PREFACE

The Division of Building Research of the National Research Council is currently investigating how computers could be used to advantage in designing air-conditioning systems. This paper augments this work by describing how Russian engineers used a computer to obtain an optimum design for a refrigerant condenser.

The results reported herein are of limited interest in Canada because the types and sizes of condenser tubes studied are not available here; but the system of equations presented is valuable. It can be used to make similar calculations for any size pipe.

The Division wishes to record its thanks to Mr. H.R. Hayes of the Translations Section of the National Research Council who translated this paper and to Dr. D.G. Stephenson of this Division who checked the translation.

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Title: Selection of optimal sizes of shell-and-tube condensers for small refrigerating machines by means of electronic computer

(Vybor optimal'nykh razmerov kozhukhotrubnykh kondensatorov dlya malykh kholodil'nykh mashin s pomoshch'yu elektronnoi vychislitel'noi mashiny)

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SELECTION OF OPTIMAL SIZES OF SHELL-AND-TUBE CONDENSERS FOR SMALL REFRIGERATING MACHINES BY MEANS OF ELECTRONIC COMPUTER

The construction of water-cooled condensers for small (mechanical) refrigerating machines for use in ships' air-conditioning installations entails the fulfilment of a number of requirements.

Principally these are: to ensure minimum condenser size, simplicity, technological dependability, long life, reliability of construction and accessibility for pipe cleaning.

To determine the minimum overall size of shell-and-tube condensers for refrigerating machines with a refrigerating capacity of 1,500 to 11,000 kcal/hr (0.5 to 3.6 tons) operating on Freon-22 with given parameters for the cooling cycle and tube length, we studied the effect of water coolant flow rate and temperature, type and parameters of tube ribbing and diameter on the size of the condenser.

Calculations were made for seven different kinds of knurled tubing with inside diameters of 10 - 23 mm, rib heights of 1.5 - 3 mm and three different kinds of tubing with diameters of 10 - 16 mm and knurled ribs 3 mm in height. (Table I).

The geometrical parameters for the knurled tubes were taken from the encyclopaedic manual "Refrigeration Engineering", Volume 1 and from VDI-Vermeatlas [sic] data. Geometrical parameters for knurled tubing produced by the Baltic factory (Leningrad) and tubes with coiled ribs produced by other Soviet plants were also used.

Data used in the calculations are as follows:

Temperature, °C:

Condensation t_c 40

Boiling t_o 3

Water coolant on entering condenser t_{w_1} 30

Specific heat capacity of (sea) water coolant c_w, kcal/kg deg 0.92

Gravitation acceleration g, m/sec^2 .... 9.81

Coefficient of heat conductivity of layer of accumulated matter λ_{cl}, kcal/m hr deg 2

Thickness of layer of accumulated matter δ_{cl}, m 0.0005

Density of water coolant γ_w, kg/m^3 1025

Coefficient of heat conductivity of tube material (cupronickel) λ, kcal/m hr deg 32.4

The flow rate of the water coolant W_w was assumed to be 1.6 - 1.8 m/sec, the variation in the temperature of the water coolant Δt_w 2 - 6°.

Table II shows the basic values of N, G_{fr}, l_t and ΔP allowed for, depending on the refrigeration capacity of the conditioner.

The heat transmitting surface, capacity and total hydraulic resistance of condensers in relation to the indicated variables are determined in the following order according to the formulae:

1. Condenser heat load (losses of heat to environment disregarded) kcal/hr .... $Q = Q_o + 860\ N$

2. Water consumption, kg/hr .... $G_w = \dfrac{Q}{c_w(t_{w_2} - t_{w_1})}$

3. Mean logarithmic difference of temperatures, °C .... $\Theta_m = \dfrac{t_{w_2} - t_{w_1}}{2.3 \log \dfrac{t_c - t_{w_1}}{t_c - t_{w_2}}}$

4. Mean temperature of water, °C .... $t_{w_m} = t_c - \Theta_m$

5. Coefficient of heat transfer from inner surface of tube to water (ref.1, p.53) kcal/m hr deg .... $\alpha_w = (1190 + 21.5\ t_{w_m} - 0.045\ t_{w_m}^2)w_w^{0.8},\ d_c^{-0.2}$

6. Average rib diameter, m $D_{av.} = 0.5(D_r + d_o)$

7. Rib height, m .... $h = 0.5(D_r - d_o)$

8. Length of rib generating line, m .... $l_o = \sqrt{h^2 + \left(\dfrac{\delta_1 - \delta_2}{2}\right)^2}$

9. Coefficient, taking into account effect of ribbing of knurled tubes (ref.5, p.110). For tubes with circular flat ribs the same formula is used in simplified form (same ref.) ....

