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G. M. Chambers
By direction

DESIGN DATA SHEET
DEPARTMENT OF THE NAVY, BUREAU OF SHIPS

SECTION DDS4601-1
STEAM CONDENSERS

References:

(a) Interim Military Specification MIL-C-15430B, Condenser, Steam, Surface (Naval Shipboard Use) of 24 February 1953.

(b) Standards of the Heat Exchange Institute, Condenser Section, third edition, 1952.

DDS4601-1-a. General design

This section has been compiled from reference (*a*). Although it specifies certain materials, tube sizes, water velocities, and temperatures, the formulae and methods of calculation given in paragraph DDS4601-1-b are applicable to any condenser.

Tubes.—Condenser tube size is ⅝ inch O. D. with 0.049 inch wall thickness (No. 18BWG); the material should be 70–30 or 90–10 Cu-Ni; the ordering length of the tubes must be a multiple of 6 inches; not more than 1/16 inch of the tube should project beyond the face of the tube sheet.

Tube sheets.—The minimum specified tube sheet thickness is ¾ inch for expanded tubes and 1 inch for packed tubes; the total area of the tube holes in the tube sheet shall not exceed 22 percent (for condensers designed for 1.25 p. s. i. absolute or less at the steam inlet) or 24 percent (for condensers designed for a higher pressure than 1.25 p. s. i. absolute at the steam inlet) of the total tube sheet area determined prior to drilling.

Water box.—The depth of the water box should be at least half the mean tube sheet diameter but not more than 45 inches.

Hotwell.—The volumetric capacity should equal the volume of condensate handled during one minute at full power.

Water velocities in tubes.—

Turbogenerator and auxiliary condensers—6 feet per second maximum.

Main condensers—6 feet per second maximum for double pass or 9 feet per second for scoop injected condensers (single pass).

Heat load.—Assume condensate rejects 950 B. t. u. per pound to cooling water unless actual figure is available from heat balance.

Injection temperature.—75° F.

Basic heat transfer coefficient.—For ⅝ inch No. 18 BWG tubes use 270 times the square root of the water velocity.

Cleanliness factor.—Use 0.85 for tube cleanliness.

Material factor.—Use 0.90 when 90–10 Cu-Ni tubes are used and 0.83 when 70–30 Cu-Ni tubes are used.

Temperature factor.—Use 1.025 for 75° F. inlet (injection) water temperature.

DDS4601-1-b. Detail design

1. The rate at which heat is transferred from steam undergoing condensation to cooling water is dependent on:

(a) Velocity of cooling water through the tubes.

(b) Temperature difference between steam and cooling (circulating) water.

(c) Temperature of cooling water.

(d) The material and surface condition of the tubes.

(e) The presence of air with steam.

(f) Amount of load on condenser.

(g) Size and means of steam distribution within the condenser.

(h) Tube size.

2. Condenser performance is not usually guaranteed at pressures under 0.7 inch of mercury absolute and/or 5° F. terminal difference due to uncontrollable minor variations in condenser plant airtightness and vacuum pump or air ejector effectiveness.

It is also recognized that condenser tube water velocities of less than approximately 3 feet per second do not build up enough flow resistance in condenser tubes to force a substantially uniform quantity through all the tubes of a surface condenser, and hence condenser performance under such conditions cannot be exactly predicted. The figure of 3 feet per second is arbitrary and will vary with tube size and length.

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3. Assume the following nomenclature:

A = total condensing surface based on the O. D. tube area, square feet.

C_p = specific heat of circulating water, B. t. u. per pound per °F.

F_1 = cleanliness correction factor for U due to fouling of tubes. Use 0.85 unless otherwise specified.

F_2 = material correction factor for U. A table of material factors is contained in table 3.

F_3 = inlet water correction factor for U due to cooling water temperatures other than 70° F. If water temperature rise is excessive, use average temperature of water to obtain correction factor in lieu of inlet temperature. See figure 1.

g = g. p. m. per tube at one foot per second water velocity. See table 2.

G = gallons of circulating (injection or cooling) water per minute.

h_{ex} = enthalpy of steam entering condenser, B. t. u. per pound.

Δh_s = subcooling or hotwell depression, B. t. u. per pound.

h_w = enthalpy of condensate leaving condenser hotwell, B. t. u. per pound. Equals enthalpy of saturated water at pressure P_s' in condenser less Δh_s for subcooling of condensate.

