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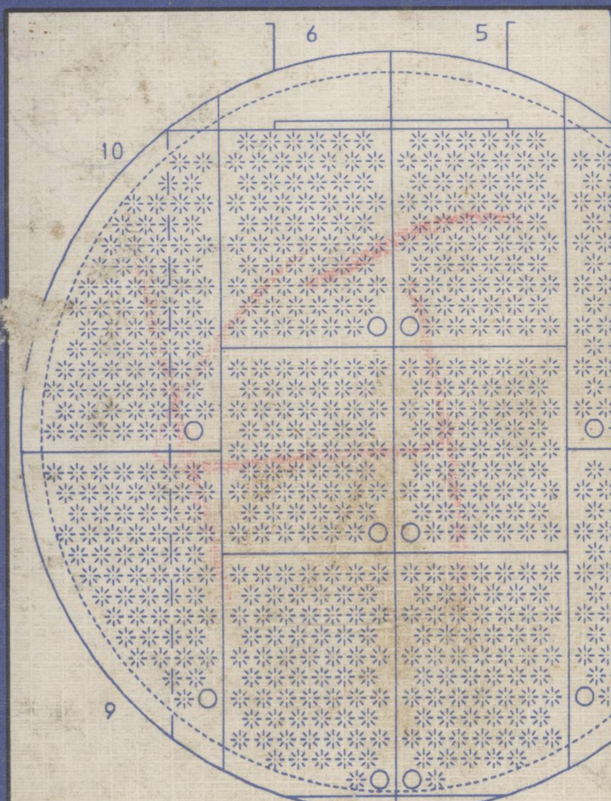

CONDENSERS: THEORY AND PRACTICE

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CONDENSERS: THEORY AND PRACTICE

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Preface

The purpose of the Symposium is to stimulate the exchange of information between researchers into condensation and the designers and users of co[redacted]ensers. The Proceedings are classified into four categories: Theory and [redacted]tice as applied in the Power and Process industries. The organising [redacted]mittee has sought to assemble a programme of papers that will be of [redacted]est to a wide audience and within each session there is a keynote [redacted]ntation of a review nature to place the individual papers in a broader context. Inevitably certain topics of current interest or importance receive stronger emphasis. Of note are heat transfer enhancement, computer modelling of flow and prediction of performance and multicomponent condensation.

Cover Illustration

Section of a computer drawing of condenser tube layout, derived from the HTFS program OPTU3 (by courtesy of the Heat Transfer and Fluid Flow Service, AERE, Harwell).

ERRATA

			error	correct version
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POWER CONDENSERS
Theory

CONDENSATION IN TUBE BANKS

Tetsu Fujii*

This report treats condensation phenomena in small, simple tube banks as a basic problem for research and development on turbine condensers. Current studies are described on the effect of air upon heat transfer, characteristics of pressure drops and mechanism of inundation, while referring to future problems. Then, variations in condensation characteristics in the directions of the steam flow and of the tube axis are predicted by a simple calculation, and a bank consisting of enhanced tubes is compared with that of smooth tubes regarding the number of rows, tube length, and depth of the bank.

INTRODUCTION

Technology for steam surface condensers was generally completed in the 1930's as seen in references (1) – (6). A synopsis of claims in patents before that time reveals that improvements on the condensers in terms of the layout of tubes had been made in accordance with the following objectives (Fujii and Uehara (7)): [1] to have steam distributed evenly and smoothly in all tubes (to prevent air pockets), [2] to reduce the thickness of liquid film on tubes (to prevent inundation), and [3] to prevent the sub-cooling of condensate (to increase thermo-cycle efficiency). These objectives are still valid today. However, condensers have been growing in capacity, rendering these objectives infeasible without concrete design schemes, more advanced basic studies on condensation, and mathematical techniques for integrating their results.

