$P_v = P_t \times \frac{\text{Mols of condensable vapor}}{\text{Total mols of mixture}}$$

$$P_v = 2.00 \times \frac{55.55}{59.00} = 1.88 \text{ in HgA}$$

The temperature corresponding to 1.88 in. HgA is 99.08°F (refer to Properties of Saturated Water Vapor, Table 2).

Outlet water temperature:
 $99.08 - 5.00 = 94.08^{\circ}\text{F}$

3.6 Condensing Water Quantity

Continuing the example in Par. 3.5, with a heat release of 950 Btu per pound of steam and 70°F inlet water temperature, the quantity of condensing water is determined as follows:

Temperature rise:

$$94.08 - 70 = 24.08^{\circ}\text{F}$$

Condensing water quantity:

$$\frac{1000 \times 950}{500 \times 24.08} = 78.9 \text{ GPM}$$

Where $C_p = 1.0$
 $S_G = 1.0$

3.7 Non-condensible Load

3.7.1 Turbine Service

Unless otherwise specified, non-condensible load due to air leakage into the system for turbine service is estimated from the maximum quantity of steam condensed and is assumed to be not less than the values tabulated in Table 1. These values are greater, however, than the maximum air leakage expected with a commercially tight system.

TABLE 1
AIR LEAKAGE ALLOWANCE

Maximum Steam Condensed Lb./Hr.	Lb./Hr. 70°F Free Dry Air
Up to 10,000	10
10,001 to 25,000	15
25,001 to 50,000	25
50,001 to 100,000	35
100,001 to 150,000	45

If the condenser serves more than one prime mover, increase the dry air capacity given in the above table for the total steam load by multiplying it by the following factors:

- 2 Prime Movers – 1.5
- 3 Prime Movers – 1.75
- 4 Prime Movers – 2.00
- 5 Prime Movers – 2.15
- 6 Prime Movers – 2.25

3.7.2 Process Service

Unless otherwise specified, non-condensible load due to air leakage into the system for process service is estimated from the system volume. Fig. 5 shows the maximum expected air leakage values for a commercially tight vacuum system without any agitator. For a system with agitator having ordinary commercial shaft seal, a reasonable amount of maintenance will keep shaft leakage down to five pounds per hour. It is recommended that the design non-condensible load be double the estimated air leakage to which must be added the non-condensibles gases produced in the process.

3.8 Air Liberated from Condensing Water

Unless otherwise specified, the quantity of air liberated from the condensing water upon admission to the vacuum space will depend on the inlet temperature and water quantity as indicated in Fig. 6.

3.9 Gas-Vapor Outlet Load

3.9.1. The load at the condenser gas-vapor outlet consists of a saturated mixture of non-condensibles gases and associated water vapor. The amount of water vapor in the mixture is a function of the temperature, the absolute pressure and the molecular weight of the non-condensibles gases. The amount of water vapor in the mixture is determined by the following formula:

$$W_v = \frac{W_n \times M_v \times P_v}{M_n \times P_n}$$

W_v = Vapor of saturation, lb. per hr.

W_n = Non-condensibles gas, lb. per hr.

M_v = Molecular weight of water vapor

P_v = Partial pressure of water vapor, in HgA

M_n = Molecular weight of non-condensibles gas

P_n = Partial pressure of non-condensibles gas, in. HgA (equals total pressure of mixture minus partial pressure of water vapor)

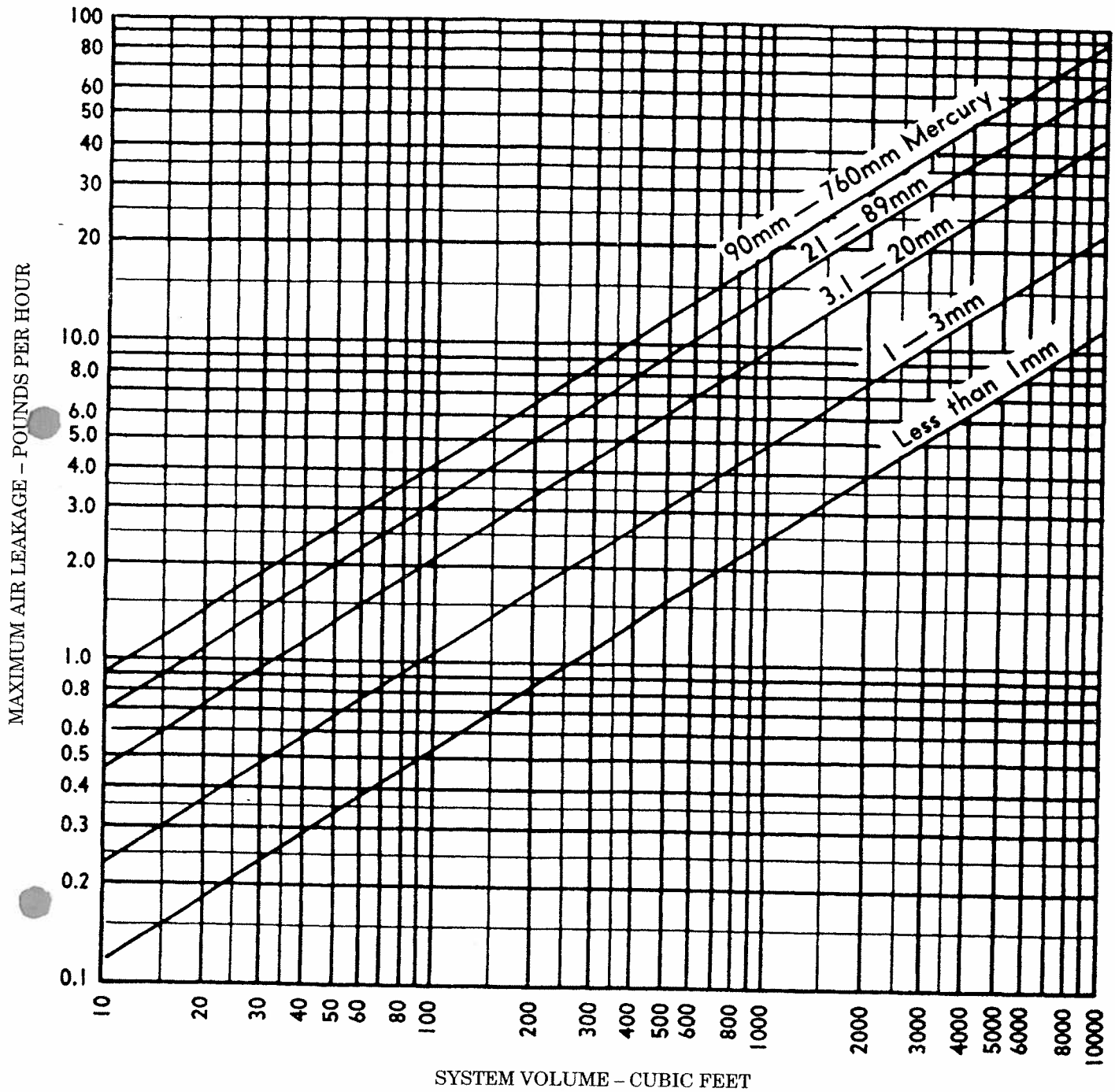
3.9.2 In a well-designed counter flow type condenser, or parallel flow type condenser equipped with a suitable direct contact spray cooler, when the relative proportion of non-condensibles to the condenser load is small, the temperature at the gas-vapor outlet is assumed to be 5°F above the temperature of the inlet water. When the relative proportion of non-condensibles to the condenser load is large the temperature at the gas-vapor outlet may be more than 5°F above the temperature of the inlet water. When the relative proportion of non-condensibles to the condenser load is small, the condenser pressure drop can be assumed to be 0.1 in. Hg. The total pressure at the gas-vapor outlet will be equal to the suction pressure less 0.1 in. Hg. When the relative proportion of non-condensibles is large, the condenser pressure drop may be greater than 0.1 in. Hg and should be considered. The value of (P) for water vapor can be obtained from Table 2, "Properties of Saturated Water Vapor." When a non-condensibles gas is all air, the water vapor component in the mixture can be obtained from Figs. 7, 8, 9 and 10. Where a mixture of non-condensibles gases is involved, the average molecular weight of the mixture is calculated and the total pounds per hour of the mixture and the average molecular weight of the mixture are used in the calculations as if the mixture were a pure gas having a molecular weight the same as the average molecular weight. The sum of the partial pressures of the non-condensibles gases and vapors equals the total pressure of the mixture, provided the gases obey perfect gas laws and are not soluble in water or each other.

TABLE 2
PROPERTIES OF SATURATED WATER VAPOR

Temp F	Absolute Pressure			Sp Vol cu. ft. per lb.	Temp F	Absolute Pressure			Sp Vol cu. ft. per lb.
	MM Hg	Inches Hg	Psi			MM Hg	Inches Hg	Psi	
58	12.33	.4856	.2385	1,292	103	53.73	2.115	1.039	321.8
59	12.78	.5033	.2472	1,249	104	55.35	2.179	1.070	312.9
60	13.25	.5216	.2562	1,207	105	57.00	2.244	1.102	304.4
61	13.73	.5405	.2654	1,167	106	58.70	2.311	1.135	296.0
62	14.22	.5599	.2750	1,129	107	60.45	2.380	1.169	288.0
63	14.73	.5800	.2849	1,092	108	62.23	2.450	1.203	280.2
64	15.26	.6007	.2950	1,056	109	64.08	2.523	1.239	272.6
65	15.80	.6221	.3055	1,022	110	65.96	2.597	1.275	265.3
66	16.36	.6441	.3164	988.6	111	67.89	2.673	1.313	258.2
67	16.94	.6668	.3275	956.8	112	69.88	2.751	1.351	251.3
68	17.53	.6902	.3390	926.1	113	71.91	2.831	1.390	244.6
69	18.14	.7143	.3508	896.5	114	73.97	2.912	1.430	238.1
70	18.78	.7392	.3630	868.0	115	76.10	2.996	1.472	231.8
71	19.43	.7648	.3756	840.5	116	78.28	3.082	1.514	225.7
72	20.09	.7911	.3886	814.0	117	80.52	3.170	1.557	219.8
73	20.78	.8183	.4019	788.4	118	82.83	3.261	1.601	214.1
74	21.50	.8483	.4156	763.7	119	85.17	3.353	1.647	208.6
75	22.23	.8751	.4298	740.0	120	87.58	3.448	1.693	203.2
76	22.98	.9047	.4443	717.0	121	90.04	3.545	1.741	197.9
77	23.75	.9352	.4593	694.9	122	92.56	3.644	1.790	192.9
78	24.55	.9667	.4748	673.5	123	95.17	3.745	1.840	187.9
79	25.38	.9990	.4907	652.9	124	97.79	3.850	1.891	183.2
80	26.21	1.032	.5070	633.0	125	100.7	3.956	1.943	178.5
81	27.08	1.066	.5238	613.8	126	103.3	4.065	1.997	174.0
82	28.00	1.102	.5411	595.3	127	106.1	4.177	2.051	169.6
83	28.91	1.138	.5589	577.4	128	109.0	4.291	2.107	165.4
84	29.85	1.175	.5772	560.1	129	112.0	4.408	2.165	161.3
85	30.84	1.214	.5960	543.3	130	115.0	4.527	2.224	157.3
86	31.83	1.253	.6154	527.2	135	131.3	5.168	2.538	138.9
87	32.84	1.293	.6353	511.6	140	149.5	5.884	2.890	123.0
88	33.91	1.335	.6557	496.5	145	169.8	6.683	3.282	109.1
89	35.00	1.378	.6788	482.0	150	192.4	7.573	3.719	97.04
90	36.12	1.422	.6984	467.9	155	217.5	8.561	4.205	86.50
91	37.28	1.467	.7206	454.3	160	245.3	9.656	4.742	77.27
92	38.46	1.514	.7434	441.1	165	276.1	10.87	5.337	69.17
93	39.65	1.561	.7668	428.4	170	309.9	12.20	5.994	62.04
94	40.89	1.610	.7909	416.1	175	347.5	13.68	6.717	55.77
95	42.19	1.661	.8156	404.2	180	388.4	15.29	7.512	50.22
96	43.49	1.712	.8410	392.7	185	433.6	17.07	8.384	45.31
97	44.83	1.765	.8671	381.5	190	483.1	19.02	9.340	40.96
98	46.23	1.820	.8939	370.7	195	537.0	21.14	10.39	37.09
99	47.65	1.876	.9214	360.3	200	596.1	23.47	11.53	33.64
100	49.10	1.933	.9496	350.2	205	660.4	26.00	12.77	30.57
101	50.60	1.992	.9785	340.4	210	730.3	28.75	14.12	27.82
102	52.15	2.053	1.008	331.0	212	759.99	29.92	14.690	26.80

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Fig. 5
**MAXIMUM AIR LEAKAGE VALUES
 FOR COMMERCIALY TIGHT SYSTEM**



NOTES:

1. The air rates indicated are guidelines only. In actual field practice, the air leakage may be significantly larger or smaller depending upon the condition of sealing, maintenance factors and applications.
2. For comparison purposes, alternate methods of air leakage estimation may be used. A selected bibliography is included in Appendix A.

Fig. 6
ALLOWANCE TO BE MADE FOR AIR IN CONDENSING WATER

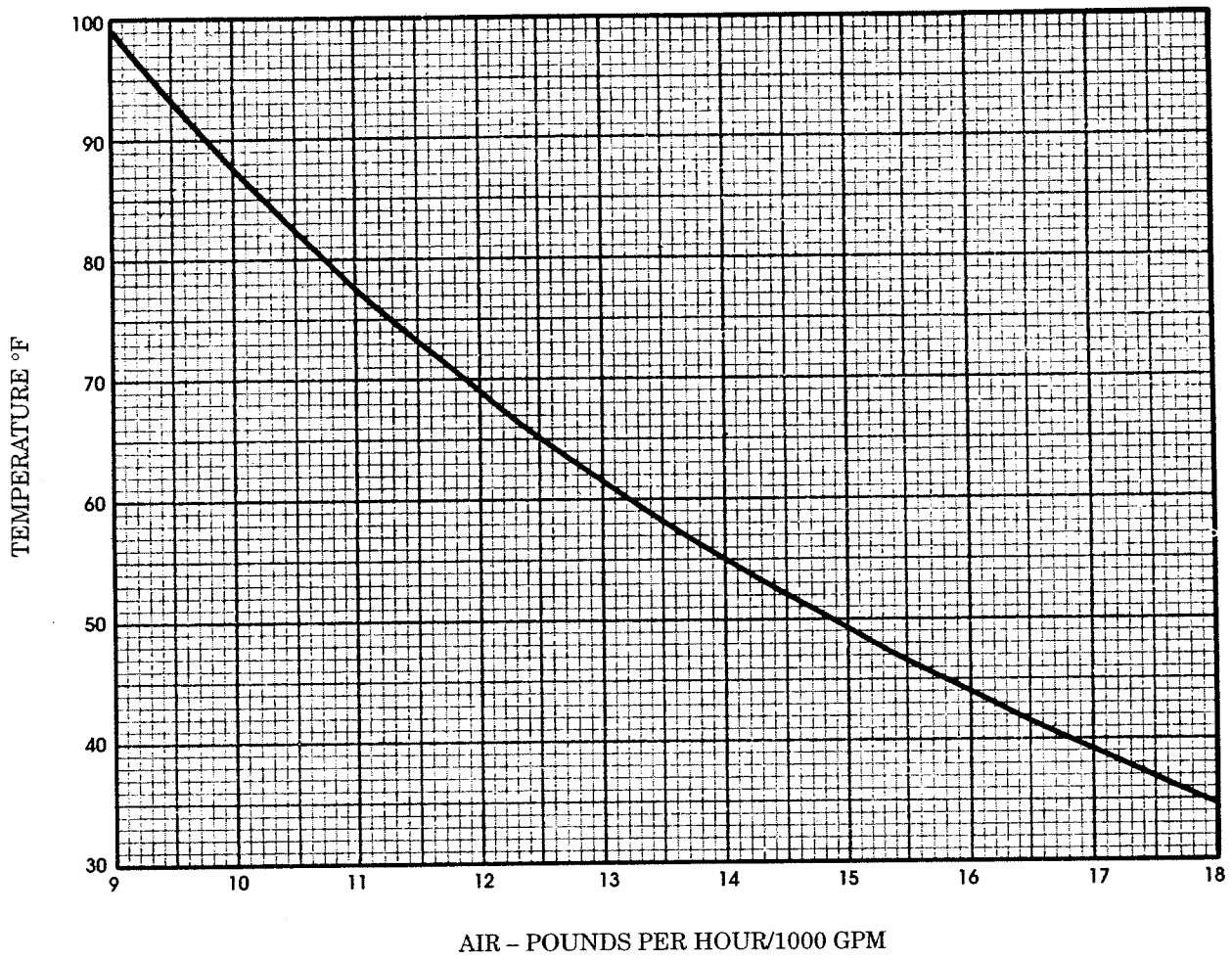


Fig. 7
AIR AND WATER VAPOR MIXTURE DATA (DALTON'S LAW)

