CONF-790631--10

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Prepared for

6th Ocean Thermal Energy Conversion Conference

Washington, D.C.

June 18-21, 1979



U of C-AUA-USDOE

ARGONNE NATIONAL LABORATORY, ARGONNE, ILLINOIS

Operated under Contract W-31-109-Eng-38 for the
U. S. DEPARTMENT OF ENERGY

PERFORMANCE TESTS OF THE 1MWt SHELL-AND-TUBE HEAT EXCHANGERS FOR OTEC*

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## Abstract

Final test results are reported for the five 1 MWt shell-and-tube heat exchangers tested at Argonne National Laboratory. These five heat exchangers are the Union Carbide flooded-bundle evaporator, the Union Carbide sprayed-bundle evaporator, the Union Carbide enhanced-tube condenser, the Carnegie-Mellon vertical fluted-tube evaporator, and the Carnegie-Mellon vertical fluted-tube condenser. Performance parameters measured include the overall heat transfer coefficient ($U_O$), the water-side pressure drop, and the vapor quality. Also measured were operational characteristics of the heat exchangers such as repeatability of results and the dependence of $U_O$ on heat duty, ammonia flow rate, and subcooling. Individual water-side and ammonia-side coefficients were deduced using the Wilson Plot method.

## Introduction

The Ocean Thermal Energy Conversion (OTEC) test program at Argonne began in 1977 with the specific purpose of measuring the heat transfer performance of OTEC heat exchangers. Preliminary test results for two of the five shell-and-tube heat exchangers were reported at the Fifth OTEC Conference in Miami Beach, Florida. In this paper, the final experimental results for all five 1 MWt shell-and-tube heat exchangers are reported. These five heat exchangers are the Union Carbide flooded-bundle evaporator, the Union Carbide sprayed-bundle evaporator, the Union Carbide enhanced-tube condenser, the Carnegie-Mellon vertical fluted-tube evaporator, and the Carnegie-Mellon vertical fluted-tube condenser. Performance parameters measured include the overall heat transfer coefficient ($U_O$), the water-side pressure drop, and the vapor quality. Also measured were operational characteristics of these heat exchangers such as repeatability of results and the dependence of $U_O$ on heat duty, ammonia flow rate, and subcooling. Individual water-side and ammonia-side heat transfer coefficients were deduced using the Wilson Plot method.

The material in this paper is presented in four parts. First, brief descriptions of the five shell-and-tube exchangers tested are given. Second, the experimental facility and instrumentation are described. Third, the formulation and methods used to deduce the overall and individual heat transfer coefficients and their possible errors are discussed. Finally, the experimental results for the five shell-and-tube heat exchangers are presented.

*The work sponsored by DOE's Ocean Systems Branch, Division of Central Solar Technology, was performed under Contract No. 09ENG38 with the University of Chicago, operator of Argonne National Laboratory.

## Test Heat Exchangers

The five 1 MWt shell-and-tube heat exchangers that have been tested are:

1. A flooded-bundle evaporator supplied by the Linde Division of Union Carbide Corporation. This unit is a horizontal three-pass shell-and-tube heat exchanger containing 279 tubes each 1.5 in. OD by 75 in. long. The tubes are made of titanium and are coated with a porous aluminum High Flux surface on the ammonia side. Water flows through the tubes, which are circular and unenhanced on the inside.

2. A horizontal sprayed-bundle evaporator supplied by Union Carbide/Linde and containing titanium tubes coated with the porous aluminum High Flux surface. Liquid ammonia is sprayed onto the outside surface of the tubes by means of perforated feed tubes in the top row of the tube bundle. Water flows through the tubes, which are unenhanced on the water side, in a four-pass arrangement. This unit has 388 tubes each 1.5 in. OD by 55 in. long.

3. A horizontal enhanced-tube condenser provided by Union Carbide/Linde containing aluminum tubes with internal axial fins on the water side and a thin-film promoter on the outside to enhance the condensation of ammonia (Fig. 1). This unit has 147 tubes each 1.5 in. OD by 155 in. long.

