The Condensing of Low Pressure Steam on Vertical Rows of Horizontal Copper and Titanium Tubes

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Heat transfer data are presented for condensing steam at 2 in. Hg absolute pressure on the outside of nine copper and nine titanium horizontal tubes in a vertical row. The condensing coefficient correction factor was maximum for the top titanium tube and was 46% higher than the correction factor for the top copper tube. The difference between the correction factors for titanium and copper tubes diminished with the number of tubes in a vertical row to 8% higher than the correction factor for copper tubes with six to nine tubes in a vertical row.

The use of titanium tubes for steam condensation in shipboard power plants and in saline water conversion processes is currently of significant interest. The low wettability surface characteristics of titanium tubes tend to give higher condensing coefficients and the high mechanical strength permits the use of thinner tube walls when compared to conventional materials. Favorable erosion and corrosion resistance properties of titanium further add to the benefits of using titanium tubes.

REVIEW OF THE LITERATURE

In 1916, Nusselt (*1*) derived the equation governing the condensation of pure saturated vapors on the outside of a horizontal tube with a wettable surface. Equation (1) was obtained by assuming laminar flow of the condensate and no vapor velocity effects:

$$h_m = 0.725 \left[\frac{k^3 \rho^2 g \lambda}{\mu D \Delta t_f} \right]^{1/4} \tag{1}$$

For laminar flow of the condensate, the film temperature t_f is given by

$$t_f = t_{sv} - \frac{3}{4} \Delta t_f \tag{2}$$

Experimental investigations of the condensation of pure saturated vapors on single horizontal tubes indicate that Equation (1) predicts values generally within $\pm$ 10% of the experimental condensing coefficients (*2*). The experimental coefficients are usually higher than the theoretical values. This is attributable to turbulence or rippling in the condensate layer. When turbulent flow of the condensate is expected, the average film temperature is often evaluated with Equation (3) (*3*).

$$t_f = t_{sv} - \frac{1}{2} \Delta t_f \tag{3}$$

When several horizontal tubes are placed in a vertical row so that condensate from the upper tubes drops on the lower tubes, the mean thickness of the condensate film on a particular tube increases from the top tube to the bottom tube. By accounting for the accumulation of condensate from tube to tube, but still assuming laminar flow of the condensate, Nusselt derived Equation (4) to predict the average condensing coefficient h_m for n tubes located in a vertical row (*1*).

$$h_m = 0.725 \left[\frac{k^3 \rho^2 g \lambda}{n \mu D \Delta t_f} \right]^{1/4} \tag{4}$$

Equation (3) would be used for calculating t_f if turbulent flow of condensate is expected. Experimental data taken on multiple horizontal tubes in a vertical row by Katz and Geist (*4*), Short and Brown (*5*), and Young and Wohlenberg (*6*) indicate that Equation (4) is very conservative. The correction for multiple tube rows of $(1/n)^{1/4}$ is much too severe in view of the high degree of turbulence and splashing with condensate dropping from tube to tube. A turbulence correction factor C_n was added to Equation (4) by Katz, Young, and Balenkjian (*3*) to give Equation (5).

$$h_m = 0.725\, C_n \left[\frac{k^3 \rho^2 g \lambda}{n \mu D \Delta t_f} \right]^{1/4} = C \left[\frac{k^3 \rho^2 g \lambda}{n \mu D \Delta t_f} \right]^{1/4} \tag{5}$$

Equation (5) corrects the basic theoretical Nusselt model with the correction factor C_n and gives a means of correlating experimental condensing data for multiple tube arrangements (*3*). The correction factor C_n varies with the number of tubes in a vertical row, the physical properties of the condensate, the tube surface, and the vapor velocity.

An extensive experimental program was completed by the British Admiralty in which condensing heat transfer data were obtained for multiple tube arrangements with film and dropwise condensation of steam (*7*). Photographic studies indicated that heat fluxes six times the average heat flux were obtained in the drop tracks formed in dropwise condensation when large drops rolled across the surface leaving a "bare" metal surface. About one-fifth of the surface had fresh drop tracks at all times. They concluded that high heat fluxes are sustained for times in the order of seconds in very narrow width tracks. The heat flow through these tracks then diverged in crossing the tube wall because the entire internal surface can be used for heat transfer. Because of this, they concluded that very thin metal walls would limit the effectiveness of dropwise condensation.