The author (8) has made suggestions as to the development of turbine condensers on the basis of his historical consideration and laboratory experience. The outline of his suggestions is as follows: [1] Design a shell or a special device so that water droplets in steam can be separated between the turbine exhaust and the condenser tube nest; [2] Cause steam to flow horizontally through in-line banks where the speed is high; [3] Provide individual tube nests for the high-speed steam, low-speed steam, and air-cooling zones, and connect them in series; and [4] Duly consider variations in characteristics in the tube axis direction. Furthermore, he has suggested that the flow field should be calculated independently for the tube nest zones and spaces between the tube nests and the shell, and that the boundary conditions in these calculations could be met on circumference of the tube nests. Most of the concepts in these suggestions have been adopted in design policy for the bell-type condenser developed by Toshiba Corporation, whose high performance are represented by Ozeki et al. (9).

Davidson and Rowe (10) and others discussed computer modeling at the workshop held in

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Monterey, California in 1980, proving that it was very effective in the diagnosis of defects in condensers in operation.

Both in the pursuit of designs and in diagnosis of existing condensers, the reliability of mathematical estimation depends on the reliability of numerical values concerning characteristics of heat transfer and steam flow in tube banks. Although such values should be reduced from the measurements on actual condensers, the values obtained by simple tube banks on a laboratory scale are more reliable at present. This report, confined to the latter case, intends to recognize the latest progress on research activities, while pointing out problems awaiting solution.

EFFECT OF AIR

Chisholm (11) reviewed the heat and mass transfer of steam-air mixture and its application to condenser designs and performance. In this review a classic method based on stagnant film models and their modifications was used. Rachko (12), Berman (13, 14, 15), Fuks and Zernova (16) and Buglaev et al. (17, 18, 19) studied the relevant subject for tube banks, and presented empirical formulas obtained by using the classic method and dimensional analysis. The physical meaning of their formulas is not quite clear, because they contain an inundation term, which is related to thickness of the liquid film but will have scarcely a direct effect on mass transfer. The method used by Rose (20) of applying the boundary layer theory to the condensation of steam-air mixture on a single tube will be superior to the classic method. Here is presented a more simple method for tube banks based on the boundary layer theory.

Theoretical Prediction of Mass Transfer

Average heat transfer coefficient for laminar forced convection to a single-phase flow along a flat plate is given by the equation

$$Nu_{\ell}{}^{+} = 0.664 Re_{\ell}{}^{\frac{1}{2}} Pr^{\frac{1}{3}} \quad \text{.......... (1)}$$

On the other hand, average heat transfer coefficients of a single-phase flow for tube banks with in-line, and staggered arrangements are derived from references (21) – (23) as

$$Nu_{d}{}^{+} = 0.52 Re_{d}{}^{\frac{1}{2}} Pr^{\frac{1}{3}} \text{ (in-line)} \quad \text{.......... (2)}$$

$$Nu_{d}{}^{+} = 0.70 Re_{d}{}^{\frac{1}{2}} Pr^{\frac{1}{3}} \text{ (staggered)} \quad \text{.......... (3)}$$

in the range $50 < Re_d < 100$. The ratios of equations (1) to (2) and equations (1) to (3) yield the following relations, respectively

$$L = 1.63D \text{ (in-line)} \quad \text{.......... (4)}$$

$$L = 0.90D \text{ (staggered)} \quad \text{.......... (5)}$$

These mean that equations (1) and (2), and equations (1) and (3) can be converted with equations (4) and (5) respectively.

The mass transfer coefficient obtained by a theoretical analysis of laminar forced convection condensation of a steam-air mixture on a flat plate is given by Fujii et al. (24) as

$$Sh_{\ell} = 0.92f(\omega,Sc)Re_{\ell}^{\frac{1}{2}} \quad (6)$$

where $f(\omega,Sc) = (1+\omega)^{-0.48}\omega^{-0.52}Sc^{0.32}$ .... (7)