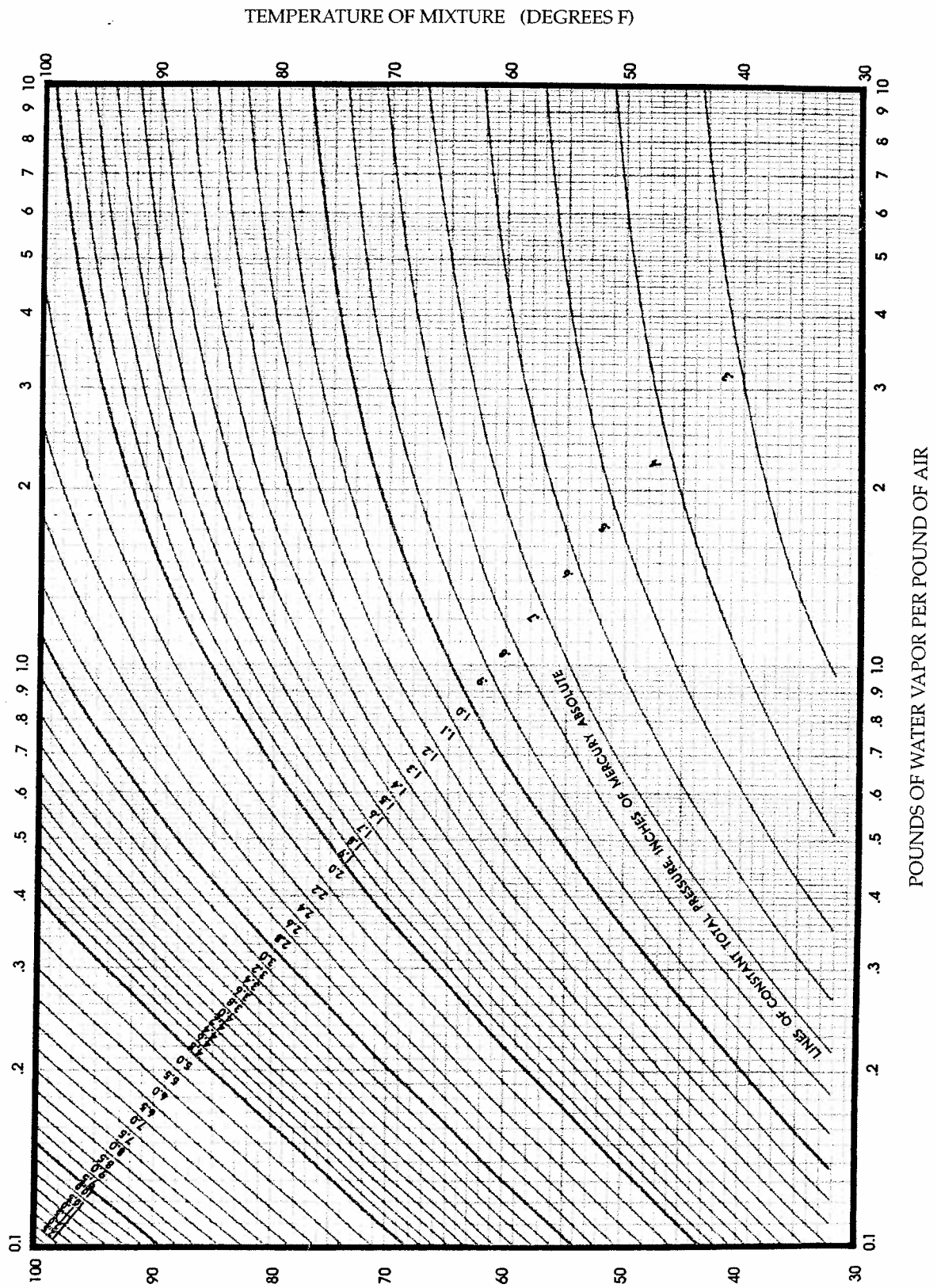


Fig. 8
AIR AND WATER VAPOR MIXTURE DATA (DALTON'S LAW)

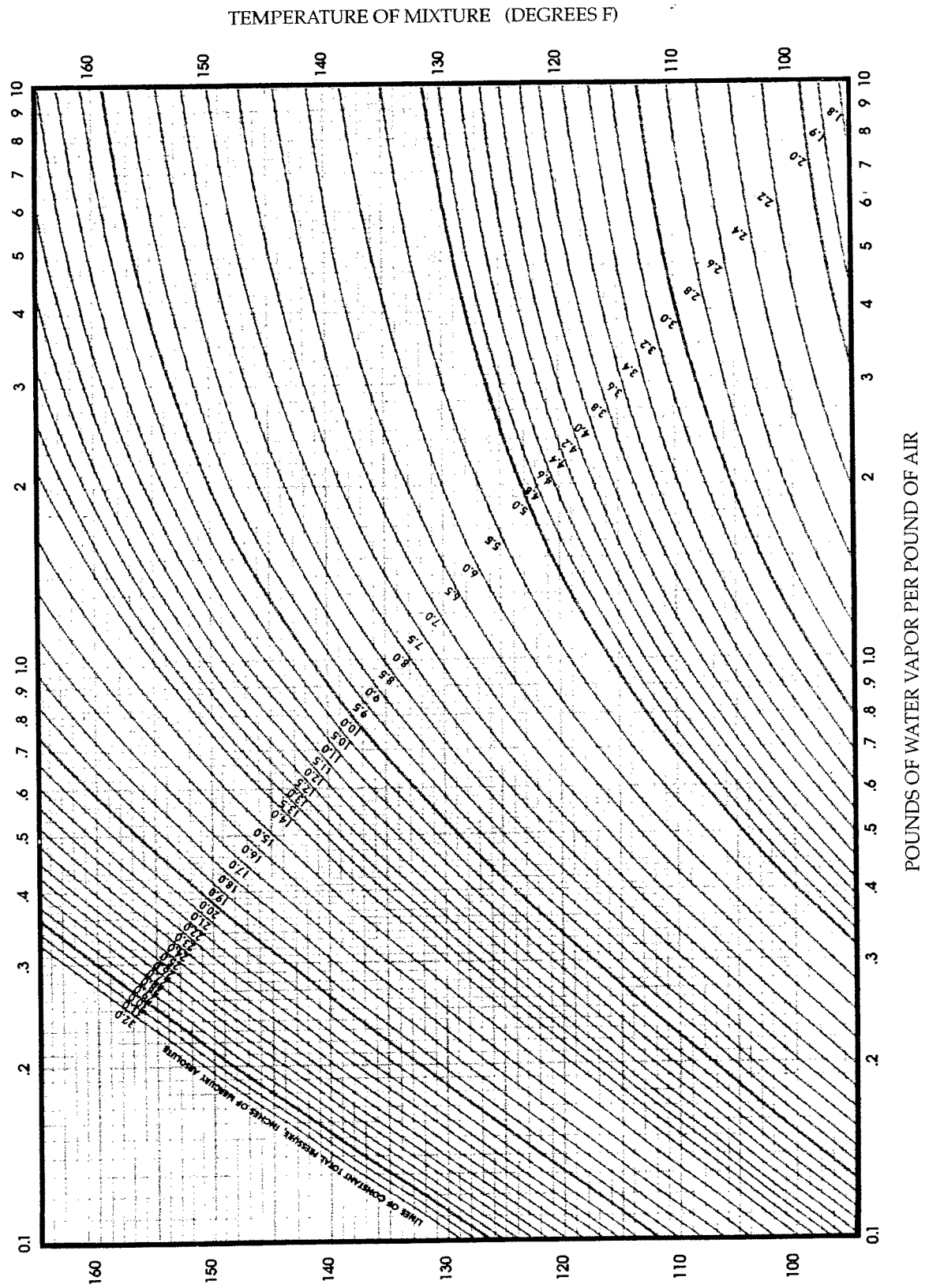


Fig. 9
AIR AND WATER VAPOR MIXTURE DATA (DALTON'S LAW)

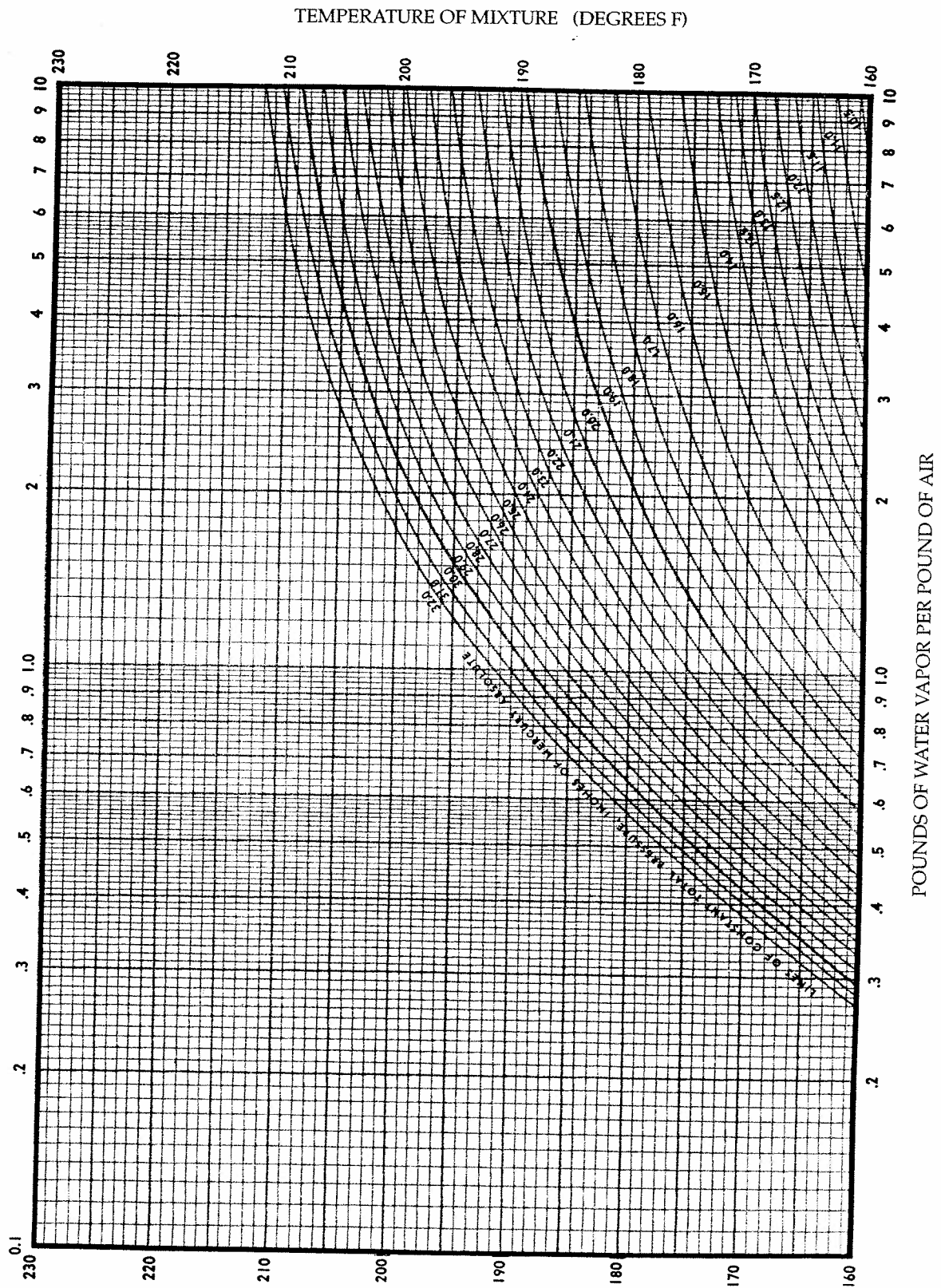
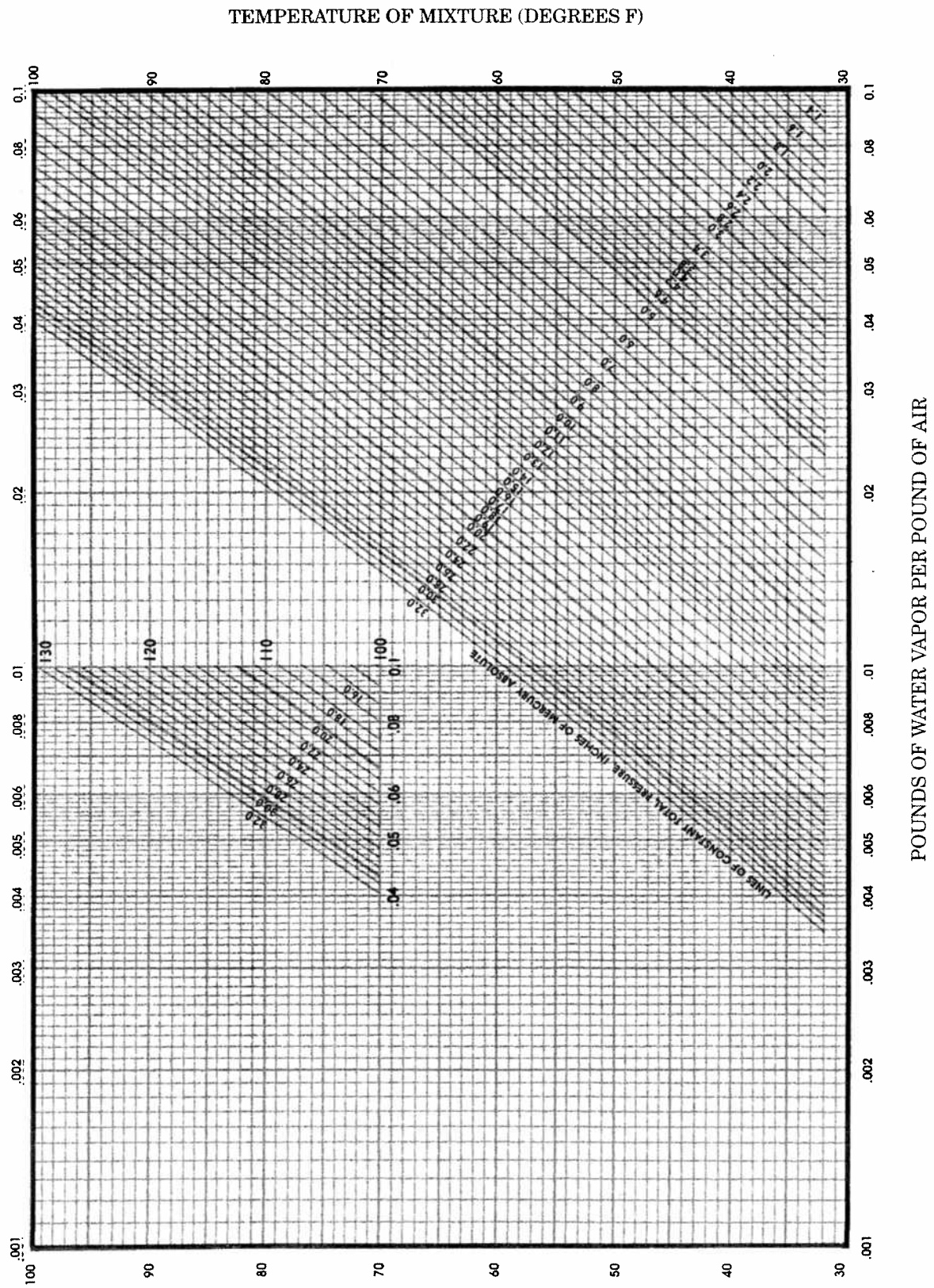


Fig. 10
AIR AND WATER VAPOR MIXTURE DATA (DALTON'S LAW)



4.0 CONDENSER ENGINEERING DATA

4.1 Condenser Water Ratings

Condenser water ratings are a function of operating conditions and condenser design.

4.2 Vapor Inlet Size

The maximum recommended vapor inlet velocities for saturated steam are shown in Table 3. These velocities are calculated by using the following equation:

$$v^2 \rho = 400$$

where:

v = Vapor inlet velocity, feet per second

ρ = Vapor density, pounds per cubic foot

For vapors other than saturated steam the above equation may be applied.

TABLE 3
MAXIMUM INLET VELOCITIES
FOR SATURATED STEAM

Suction Pressure in HgA	Velocity at Vapor Inlet, Feet per Second
1.0	500
1.5	420
2.0	370
3.0	300
4.0	260
6.0	220
8.0	190
10.0	170

4.3 Vapor Piping

The size of the vapor piping should be determined from the allowable pressure loss. (See Figs. 11, 12, and 13.) When the size of the vapor piping is not the same as the condenser vapor inlet, a suitable tapered transition piece with a maximum total divergence angle of eleven degrees should be installed between the vapor piping and the condenser.