The investigators further determined the effect of condensate inundation on the condensing heat transfer coefficient. By pumping condensate through a perforated tube placed above the test section, the tube on which data were taken could effectively simulate any tube in a vertical row of twenty-two tubes. For filmwise condensation, the condensing coefficient first decreased with inundation due to a thicker condensate film, and then reversed the trend due to increased turbulence at about the fourteenth or fifteenth tube.

In dropwise condensation, the effect of inundation was first to increase the condensing coefficient due to enhanced wiping action for the top six or seven tubes followed by a gradual decrease. The coefficient for the simulated twenty-second tube in a vertical row was higher than for the top tube.

The existence of any noncondensible gas in the condensing vapor significantly reduces the rate of heat transfer due to the buildup of noncondensible gas around the condensing surface. Experimental work by Othmer (*8*) and Hampson (*9*) indicates that as little as 1.5% air by volume can reduce the condensing coefficient by 50%. The greatest effect occurs when there is little motion of vapor across the tubes. Under these conditions, most of the noncondensible gases eventually migrate to the vicinity of the tube.

EQUIPMENT AND TEST PROCEDURE

The equipment in this investigation consisted of a condenser, inlet and outlet water headers, reboiler, make-up tank, water preheater, pump, two-stage steam jet ejector, and automatic controllers. Figure 1 is a line diagram showing the flow of steam and water. Steam was generated by boiling distilled water in the reboiler with 150 lb./sq. in. gauge steam. The vapor flowed to the condenser where it condensed on the test tubes. The condensate was returned to the reboiler. Water from the cooling tower system was used as the coolant.

The condenser was 6 ft. long and 18 in. in diameter. O rings were used to seal the tubes in the tube sheets to permit changing. An impingement baffle was placed over and 2 in. above the tubes in the condenser to prevent direct impingement of steam onto the tubes.

The reboiler was 6 ft. long and 24 in. in diameter. High pressure steam (150 lb./sq. in. gauge) was used to vaporize the water in the reboiler. The condensate was returned to the high pressure boiler through a steam trap.

Four automatic controllers were installed to assist in the operation of the equipment when taking data. One instrument controlled the water flow rate. A second instrument served as an inlet cooling water temperature controller. The controller pneumatically actuated a steam valve which regulated the amount of steam entering the water preheater. The remaining two instruments were absolute pressure controllers. One sensing element was connected to the condenser. The controller used the pressure signal to regulate the amount of steam entering the reboiler through a ¾ in. pneumatically operated valve so that the desired pressure in the condenser could be maintained. The second pressure controller was installed in the steam jet ejector system to minimize fluctuations in pressure at the ejector due to variations in the steam flow rate. The control instrument controlled a small bleed valve. By bleeding in small amounts of air, the pressure in the ejector header could be kept relatively constant.

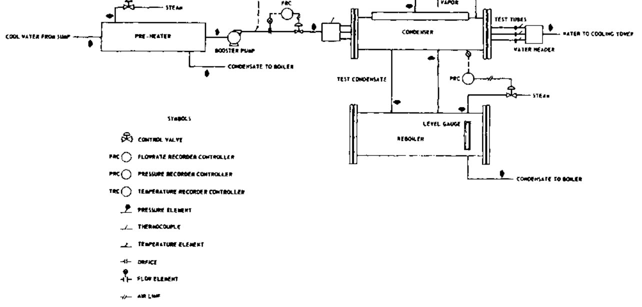

Fig. 1. Line diagram of equipment showing the flow of steam and water.

The water flow rates in each tube were measured by calibrated orifices placed in orifice holders which were located at the outlet end of the test tubes between the condenser and the exit water header. The orifices were calibrated for each tube tested. The pressure drop across the orifices was measured with water over mercury manometers. Both 50- and 100-in. manometers were used. A manifolded system permitted the same manometer to be used for several orifices. The accuracy of the flow rate measurement was between ¼ and ½%.