Δh = heat removal from steam, B. t. u. per pound.

Δh = $(h_{ex} - h_w)$.

k = tube constant $= \frac{s}{g}$, see table 2.

L = effective length of each tube in feet per pass times number of water passes.

n = number of passes.

N = total number of tubes per water pass.

P_s = pressure at turbine exhaust flange, inches mercury absolute.

P_s' = average pressure in steam side of condenser, inches mercury absolute.

Q = B. t. u. per hour given up by exhaust steam to circulating water.

s = square feet of external tube surface per foot of length. See table 2.

t_i = inlet (injection) circulating water temperature, °F.

t_o = outlet (overboard) circulating water temperature, °F.

t_s = saturated steam temperature at P_s, °F.

t_s' = saturated steam temperature at P_s' °F.

Δt_m = logarithmic mean temperature difference between circulating water and condensing steam, °F.

U = basic heat transfer coefficient in B. t. u. per hour per square foot per °F. log mean temperature difference. See figure 1.

U_c = corrected heat transfer coefficient in B. t. u. per hour per square foot per °F. log mean temperature difference $= UF_1F_2F_3$.

V = velocity of circulating water through tubes, feet per second.

w = density of circulating water, pounds per gallon.

W = steam condensed, pounds per hour.

4. The following equations hold for condensers:

$Q = U_c A \Delta t_m$	Equation (1)
$Q = W \Delta h$	Equation (2)
$Q = 60\ w\ C_p\ G\ (t_o - t_i)$	Equation (3)
$^* = 500\ G\ (t_o - t_i)$	
$\Delta t_m = \dfrac{t_o - t_i}{\log_e\left(\dfrac{t_s - t_i}{t_s - t_o}\right)}$	Equation (4)
$A = L N s$	Equation (5)
$G = N g V$	Equation (6)
$k = \dfrac{s}{g}$	Equation (7)

In condenser design, it is preferable to select a tube length suitable for the application and solve for the resulting temperature rise, surface area, cooling water flow, and other desired characteristics. The following solution is based on this procedure.

By setting (1) = (3) and substituting for A, G, and Δt_m, the following is obtained:

$$\log_e \frac{t_s - t_i}{t_s - t_o} = \frac{U_c L k}{500\ V}$$

Thus:

$$\frac{t_s - t_i}{t_s - t_o} = e^a \qquad \text{Equation (8)}$$

(See table I for values of e^a)

$$\text{where } a = \frac{U_c L k}{500\ V} \qquad \text{Equation (9)}$$

This reduces to:

$$t_o = t_s - \frac{t_s - t_i}{e^a} \qquad \text{Equation (10)}$$

From equations (3) and (2):

$$G = \frac{Q}{500\ (t_o - t_i)} = \frac{W \Delta h}{500\ (t_o - t_i)} \qquad \text{Equation (11)}$$

From equations (5), (6) and (7):

$$A = \frac{L G k}{V} \qquad \text{Equation (12)}$$

From equation (6):

$$N = \frac{G}{g V} \qquad \text{Equation (13)}$$

For part load conditions, the water velocity in the tubes will approximately vary directly as ship's speed for condensers provided with scoops. The cooling water rate, equation (6), varies directly as the water velocity. The heat given up to the condenser at part load can be found from a heat balance.

*Actually $60 w c_p = 60 \times 8.55 \times 0.94 = 482$ for salt water. However, the value of $60 \times 8.33 \times 1 = 500$ for fresh water is used in this method since this is the value used by industry in computing condenser sizes. The difference between surfaces calculated for fresh and salt cooling water is of the order of ½ percent. This is considered within the accuracy of the other data.

Under part load conditions, the area and number of tubes are known. The unknowns are the cooling water outlet temperature and the saturation temperature corresponding to the absolute pressure in the condenser.

From equation (3):

$$t_o=t_i+\frac{Q}{500\,G}=t_i+\frac{W\Delta h}{500\,G}\qquad \text{Equation (14)}$$

From equation (8):

$$t_s=\frac{e^a t_o-t_i}{e^a-1}\qquad \text{Equation (15)}$$

The condenser vacuum can now be found from a book on steam tables. (Keenan and Keyes)

Sample calculations;

Given:

Steam from turbine: 247,000 pounds per hour.