4.3.1 Turbulent Flow of Air or Steam

Figure 11* can be used to determine the diameter of the cylindrical pipe required for a low pressure drop with turbulent flow of air or steam at a subatmospheric pressure, provided the pressure drop is limited to 10 percent of the final pressure. The lower limit for the application of this data is determined by the limit of the ratio:

$$\frac{w}{d} \geq 20$$

where:

w = flow rate, pounds per hour

d = inside diameter of pipe, inches

If the ratio is less than 20, the flow is streamline and Figure 11 will not apply.

For example:

Pipe length = 300 ft.
Pipe size = 12 in. nominal
Steam flow = 275 lbs. per hr.
Steam temperature = 400°F
Initial pressure = 0.5 in. HgA

Pressure drop is determined as follows:

$$F = \frac{(F_1 \times C_{D1} \times C'_{T1}) + (F_2 \times C_{D2} \times C'_{T2})}{P_1}$$

w = 275 lbs. per hr.

F_1 = 1.78×10^{-2}

F_2 = 8.45×10^{-2}

d = 12.09 in.

C_{D1} = 4.10×10^{-1}

C_{D2} = 4.25×10^{-1}

t = 400°F

C'_{T1} = 2.61

C'_{T2} = 2.515

Then,

$$F = \frac{300}{100} \times \left\{ \frac{(1.78 \times 10^{-2})(4.10 \times 10^{-1})(2.61)}{0.5} + \frac{(8.45 \times 10^{-2})(4.25 \times 10^{-1})(2.515)}{0.5} \right\}$$

$$F = \frac{0.656}{13.6^{**}} = 0.048 \text{ in. Hg and the pressure drop}$$

is 9.60 percent of the final pressure. Note that

$$\frac{w}{d} = 22.8$$

If the pressure drop is greater than 10 per cent of the final pressure, it is possible to contain the error introduced by the excessive pressure drop by separating the length of given pipe into increments such that the pressure drop in each increment is less than the tolerable 10 per cent of the final pressure.

For example:

Pipe length = 100 ft.

Pipe size = 6 in. nominal

Steam flow = 180 lbs. per hr.

Steam temperature = 100°F

Initial pressure = 0.5 in. HgA

Pressure drop is determined as follows:

w = 180 lbs. per hr.

F_1 = 7.7×10^{-3}

F_2 = 4.1×10^{-2}

d = 6 in.

C_{D1} = 12.7

C_{D2} = 11.1

t = 100°F

C'_{T1} = 1.700

C'_{T2} = 1.455

Then,

$$F = \frac{100}{100} \times \left\{ \frac{(7.7 \times 10^{-3})(12.70)(1.700)}{0.5} + \frac{(4.1 \times 10^{-2})(11.10)(1.455)}{0.5} \right\}$$

*The basis for this data is given in "A Graphical Representation of the Frictional Losses in Commercial Pipe of Air and Steam Flowing Turbulently at Low Pressures", W. C. Knapp and J. W. Metzger, *Transactions, Volume 77, No.15, July, 1955, pages 675-681.*

**13.6 is a conversion factor for in. H₂O to in. Hg.

$$F = 1.657 \text{ in. H}_2\text{O}$$

$$= \frac{0.1216}{13.6} = 0.1216 \text{ in. Hg}$$

and the pressure drop is 32.14% of the final pressure.

Since this pressure drop exceeds the 10 per cent limit for which the equation is suitable, the pressure drop should be calculated by the increment method.

The constants from Figure 11 do not change and may be combined as the constant 0.8284. This reduces the equation to the form

$$F = \frac{L \times 0.8284}{100 \times p_1 \times 13.6}$$

Then, introduce incremental lengths of 25 feet

$$1. \quad \frac{25 \times 0.0608}{100 \times 0.50} = 0.0304 \text{ in. Hg}$$

$$0.500 - 0.0304 = 0.4696$$

$$2. \quad \frac{25 \times 0.0608}{100 \times 0.4694} = 0.0324 \text{ in. Hg}$$

$$0.4696 - 0.0324 = 0.4372$$

$$3. \quad \frac{25 \times 0.0608}{100 \times 0.4372} = 0.0348 \text{ in. Hg}$$

$$0.4372 - 0.0348 = 0.4024$$

$$4. \quad \frac{25 \times 0.0608}{100 \times 0.4024} = 0.0378 \text{ in. Hg}$$

$$0.4024 - 0.0378 = 0.3646$$

The total pressure drop will be:

$$0.0304 + 0.0324 + 0.0348 + 0.0378 = 0.1354 \text{ in. Hg}$$

and the final pressure will be:

$$0.500 - 0.1354 = 0.3646 \text{ in. Hg}$$

4.3.2 Streamline Flow of Air

Calculations to determine the diameter of the cylindrical pipe with streamline flow of air at any initial absolute pressure between 50 microns and 1 millimeter of mercury can be made by using the proper value of "f" from Figure 12* in the Fanning equation provided the pressure drop is limited to 10 per cent of the final pressure. For example, the calculations to determine the size pipe required to limit the pressure drop to around 9 microns for a flow of 5 pounds per hour of 70°F dry air at 100 microns HgA pressure through a 15 foot length of pipe would be as follows:

$$R_D = \frac{D v \rho}{\mu}$$

where:

D = Inside diameter of pipe, feet

v = Average velocity, feet per second

ρ = Average density, pounds per cubic foot

μ = Absolute viscosity of air, pounds per foot-second, from Figure 13

Since D = 17.182 inches or 1.432 feet, the area of the cross section of pipe = 1.610 sq. ft.

The specific volume at 70°F and average pressure of 95.5 microns = 106,206 cu. ft. per lb.

$$\begin{aligned} \text{Average velocity} &= \frac{5 \times 106,206}{3600 \times 1.610} \\ &= 91.6 \text{ ft. per sec.} \end{aligned}$$

$$\begin{aligned} \text{Average density} &= \frac{1}{106,206} \\ &= 94.16 \times 10^{-7} \text{ lbs. per cu. ft.} \end{aligned}$$

Viscosity at 70°F = 122×10^{-7} lbs. per ft.-sec.** and therefore:

$$R_D = \frac{1.432 \times 91.6 \times 94.16 \times 10^{-7}}{122 \times 10^{-7}} = 101$$

For $R_D = 101$

f = 0.48 (from Figure 12)

The Fanning equation is:

$$P_1 - P_2 = \frac{4fL\rho v^2}{2gD \times 144}$$

where:

P_1 = Upstream static pressure, pounds per square inch absolute

P_2 = Downstream static pressure, pounds per square inch absolute

f = Friction factor, dimensionless

L = Length of pipe, feet

ρ = Average density, lbs. per cu. ft.

v = Average velocity, ft. per sec.

g = Acceleration due to gravity

= 32.17 ft. per sec. per sec.

D = Inside diameter of pipe, feet

then:

$$\begin{aligned} P_1 - P_2 &= \frac{4 \times 0.48 \times 15 \times 94.16 \times 10^{-7} \times 91.6^2}{2 \times 32.17 \times 1.432 \times 144} \\ &= 1.715 \times 10^{-4} \text{ lbs. per sq. in.} \\ &= 1.715 \times 10^{-4} \times 51,714.7 \\ &\quad \text{(from Appendix B)} \\ &= 8.87 \text{ microns} \end{aligned}$$

4.3.3 Equivalent Length of Pipe

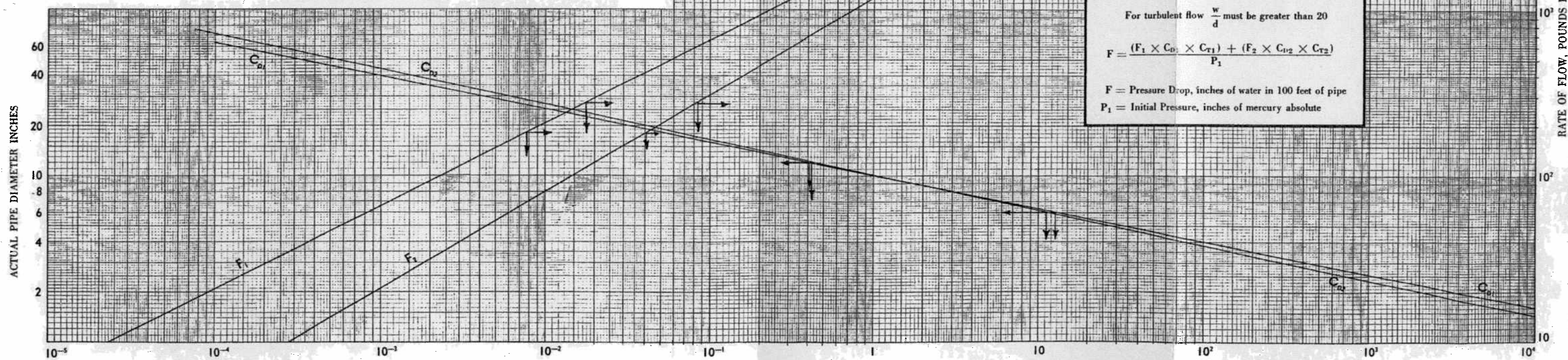
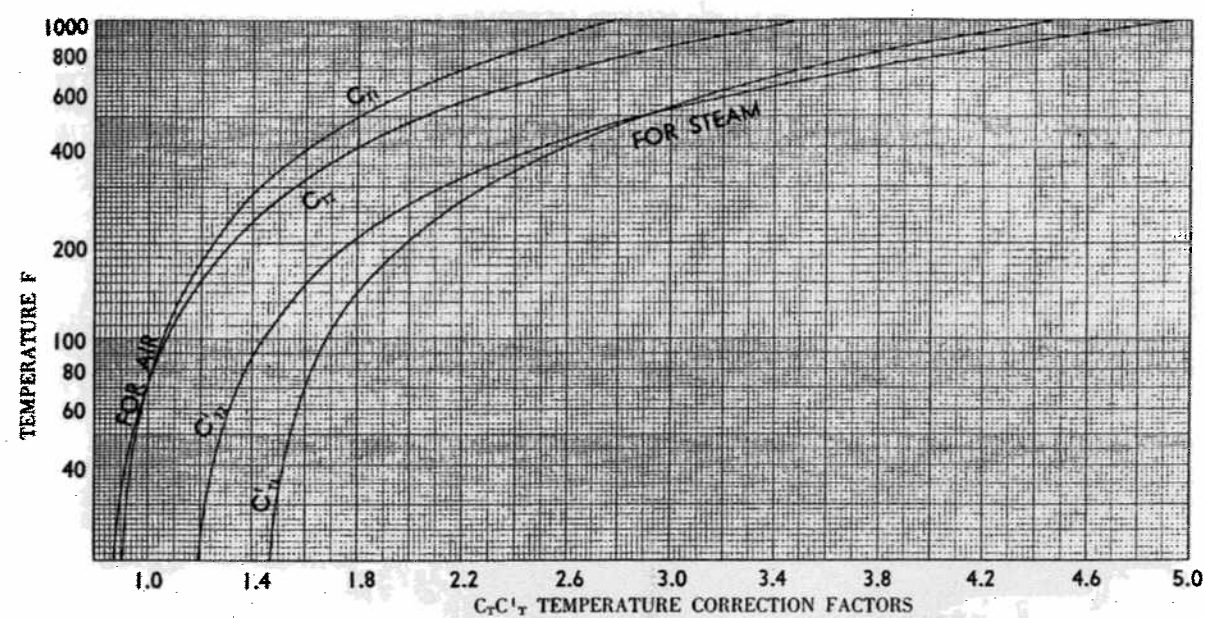
The equivalent length of pipe for various valves and fittings can be determined from existing data as published in many handbooks.

4.4 Water Outlet Size

Water outlet size is a function of operating conditions and condenser design. The tailpipe diameter should not be smaller than the condenser water outlet.

*These friction factors were developed as a result of an extensive investigation of pressure drops at low absolute pressures which was reported in detail in "Fanning Friction Factors for Air Flow at Low Absolute Pressures in Cylindrical Pipes." W. J. Bohnet and L. S. Stinson, *ASME Transactions*, Volume 77, No. 15, July, 1955, pages 683-692.

**Values for viscosity should be taken from Figure 13 to correspond to the viscosity values used in Figure 12.

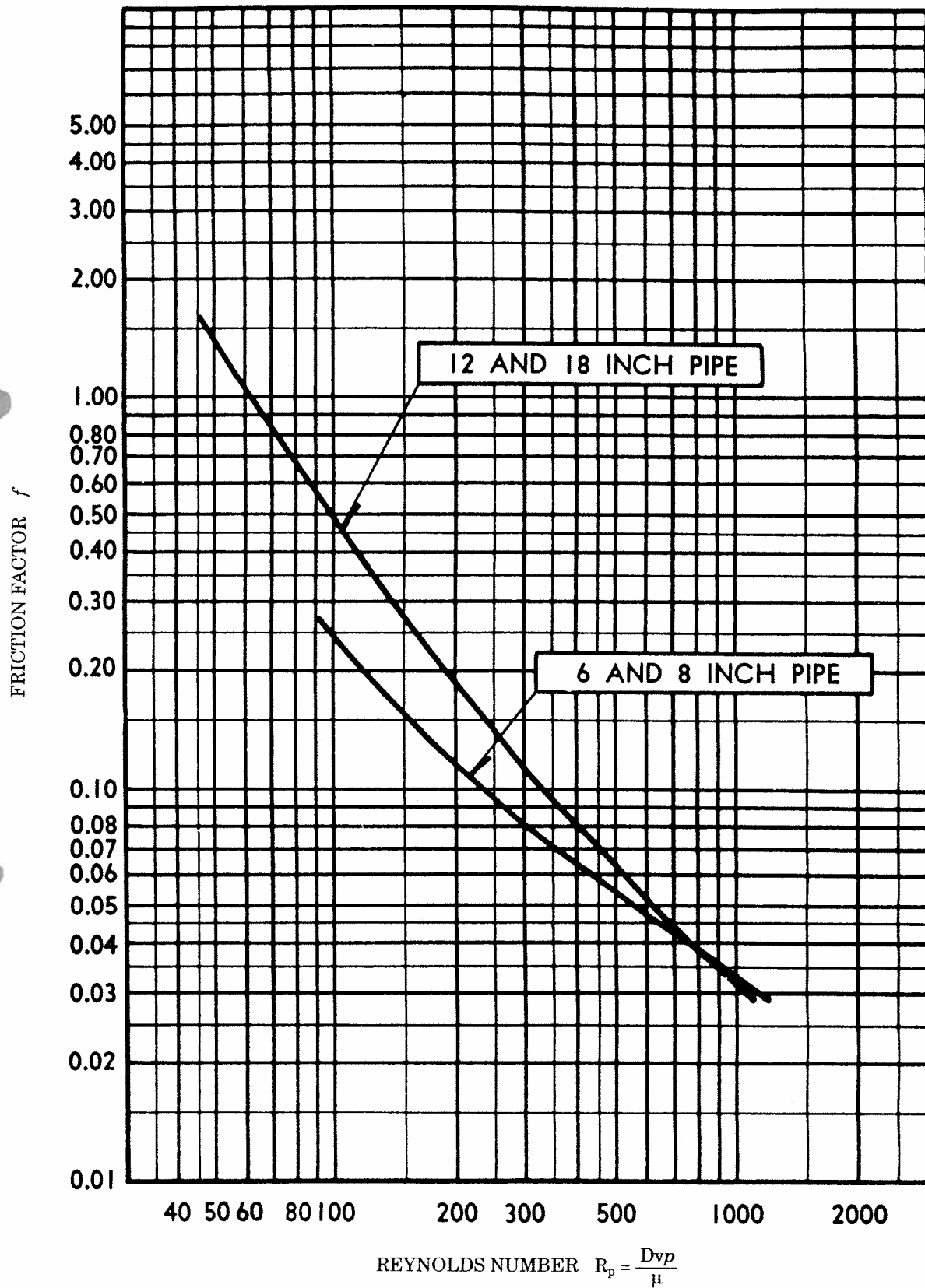


Note: Friction Factors F_1 and F_2 are based on rate of flow, while Factors C_{D1} and C_{D2} are based on actual pipe diameter.
 EVALUATION CURVES FOR FRICTION LOSSES OF AIR AND STEAM FLOWING TURBULENTLY IN COMMERCIAL PIPE AT LOW PRESSURES