Inlet water, outlet water, and condenser steam temperatures were measured with calibrated 30 gauge copper-constantan thermocouples with a Leeds and Northrup K-3 potentiometer. Temperatures could be measured to 0.01°F. The inlet water thermocouple was placed in the inlet water header. The exit water thermocouples were located in the orifice holder assemblies within stainless steel sheaths extended upstream along the tube axis for 1 in. Thermocouples were placed in two places in the back of the condenser to permit the measurement of the steam temperature.

The condenser absolute pressure was determined with a mercury manometer and calibrated barometer.

During normal operation, the reboiler was one-half to two-thirds filled with distilled water through the water make-up tank. Once the reboiler was filled to the desired level, steam and water to the steam jet ejector were turned on and adjusted to give the maximum evacuation rate. The ejector was allowed to operate for approximately 30 to 45 min. to evacuate thoroughly noncondensible gases from the condenser-reboiler system. The pressure in the condenser rapidly approached the vapor pressure of the water in the reboiler during this period. With the ejector still pulling a vacuum on the system, the condenser pressure controller was set at the desired pressure setting. The automatically controlled steam valve in the reboiler steam line then opened, allowing the water in the reboiler to be heated until the vapor pressure of the water equalled the set point pressure. The system was operated under these conditions for approximately 20 to 30 min. This further assisted in degassing the water and evacuating the system. The cooling water controller was next set to the desired total water flow rate and the inlet water temperature controller set at the desired inlet water temperature. The steam jet ejector manifold pressure controller was set at a pressure somewhat below the condenser pressure. This minimized the air bleed and permitted maximum removal of noncondensible gases during the period when data were taken. There was a very small amount of air leakage into the system. Before data were taken, the saturated steam temperature was calculated on the basis of the absolute pressure indicated by the manometer and compared to the steam temperature measured with a thermocouple. If the two temperatures agreed within ½°F., the system was considered ready for taking data. If the temperature calculated from the absolute pressure in the condenser was greater than ½°F. above the measured temperature, an excessive amount of air still remained in the system and evacuation was continued until satisfactory agreement was obtained. Heat transfer data were taken when the automatic controllers had stabilized all the control variables at the desired set points.

MULTIPLE TUBE DATA PROCESSING AND RESULTS

The overall heat transfer coefficient for each tube was calculated from

$$U_o = \frac{Q}{A_o \, \Delta T_m} \tag{6}$$

where the heat duty Q was obtained experimentally from

$$Q = W \, c_p \, (t_{out} - t_{in}) \tag{7}$$

TABLE 1. TUBE DIMENSIONS AND CHARACTERISTICS

	Copper		Titanium	
	Top tube	Average for vertical row	Top tube	Average for vertical row
Tube type	plain	plain	plain	plain
Tube O. D., in.	0.6252	0.6252	0.6287	0.6271
Tube I. D., in.	0.5550	0.5550	0.5592	0.5581
Tube wall thickness, in.	0.0351	0.0351	0.0347	0.0345
Tube length, in.	72.156	72.156	72.156	72.156
Thermal conductivity, B.t.u./(hr.) (sq. ft.)(°F.)/ft.	196	196	10	10

A modified Wilson plot technique (*10*) was used to obtain a realistic value of the coefficient in Equation (8) for the particular experimental equipment used in the investigation. This was done in preference to using an arbitrary value from the technical literature. The modified Wilson plot technique took into account the variation in the average temperature drop across the condensing film with changes in water velocity. It did not take into account the variation of the temperature drop across the condensing film from one end of a tube to the other end. The intent of the investigation was to look at the average condensing heat transfer coefficients with titanium and copper tubes. Wilson plot data were obtained on the top tube in the vertical row for each alloy tube. The tube dimensions are given in Table 1. With the use of the average value of C_i for the two sets of data, the inside heat transfer coefficient became

$$\frac{h_i D_i}{k_i} = 0.0248 \left[\frac{D_i G}{\mu}\right]^{0.8} \left[\frac{c_p \mu}{k}\right]^{1/3} \left[\frac{\mu_i}{\mu_w}\right]^{0.14} \quad (8)$$

The condensing coefficient and the condensing coefficient constant were calculated with Equations (9) and (10), respectively, for all the Wilson plot data.