$$\varepsilon_r = \frac{S-\delta_1}{S} + \frac{\delta_2}{S}\left(\frac{D_r}{d_o}\right)^{0.75} th\sqrt{\frac{\delta_1 S}{h(S-\delta_1)}\left(\frac{d_o \delta_2}{\delta_1 D_r}\right)^{0.2}} + \frac{2\ D_{av.}\ l_0}{d_o S}\left(\frac{2\ d_o h}{\pi\ D_{av.}\ l_0}\right)^{0.25} th\sqrt{\frac{2\ \delta_1 S}{h(S-\delta_1)}\left(\frac{\delta_{av.} d_o}{\delta_1 D_r}\right)^{0.2}}$$

10. Average number of tubes, verical .... Initially assign value n_1, then find precise value by formula $n_1 = \frac{m}{z}$

11. Coefficient, taking into account average number of tubes in vertical bundle (ref.3, p.131, Fig. 21) .... $\bar{\varepsilon} = 1 - [0.1(n_1 - 1) - 0.00375\, n_1^2]$

12. Mean temperature of condensate film, deg ... $t_m = \frac{t_c + t_{\underline{w}}}{2}$ (initial value $t_{\underline{w}_0} = 0.5(t_c + t_{w_m})$)

13. Coefficient b (ref.3, p.131, Table 13), kcal$^{3/4}$ kg$^{1/4}$ m$^{1/4}$ hr deg$^{3/4}$ ...... $b = 591.2 - 2.2\, t_m$

14. Specific heat load in condenser, kcal/kg ..... $r' = \frac{Q}{G_{fr}}$

15. Specific heat flow, transferred to primary surface of tubes, from Freon to wall (ref.3, p.131) kcal/m² hr ...... $q_{fr} = 0.725\, b\, \varepsilon_r \bar{\varepsilon} \sqrt[4]{\frac{r'}{d_o}}\, (t_c - t_{\underline{w}})^{0.75}$

16. Specific heat flow, transferred to primary surface of tubes not allowing for formation of scale (from wall to water), kcal/m² hr ..... $q_w = \alpha_w \frac{d_i}{d_o} (t_{\underline{w}} - t_{w_m})$

17. Specific heat flow in condenser, not allowing for formation of scale, q kcal/m² hr .... Equations of para. 15 and 16 are solved with respect to $t_{\underline{w}}$, then the value of $t_{\underline{w}}$ is substituted in equation 15 or 16.

18. Determination of wall temperature t_w, deg .... Substitute $t_{\underline{w}}$ in para. 12 equation and repeat para. 12, 13, 15, 16, 17.

19. Heat transferring surface of condenser, not allowing for formation of scale, m² .... $F = \frac{Q}{q}$

20. Coefficient of heat transfer in clean condenser, kcal/m² hr deg . $c = \frac{q}{\Theta_m}$

21. Specific heat flow, transferred to primary surface of tubes allowing for formation of scale (from wall to water), kcal/m² hr ..... $q_w' = \frac{t_{\underline{w}} - t_{w_m}}{\left(\frac{1}{\alpha_w} + \frac{\delta_{fs}}{\lambda_{fs}}\right)\frac{d_i}{d_o}}$

22. Specific heat flow in condenser allowing for formation of scale q_{fs}, kcal/m² hr .... Solve equations at para. 15 and 21 with respect to t_w then substitute resulting value of t_w in one of equations 15 or 21

23. Determination of wall temperature t_w, deg .... Substitute t_w in para. 12 equation and repeat valculation for para. 12,13,15,21 and 22

24. Heat transferring surface of condenser, allowing for formation of scale, m² .... $F_{fs} = \frac{Q}{q_{fs}}$

25. Overall length of tubing, m .... $L = \frac{F_{fs}}{\pi d_o}$

26. Number of tubes per pass .... $n_w = \frac{4G_w}{\pi d^2{}_1 3600\ w_w \gamma_w}$

27. Total number of condenser tubes .... $m = \frac{L}{l_t}$

28. Tube spacing (with cross-section arrangement of equilateral triangle), m .... $S_t = D_r + 0.003$

29. Diameter of circle passing through centres of outer tubes (ref.4), m .... $D' = S_t \sqrt{0.94 + \frac{m - 3.7}{0.907}}$