Since equation (6) is derived from the characteristics of the vapour phase boundary layer, it can be assumed that the same conversion as that with the single-phase flow is possible with respect to a representative length. Thus, by converting equation (6) with equations (4) and (5), mass transfer in tube banks can be predicted by the equations

$$Sh_d = 0.72f(\omega,Sc)Re_d^{\frac{1}{2}} \text{ (in-line)} \quad (8)$$

$$Sh_d = 0.97f(\omega,Sc)Re_d^{\frac{1}{2}} \text{ (staggered)} \quad (9)$$

The validity of this method has been proved by experimental results with single tubes by Fujii and Kato (25), and the thus obtained equation coincides numerically with Rose's equation. Also, an equation corresponding to body force convection condensation can be obtained by using a similar method. It reveals that the velocity of the main flow has such a strong effect on mass transfer, that presumably, equations (8) and (9) can be applied when $u_\infty{}^2T_\infty/g_nD(T_\infty-T_i) \gtrsim 10$.

Comparison Between Theory and Experiment

Figure 1 shows experimental results for heat transfer of steam containing air in a tube bank (D = 14mm, L = 100mm, P_T/D = 22/14, 15 rows 5 lines) by Fujii and Oda (26). The solid lines in the Figure denote average curves for data for pure steam obtained with the same equipment (Fujii and Oda (27)). The chain lines represent a theoretical solution by Fujii et al. (28) as

$$Nu = 0.69\left(\frac{GaPr_L}{Ph}\right)^{\frac{1}{4}} \quad (10)$$

for body force convection condensation of pure steam on a single tube with uniform surface heat flux density. Where the solid lines lie below the chain lines, there exists an effect of inundation. Different runs are identified by such symbols as ○, □, ◇, △, and ▽. Various patterns appearing in each symbol such as ○, ◐, ... ● represent ranges of the mass concentration of air $x_{g\infty}$ as shown in the Figure. The left and right ends of data for the same run correspond to the inlet and outlet of the tube bank. The data corresponding to a few tube rows from the inlet coincide with those for pure steam. In addition, the effects of $Pr_L/FrPh$ and $x_{g\infty}$ on $Nu/\sqrt{Re_L}$ display similar trends with those for a single tube.

Figure 2 shows Sh obtained from the data given in Fig. 1. In calculating the values of Sh, temperature T_i at the vapour-liquid interface was obtained by substituting the measured values of $\dot{q}$ and T_w and the calculated value α, which is obtained by substituting $T_i = T_s$ into empirical formulas that represent solid lines drawn in Figure 1, into $\dot{q} = \alpha(T_i-T_w)$. In other words, T_i corresponding to x_{gi} was obtained on the assumption that characteristics of the liquid film are not affected by the mixed air.

Thick solid lines in the Figure represent average lines for data drawn in parallel with thin lines corresponding to equations (8) and (9). Data shows dispersion to an extent of about ±(20 ~ 30)% of the average lines. This depends on the accuracy of equations for pure steam as well as on the accuracy of the actual data in Fig. 1. Experimental values are about 30 ~ 50% higher than the theoretical estimations for horizontal flow, about 40% for vertical down flow, and about 25% for vertical up flow. These differences may be considered to be caused by turbulence in the main flow. In addition, note that $x_{g\infty}$ to be used in the calculation of data was calculated as a value averaged over the widest cross section.

When the condensation rate ratio of the experimental value to the theoretically estimated value corresponding to the experimental conditions T_∞, u_∞, $x_{g\infty}$, and T_W was calculated, it averaged about 1.2 for the horizontal flow and downflow, and about 1.1 for the upflow.