Tube ordering length: 10′–6″

Tube size ⅝″–18 Gage

Material: 70-30 Cu-Ni

Exhaust pressure: 5″ mercury absolute

Single pass ($n=1$)—Scoop injection

From Mil Spec. MIL–C–15430B:

$\Delta h=950$ B. t. u. per pound

$t_i=75°F$.

Effective tube length $=10'6''-(1''$ per tube sheet $+\frac{1}{16}''$ per tube sheet for rolling)

$=10'6''-2\frac{1}{8}''=10'3\frac{7}{8}''=10.32'$

$V=9$ feet per second

$F_1=0.85$

$F_2=0.83$

$F_3=1.025$

Required:

Condensing surface (A), ft²

Circulating water (G) g. p. m.

Circulating water temperature rise, (t_o-t_i)

Terminal temperature difference, (t_s-t_o), shall not be less than 5°F.

Number of tubes per pass, (N)

Approximate overall dimensions of condenser

Performance at part load

Solution:

$U=270\sqrt{V}=270\sqrt{9}=810$ B. t. u./hr-ft²-°F

$U_c=UF_1F_2F_3=810\times0.85\times0.83\times1.025$

$=586$ B. t. u./hr-ft²-°F

From Table 2, $k=0.2406$

$t_s=133.76°F$ at 5 inches of mercury abs. from steam tables

Note: In preliminary design calculations, P_s-P_s' is assumed to be zero. Therefore, t_s is taken at P_s'.

From equation (9)

$$a=\frac{U_cLk}{500\,V}=\frac{586\times10.32\times0.2406}{500\times9}=0.324$$

$e^a=1.383$ from table I.

From equation (10)

$$t_o=t_s-\frac{t_s-t_i}{e^a}=133.76-\frac{133.76-75}{1.383}$$

$=133.76-42.49=91.27°F$

$t_o-t_i=91.27-75=16.27\ °F$

$t_s-t_o=133.76-91.27=42.49\ °F$—This is greater than 5°F and conforms to reference (a).

From equation (11)

$$G=\frac{W\Delta h}{500\,(t_o-t_i)}=\frac{247{,}000\times950}{500\,(91.27-75)}=28{,}840\text{ g. p. m.}$$

From equation (13)

$$A=\frac{LGk}{V}=\frac{10.32\times28{,}840\times0.2406}{9}=7{,}960\text{ ft}^2$$

From equation (6)

$$Nn=\frac{Gn}{gV}=\frac{28{,}840\times1}{0.68\times9}=4{,}710\text{ tubes}$$

To find approximate overall dimensions;

$P_s=5$ inches of mercury absolute $\times0.491=2.46$ p. s. i. absolute.

Reference (a) states that the area of tube holes in the tube sheet shall not exceed 24 percent of the total tube sheet area before drilling, for P_s greater than 1.25 p. s. i. absolute.

Area per tube $=\frac{\pi d^2}{4}=0.7854(\frac{5}{8})^2=0.3068$ square inches

Area of holes $=4{,}710\times0.3068=1{,}450$ square inches

Minimum tube sheet area $=\frac{1{,}450}{0.24}$

$=6{,}040$ square inches $\diamond$ 42 square feet.

Reference (a) also states that the water box depth shall not be less than one-half the mean tube sheet diameter but should not exceed 45 inches.

For a circular tube sheet:

$$d^2=\frac{42}{0.7854}=53.5$$

$d=7.32$ ft. (diameter)

Water box depth $=\frac{7.32}{2}=3.66$ ft. $\diamond$ 43.9 in. $\diamond$ 3 ft.–8 in.

The total length of the condenser including water boxes is approximately:

$(10'\text{–}6'')+2\,(3'\text{–}8'')=10'\text{–}6''+7'\text{–}4''=17'\text{–}10''$

Reference (a) specifies that the hotwell volumetric capacity at maximum designed working level shall be at least equal to the volume of condensate for one minute's operation at rated full load.

$\frac{247{,}000}{60}=4{,}120$ pounds of steam per minute

From steam tables: For saturated condensate, $V=0.0163$ cubic feet per pound at $t=133.8°$ F.

$4{,}120\times0.0163=67.2$ cubic feet of hotwell capacity required.