$$\frac{1}{h_m} = \frac{1}{U_o} - \frac{A_o}{A_i h_i} - r_m \quad (9)$$

From Equation (5) with $n = 1$ for the top tube

$$C = \frac{h_m}{\left[\dfrac{k^3 \rho^2 g \lambda}{\mu D \Delta t_f}\right]^{1/4}} \quad (10)$$

Two sets of heat transfer data were taken on the nine tubes in a vertical row. The first set of data were taken on copper tubes and the second set of titanium tubes. The original data appear in reference *10.*

The purpose of taking multiple tube data was to obtain the correction factor C_n for Equation (5) as a function of the number of tubes in a vertical row for the condensation of steam at 2 in. Hg absolute pressure. A computer program was written for the IBM 7090 digital computer to process the data. The computer program consisted of three sections. In the first section the input data, including the average value of the inside heat transfer coefficient constant, were read into the computer and preliminary calculations were made. These operations included the calculation for each tube of the heat duty, logarithmic temperature difference, overall heat transfer coefficient, water velocity, bulk water physical properties, inside heat transfer coefficient, and condensing coefficient from Equation (9). From the condensing coefficient and physical properties of the condensate film, the condensing coefficient constant was calculated from Equation (10). The average inlet water temperature, water velocity, and steam temperature for all nine tubes were also calculated. A printout of the results completed the first section.

In the second section of the program, the average inlet water temperature, water velocity, steam temperature, and condensing coefficient constants for each tube were used to predict for each tube what the heat duty, exit water temperature, logarithmic temperature difference, overall heat transfer coefficient, inside heat transfer coefficient, and condensing coefficient would have been had the inlet water temperature, water velocity, and steam temperature been equal to the average values. These calculations put all the tubes on a consistent basis. A printout of the results completed the second section.

The condensing coefficient correction factor was calculated in the third section of the computer program. The correction factor is by definition that factor which makes Equation (5) an equality and is calculated from Equation (11):

$$C_n = \frac{h_m}{0.725 \left[\dfrac{k^3 \rho^2 g \lambda}{n \mu D \Delta t_f}\right]^{1/4}} \quad (11)$$

In Equation (11), the mean condensing coefficient h_m is the mean condensing coefficient for the top n tubes calculated from the experimental data. The correction factor for the top tube was calculated with the values of the heat duty and exit water temperature calculated in the previous section. The overall heat transfer coefficient, logarithmic temperature difference, and inside heat transfer coefficients were then calculated and the mean condensing coefficient computed from Equation (9). Equation (12) was used to calculate the temperature drop across the condensing film

$$\Delta t_f = \frac{U_o \Delta t_m}{h_m} \quad (12)$$

and Equation (3) was used to calculate the film temperature. Once the film temperature was known, the quantity with $n = 1$

$$0.725 \left[\frac{k^3 \rho^2 g \lambda}{1 \mu D \Delta t_f}\right]^{1/4}$$

was calculated and C_n computed from Equation (11) for the top tube.

To determine C_n for the top two tubes in the vertical row, the heat duties calculated in the second section of the computer program for the top two rows were added to give the total heat transferred. With the mean values of the water density and heat capacity for the top two tubes, we calculated the average exit water temperature for the top two tubes. The logarithmic temperature difference, overall heat transfer coefficient, and inside heat transfer coefficient were calculated next and the mean condensing coefficient was calculated from Equation (9), the temperature drop across the condensing film was calculated from Equation (12), and the film temperature was calculated from Equation (3). The quantity

$$0.725 \left[\frac{k^3 \rho^2 g \lambda}{2 \mu D \Delta t_f}\right]^{1/4}$$

was computed and the correction factor for two tubes in a vertical row was calculated from Equation (11). The correction factors for 3, 4. . . .9 tubes in a vertical row were calculated by adding the heat duties for the top n tubes and by following the procedure previously outlined.

The computer program and typical calculated results can be found in reference *10.*

DISCUSSION OF RESULTS

Tables 2* and 3 give the values of the condensing coefficient correction factors for a vertical row of one to nine copper and titanium tubes, respectively. The results were obtained from experimental data taken at a steam pressure of approximately 2 in. Hg absolute and an inlet water temperature of 75°F. Average values of C_n for each water velocity are given in Tables 4 and 5 for copper and titanium tubes, respectively. The results are also presented in Figures 2 and 3. Average values of C_n for all the data as a function of the number of tubes in a vertical row are given in Table 4 for copper tubes and Table 5 for titanium tubes.