Method of Application

Within a limited operational range of turbine condensers, an equation for calculating Nu without repeated calculation can be obtained. An example is shown for downflow through an in-line bank. Heat transfer by pure steam is given by the equation which represents the corresponding solid line in Fig. 1.

$$Nu_s = 0.77\left(\frac{Pr_L}{FrPh}\right)^{0.15} Re_L^{\ 0.5} \quad \ldots\ldots (11)$$

Sh is given by the following equation in reference to Fig. 2

$$Sh = 1.4 \times 0.72 f(\omega, Sc) Re^{0.5} \quad \ldots\ldots (12)$$

Repeated calculation with a combination of equations (11) and (12) gives numerical values of Nu for given conditions of steam and cooling tube. Such kind of data can be approximated within an accuracy of a few per cent by the equation–

$$\frac{Nu}{Nu_s} = A\exp(-mx_{g\infty}) + B\exp(-nx_{g\infty}) \quad \ldots\ldots (13)$$

where $A = 0.83(T_\infty - T_W)^{-0.15} + 0.1\ln u_\infty$, $B = 0.21(T_\infty - T_W)^{0.25} - 0.09\ln u_\infty$, $m = 3.7u_\infty^{-0.12}$ and $n = 19(T_\infty - T_W)^{0.2} u_\infty^{-0.3}$; applicable ranges are $0 < x_{g\infty} < 0.3$, $2 < u_\infty < 20$ m/s, $0 < T_\infty - T_W < 15$K, $20 < T_\infty < 40$°C and D = 14mm.

The problem that remains to be solved in this section may be to discover experimentally the effect of the tube arrangement on equations (8) and (9). It will be related to the turbulent diffusion of air in theory.

PRESSURE DROP

Since it is still difficult to strictly mathematically solve turbulent flow with suction in a tube bank, only empirical knowledge is available.

Drag coefficient ξ is defined by the equation–

$$\xi = \frac{\bar{\rho}}{2n\dot{m}_{max}^2}\left\{(p_i - p_j) + \left(\frac{\dot{m}^2}{\rho}\right)_i - \left(\frac{\dot{m}^2}{\rho}\right)_j\right\} \quad \ldots\ldots (14)$$

Note here that equation (14) was derived, within a control volume surrounded by i-th and j-th widest cross sections and side walls, from the momentum law on the assumptions that [1] on cross sections i and j, pressure is uniform (variations in pressure in the cross sections are very small compared with $p_i - p_j$); [2] on cross sections i and j, the patterns of steam velocity distribution are identical (this does not hold true with the first and the last tube rows); [3] force of the side walls on the fluid is negligible in comparison with force of the tubes on the fluid; [4] momentum change of the condensate is negligible in comparison with that of steam.

Experimental Results for Drag Coefficient ξ

Figure 3 shows experimental results for local value of ξ obtained by Fujii and Oda (29). The data have a large dispersion as shown in the Figure (the accuracy of measurement is estimated to be about ±20% with higher Re, and about ±40% with lower Re). However, it is clear that the value for the first row is high, and that there is a characteristic difference between the in-line and staggered banks in comparison with ξ_D for dry tubes in the core part. It seems certain that data on upflow, which is not shown in the Figure because of its low reliability, will be higher than ξ_D in the Figure. This may be due to the effect of inundation.

Table 1 lists measured values of ξ_D for the first row and the exit of tube banks without condensation in relation to those for the core part. The fact that the values of ξ_D for the first row differ among researchers may be due to accuracy of the measurement.

TABLE 1 - Drag Coefficients at the 1st and last Rows against Those in the Core Part for dry Tube Banks.

Authors	Fluid	Tube Arrangement	D (mm)	P_T/D	1st Row	Core Part	Last Row
Fujii et al. (31, 32)	air	in-line	15.6	1.6	3.5	1	-1.7
		staggered	15.6	1.6	2.0	1	-0.8
Fujii et al. (29)	steam	in-line	14.0	1.57	3	1	–
Nicol et al. (33)	steam	in-line	9.525	1.736	2.2	1	–

Table 2 lists relative values of ξ to ξ_D obtained by Fujii and Oda (29), Nicol et al. (33), Lee et al. (34) and Lee (35). The values for the former two obtained with condensation of steam are qualitatively identical. Particularly, the prominent decrease of ξ for the in-line banks is quite characteristic. Also, the ratio of ξ for the first row to that for core part becomes lower as a result of condensation (cf. Table 1). The test by Lee et al. consists of a simulation of condensation with the suction of air through seven rows of porous tubes. The results agree qualitatively with the above two

TABLE 2 - Drag Coefficients in Condensation of Steam and Suction of Air against Those without Condensation and Suction.