4. A vertical fluted-tube evaporator designed by Carnegie-Mellon University (C-MU). This heat exchanger contains 240 fluted aluminum tubes, 1 in. in diameter (nominal) by 172 in. long. The tubes are fluted on both the inside and outside. Ammonia is fed in a thin film onto the outside of the tops of the tubes, and water flows downward on the inside.

5. A vertical fluted-tube condenser, also designed by Carnegie-Mellon. This unit is identical in all respects to the fluted-tube evaporator described above, except that the external (ammonia-side) surface of each tube contains 33 flutes, rather than 42 (Fig. 2).

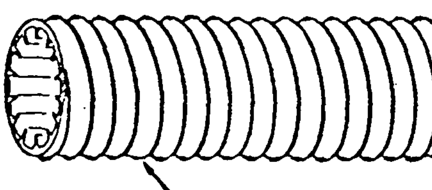


Fig. 1 Linde condenser tube with internal axial fins and external wrapped wire enhancement.

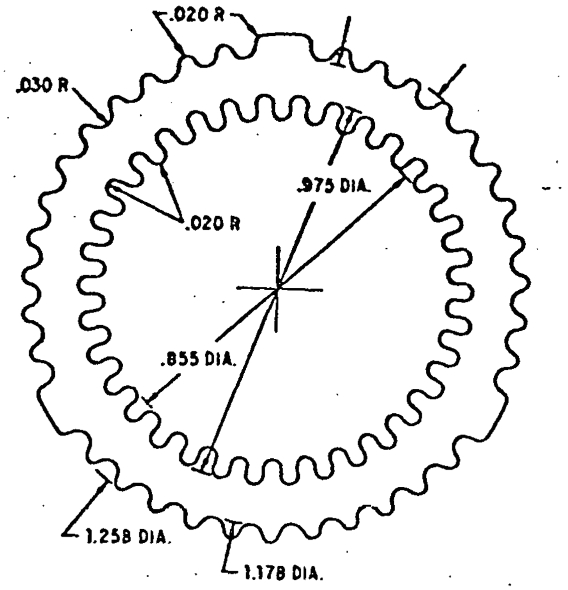


Fig. 2 Cross section of C-MU fluted condenser tube (all dimensions in inches).

Future plans include tests of the following heat exchangers: the Applied Physics Laboratory folded-tube evaporator in which the ammonia is on the inside of the tubes, an aluminum plate-fin evaporator built by the Trane Company, and the Variflux concentric-tube evaporator developed by Rockwell International.

## Test Facility

During 1977 a test facility was designed and constructed at Argonne in order to measure the performance of test heat exchangers operating under simulated OTEC conditions. The test facility, shown in Fig. 3, is designed to measure heat transfer rates and water pumping requirements of evaporators and condensers sized to handle a nominal heat duty of 3.2 million Btu/hr. The facility is designed for steady-state operation over a range of temperatures and flow rates of warm and cold water shown in Table 1. A detailed description of this facility is reported elsewhere.[1,2] Provision is made for control of temperatures and flow rates at preselected values and for measurement of all quantities necessary for calculation of the overall heat transfer coefficient and water-side pressure drops of the test units. The operating conditions of the heat exchangers during the test are typical in all respects of those expected of actual OTEC plants except that deionized water, chemically treated to inhibit the effects of biofouling and corrosion, is used instead of ocean water. The data obtained on clean-tube performance, combined with information on the effects of biofouling, provide a means for predicting the performance of similar evaporators and condensers in an ocean environment.

The accuracy goals for the measurements characterizing the overall performance of the test units are ±3% for the overall heat transfer coefficients and ±5% for the water-side pressure drops across the heat exchangers. The equipment used for making these measurements is described in the Data Reduction section. In general, the actual measurement accuracy far exceeds the accuracy goals of the test program.