Fig. 11

Fig. 12

FRICION FACTOR FOR STREAMLINE FLOW OF AIR AT ABSOLUTE PRESSURES FROM 50 MICRONS MERCURY TO 1 MILLIMETER MERCURY



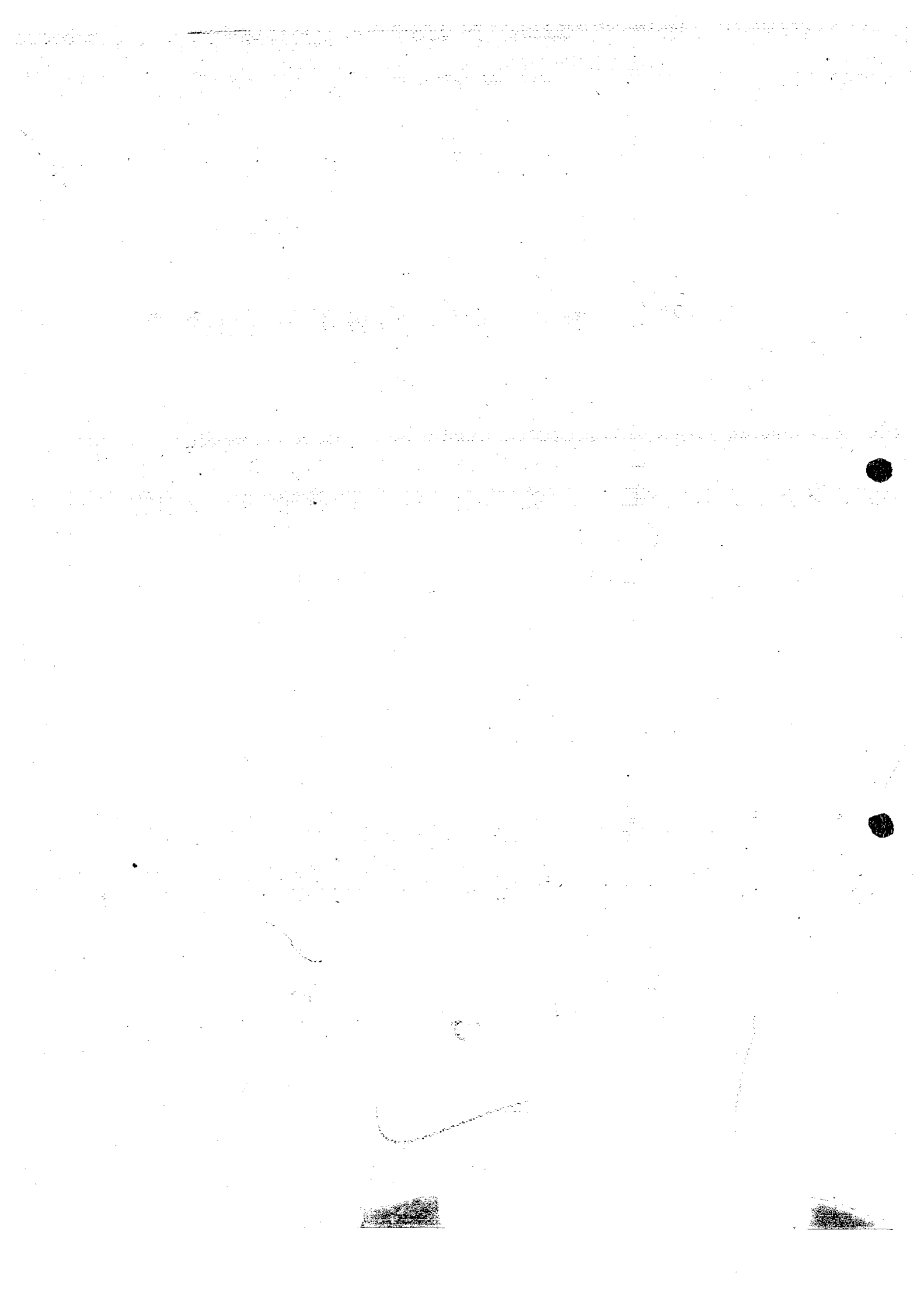
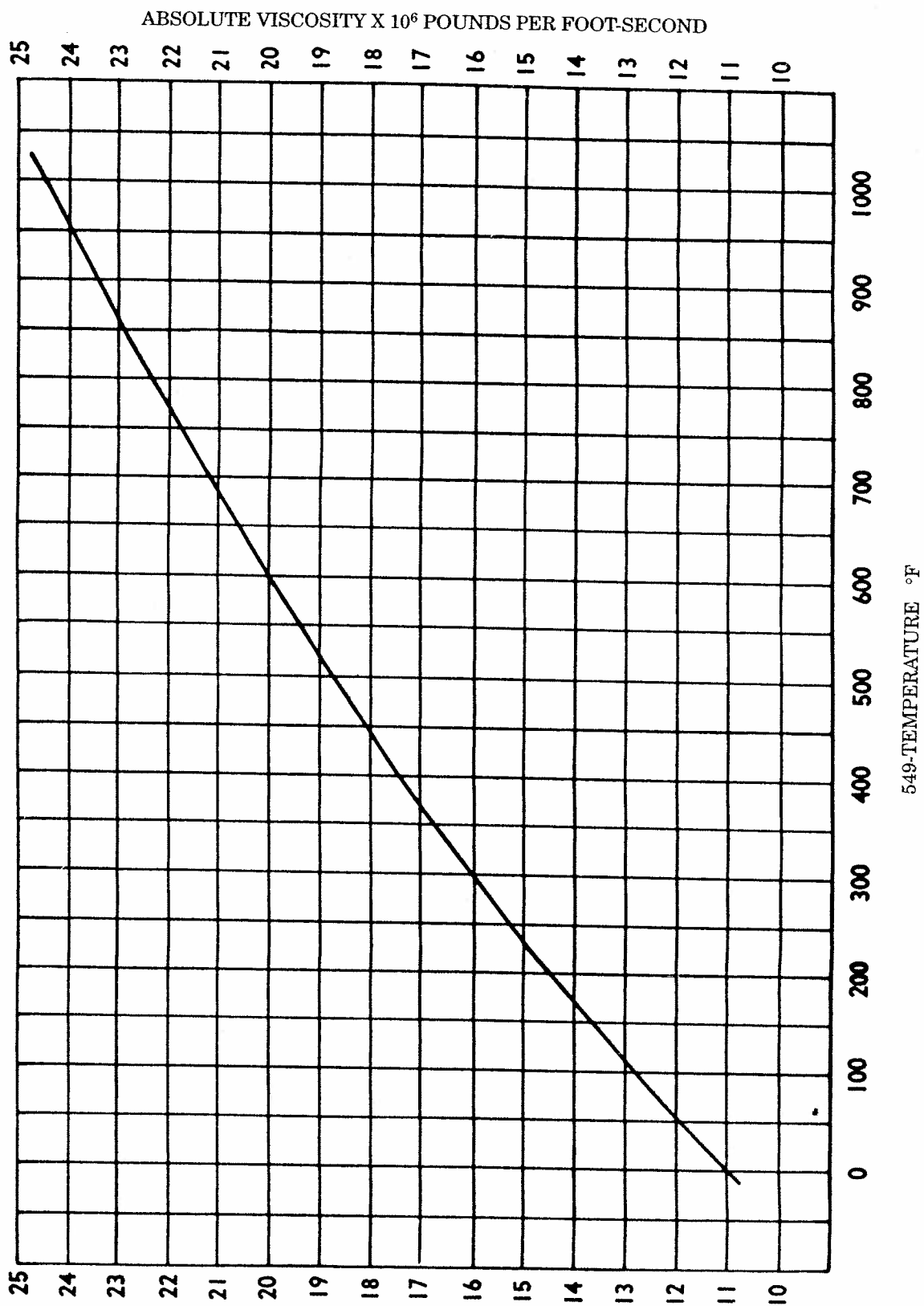


Fig. 13
ABSOLUTE VISCOSITY OF AIR



5.0 INSTALLATION

5.1 Condensing Water Inlet

5.1.1 Condensing Water Pumping Head

The condensing water pumping head is determined by the following formula:

$$H = h_1 + h_2 + h_3 + h_4 + h_5 - 0.80h_6$$

where: (see Fig. 14)

H = Total required pumping head, feet of water

h_1 = Pump suction lift

h_2 = Vertical distance from centerline of pump to water inlet

h_3 = Pipe friction for all piping between pump and condenser

h_4 = Pump suction pipe friction

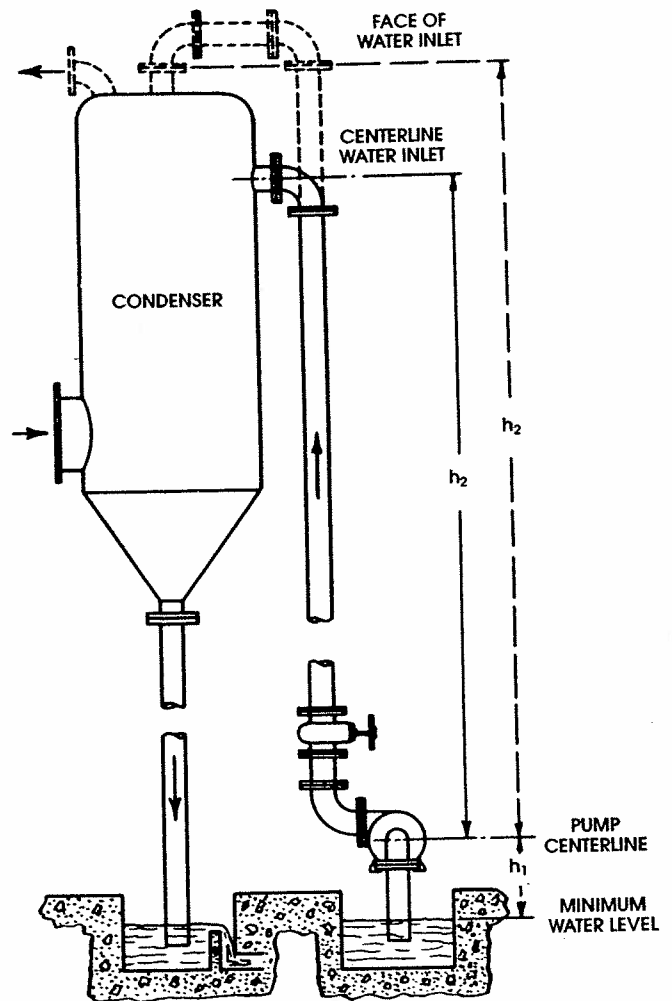
h_5 = Velocity head in piping between pump and condenser

h_6 = Suction lift due to vacuum in condenser, Table 4

Where spray nozzles are used instead of a weir for the distribution of the condensing water, add the value (h_7) representing the spray nozzle head measured from the water inlet opening. If it is necessary to establish the flow of water into the condenser before the vacuum is created, the pump characteristic must be such that the pump will provide approximately 20 per cent flow when (h_6) equals zero.

5.1.2 Condensing Water Suction Lift

The maximum theoretical suction lift of the condensing water is indicated in Table 4. When using this Table, it is essential to determine the maximum expected condensed pressure at the maximum load and condensing water temperature. To insure reliability under all possible operating conditions, it is recommended that only 80 per cent of the theoretical suction lift as derived from Table 4 be used in computing the allowable suction lift. It should be noted that suction lift is greatly influenced by the barometric pressure in addition to the absolute pressure being maintained in the condenser. In order to make due allowance for all possible operating conditions likely to be encountered, it is recommended that the minimum recorded barometric pressure be used.



**BAROMETRIC CONDENSER
INSTALLATION**

Fig. 14

TABLE 4
MAXIMUM THEORETICAL SUCTION LIFT IN FEET OF WATER AT 32°F

Absolute Pressure in Condenser (Inches Mercury)	Barometric Pressure – Inches of Mercury at 32°F								
	31	30	29	28	27	26	25	24	23
1	34.0	32.9	31.6	30.7	29.5	28.4	27.2	26.1	25.0
2	32.9	31.8	30.7	29.5	28.4	27.2	26.1	25.0	23.8
3	31.8	30.6	29.5	28.4	27.2	26.1	25.0	23.8	22.7
4	30.6	29.5	28.4	27.2	26.1	25.0	23.8	22.7	21.5
5	29.5	28.4	27.2	26.1	25.0	23.8	22.7	21.5	20.4
6	28.4	27.2	26.1	25.0	23.8	22.7	21.5	20.4	19.3
7	27.2	26.1	25.0	23.8	22.7	21.5	20.4	19.3	18.1
8	26.1	25.0	23.8	22.7	21.5	20.4	19.3	18.1	17.0
9	25.0	23.8	22.7	21.5	20.4	19.3	18.1	17.0	15.9
10	23.8	22.7	21.5	20.4	19.3	18.1	17.0	15.9	14.7
12	21.5	20.4	19.3	18.1	17.0	15.9	14.7	13.6	12.5
15	18.1	17.0	15.9	14.7	13.6	12.5	11.3	10.2	9.1

5.2 Barometric Condenser Water Outlet

5.2.1 Tailpipe Height

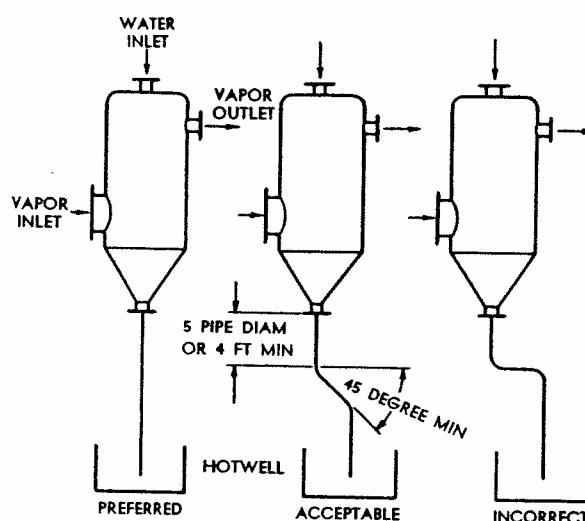
To provide properly for the removal of condensing water under all possible operating conditions likely to be encountered, it is recommended that the minimum effective height of the tailpipe be as shown in Table 5. The minimum height recommended is based entirely on the maximum recorded barometric pressure, regardless of the contemplated operating pressure.

TABLE 5
RECOMMENDED MINIMUM EFFECTIVE HEIGHT OF TAILPIPE

Maximum barometric pressure, inches of Mercury at 32°F	Height in Feet (Water at 32°F)
31	35.2
30	34.0
29	32.9
28	31.7
27	30.6
26	29.5
25	28.4
24	27.2
23	26.1

5.2.2 Tailpipe Arrangement

Preferred, acceptable and incorrect tailpipe arrangements are illustrated in Fig. 15.

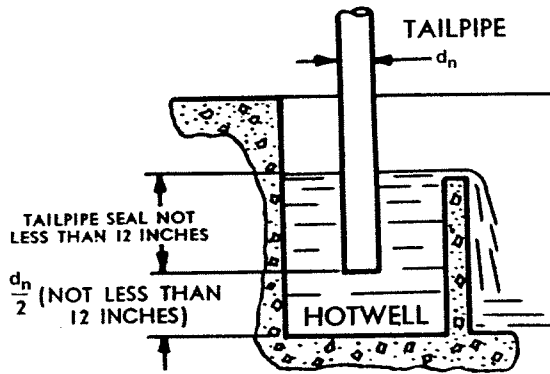


TAILPIPE ARRANGEMENT

Fig. 15

5.2.3 Hotwell

The recommended tailpipe seal and clearance from tailpipe outlet to the bottom of hotwell are shown in Fig. 16. The area of the hotwell shall be such that the volume of water measured from the bottom of the tailpipe to the point of overflow is at least equal to 1.5 times the volume of the minimum recommended height of the tailpipe.



HOTWELL DESIGN

Fig. 16

5.3 Low Level Condenser Water Outlet

5.3.1 Suction Lift

The maximum theoretical and allowable suction lifts for low level condensers are determined in the same manner as for barometric condensers. (See Sec. 5).

5.3.2 Removal Pump

The removal pump must be capable of handling the maximum flow of condensing water plus condensate against the total head at the available net positive suction head. The total head is determined by the following formula:

$$TH = h_1 + h_2 + h_3 + h_4 + h_5 - h_6$$

where: (see Fig. 17)

TH = Total head, feet of water

h_1 = Static discharge head

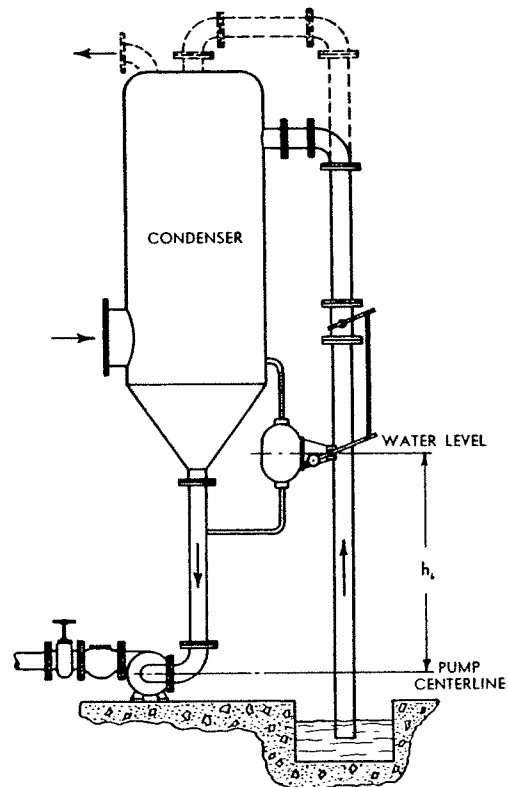
h_2 = Friction head in pump discharge piping

h_3 = Velocity head in discharge piping

h_4 = Suction head due to condenser pressure (use Table 4 for minimum condensing pressure obtained at zero load and minimum condensing water temperature)

h_5 = Friction head in pump suction piping

h_6 = Submergence measured vertically above centerline of pump impeller



LOW LEVEL CONDENSER INSTALLATION

Fig. 17

Net positive suction head is determined by the following formula:

$$h_{sv} = h_s + 1.33(h_s - h_{vp}) - h_f$$

where:

h_{sv} = Net positive suction head, feet of water

h_s = Static head of water above centerline of first stage impeller, feet of water

h_a = Absolute pressure acting on the surface of the pump suction supply, inches of mercury

h_{vp} = Vapor pressure of the water at the prevailing water temperature, inches of mercury

h_f = Friction losses in suction piping, feet of water

6.0 ATMOSPHERIC RELIEF DEVICES

6.1 Determination of Atmospheric Relief Device Sizes

6.1.1 The size of atmospheric relief devices is dependent upon the local operating conditions. It is always understood that they must be of sufficient size to pass all of the steam which can be admitted to a condenser through any openings, except from the lines which are already protected by relief devices set to open at pressures not exceeding 10 lbs./sq. in. gauge. For example, an extraction turbine would normally require an atmospheric relief device of sufficient size to pass the full throttle steam flow to the condenser under normal operation.