Authors	Tube Arrangement	D (mm)	P_T/D (trans./ longi.)	1st Row	Core Part	
					Horizontal	Downflow
Fujii et al. (29)	in-line	14.0	1.57/1.57	$2.3\xi_D$	$0.4\xi_D$	$0.35\xi_D$
(condensation	staggered	14.0	1.57/1.57	$1.8\xi_D$	ξ_D	$0.85\xi_D$
at Re = 10^4)						
Nicol et al. (33)	in-line	9.525	1.736/1.736	$1.4\xi_D$	$0.27\xi_D$	–
(condensation	staggered	9.525	1.58/1.80	–	$0.75\xi_D$	–
at Re = 10^4)						
Lee (35)	in-line	19.0	1.25/1.25	–	ξ_D or more	
(suction at	staggered	19.0	1.25/1.083	–	$0.9\xi_D$	
Re = 2.5 x 10^4)	staggered	19.0	1.25/0.935	–	$(0.85–0.9)\xi_D$	

cases for the staggered bank, while ξ for the in-line bank is increased by suction (this characteristic is independent of the representative value of $\bar{\dot{m}}_{max}$ in equation (14).)

The foregoing results raise the problem as to whether or not condensation can be simulated by suction. There is a possibility that the change of shape of liquid-vapour interface promoted a drastic decrease of the value of ξ for the in-line bank. This doubt will be resolved by an experiment with the same tube arrangement, the same Reynolds number, and the same suction rate as those of the condensation case.

Comparison with Recommended Values for Pressure Drop

Figure 4 shows experimental results obtained by Fujii and Oda (29) for pressure distribution in small tube banks, whose dimensions are shown in Table 2. These data are compared with predicted values. The prediction is made by substituting recommended values of Δp_f into the equation

$$p_{j+1} = p_j - [\Delta p_f - \left\{\left(\frac{\dot{m}^2}{\rho}\right)_j - \left(\frac{\dot{m}^2}{\rho}\right)_{j+1}\right\}] \qquad (15)$$

The values expressed by solid lines in the Figure were obtained by substituting the values of ξ in the 1st and second lines in Table 2 into $\Delta p_f = 2\xi\dot{m}^2_{max}/\rho$. The slight difference between the measured and the predicted values is attributed to the fact that the values of ξ in the Table have been roughly determined. The values expressed by symbols + and x were obtained by using the values of Δp_f which were derived from experiments with adiabatic two-phase flow of air and water by Diehl and Unruh (36) and Ishihara et al. (37) respectively. In every calculation the values of $\dot{m}_j$ were obtained from calculated values of heat exchange for individual runs.

While the recommended values by Ishihara et al, offer a fairly good approximation for the staggered bank, they cannot be applied to the in-line bank. Note here that the experimental data for runs SA and SB were obtained with pressure at the exit of the tube bank greatly lowered. Calculations with an increasing number of tube rows show that the temperature of steam approaches that of cooling water before the completion of condensation. In other words, pressure drop per row as large as those will not occur under actual conditions.

The equation of total pressure drops for tube banks with n rows presented by Rowe et al. (38) can be transformed as

$$p_o - p_n = 0.0667\left\{\frac{\dot{m}_o p_T}{(p_T - D)}\right\}^2 \frac{n}{\rho} \qquad (16)$$

The values obtained by this equation compare fairly well with pressure drops for the staggered bank, which are obtained by extending such curves as shown in Fig. 4 (except for Runs SA and SB) to the complete condensation, but predict nearly one-half the values of those for the in-line bank.