30. Outside diameter of condenser flange, m .... $D_c = (D' + d_1)1.08 + 0.052$

31. Number of vertical series of tubes .... $z = \frac{D'}{S_t} + 1$ (determine from para. 10)

32. Volume of condenser, m³ .... $V = \frac{\pi}{4} D_c^2(l_t + 0.126)$

33. Coefficient of heat transfer in condenser, allowing for formation of scale, kcal/m² hr deg .... $c_{fs} = \frac{q_{fs}}{\Theta_m}$

34. Coefficient of kinematic viscosity of water (ref. 1), m²/sec .... $\nu_w = \frac{0.00178}{1 + 0.0337\ t_{w_m} + 0.000221\ t_{w_m}^2} \cdot \frac{1}{\gamma_w}$

35. Reynolds number .... $Re_w = \frac{w_w d_1}{\nu_w}$

36. Frictional resistance coefficient when $3{,}000 \leqslant Re_w \leqslant 100{,}000$ .. $\lambda_t = \frac{0.316}{Re_w^{0.25}}$

37. Frictional resistance coefficient when $Re_w <$ 3,000 (ref.3, p.144) ... $\lambda_t = \frac{64}{Re_w}$

38. Frictional resistance of water circuit, m water gauge .... $\Delta P_t = 1.17\ \lambda_t \frac{L}{n_w d_1} \cdot \frac{w_w^{1.8} \gamma_w 10^{-3}}{2\ g}$

39. Resistance at bends, m water gauge .... $\Delta P_b = 0.90 \left(\frac{m}{n_w} - 1\right) \frac{w_w^2 \gamma_w 10^{-3}}{2\ g}$

40. Intake and discharge resistance, m water gauge .... $\Delta P_{in\ and\ dis} = 19 \cdot 10^4 Re_w^{-1.2} \frac{w_w^2 \gamma_w 10^{-3}}{2\ g}$

41. Total hydraulic resistance, m water column ... $\Delta P_{tot} = 1.1(\Delta P_t + \Delta P_{in\ and\ dis} + \Delta P_b)$

Unlike the graph analysis method normally employed, the cited method permits the temperature of the tube wall to be determined jointly with the solution of the equation for heat flow from the Freon condensation vapour to the main surface of the tubes (para. 15) and the equation for heat flow from the wall to the water coolant (para. 16). The system of equations is solved by means of successive approximations with a given accuracy of 0.1°.

A programme for computation on the "Ural-2" digital computer was carried out in accordance with this method at the technological computing centre (in the town of Nikolaev).

Approximately 1,200 computation variants were worked out, the only results selected for printing being those of variants with a hydraulic resistance $\Delta P < 5$ m water column. In accordance with a computed value V for each standard tube size, graphs were drawn for a constant flow rate of water coolant W_w and variable values ΔT_w, d_1 and l_t (Fig. 1 - 3).

Analysis of the resulting graph relationships indicated the following:

With an increase in the water coolant flow rate the hydraulic resistance of the condenser increases very rapidly in comparison with reduction of its volume, especially when refrigerating capacity is reduced. An increase in the sea-water coolant flow rate to more than 2 m/sec results in a substantial drop in the heat transfer coefficient in the operation of the heat exchanger owing to an intensification of the salt deposition process and a reduction in the erosion resistance of the condenser materials. Therefore, a flow rate of 1.6 m/sec was used in subsequent analysis.

With an increase in the temperature drop of the water coolant Δt_w when W_w = const, the condenser volume and hydraulic resistance increase. For a fixed tube length this results in an increase of the total number of tubes and a reduction of the number of tubes per pass, which complicates the design of the condenser cover to the detriment of technological dependability and reliability of construction.

Minimum condenser volume and hydraulic resistance are achieved with a minimum water-coolant temperature drop ($\Delta t_w = 2°$), but in this case the coolant flow rate increases.

In order to reduce the coolant flow rate and the diameter of the tubes, it is expedient to assume that $\Delta t_w = 3°$ for condensers used in ships' air-conditioning plants. This would involve a very small increase in the volume of the condenser (approximately 5%).

The type of tube ribbing has an appreciable effect on condenser volume.

Heat exchange conditions are better with knurled tubing than with tubing which has coiled ribs (the efficiency of the knurled tube ribbing is greater by 27 - 31%) and consequently, its heat transmitting surface may be reduced. For the same heat load the condenser capacity with knurled piping is reduced by 10 - 15%.

The diameter of the tubes also has a considerable effect on the volume and hydraulic resistance of the condenser. Comparison of the volumes of condensers used with different sizes of knurled tubing (at a constant hydraulic resistance) indicated that minimal volumes are attained when $d_o \times d_i = 16 \times 14.4$ mm; $D_r = 20.4$ mm.