6.1.2 When relief devices are designed for maximum non-condensing operation, they should be sized to pass all incoming steam at the design non-condensing pressure. It is suggested that in lieu of an automatic atmospheric relief device being sized for this flow, a relief device for protection be provided and a separate gate or butterfly valve be provided for additional flow for non-condensing operation.

6.1.3 The size of atmospheric relief devices for normal operation of condensing turbines should be based on the following criteria for selection:

6.1.3.1 Device size and associated piping should be selected to prevent pressure in condenser from exceeding 10 psig.

6.1.4 Relief devices should be located and installed so they are readily accessible for inspection and repair and so they cannot be rendered inoperative. The protective devices need not be directly installed on the condenser but may be installed on turbine exhaust hood or source of pressure external to the condenser.

6.2 Design Features Recommended

6.2.1 A water seal around the valve disc of ample depth to insure proper sealing of the seat with provision for adequate drainage should be provided.

6.2.2 If pilot valve or other auxiliary operator is used, design should be such that valve will open and discharge full rated capacity even though some essential part of pilot valve or auxiliary device should fail. In addition, when pilot valve or auxiliary device is used, protected side of relief valve should be equipped with a warning device set to sound when valve is open and/or pressure is above set point.

6.2.3 Valve should be equipped with manual lifting or opening device.

6.2.4 Rupture disks must be supplied with adequate support for full vacuum and protection from mechanical damage or corrosion.

6.2.5 Exhaust from all relief devices must be vented to avoid injury to personnel or equipment.

6.2.6 Relief devices and/or supplementary valves should be supported and provision made to eliminate discharge thrust and/or thermal expansion forces from being transmitted to condenser shell.

TABLE 6

ATMOSPHERIC RELIEF VALVE SIZES

Maximum Steam Flow Pounds Per Hour	Sizes of Atmospheric Relief Valves Inches	
	For Production*	For Maximum Non- Condensing Operation*
Up to 7,500	6	8
7,501 to 11,800	8	10
11,801 to 17,000	8	12
17,001 to 20,000	8	14
20,001 to 23,100	10	14
23,101 to 30,200	10	16
30,201 to 38,200	12	18
38,201 to 45,000	12	20
45,001 to 47,200	14	20
47,201 to 62,000	14	24
62,001 to 68,000	16	24
68,001 to 82,000	16	30
82,001 to 106,000	18	30
106,001 to 120,100	18	-
120,001 to 170,000	20	-
170,001 to 250,000	24	-
250,001 to 380,000	30	-
380,001 to 550,000	36	-

*The sizes listed "For Protection" are normally used under ordinary condensing operation and are for general reference only. If it is desired to operate the turbine temporarily non-condensing at its maximum non-condensing capacity, the sizes listed under "For Maximum Non-Condensing Operation" should be used. Actual design conditions, i.e., flow, relieving pressure should be established by the Purchaser and condenser Manufacturer. The valve relieving capacity and design should be certified by the valve supplier.

7.0 CONSTRUCTION STANDARDS

7.1 Mechanical Design Criteria

7.1.1 Scope

A. These Standards cover the minimum construction requirements for the design and fabrication of direct contact condensers.

B. These Standards have been formulated on the basis of design principles and construction practices applicable to vessels designed for vacuum service with an internal pressure equal to or less than atmospheric pressure.

C. The recommendations of these Standards terminate at the connections to external piping.

7.1.2 Nozzle Loads

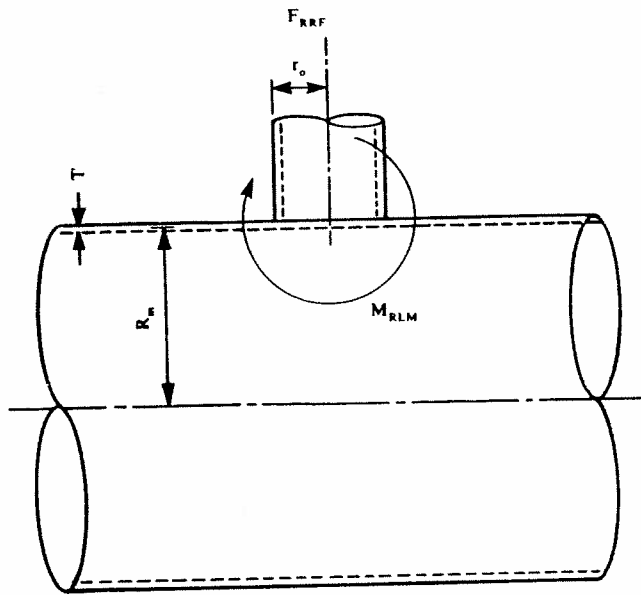
The determination of acceptable nozzle loads is a complex problem involving the interaction of external forces and moments applied at the vessel wall. These loads are functions of the piping mechanical and thermal design. Frequently, the piping designer had a need to know the allowable loads at the nozzle in order to determine the piping configuration and generate the actual loads. The procedure below permits estimating nozzle loads for cylindrical shells. (The procedure is

based in part on the design data included in Welding Research Council Bulletin 107.5) The allowable loads have been linearized to show the interaction between the maximum permitted external radial load and the maximum permitted applied moment vector.

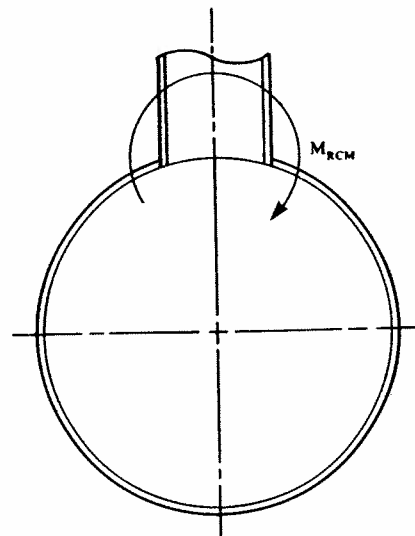
The procedure represents a simplification of the method of WRC 107 and users of the procedure included in this standard are cautioned that more exact analysis is required to verify the adequacy of final designs. The stresses considered in developing the procedure have been defined as secondary stresses with stress limits established according to that definition. Although the effect of internal pressure has been included in the combined stresses, the effect of pressure on nozzle thrust has not been included and requires combination with other radial loads.

Loads exceeding those calculated by this method usually require additional reinforcement. The user is cautioned that the higher allowable loads obtained through design modification may require strengthening other parts such as supports, supporting structure and floors. It should be understood by the user that the condenser is not intended to serve as an anchor point for the piping.

Fig. 18
NOMENCLATURE



Elevation or Plan



Transverse Section

**PROCEDURE FOR CALCULATING NOZZLE
EXTERNAL FORCES AND MOMENTS IN CYLIN-
DRICAL VESSELS**

7.1.2.1 Nomenclature

- P = Design Pressure, pounds per square inch
 r_o = Nozzle Outside Radius, inches
 R_m = Mean Radius of Shell, inches
 T = Shell Thickness, inches
 S_Y = Yield Strength of Material at Design Temperature, pounds per square inch
 σ = Stress Due to Design Pressure, pounds per square inch
 S_a = ASME Section VIII, Division 1 Code Allowable Stress, pounds per square inch
 β = Dimensionless Numbers
 γ = Dimensionless Numbers
 α = Dimensionless Numbers
 Σ = Dimensionless Numbers
 Δ = Dimensionless Numbers
 F_{RRF} = Maximum Resultant Radial Force, pounds*
 M_{RCM} = Maximum Resultant Circumferential Moment, inch-pounds*
 M_{RLM} = Maximum Resultant Longitudinal Moment, inch-pounds*
 F_{RF} = Maximum Resultant Force, pounds*
 M_{RM} = Maximum Resultant Moment, inch-pounds*

*Use absolute values.

7.1.2.2 External Forces and Moments**

To calculate the maximum force and moment, first evaluate β and γ . Then determine α , Σ , and Δ from Figures 19, 20 and 21 for the specified β and γ , substitute into the equations below and calculate RRF, R_{CM} and R_{LM} .

$$\beta = 0.875 \left(\frac{r_o}{R_m} \right) \gamma = \frac{R_m}{T}$$

** Local Stresses in Spherical and Cylindrical Shells Due to External Loadings, K.R. Wichman, A.G. Hopper and J.L. Mershon-Welding Research Council. Bulletin 107/August, 1965 - Revised Printing - December 1968.

Determine β , Σ , and Δ from Figures 19, 20 and 21. Calculate Pressure Stress (σ).

$$\sigma = \frac{2P}{T} \left(R_m - \frac{T}{2} \right)$$

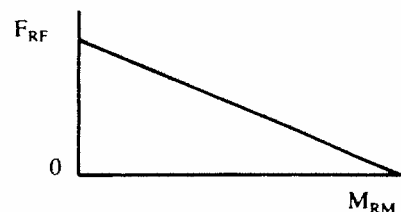
If (σ is greater than S_a , then use S_a as the stress due to design pressure:

$$F_{RRF} = \frac{R_m^2}{\alpha} (S_Y - \sigma)$$

$$M_{RCM} = \frac{R_m^2 r_o S_Y}{\Sigma}$$

$$M_{RLM} = \frac{R_m^2 r_o}{\Delta} (S_Y - \sigma)$$

Plot the value of F_{RRF} as F_{RF} and the smaller of M_{RCM} and M_{RLM} as M_{RM} . The allowable nozzle loads are bounded by the area of F_{RF} , 0, and M_{RM} .



7.1.2.3 Sample Problem

Determine Resultant Force and Moment

$$R_m = 37.5$$

$$S_Y = 31,500 \text{ psi at } 460^\circ$$

$$r_o = 15"$$

$$S_a = 17,500 \text{ psi}$$

$$T = 0.75"$$

$$P = 150 \text{ psi}$$

$$\beta = 0.875 \left(\frac{r_o}{R_m} \right) = 0.875 \left(\frac{15}{37.5} \right) = .35$$

$$\gamma = \frac{R_m}{T} = \frac{37.5}{0.75} = 50$$

From Figure 19, $\alpha = 440$

From Figure 20, $\Sigma = 1.070$

From Figure 21, $\Delta = 340$

Calculate Pressure Stress

$$\alpha = \frac{2P}{T} \left(R_m - \frac{T}{2} \right) = \frac{2(150)}{0.75} \left(37.5 - \frac{0.75}{2} \right)$$

$$= 14,850 \text{ psi} < S_n = 17,500 \text{ psi}$$

Use $\sigma = 14,850$ in the equations for calculating F_{RRF} and M_{RLM} .

Calculate Allowable Forces and Moments

$$F_{RRF} = \frac{R_m^2}{\alpha} (S_Y - \sigma)$$

$$= \frac{(37.5)^2}{440} (31,500 - 14,850) = 53,214 \text{ lbs.}$$

$$M_{RCM} = \frac{R_m^2 r_o S_Y}{\Sigma}$$

$$= \frac{(37.5)^2 (15) (31,500)}{1,100} = 604,048 \text{ in.-lbs.}$$

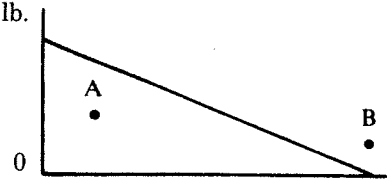
$$M_{RLM} = \frac{R_m^2 r_o}{\Delta} (S_Y - \sigma)$$

$$= \frac{(37.5)^2 (15)}{340} (31,500 - 14,850)$$

$$= 1,032,973 \text{ in.-lbs.}$$

Plot the value of F_{RRF} as F_{RF} and the smaller of M_{RCM} and M_{RLM} as M_{RM} . The allowable nozzle loads are bounded by the area of F_{RF} , 0, and M_{RM} .

$$F_{RF} = 53,214 \text{ lb.}$$



$$M_{RM} = 604,048 \text{ in.-lbs.}$$

Therefore, a nozzle reaction of $F = 20,000$ lbs. and $M = 100,000$ in.-lbs. would be allowable (point A) but a nozzle reaction of $F = 5,000$ lbs. and $M = 604,000$ in.-lbs. would not be allowable (point B).

*Note: Use absolute values in the graph.