In short, existing knowledge makes it possible to estimate pressure drops in the staggered bank fairly accurately. On the other hand, experimental data is insufficient to estimate pressure drops in the in-line bank. Considering that the latter tube arrangement is very effective in its application to actual condensers, there is a great demand for detailed measurement of the effect of pitch to diameter ratio upon the ξ value. Moreover, there are a few problems to be studied, such as tube arrangements in a central flow-type condenser and with various kinds of by-pass lanes.

INUNDATION

Modification of Nusselt's Solution

The basic assumption in Nusselt's (39) inundation theory on a vertical row of horizontal tubes laid in stagnant vapour is that all droplets of condensate from the n-th tube from the top tube are caught by the (n+1)th tube and spread uniformly over the top of the latter tube. Therefore, calculation was made with the physical model by assuming that the condensate continuously falls, forming a sheet between n-th and (n+1)th tubes. The disagreement of this model with real phenomena was described in detail by Young and Wohlenberg (40), and has been reconfirmed by a number of observant researchers. While selecting only the extent of the droplet spread over the tube surface out of the complicated behavior of the liquid film, Fujii and Oda (41) proposed the two simple models shown in Fig. 5.

[Model I] If the heat transfer rate per unit tube length of the first tube is represented by $\dot{Q}_1$, then the flow rate of falling condensate is given by $\dot{M}^+ = \dot{Q}_1/\Delta h_v$. When the condensate reaches to the 2nd tube, $\dot{M}_1 = \zeta\dot{M}^+$ spreads axially, $(1-\zeta)\dot{M}_1{}^+$ flows to the tube bottom without affecting thickness of the liquid film. These join with the condensate $\dot{Q}_2/\Delta h_v$ generated on the 2nd tube, amounting to $\dot{M}_2{}^+ = (\dot{Q}_1+\dot{Q}_2)/\Delta h_v$ and falls down to the 3rd tube. Of this amount, $\dot{M}_2 = \zeta\dot{M}_2{}^+$ spreads axially, $(1-\zeta)\dot{M}_2{}^+$ without affecting the liquid film thickness. Further, it proceeds likewise.

Upon determination of an integral constant for the solution of Nusselt's equation on the basis of this model, $(Nu_n)_I$ for the n-th tube is given by the equation

$$(Nu_n)_I = \left(\zeta\sum_{i=1}^{n-1} Nu_i\frac{Ph_i}{Ph_n}\right)\left[\left\{1 + \frac{\left(\frac{Ph_1}{Ph_n}\right)^{\frac{1}{3}} Nu_1^{\frac{4}{3}}}{\left(\zeta\sum_{i=1}^{n-1} Nu_i\frac{Ph_i}{Ph_n}\right)^{\frac{4}{3}}}\right\}^{\frac{3}{4}} - 1\right] \qquad (17)$$

[Model II] On the assumption that the fraction $(1-\zeta)$ of condensate falling from any tube passes over the tube bank without affecting film thickness on the subsequent tubes, $(Nu_n)_{II}$ is given, similarly to Model I, by the equation

$$(Nu_n)_{II} = \sum_{i=1}^{n-1}\zeta^{n-i} Nu_i\frac{Ph_i}{Ph_n}\left[\left\{1 + \frac{\left(\frac{Ph_1}{Ph_n}\right)^{\frac{1}{3}} Nu_1^{\frac{4}{3}}}{\left(\sum_{i=1}^{n-1}\zeta^{n-i} Nu_i\frac{Ph_i}{Ph_n}\right)^{\frac{4}{3}}}\right\}^{\frac{3}{4}} - 1\right] \qquad (18)$$

Values of $(Nu_n)_I$ and $(Nu_n)_{II}$ in equations (17) and (18) can be successively calculated from

$$(Nu_1)_I = (Nu_1)_{II} = 0.725\left(\frac{GaPr_L}{Ph_1}\right)^{\frac{1}{4}} \qquad (19)$$

, and both agree with Nusselt's solution when $Ph_n = Ph_1$ and $\zeta = 1$.