Variations of condenser volumes in relation to the inside diameter of the tubes are shown at Fig. 2.

For tubing with coiled ribs the smallest condenser is attained when $d_o \times d_i = 16 \times 14.4$ mm and $D_r = 22$ mm. The inside diameter of the tubes is such as to permit easy access for cleaning out deposits which form during use.

In machines with a refrigerating capacity of 1,500 - 2,500 kcal/hr (.5 - .8 tons) the tube diameter has little effect on the volume of condensers of a given length, since large diameter pipes do not fit into the small diameter of the tube panel as well as small diameter tubes. Thus, in both cases the volumes are approximately the same.

With an increase of refrigerating capacity the diameter of the tube panel is increased and tubes of a larger diameter fit into it better, which permits a more efficient arrangement of the bundle than is possible with smaller diameter tubes.

The question of the effect of tube length l_t on the volume of the condenser is of considerable importance. Preliminary calculations made with a variable l_t for knurled tubes and the graph relationships plotted according to these calculations (see Fig. 3) permit the assumption that when l_t increases, the hydraulic resistance of the condenser and its volume decrease at first, but when $l_t > l_{t\ opt}$ the volume increases.

For a condenser with a refrigerating capacity of 4,400 kcal/hr (1.5 tons), $l_{t\ opt} \approx 0.4$ m. The minimum condenser volumes for different values of l_t are obtained when $d_o \times d_i = 16 \times 14.4$; $D_r = 20.4$ mm and $d_o \times d_i = 15 \times 13$ mm; $D_r = 19.4$ mm.

Definitive conclusions as to the selection of optimum tube lengths will be drawn after the data for the entire series of condensers have been processed.

Thus, taking into consideration simplicity, technological dependability, reliability of construction and ease of access for cleaning purposes, optimum size is ensured for a range of condensers fitted in ships' air-conditioning plants with accepted uniform tube lengths on selection of:

(a) knurled tubes of the following sizes (in mm):

d_o	16
d_i	14.4
D_r	20.4
δ_1	1.81
δ_2	1.14
h	2.2
S	2.04

(b) a flow rate of water coolant in the condenser tubes equal to 1.6 m/sec;

(c) a water coolant temperature drop $\Delta t_w = 3°$.

Tests were carried out on an experimental model of a condenser constructed in accordance with the calculation results at a refrigerating capacity of 1,700 kcal/hr (.56 tons). The resulting experimental data corresponds closely to the calculations.

References

1. Tsygankov, A.S. Raschety teploobmennykh apparatov (Heat exchanger designs). Sudpromgiz, 1948.

2. Kutateladze, S.S. and Borishanskii, V.M. Spravochnik po teploperedache (Manual of heat transfer). Gosenergoizdat, 1959.

3. Encyclopaedic manual "Kholodil'naya tekhnika" (Refrigeration Engineering), Vol. 1. Gostorgizdat, 1960.

4. Tadeush, Khobler. Teploperedacha i teploobmeniki (Heat transfer and heat exchangers). Goskhimizdat, 1961.

5. Ioffe, D.M. and Yakobson, V.B. Malye kholodil'nye mashiny i torgovoe oborudovanie (Small refrigerating machines and the equipping of merchant ships). Gostorgizdat, 1961.

6. Rozenfel'd, L.M. and Tkachev, A.G. Kholodil'nye mashiny i apparaty (Refrigerating machines and systems). Gostorgizdat, 1960.

Table I

Characteristics	Type of tube ribbing for variants									
	Knurled							Coiled		
	1	2	3	4	5	6	7	8	9	10
Outside diameter of tube d_o, mm	12	12	17.7	23	16	14	15	10	16	15
Inside diameter of tube d_1, mm	8	10	15.7	20	14.4	12	13	8	14.4	13
Outside diameter of rib D_r, mm	18	18	21.5	26	20.4	20	19.4	16	22	21
Thickness of rib at base δ_1, mm	1.7	1.6	1.6	1.6	1.81	1.8	1.81	0.5	0.5	0.4
Thickness of rib at top δ_2, mm	0.7	0.8	0.99	1.05	1.14	1.0	1.14	0.5	0.5	0.4
Height of rib h, mm	3	3	1.9	1.5	2.2	3.0	2.2	3.0	3.0	3.0
Rib spacing S, mm	3	3	2.03	2.0	2.04	3.0	2.04	3.5	3.5	3.4