Fig. 19
ALLOWABLE NOZZLE LOADS

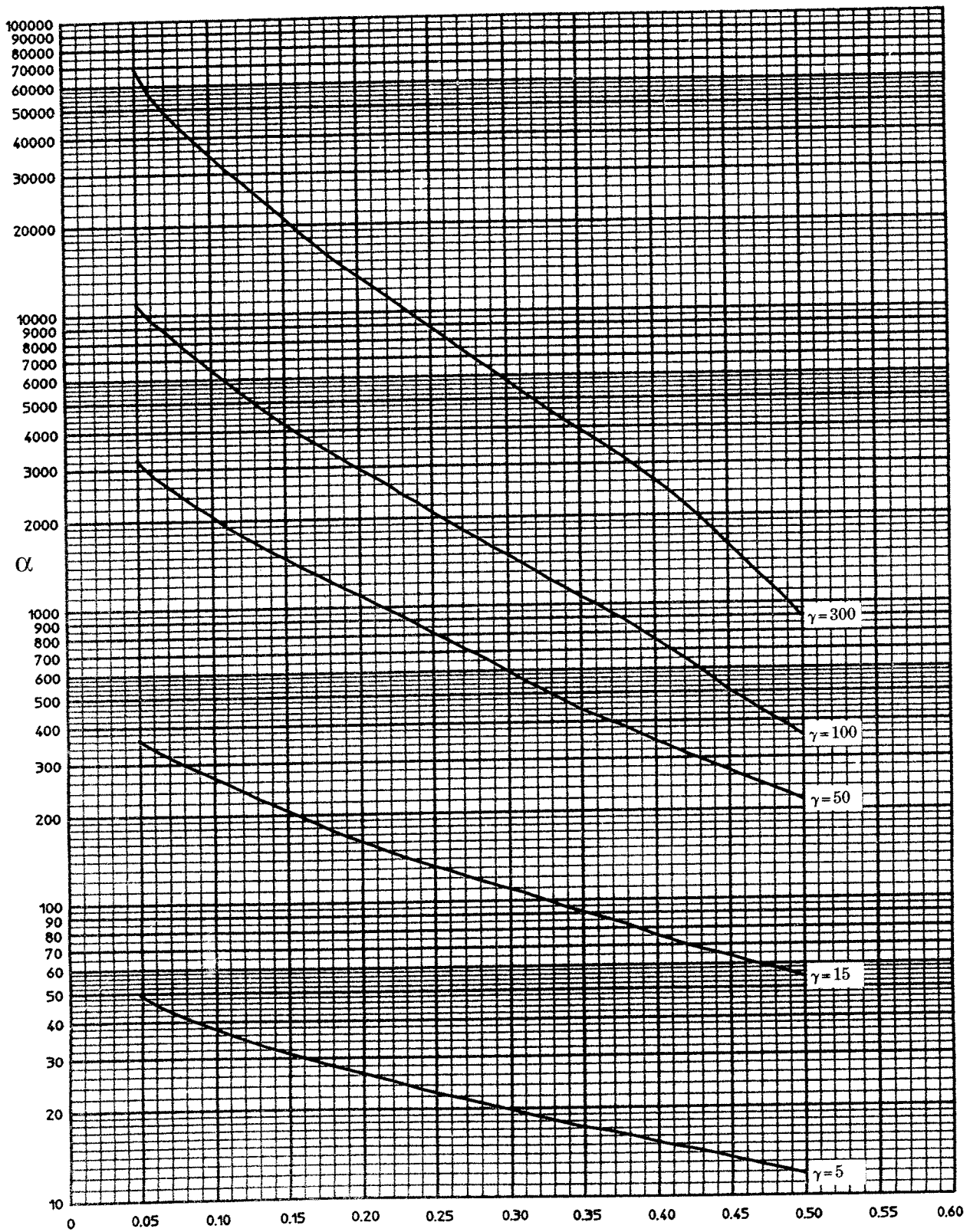


Fig. 20
ALLOWABLE NOZZLE LOADS

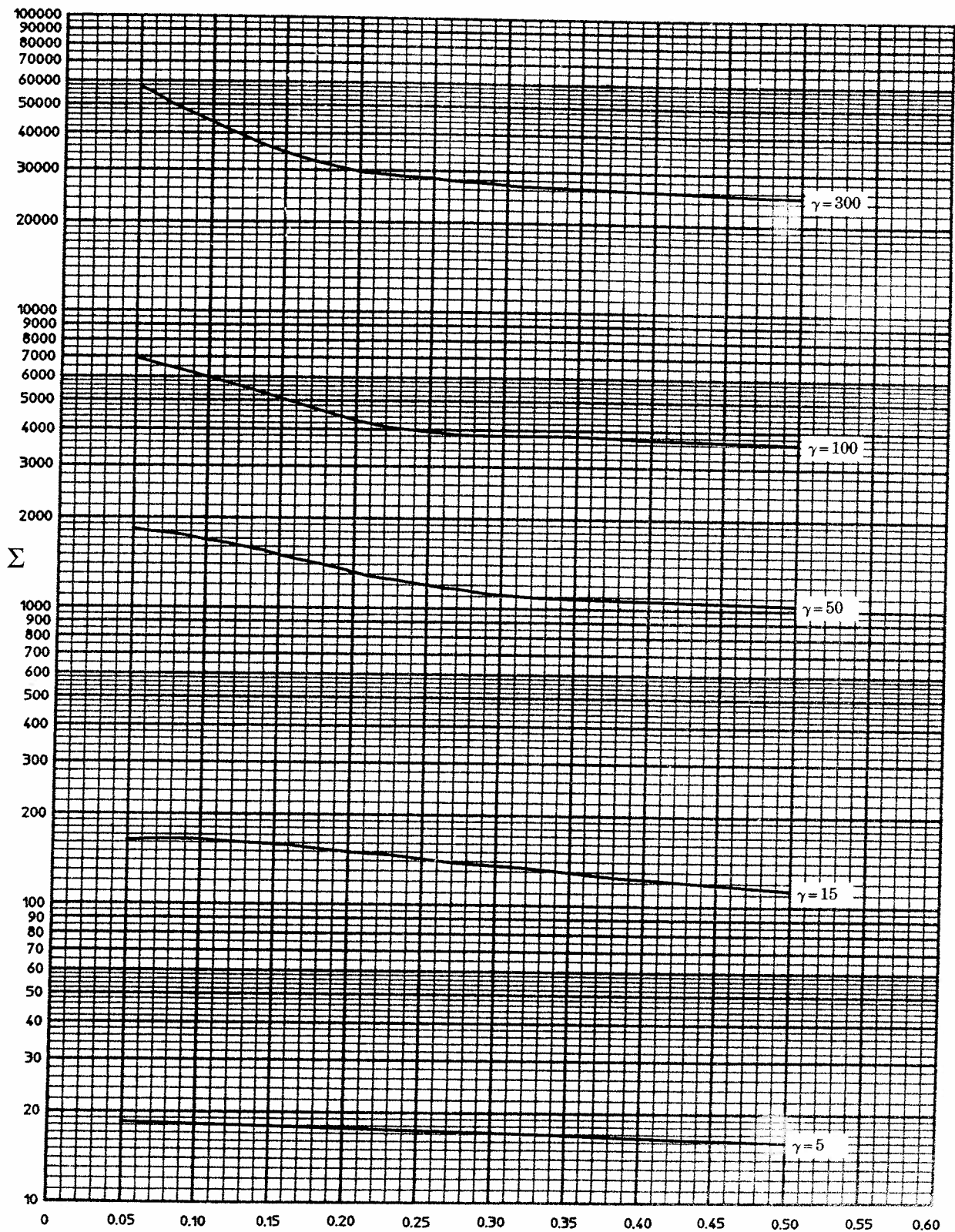
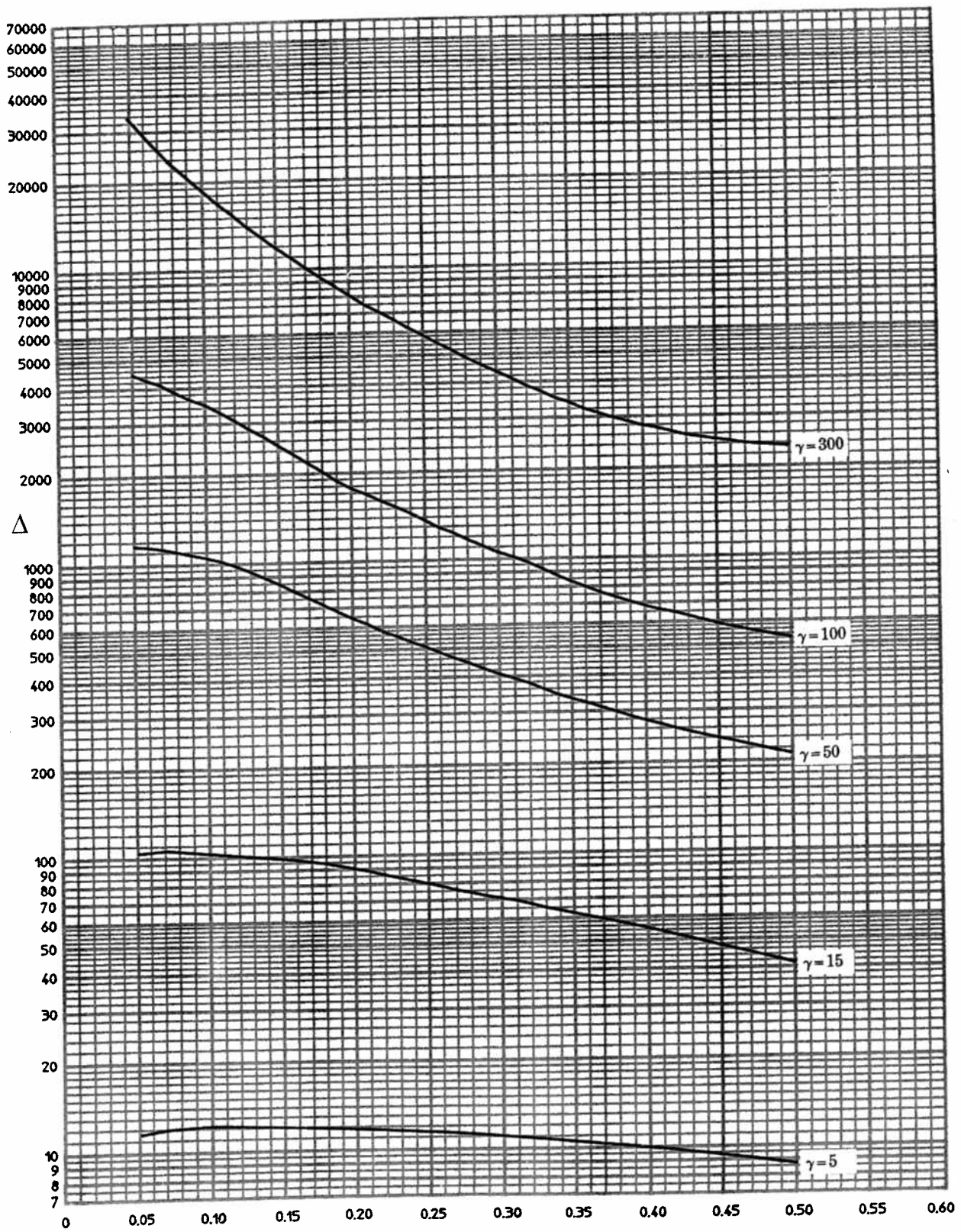


Fig. 21
ALLOWABLE NOZZLE LOADS



7.1.3 External Loading Connecting Piping

A. The piping connections on the condensers are designed to withstand the normal loading required to make up the connection on axial alignment and parallel to the connection on the condenser.

B. For all connections, except the tailpipe connection, the maximum forces and moments from external loading, acting on each connection, shall not exceed the following:

$$\text{Force} = 50d_n \quad \text{Moment} = 100d_n$$

where:

Force = Resultant of all forces from external loading-lbs.

Moment = Resultant of all moments from external loading-ft. lbs.

d_n = Nominal pipe size, inches.

C. The tailpipe connections are designed for either one of the following conditions:

1. Hanging Tailpipe

When the weight of the condenser is supported by feet attached to the condenser shell, the tailpipe connection is designed to carry the hanging weight of a normal length of vertical tailpipe. The tailpipe should be restrained from swaying by a loose fitting collar at or near the hotwell. The sway restraint must not provide any vertical support to the tailpipe.

2. Supporting Tailpipe

When the weight of the condenser is supported by the tailpipe, the tailpipe connection on the condenser shell is designed to carry the static weight of the condenser. It is not designed to carry any turning moment.

7.1.4 Wind and/or Seismic Loading

A. Condenser Supported on Feet. When the weight of the condenser is supported by feet attached to the condenser shell, the feet shall be designed to withstand the wind and/or seismic loading specified.

B. Condenser Supported on Tailpipe. When the weight of the condenser is supported on the tailpipe, the tailpipe connection on the condenser shell is not designed to withstand any turning moment caused by wind or seismic loading. The condenser shall be provided with guy wires, or it shall be horizontally attached to some structure with suitable bracing.

7.2 Typical Materials of Construction Fabricated Direct Contact Condensers

Material	Specification
Carbon Steel Plate	ASTM A36 ASTM A283 ASTM A285 ASTM A515 ASTM A516
Stainless Steel Plate	ASTM A240 UNS S30403 ASTM A240 UNS S31603
Carbon Steel Pipe	ASTM A53 ASTM A106

Stainless Steel Pipe	ASTM A312 ASTM A376
Carbon Steel Bar	ASTM A36 ASTM A675
Stainless Steel Bar	ASTM A276 UNS S30400 ASTM A276 UNS S31600

Note: Allowable stress values should be taken from Section II, Part D of the ASME Pressure Vessel Code.

7.3 Corrosion Allowance

The following are minimum recommended corrosion allowances unless otherwise specified by the customer.

A. Carbon Steel. Shell pressure boundary plates and welds shall be provided with a $\frac{1}{32}$ " corrosion allowance on each wetted side.

B. Lined Material. No corrosion allowance, unless otherwise specified by the customer.

C. Corrosion Resisting Alloys. No corrosion allowance, unless otherwise specified by the customer.

7.4 Name Plates

A suitable resistant name plate, giving manufacturer's name, serial number, and other pertinent data shall be attached to the condenser.

7.5 Manufacturer's Testing for Leakage

When practical, condensers are to be hydrostatically tested to a pressure of 1-1/2 times the design pressure. In cases where it is difficult to blank off the condenser connection openings for reasons of their size, shape, or weld ends, it is considered satisfactory, in lieu of the normal hydrostatic test, to fill the condenser with water to the highest possible level or by other methods mutually agreed upon with the customer.

7.5.1 Impact Properties

It is recommended that carbon steel shells shall not be subjected to hydrostatic test conditions where the material temperature will be below 60 degrees F. If the customer anticipates lower test temperatures, he shall specify the material to be used.

7.6. Preparation for Shipment

7.6.1 Cleaning Exterior Surfaces

Exterior unmachined surfaces are to be cleaned with a suitable method to remove oil and dirt.

7.6.2 Painting Exterior Surfaces

A. Carbon Steel – One coat of primer or machinery enamel

B. Other Materials – Painting not required for corrosion resistant materials.

7.6.3 Protection of Machined Flanged Surfaces

A. Apply protective coating

B. Openings to be closed with suitable covers

7.6.4 Protection of Weld End Connections

Ends are to be covered with suitable water-proof paper, tape sealed.

8.0 FABRICATION STANDARDS

8.1 Shell Thickness

8.1.1 Fabricated shells for direct contact barometric condensers shall be designed when specified in accordance with the ASME Code, Paragraph UG-27 for internal pressure and Paragraph UG-28 for external pressure. To determine if nozzle reinforcing is required, refer to the ASME Code, paragraph UG-37. The required corrosion allowance, if any, must be added to the calculated thickness.

8.1.2 When a condenser has been designed with an unsupported length, *L*, dependent upon the use of circumferential stiffening rings (either internal or external), the required moment of inertia of such rings shall be determined from Fig. 22. In the case of internal rings, the section providing the required moment of inertia must have corrosion allowance, if any, added to all exposed surfaces.

8.2 Flat Covers

8.2.1 Flat Covers shall be designed in accordance with the Code, Paragraph UG-34, and all applicable appendices.

8.2.2 The thickness of bolted covers, shown in Fig. 23, is based on the use of the flanges shown in Table 7 of this section. Covers bolted to flanges with larger bolt circle diameters than those shown in Table 7 will require additional thickness.

8.3 Dished Heads

A. Flanged and dished heads operating under internal pressure shall be designed in accordance with the Code, Paragraph UG-32, and all applicable appendices.

B. Flanged and dished heads operating under external pressure shall be designed in accordance with the Code, Paragraph UG-33, and all applicable appendices.

Required corrosion allowance, if any, shall be added to the thickness determined from these formulae. The total thickness of flanged and dished heads shall not be less than the thickness of the shell to which it is attached.

8.4 Conical Heads

A. Conical heads operating under internal pressure shall be designed in accordance with the Code, Paragraph UG-32, and all applicable appendices.

B. Conical heads operating under external pressure shall be designed in accordance with the Code, Paragraph UG-33, and all applicable appendices.

Required corrosion allowance, if any, shall be added to the thickness determined from these formulae. The total thickness of conical heads shall not be less than the thickness of the shell to which it is attached.

8.5. Manholes and Inspection Openings

Manholes and/or inspection openings are not standard. When specified, manholes should be a minimum of 18 inches in diameter and inspection openings 4 inches in diameter.

8.6 Interface Configurations

Flanges (Note: Refers only to bolting geometry of interface between customer's piping and manufacturer's connection).

ANSI B-16.5. Pipe Flanges and Flanges Fittings (Including rating for class 150, 300, 400, 600, 900, 1500 and 2500).

ANSI B-16.1. Cast Iron Pipe Flanges and Flange Fittings Class 25, 125, 250 and 500.

Other mutually agreed upon flange design.

8.6.1 Weld Ends

To suit required installation.

8.6.2 Threaded Connections

In accordance with ANSI Standard B1.20.1-1983 Pipe Threads, General Purpose (inch).

8.7 Supports

Barometric condensers are to be provided with supporting feet which are designed to avoid undue stress in the supports or shell.

8.7.1 Number of Supporting Feet

A. Two (2) feet are to be provided on condensers of shell diameter 48 inches or less.

B. Four (4) feet are to be provided on condensers of shell diameter greater than 48 inches.

8.7.2 Continuous Ring

A continuous ring can be used for support in place of the supporting feet.

8.8 Welding

For applicable joint efficiencies to be used with formulas in the Standard, refer to Table UW-12 in the Code.

For fillet welds, the stress shall not exceed 0.55 times allowable (S_A) stress of base material based upon minimum leg length ($0.55 S_A = 0.78 S_A \times \text{Sine of } 45^\circ$).

For designs not specifically covered in this section, all welds should be of sufficient size and configuration to transfer the loads in the adjoining plates at an allowable stress level that is not greater than that of the base material.

8.8.1 Types of Welds and Joints

Pressure boundary welds may be either full depth or properly designed fillet welds. For one sided full depth welds, a backing strip shall be used. Where fillet welds are utilized, the weld configuration must be such as to be adequate to support whatever loads are present at the joint.

For structural joints other than pressure boundary, the minimum leg distance for fillet welds shall be 5/16" or the thickness of the thinner member of the joint, whichever is less.

Non-pressure boundary and non-structural welds shall be designed in accordance with good engineering practice.

All butt-type field welds used for assembly of the pressure boundary will be full depth welds. When the weld is made without a backing, it must be welded from both sides unless otherwise specified.