Figures 6 (a) and (b) show calculations with $Ph_n = Ph_1$ in equations (17) and (18) respectively, and Figures 6 (c) and (d) show average Nusselt numbers $(\overline{Nu}_n)_I$ and $(\overline{Nu}_n)_{II}$ obtained from these equations for the 1st through the n-th tube respectively. As the values of ζ decrease, Nusselt numbers become higher than Nusselt's solution (ζ = 1). Model II reveals this effect more prominently with the increase of n, showing a different trend from the Nusselt's solution.

Comparison Between Theory and Experiment

Figures 7 and 8 show existing experimental data (Young and Wohlenberg (40), Katz and Geist (42), Short and Brown (43), Katz et al. (44), Young and Briggs (45) and Smirnov and Lukanov (46)) and recommended equations (Kern (47) and Kutateladge (48)) for $Nu_n/(Nu_1)_{th}$ and $\overline{Nu}_n/(Nu_1)_{th}$ respectively. In Fig. 7 average values from experiments with R-12 by Young and Wohlenberg fairly agree with the values of Model I using ζ = 0.3. In Fig. 8, the equation recommended by Kern

$$\overline{\alpha_n} = \alpha_1 n^{-\frac{1}{6}} \quad \text{(20)}$$

nearly agrees with Model I using ζ = 0.4. Butterworth (49) has pointed out that equation (20) nearly agrees with the test results by Grand and Osmet (50), represented by the equation

$$\frac{Nu}{(Nu_1)_{th}} = \left(\frac{\Sigma \dot{m}_n}{\dot{m}_n}\right)^{-0.332} \quad \text{(21)}$$

As a matter of course, nearly the same relationship as equation (21) can be derived from the solution of Model I using ζ = 0.4. In Figs. 7 and 8, no similar data characteristic of Model II can be found.

Figure 9 compares experimental results for R-12 by Gogonin et al. (51) with $(\overline{Nu}_n)_{II}/(Nu_1)_{th}$ in Model II. Symbols ○, □, ◇, and ▽ in the Figure denote data under conditions θ_s = 40°C, $\theta_s - \theta_w$ = 3 – 6°C and $\dot{q}_w$ = 6 – 13 kW/m^2, ●, ■, ◆, and ▼ under conditions θ_s = 40°C, $\theta_s - \theta_w$ = 8 – 21°C, and $\dot{q}_w$ = 12 – 32 kW/m^2 and +, X, Y, T, and V under conditions θ_s = 90°C, $\theta_s - \theta_w$ = 21 – 56°C, and $\dot{q}_w$ = 31 – 76 kW/m^2. Let us designate these three groups as A, B, and C. The data in groups A and B and most of the data in group C nearly agree with theoretical solutions using ζ = 0.6 and 0.1 in the 3rd through the 10th row.

As shown in Figs. 7 to 9, existing experimental results disagree so widely with each other that it is difficult to proceed with further comparison with theory. Either pattern of condensation—Model I or Model II—can occur. In the future, it will be possible to discover unknown factors to be introduced by closely observing phenomena in highly accurate experiments that can vary such factors as heat flux density, surface tension, and geometrical patterns of tube rows.

Trends of Studies on Inundation

Inundation without vapour shear can only be applied to a part of a large-capacity condenser and to a small-capacity condenser, because the effect of steam velocity is added in large-capacity condensers. However, most of the experimental results are represented in the same form as equation (20) without this distinction, as—

$$\alpha_n = \alpha_1 n^{-\gamma} \quad \text{or} \quad \overline{\alpha}_n = \alpha_1 n^{-\gamma'} \quad \text{(22), (23)}$$

where the values of the indices, which are related to the level of steam velocity and the inundation rate, are in the ranges of γ = 0 – 0.15 and γ' = 0 – 0.25. Some of the experimental results on inundation are plotted in terms of $\alpha_n(\nu^2/g_n)^{\frac{1}{3}}/\lambda$ versus $\pi D \Sigma \dot{q}_i/(\rho \nu \Delta h_v)$. However, the accuracy of the correlation is of the same order as that in equation (22) or (23).