Part load performance:

Given $W=123{,}500$ pounds per hour

$V=7.5$ feet per second

$\Delta h=950$ B. t. u. per pound

Required: Absolute condenser pressure, P_s'

Solution:

$$G=\frac{28,100\times 7.5}{9}=23,400 \text{ g. p. m.}$$

$$U=270\sqrt{7.5}=270\times 2.74=738 \text{ B. t. u./hr-ft}^2\text{-°F}$$

$$U_c=738\times 0.86\times 0.83\times 1.025=534 \text{ B. t. u./hr-ft}^2\text{-°F}$$

From equation (14):

$$t_o=t_i+\frac{W_a h}{500G}=75+\frac{123,500\times 950}{500\times 23,400}=75+10=85\text{°F.}$$

From equation (9):

$$a=\frac{U_c L k}{500V}=\frac{534\times 10.32\times 0.2406}{500\times 7.5}=0.355$$

From table 1:

$$e^a=1.427$$

From equation (15):

$$t_s=\frac{e^a\, t_o-t_i}{e^a-1}=\frac{1.427(85)-75}{1.427-1}=\frac{121-75}{0.427}=108.2\text{° F.}$$

From steam tables at 108° F. saturation temperature:

$P_s'=2.5$ **inches of mercury, absolute.**

$t_s-t_o=108.2-85=23.2$**° F. which is greater than 5° F. terminal temperature difference and is acceptable.**

TABLE 1

Table of e^a

For intermediate values use direct interpolation or slide rule log—log scales.

a	e^a	a	e^a
0.1	1.105	1.1	3.00
.15	1.159	1.15	3.16
.2	1.222	1.2	3.32
.25	1.284	1.25	3.49
.3	1.350	1.3	3.67
.35	1.418	1.35	3.86
.4	1.402	1.4	4.06
.45	1.568	1.45	4.27
.5	1.648	1.5	4.48
.55	1.734	1.55	4.71
.6	1.822	1.6	4.95
.65	1.916	1.65	5.21
.7	2.015	1.7	5.47
.75	2.138	1.75	5.76
.8	2.225	1.8	6.05
.85	2.34	1.85	6.36
.9	2.46	1.9	6.69
.95	2.58	1.95	7.03
1.0	2.72	2.0	7.39
1.05	2.86	--------	--------

TUBING CHARACTERISTICS

TABLE 2

O. D. of tubing (in.)	Gage BWG	Thickness (in.)	I. D. of tubing (in.)	Surface external, sq. ft per linear ft. (s)	Water g. p. m. at 1 ft. per sec velocity (g)	Tube constant	Weight per foot (lbs.)*
⅝	16	0.065	0.4[illegible]	0.16[illegible]	0.60	0.2728	0.4432
	17	.058	.509	.1636	.67	.2[illegible]	.4006
	18	.049	.527	.1636	.68	.2407	.3430
	20	.035	.555	.1636	.75	.2170	.2514
¾	16	.065	[illegible]	[illegible]	.94	.2087	.5121
	17	.058	.634	.1963	[illegible]	.1995	.4888
	18	.049	.652	.1963	1.0[illegible]	.1837	.4182
	20	.035	.680	.1963	1.13	.17[illegible]	[illegible]047
⅞	16	.065	.745	.22[illegible]	1.36	.1686	[illegible]10
	17	.058	.759	.2291	1.41	.1624	.5771
	18	.049	.777	.2291	1.48	.1550	.4928
	20	.035	.805	.2291	1.59	.1444	.3579
1	16	.065	.870	.2618	1.85	.1413	.7400
	17	.058	.884	.2618	1.91	.1369	.6654
	18	.049	.902	.2618	1.99	.1314	.5674
	20	.035	.930	.2618	2.12	.1236	.4112

*Weights in this table are [illegible] weight of 0.333 pounds per cubic inch which [illegible] and 90–10 Cu-Ni and on the exact dimensions indicated [illegible] allowance has been made for either plus or minus tolerances occurring in manufacture.

The heat transfer curves in figure 1 are based on new, clean, bright, and [illegible] tubes and on an inlet circulating water temperature of 70° F. For circulating water inlet temperatures other than 70° F., the basic heat transfer coefficient should be multiplied by the inlet water temperature correction factor on figure 1. The heat transfer coefficients are based on No. 18 BWG Admiralty metal tubing and should be multiplied by the following correction factors for tubes of other material or gage.