Fig. 22
STIFFENING RINGS
REQUIRED MOMENT OF INERTIA I

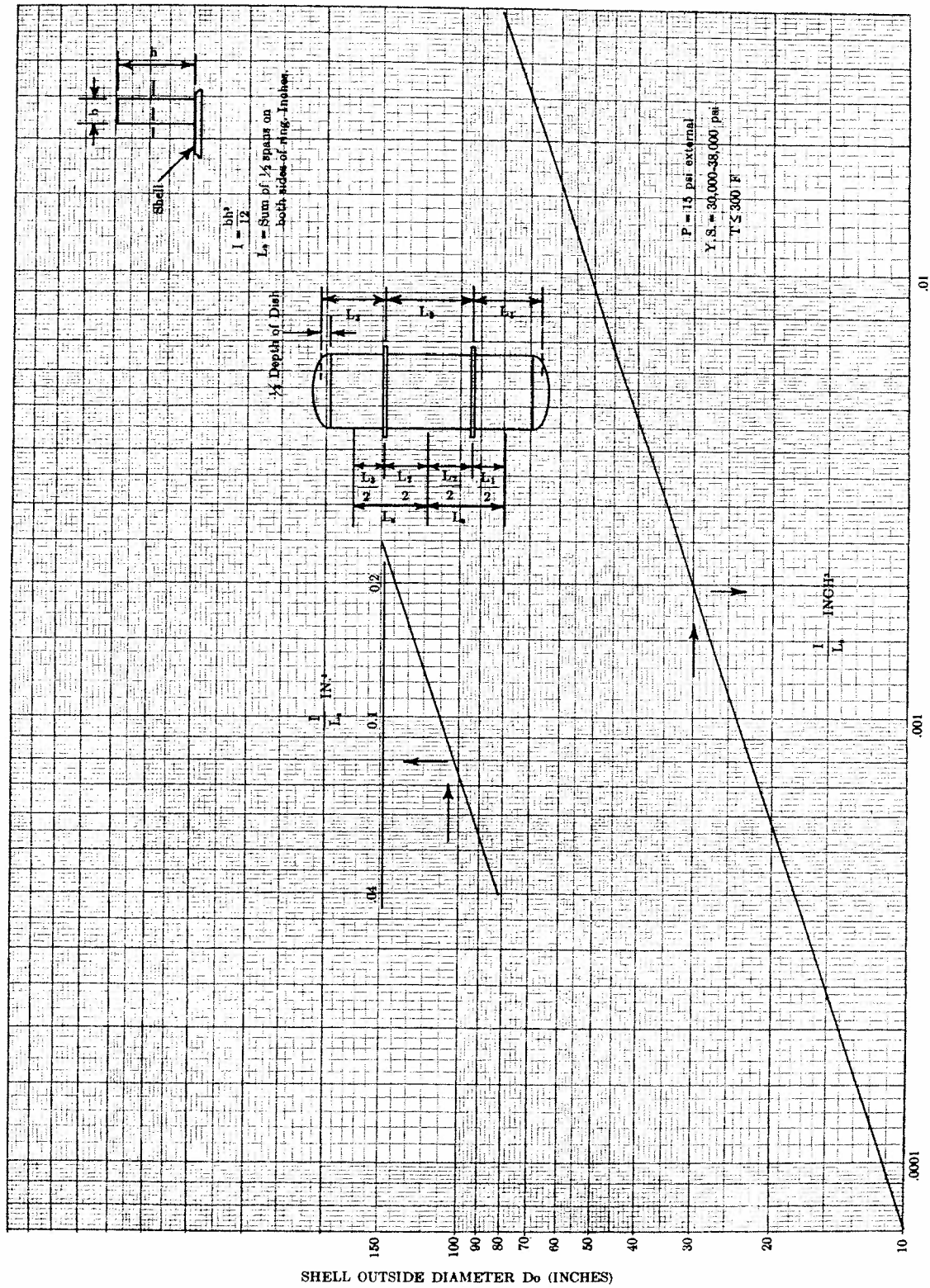


Fig. 23
FLAT COVERS

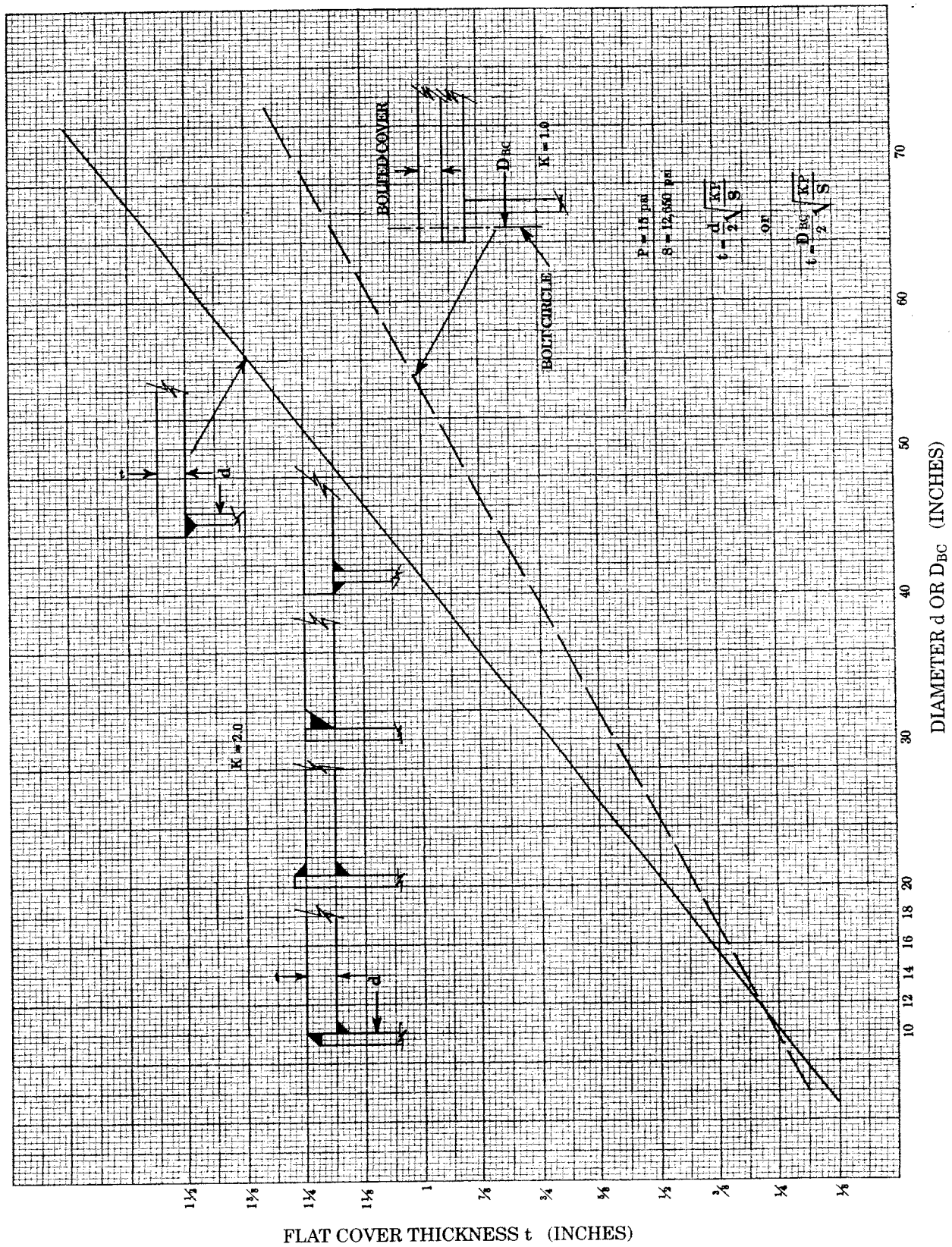
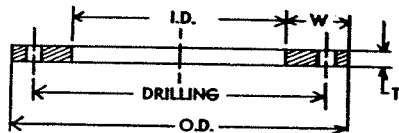


TABLE 7
STANDARD FLANGES FOR VACUUM SERVICE

Material: See Part II,
Fabrication Standards



Following notes apply to Table below:

- (1) Dimensions W and T above 24 inches are rectangular bar sizes.
- (2) Use medium carbon steel heat treated bolting, A325 (SAE grade 5) over 24 inch size. Bolts are 1/8 inch smaller than hole size.
- (3) 1/8 inch thick compressed asbestos gaskets to be used.

Size	O.D.	I.D.	W(1)	T(1)	Drilling(2)		Gasket(3) Size
					No. & Size Holes	B.C.	
4	STANDARD 125 LB ANSI DIMENSION			5/8	STANDARD 125 LB ANSI WITH CARBON STEEL BOLTS		9 x 6
5				5/8			10 x 7
6				5/8			11 x 8
8				5/8			13 1/2 x 10
10				5/8			16 x 12 1/2
12				5/8			19 x 15
14				3/4			21 x 16 1/2
16				3/4			23 1/2 x 19
18				3/4			25 x 20 1/2
20				3/4			27 1/2 x 22 1/2
24				3/4			32 x 27
30	36 1/4	30 1/4	3	1	28—3/4	34 3/8	36 1/4 x 32 1/2
36	42 1/4	36 1/4	3	1	32—3/4	40 3/8	42 1/4 x 38 1/2
42	48 1/4	42 1/4	3	1 1/4	40—3/4	46 1/4	48 1/4 x 44 1/4
48	54 1/4	48 1/4	3	1 1/4	44—3/4	52 1/4	54 1/4 x 50 1/4
54	60 1/4	54 1/4	3	1 1/2	44—7/8	58 1/4	60 1/4 x 56 1/4
60	66 1/4	60 1/4	3	1 1/2	52—7/8	64	66 1/4 x 61 3/4
66	72 1/4	66 1/4	3	1 3/4	52—7/8	70	72 1/4 x 67 3/4
72	78 1/4	72 1/4	3	1 3/4	60—7/8	76	78 1/4 x 73 3/4
78	84 1/4	78 1/4	3	2	60—1	81 3/4	84 1/4 x 79 1/4
84	90 1/4	84 1/4	3	2	64—1	87 3/4	90 1/4 x 85 1/4
90	96 1/4	90 1/4	3	2	68—1	93 3/4	96 1/4 x 91 1/4
96	102 1/4	96 1/4	3	2 1/4	68—1	99 3/4	102 1/4 x 97 1/4
102	109 1/4	102 1/4	3 1/2	2 1/2	68—1 1/8	106 1/4	109 1/4 x 103 1/4
108	115 1/4	108 1/4	3 1/2	2 1/2	72—1 1/8	112 1/4	115 1/4 x 109 1/4
114	121 1/4	114 1/4	3 1/2	2 1/2	72—1 1/4	118 1/4	121 1/4 x 115 1/4
120	127 1/4	120 1/4	3 1/2	2 1/2	76—1 1/4	124 1/4	127 1/4 x 121 1/4

8.8.2 Inspection Standard for Welds

This section establishes minimum standards for visual inspection of ejector components. The visual acceptance criteria were developed using recognized codes and standards such as ASME codes, ANSI standards, AWWA, and AWS as a guide. More stringent requirements may be specified by the purchaser and would be governing.

Non-destructive examination, where not required by the Code, is not performed except by special arrangement between the manufacturer and the purchaser.

The welding shall be performed using welders and written weld procedures which have been qualified in a manner comparable to that defined in Section IX of the ASME Unfired Pressure Vessel Code.

Welds shall be examined in the "as welded" condition preceded only by normal cleaning.

8.8.3 Weld Inspection Methods and Equipment

Personnel performing inspections shall be qualified to perform eye examinations in accordance with Society of Nondestructive Testing (SNT) requirements.

Inspections shall be performed with the naked eye or corrective lenses as required to obtain the visual acuity required by SNT. Magnifying glasses, microscopes, and other optical devices shall not be used.

Any required measurements of welds shall be made using instruments such as scales, rules, calipers, gauges (height and depth), fillet gauges, and dial-type equipment. Micromasuring or vernier-type equipment is not required.

Accuracy of measurement equipment shall be to 1/64". All measuring equipment shall be maintained and calibrated in accordance with the manufacturer's approved quality control manuals and procedures.

8.8.4 Weld Categories

The following categories are established considering the service requirements of specific types of welds.

These criteria apply to shop welds and to field welds in the apparatus except for pipe welds made to connection stubs.

Category I includes pressure boundary welds.

Category II includes structural welds.

Category III includes all other welds.

8.8.5 Acceptance Levels

Acceptance levels for various types of welds in each of Categories I, II, III are tabulated in Table 8. Refer to Figure 24 for nomenclature.

8.9 Tolerances

A. Flange face shall not deviate from normal plane as measured at O.D. of flange by more than the following:

Size	Deviation
0-4 inches	1/16 inch
4-12	3/32 inch
Above 12 inches	3/16 inch

B. Vapor inlet and outlet connections shall not deviate more than $\pm 1/8$ inch from the centerline to the face of the connection. Water inlet and outlet connections shall not deviate more than 3/16 inch. Length dimensions shall not deviate more than the following:

Size	Deviation
Up to 24 inches	$\pm 1/8$ inch
24 to 60 inches	$\pm 1/4$ inch
Above 60 inches	$\pm 3/8$ inch

C. Supports should be held within $+1/4 -0$ inch when measured above the vapor inlet. It is suggested that all vertical dimensions be referenced to vapor inlet.

TABLE 8
WELD ACCEPTANCE CRITERIA*

SUBJECT	REFERENCE	ACCEPTANCE CRITERIA		
		CATEGORY I	CATEGORY II	CATEGORY III
Fillet Weld Fit-Up	S Sketch 1	$x \leq \frac{S}{4}$ (1/8" Maximum) See Notes 4 and 5.	For $T < 3"$ $x \leq 3/16"$ For $T \geq 3"$ $x \leq 3/16"$ See Notes 4 and 5.	
Fillet Weld Size Minimum	S Sketch 1	$S_{Min} = S - 0"$ See Note 5.	$S_{Min} = S - 1/16"$ For .1 L_w See Note 5.	$S_{Min} = S - 1/16"$ For .25 L_w See Note 5.
Fillet Weld Size Maximum	S Sketch 1	$S_{Max} = S + T$ or $S + 3/8"$ (Whichever is greater) See Note 5.		
Fillet Weld Concavity	(-) C Sketch 1	Concavity shall not encroach upon minimum required weld throat.	$(-)C \leq 1/16"$ for .1 L_w (Not to exceed 3" in any foot.)	$(-)C \leq 1/16"$ for .25 L_w (Not to exceed 6" in any foot.)
Fillet Weld Convexity	C Sketch 1	$C \leq \frac{S}{6} + 1/32"$	$C \leq \frac{S}{6} + 1/16"$	
Weld Overlap	ϕ Sketch 2	None Allowed	$\phi \leq 3/32"$ Depth for a .1 L_w Length of ϕ not to exceed 6" in any 60" length of weld. The fusion line must be visible.	
Butt Weld Fit-Up Misalignment	T Sketch 3	$M = \frac{T}{4}$ (1/4" Maximum)		$M = \frac{T}{4}$ or 1/8" (Whichever is greater)
Taper Required for Butt Weld Misalignment	M Sketch 3	3 to 1, Width of weld to M (Both Sides)	3 to 2, Width of weld to M, both sides only when M exceeds $\frac{T}{8}$	Not Required
Butt Weld Root Open with Backup Bar	G Sketch 5	G_{Min} = as qualified The weld geometry should be such as to permit complete joint penetration where required.		

*See Notes

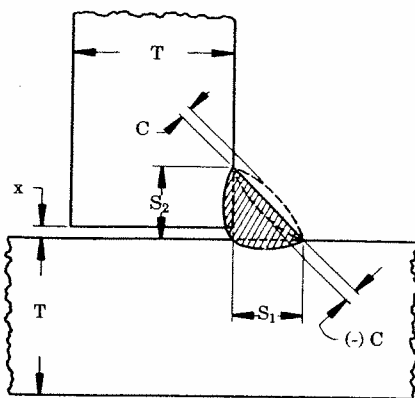
TABLE 8
WELD ACCEPTANCE CRITERIA*

SUBJECT	REFERENCE	ACCEPTANCE CRITERIA		
		CATEGORY I	CATEGORY II	CATEGORY III
Butt Weld Root Opening Without Backup Bar	G Sketch 5	G _{Min} = as qualified Oversize gaps shall be welded with a technique which will close the gap without excessive bridging of the open root. The weld geometry should be such as to permit complete joint penetration where required.		
Butt Weld Size	B Sketch 4	B ≥ T See Note 5.		B = T - $\frac{T}{4}$ ($\frac{T}{4}$ not to exceed 1/8") See Note 5.
Butt Weld Capout Pass Excluding Weld Metal Used For Tapering and Blending	W Sketch 4	W _{Min} = Equal to Width of Weld Geometry W _{Max} ≤ 2-1/2 × thickness of the thicker of the members being joined. Note: Capout may consist of multiple passes.		
Butt Weld Reinforcement on Both Sides	R Sketch 4	When T ≤ 1/2" R = 1/8" Max. T > 1/2" or ≤ 1", R = 3/16" Max. T > 1" or ≤ 2", R = 1/4" Max. T > 2", R = 5/16" Max. Where T is the thinner of the members being joined.		
Undercut	U Sketch 2	1/32" Continuous	1/32" undercut is permitted for full length weld; up to 1/16" for .25 L _w .	
Visible Porosity (Defined as a Hole 1/32" in Diameter or Greater)		No Visible pore 1/32" is allowed.	The sum of the diameters shall not exceed 3/8" in any linear in. or 3/4" in any 12" length of weld. the maximum single pore diameter is 1/8". Linear porosity must be separated by a minimum of one pore diameter.	
Visible Cracks		None Allowed		
Visible Cracks		None Allowed		

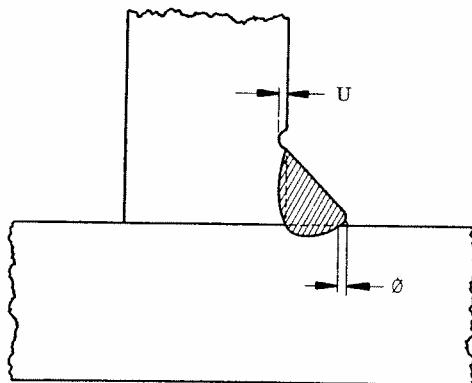
NOTES:

1. Highly localized concavity or weld reinforcement such as caused by weld restarts or where welds join or cross another weld are not to be considered unacceptable under these criteria.
2. Weld ripples transverse to the direction of electrode travel are permitted providing that the crest-to-trough height is not greater than 3/22" and providing that the weld throat dimension is within acceptable range. Valleys between successive weld beads shall not exceed 1/8" providing that minimum weld size is maintained.
3. On weldments made by more than one welder and when combinations of stringer and weave techniques are used, the quality of the weld shall be judged by the visual acceptance criteria defined.
4. If fit-up root gap of fillet welds in any category exceeds 1/16", the fillet weld size shall be locally increased (in the area where fit-up is beyond this criteria) by an amount equal to the size of the separation.