As can be seen in Figures 2 and 3, the condensing coefficient correction factor C_n is higher for titanium tubes than for copper tubes. The maximum value of C_n for titanium tubes occurs for the top tube where C_n is 46% higher than the value for the top copper tube. The difference in C_n diminishes to a more or a less constant value of approximately 8% greater with six to nine tubes in a vertical row. Inundation drastically reduces the effectiveness of titanium tubes with essentially all the improvement being a result of the increase on the top tube.

In Figures 2 and 3, there appears to be a consistent trend in which for a given number of tubes in a vertical row C_n varies with the tube-side water velocity (or condensate loading as both are directly related). For low and high velocities the correction factor is higher than for intermediate velocities. This is attributable in part to the varying degrees of turbulence in the condensate film depending upon the tube condensate flow rate, as previously mentioned. The maximum deviations from the mean values in Figures 2 and 3 are 5.3 and 6.3%, respectively.

Visual observation of low pressure steam condensing on the nine titanium tubes in vertical row revealed that as could best be seen, only filmwise condensation was occurring. Visual observation of the top tube was limited and it could be possible that partial dropwise condensation was occurring on this tube. The generally lower wettability of the titanium tube surface increases the condensing coefficient but not to the point where dropwise condensation will persist for multiple tube arrangements.

* Tables 2 through 5 have been deposited as document 8560 with the American Documentation Institute, Photoduplication Service, Library of Congress, Washington 25, D. C., and may be obtained for $1.25 for photoprints or 35-mm. microfilm.

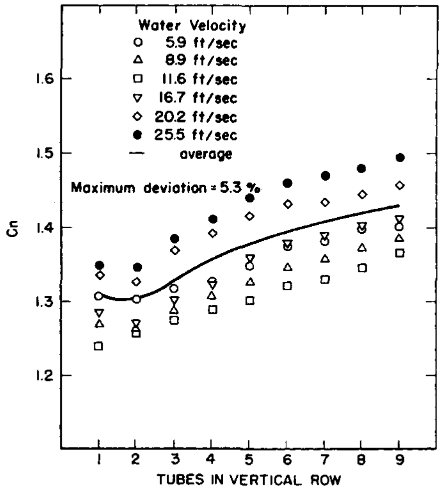

Fig. 2. Condensing coefficient correction factors for condensation of steam at 2 in. Hg absolute pressure on one to nine copper tubes in a vertical row.

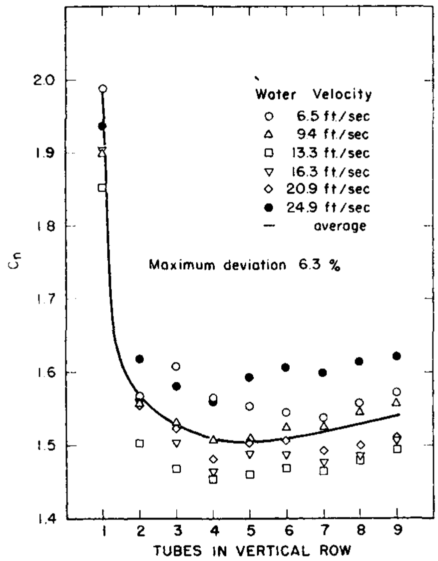

Fig. 3. Condensing coefficient correction factors for condensation of steam at 2 in. Hg absolute pressure on one to nine titanium tubes in a vertical row.