Table II

Refrigeration capacity of conditioner, kcal/hr	Input required by pneumatic compressor motor N, kw	Freon consumption G_{fr}, kg/hr	Length of one condensor tube l_t, m
1700	0.8	42.5	0.20
2700	1.1	67.5	0.25
4400	1.8	110	0.30
6900	2.9	173	0.35
11000	5.2	280	0.40

Note: The hydraulic resistance of the condenser ΔP must not exceed 5 m water column

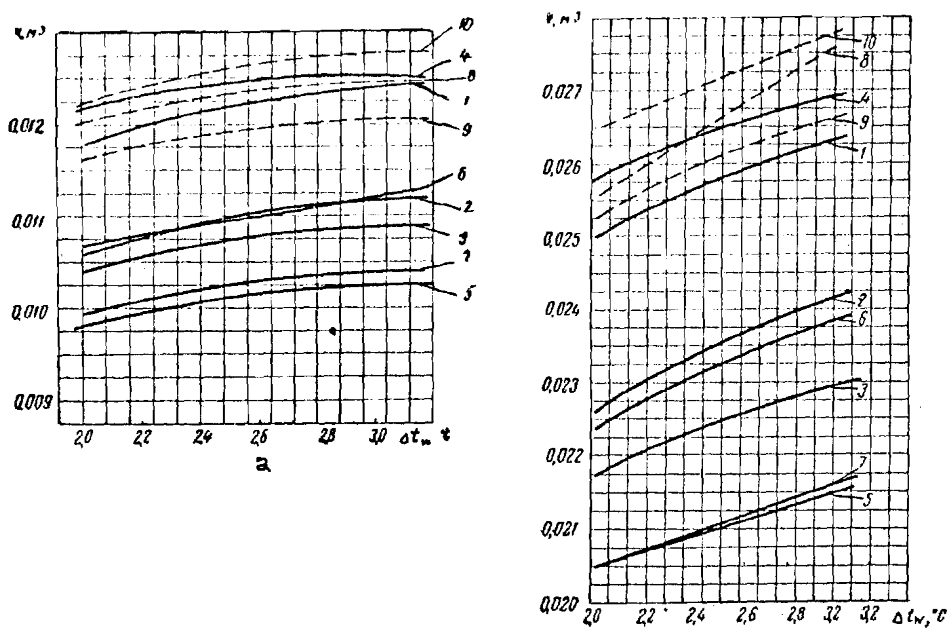

Fig. 1

Relationship of condenser volume V to water coolant temperature drop Δt_w

a - Q_o = 2,700 kcal/hr; b - Q_o = 6,900 kcal/hr when w_w = 1.6 m/sec (interpretation of numerals 1 - 10 cm in Table I)

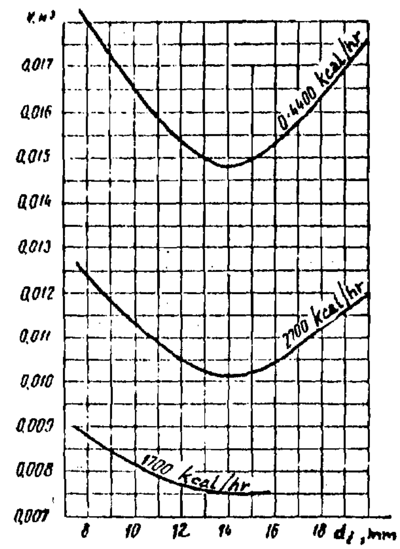

Fig. 2

Relationship of condenser volume V to inside diameter of tube d_i when w_w = 1.6 m/sec

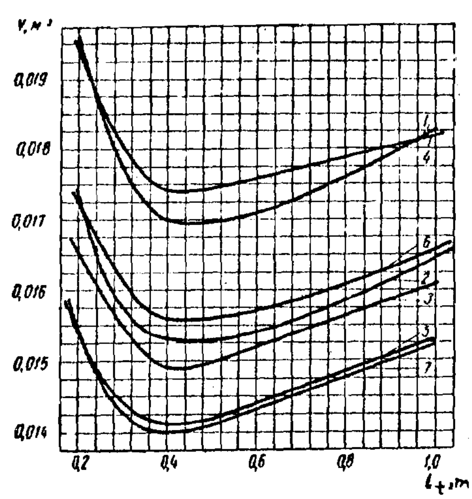

Fig. 3

Relationship of condenser volume V (refrigeration capacity Q_o = 4,400 kcal/hr) to length of of tube l_t when Δ_{tw} = 2°, w_w = 1.6 m/sec