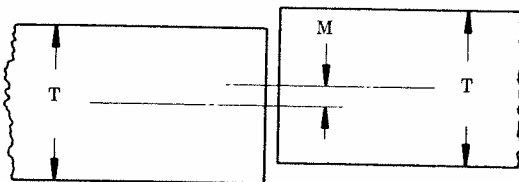
Figure 24
WELDING NOMENCLATURE



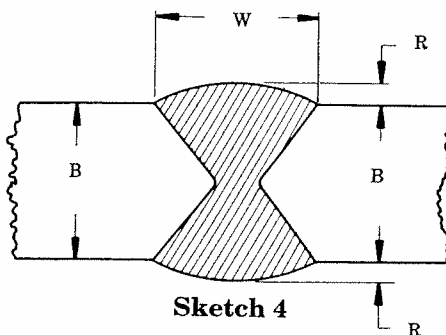
Sketch 1



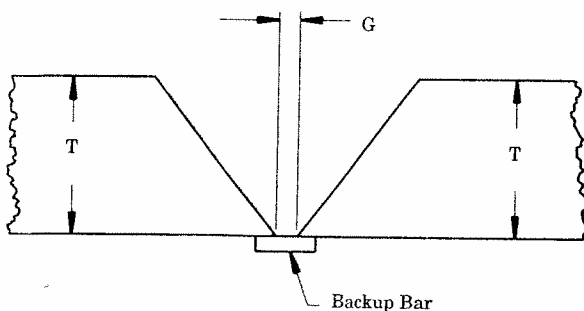
Sketch 2



Sketch 3



Sketch 4



Sketch 5

- S – Min. Specified Weld Leg
- S_1 – Weld Leg Size
- S_2 – Weld Leg Size
- x – Weld Gap (Excluding Local gouges)
- T – Material Thickness of the Thinnest Member
- C – Convexity
- (-) C – Concavity
- L_w – Length of the Weld Described by the Weld Symbol

- Ø – Overlap
- U – Undercut

- M – Misalignment

- B – Size of Butt Weld
- R – Reinforcement
- L_w – Length of the Weld Described by the Weld Symbol
- W – Width of Cap Pass

- G – Root Gap

9.0 QUALITY ASSURANCE

The manufacturer shall have an organized and operational Quality Assurance Program. This program shall be outlined in a Quality Assurance Manual which shall be available to the purchaser and his representatives upon request. The system shall provide for control of quality in both the manufacturer's plant and that of any subcontractor fabricating ejector components. Quality Assurance for work performed in the field by other than the manufacturer shall be the responsibility of the purchaser and/or his agent.

The Quality Assurance Program shall provide for compliance with, but not limited to, the manufacturer's and the Heat Exchange Institute Standards, which shall provide as a minimum:

- A. Base material control (e.g., plate, weld wire, etc.).
- B. Controlled distribution of materials.
- C. Control of purchasing requirements.
- D. Written weld procedures which have been qualified in a manner comparable to that defined in Section IX of the ASME Unfired Pressure Vessel Code.
- E. Qualification of welders and control of welder assignments to assure that the personnel doing welding are qualified.
- F. Audit system for control of procedures.
- G. Detail drawings that contain definition of size and applicable weld procedures.

APPENDIX A

BIBLIOGRAPHY FOR AIR LEAKAGE ESTIMATION METHODS

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APPENDIX B

CONVERSION FACTORS FOR PRESSURE AND VOLUME

CONVERSION FACTORS FOR PRESSURE

Multiply Number of By → To Obtain ↓	Microns	MM Hg	In. Hg	Psi	KG/CM ²
Microns	1	1000	25,400	51714.7	735,600
MM Hg	10^{-3}	1	25.40	51.7147	735.6
In. Hg	3.937×10^{-5}	3.937×10^{-2}	1	2.0359	28.96
Psi	1.934×10^{-5}	1.934×10^{-2}	0.4912	1	14.22
KG/CM ²	1.36×10^{-6}	1.36×10^{-6}	3.453×10^{-2}	7.0304×10^{-2}	1

CONVERSION FACTORS FOR VOLUME

Multiply Number of By → To Obtain ↓	Cubic Centi- meters	Cubic Feet	Cubic Inches	Cubic Meters	Cubic Yards	Gallons (Liquid)	Liters
Cubic Centimeters	1	2.832×10^4	16.39	10^6	7.646×10^5	3785	1000
Cubic Feet	3.531×10^{-5}	1	5.787×10^{-4}	35.31	27	0.1337	3.531×10^{-2}
Cubic Inches	6.102×10^{-2}	1728	1	6.102×10^4	46.656	231	61.02
Cubic Meters	10^{-6}	2.832×10^{-2}	1.639×10^{-5}	1	0.7646	3.785×10^{-3}	0.001
Cubic Yards	1.308×10^{-6}	3.074×10^{-2}	2.143×10^{-5}	1.308	1	4.951×10^{-3}	1.308×10^{-3}
Gallons (Liquid)	2.642×10^{-4}	7.481	4.329×10^{-2}	264.2	202.0	1	0.2642
Liters	0.001	28.32	1.639×10^{-2}	1000	764.6	3.785	1

Conversion equation for
Centigrade and Fahrenheit

$$F = 1.8C + 32$$

$$C = \frac{F - 32}{1.8}$$

APPENDIX C

UNITS OF MEASURE AND CONVERSION FACTORS NOMENCLATURE

NAME		SYMBOL	OTHER UNITS
inch/inches		in	
foot/feet		ft	
meter		m	
millimeter	(SI)	mm	
square inch		in ²	
square foot		ft ²	
square meter		m ²	
square centimeter		cm ²	
square millimeter		mm ²	
cubic inch		in ³	
cubic foot		ft ³	
gallon (US liquid)		gal	
cubic meter	(SI)	m ³	
liter		L	
pound mass (avoirdupois)		lbm	
kilogram	(SI)	kg	
pound force (avoirdupois)		lbf	
kilogram force		kgf	
newton	(SI)	N	
degree Fahrenheit		°F	$m \cdot kg/s^2$
kelvin	(SI)	K	
degree Celsius	(SI)	°C	
British thermal unit (International Table)		Btu	
kilocalorie (International Table)		kcal	
joule	(SI)	J	$N \cdot m, m^2 \cdot kg/s^2$
kilojoule		kJ	
second (customary)		sec	
second	(SI)	s	
minute		min	
hour (customer)		hr	
hour (metric)		h	
watt	(SI)	W	$Js, N \cdot m/s, m^2 \cdot kg/s^3$
pound force/square inch		psi	lbf/in^2
inches of mercury		inHg	
feet of water		ftH ₂ O	
pascal	(SI)	Pa	$N/m^2, kg/(m \cdot s^2)$
kilopascal		kPa	
bar (metric)		bar	
millimeter of mercury		mmHg	
torr		torr	
centipose		cp	

Notes:

1. (SI) Denotes an "International System of Units" unit.
2. Pressure should always be designated as gage or absolute.
3. The acceleration of gravity, g , is taken as 9.80665 m/s^2 .
4. One gallon (US liquid) equals 231 in^3 .

APPENDIX C (continued)

PREFIXES DENOTING DECIMAL MULTIPLES OR SUBMULTIPLES

PREFIX	SYMBOL	MULTIPLICATION FACTOR
micro	μ	0.000 001 = 10^{-6}
milli	m	0.001 = 10^{-3}
centi	c	0.01 = 10^{-2}
deci	d	0.1 = 10^{-1}
deca	da	10 = 10^1
hecto	h	100 = 10^2
kilo	k	1000 = 10^3
mega	M	1 000 000 = 10^6
giga	G	1 000 000 000 = 10^9

CONVERSION FACTORS

LENGTH

MULTIPLY	BY	TO OBTAIN	
in	2.540×10^{-2}	m	(SI)
in	2.540×10^1	mm	
ft	3.048×10^{-1}	m	(SI)
ft	3.048×10^2	mm	

AREA

MULTIPLY	BY	TO OBTAIN	
in ²	6.451600×10^{-4}	m ²	(SI)
in ²	6.451600×10^2	mm ²	
ft ²	9.290304×10^{-2}	m ²	(SI)
ft ²	9.290304×10^4	mm ²	

VOLUME

MULTIPLY	BY	TO OBTAIN	
in ³	1.638706×10^{-5}	m ³	(SI)
in ³	1.638706×10^{-2}	L	
ft ³	2.831685×10^{-2}	m ³	(SI)
ft ³	2.831685×10^1	L	
gal	3.875412×10^{-3}	m ³	(SI)
gal	3.87541	L	

MASS

MULTIPLY	BY	TO OBTAIN	
lbm	4.535924×10^{-1}	kg	(SI)

FORCE

MULTIPLY	BY	TO OBTAIN	
lbf	4.448222	N	(SI)
lbf	4.535924×10^{-1}	kgf	(SI)
kgf	9.806650	N	

TEMPERATURE

$^{\circ}\text{K} = (^{\circ}\text{F} + 459.67)/1.8$	(SI)
$^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$	(SI)
$^{\circ}\text{F} = 1.8^{\circ}\text{K} - 459.67$	
$^{\circ}\text{F} = 1.8^{\circ}\text{C} + 32$	

ENERGY, WORK OR QUANTITY OF HEAT

MULTIPLY	BY	TO OBTAIN	
Btu	1.055056×10^3	J	(SI)
Btu	2.519958×10^{-1}	kcal	
ft · lbf	1.355818	J	(SI)
ft · lbf	3.238316×10^{-4}	kcal	

APPENDIX C (continued)

POWER (ENERGY/TIME)			
MULTIPLY	BY	TO OBTAIN	
Btu/hr	2.930711×10^{-1}	W	(SI)
PRESSURE PR STRESS (FORCE/AREA)			
MULTIPLY	BY	TO OBTAIN	
psi	6.894757×10^3	Pa	(SI)
psi	6.894757	kPa	
psi	6.894757×10^{-2}	bar	
psi	7.030696×10^{-2}	kgf/cm ²	
lbf/ft ²	4.788026×10^1	Pa	(SI)
lbf/ft ²	4.788026×10^{-2}	kPa	
lbf/ft ²	4.788428	kgf/cm ²	
inHg (32°F)	3.38638×10^3	Pa	(SI)
inHg (32°F)	3.38638	kPa	
inHg (32°F)	3.38638×10^{-2}	bar	
inHg (32°F)	3.38638×10^{-2}	kgf/cm ²	
inHg (32°F)	3.38638×10^1	mmHg	
torr (0°C)	1.33322×10^2	Pa	
torr (0°C)	1.0	mmHg	
ftH ₂ O (39.2°F)	2.98898×10^3	Pa	(SI)
ftH ₂ O (39.2°F)	2.98898	kPa	
ftH ₂ O (39.2°F)	3.047915×10^2	kgf/m ²	
VELOCITY (LENGTH/TIME)			
MULTIPLY	BY	TO OBTAIN	
ft/sec	3.048000×10^{-1}	m/s	(SI)
ft/min	5.080000×10^{-3}	m/s	(SI)
MASS FLOW RATE (MASS/TIME)			
MULTIPLY	BY	TO OBTAIN	
lbm/hr	1.259979×10^{-4}	kg/s	(SI)
lbm/hr	4.535924×10^{-1}	kg/h	
VOLUME FLOW RATE (VOLUME/TIME)			
MULTIPLY	BY	TO OBTAIN	
ft ³ /min	4.719474×10^{-4}	m ³ /s	(SI)
ft ³ /min	1.699011	m ³ /h	
gal/min	6.309020×10^{-5}	m ³ /s	(SI)
gal/min	2.271247×10^{-1}	m ³ /h	
gal/min	3.785412	L/min	
MASS VELOCITY (MASS/TIME-AREA)			
MULTIPLY	BY	TO OBTAIN	
lbm/(h · ft ²)	1.35623×10^{-3}	kg/(s · m ²)	(SI)
lbm/(h · ft ²)	4.882428	kg/(h · m ²)	
lbm/(s · ft ²)	4.882428	kg/(s · m ²)	(SI)
SPECIFIC VOLUME (VOLUME/MASS)			
MULTIPLY	BY	TO OBTAIN	
ft ³ /lbm	6.242797×10^{-2}	m ³ /kg	(SI)
ft ³ /lbm	6.242797×10^1	L/kg	
gal/lbm	8.345406×10^{-3}	m ³ /kg	(SI)
gal/lbm	8.345406	L/kg	

APPENDIX C (continued)

DENSITY (MASS/VOLUME)

MULTIPLY	BY	TO OBTAIN	
lbm/in ³	2.767990×10^4	kg/m ³	(SI)
lbm/in ³	2.767990×10^1	kg/L	(SI)
lbm/ft ³	1.601846×10^1	kg/m ³	(SI)
lbm/ft ³	1.601846×10^{-2}	kg/L	(SI)
lbm/gal	1.198264×10^2	kg/m ³	(SI)
lbm/gal	1.198264×10^{-1}	kg/L	(SI)

ENTHALPY (ENERGY/MASS)

MULTIPLY	BY	TO OBTAIN	
Btu/lbm	6.894757×10^3	J/kg	(SI)
Btu/lbm	2.326000	kJ/kg	(SI)
Btu/lbm	5.555556×10^{-1}	kcal/kg	(SI)

HEAT CAPACITY AND ENTROPY (ENERGY/MASS · TEMPERATURE)

MULTIPLY	BY	TO OBTAIN	
Btu/(lbm · °F)	4.186800×10^3	J/(kg · K)	(SI)
Btu/(lbm · °F)	4.186800	kJ/(kg · K)	(SI)
Btu/(lbm · °F)	1.000000	kcal/(kg · °C)	(SI)

THERMAL CONDUCTIVITY (ENERGY · LENGTH/TIME · AREA · TEMPERATURE)

MULTIPLY	BY	TO OBTAIN	
Btu · in/(hr · ft ² · °F)	1.442279×10^{-1}	W/(m · K)	(SI)
Btu · in/(hr · ft ² · °F)	1.240137×10^{-1}	kcal · m/(h · m ² · K)	(SI)
Btu · in/(hr · ft ² · °F)	1.730735	W/(m · K)	(SI)
Btu · in/(hr · ft ² · °F)	1.488164	kcal · m/(h · m ² · K)	(SI)

DYNAMIC VISCOSITY (MASS/TIME · LENGTH OR FORCE · TIME/AREA)

MULTIPLY	BY	TO OBTAIN	
cp	1.000000×10^{-3}	Pa · s	(SI)
lbm/(h · ft)	4.133789×10^{-4}	Pa · s	(SI)
lbm/(h · ft)	4.133789×10^{-5}	cp	(SI)
lbm/(s · ft)	1.488164	Pa · s	(SI)
lbm/(sec · ft)	1.488164×10^3	cp	(SI)
lbm · sec/ft ²	4.788026×10^1	Pa · s	(SI)
lbm · sec/ft ²	4.788026×10^4	cp	(SI)

HEAT FLUX DENSITY (ENERGY/TIME · AREA · TEMPERATURE)

MULTIPLY	BY	TO OBTAIN	
Btu/(h · ft ²)	3.154591	W/m ²	(SI)
Btu/(h · ft ²)	2.712460	kcal/(h · m ²)	(SI)

HEAT TRANSFER COEFFICIENT (ENERGY/TIME · AREA · TEMPERATURE)

MULTIPLY	BY	TO OBTAIN	
Btu/(h · ft ² · °F)	5.678263	W/(m ² · K)	(SI)
Btu/(h · ft ² · °F)	4.882428	kcal/(h · m ² · K)	(SI)

FOULING RESISTANCE (TIME · AREA · TEMPERATURE/ENERGY)

MULTIPLY	BY	TO OBTAIN	
h · ft ² · °F/Btu	1.761102×10^{-1}	m ² · K/W	(SI)
h · ft ² · °F/Btu	2.048161×10^{-1}	h · m ² · K/kcal	(SI)