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NOTATION

A_i = total internal heat transfer surface, sq. ft.
A_o = total external heat transfer area, sq. ft.
C = condensing coefficient constant which is equivalent to 0.725 C_n, dimensionless
C_i = inside heat transfer coefficient constant, dimensionless
C_n = turbulence correction factor, dimensionless
c_p = specific heat of water, B.t.u./(lb.)(°F.)
D = outside diameter of tube, ft.
D_i = tube inside diameter, ft.
G = mass flow rate, lb./(hr.)(sq. ft.)
g = acceleration due to gravity, taken as 4.17×10^8 ft./hr.^{2}
h_i = inside heat transfer coefficient, B.t.u./(hr.)(sq. ft.)(°F.)
h_m = mean condensing coefficient, B.t.u./(hr.)(sq. ft.)(°F.)
k = thermal conductivity of condensate evaluated at film temperature, B.t.u./(hr.)(sq. ft.)(°F.)/ft.
k_i = water thermal conductivity at bulk water temperature, B.t.u./(hr.)(sq. ft.)(°F.)/ft.
n = number of tubes in a vertical row
Q = total heat transfer, B.t.u./hr.
r_m = metal resistance, hr./sq. ft. (outside area) °F./B.t.u.
t_{in} = inlet water temperature, °F.
t_f = average condensing film temperature, °F.
t_{out} = outlet water temperature, °F.
t_s = outside wall temperature of the tube, °F.
t_{sv} = temperature of the saturated vapor, °F.
U_o = overall heat transfer coefficient, B.t.u./(hr.)(sq. ft.)(°F.)
W = water flow rate, lb./hr.

Greek Letters

Δt_f = temperature drop across condensate film, $t_{sv} - t_s$, °F.

ΔT_m = logarithm temperature difference, °F.
λ = latent heat at saturation temperature, B.t.u./lb.
μ_t = water viscosity at bulk water temperature, lb./(ft.)(hr.)
μ = viscosity of condensate evaluated at film temperature, lb./(ft.)(hr.)
μ_w = water viscosity at average inside wall temperature, lb./(ft.)(hr.)
ρ = density of condensate evaluated at film temperature, lb./cu. ft.

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Flow and Turbulence in a Stirred Tank

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Photographic measurements have been made of the mean and fluctuating components of velocity of water in a fully baffled stirred tank. Confirmation of much of the photographic data was obtained with a Kiel impact tube. Eulerian correlation coefficients and also Eulerian scales of turbulence were calculated from the photographic data. The Eulerian scale was of the same order as the blade dimensions, a result consistent with earlier measurements on the turbulence behind grids.

Equations have been developed to describe the flow of energy and the conservation of angular momentum in the impeller stream of a stirred tank with a radial flow impeller and vertical baffles. These are simplifications of the Navier-Stokes equations and the energy equation. They relate energy, angular momentum, and pressure to the mean and fluctuating components of velocity in the impeller stream.

The equations derived are used with the photographic data on mean and fluctuating velocities to estimate the angular momentum at different radial sections of the tank and to calculate the flow of energy through these sections. The estimates are compared with more accurate values of the total torque and energy determined with a torque table.

The objective of the work presented here is the description of the flow of a Newtonian fluid (water) in a fully baffled stirred tank, with particular emphasis on local rates of energy dissipation.

Concern with local rates of energy dissipation is based on the frequently made observation that power per unit volume (or power per unit mass) is often a very useful criterion of agitation, particularly in processes in which one phase is dispersed in another. Kolmogoroff's (*13*) theory of local isotropy provides an explanation of this observation in terms of the intensity of the small-scale eddies in the turbulent flow, since his theory predicts that these eddies are isotropic and that their intensity in a turbulent flow of sufficiently high Reynolds number is dependent only on the local rate of energy dissipation and the viscosity of the fluid. He also predicts that for turbulence of a sufficiently high Reynolds number (higher than is necessary for the first hypothesis to apply) there will be a subrange of eddies, sufficiently large so that they account for a negligible proportion of the viscous energy dissipation and at the same time sufficiently small so that they satisfy the conditions of the first hypothesis. The intensity of eddies in this intermediate range is a function only of their size and of the local rate of energy dissipation per unit mass, ϵ.

Hinze (*11*) has shown that these hypotheses can be used to correlate data by Clay (*5*) on the breakup of droplets between a rotating and stationary cylinder.

Calderbank (*4*) and Shinnar (*20*) have applied these ideas to the breakup of liquid droplets in another liquid phase and also to the breakup of gas bubbles in a liquid. Shinnar showed that Vermeulen's (*26*) data on the breakup of liquid droplets and gas bubbles could be correlated in terms of a Weber number characteristic of the turbulent stresses predicted by Kolmogoroff's hypotheses.

If Kolmogoroff's theory is to be used extensively, and particularly if it is to be used for scale-up, it becomes important to obtain information about the distribution of energy dissipation rates in a stirred tank as well as the average value.

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