TABLE 3

Tube materials	Tube wall gage (BWG)		
	No. 18	No. 17	No. 16
Admiralty metal	1.0	0.98	0.9[illegible]
Arsenical copper	1.0	.98	.95
Aluminum brass	.96	.94	.91
Muntz metal	.96	.94	.91
Aluminum bronze	.90	.87	.84
90–10 copper nickel	.90	.87	.84
70–30 copper nickel	.83	.80	.76

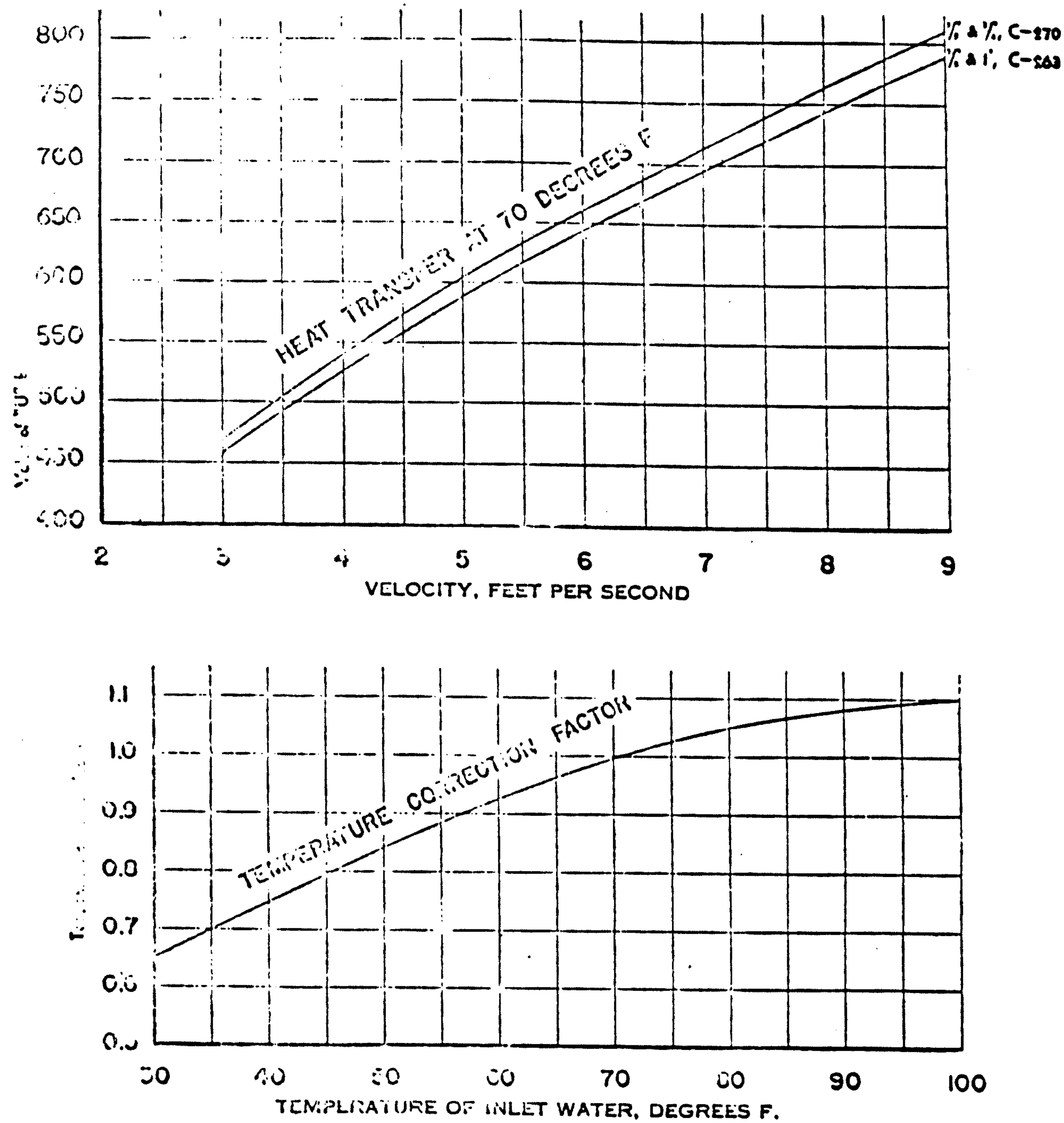

FIGURE 1.

NOTE: HEAT TRANSFER COEFFICIENT "U" BASED ON $C\sqrt{\text{VELOCITY}}$. CURVES APPLY TO CONDENSERS SERVING STEAM TURBINES. FOR CONDENSERS SERVING STEAM ENGINES, USE 65 PERCENT OF ABOVE VALUES. CURVES TAKEN FROM REFERENCE (b).