The test facility has a computer-based data system that can, on command, read the sources of data, apply calibration and correction factors to these data, compute heat balances and overall heat transfer coefficients, and produce hard-copy records of all measured and calculated quantities.

## Data Reduction

There are three independent ways of determining the heat duty (q) of the heat exchangers: a) from the temperature change of the water passing through the heat exchangers; b) from an energy balance on the ammonia side; and c) from an overall energy balance on the warm water and cold water loops.

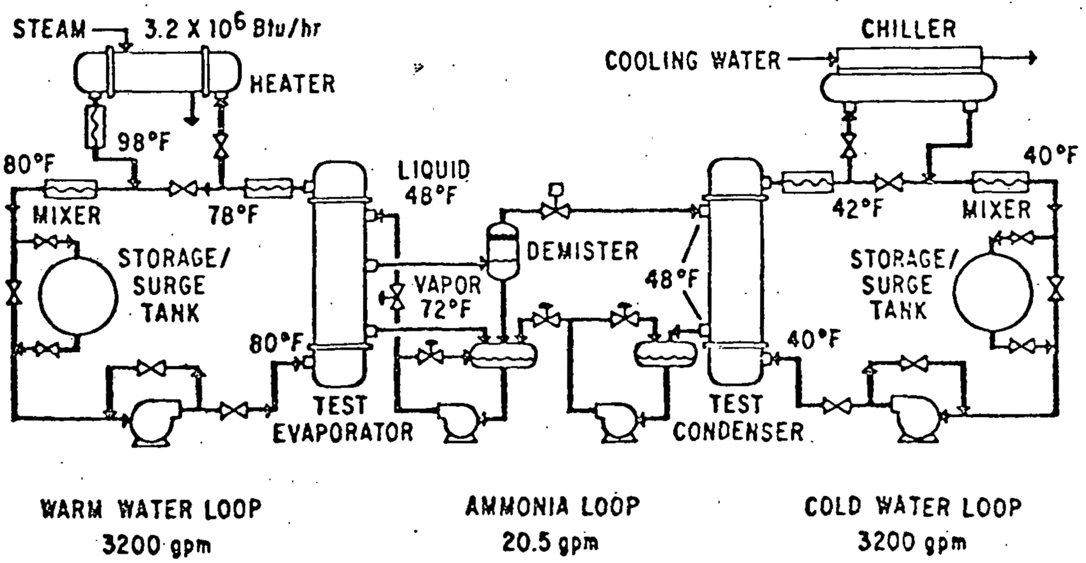


Fig. 3 OTEC heat exchanger test loop.

Table 1 Test Facility Operating Conditions

| Parameter | Nominal Value | Range |
|---|---|---|
| Heat Duty of Test Units, Btu/hr | 3.2 million | 2.4 million–4.0 million |
| Warm Water Supply Temperature, °F | 80° | 75–85° |
| Cold Water Supply Temperature, °F | 40° | 38–45° |
| Water Temperature Change, °F | 2° | 2–4° |
| Log-Mean Temperature Difference, °F | 7° | 4–10° |
| Water Flow Rate, gpm | 3200 | 1500–5000 |

Of these three possible methods, the first was determined to be the most accurate and is used exclusively in all subsequent calculations.

## Overall Heat Transfer Coefficient

Once a value of the heat exchanger heat duty, q, is obtained, the overall heat transfer coefficient, $U_o$, is calculated from the conventional definition:

$$U_o = \frac{q}{A_o \Delta T_{1m}} \tag{1}$$

where:

$A_o$ = the outside surface area of the tubes (mean outside surface area for fluted tubes), and

$\Delta T_{1m}$ = the log-mean temperature difference between the water and ammonia streams.

$\Delta T_{1m}$ can be expressed in equation form as follows:

$$\Delta T_{1m} = \frac{T_i - T_o}{\ln\left[\frac{T_i - T'_o}{T_o - T'_o}\right]} \tag{2}$$

where:

$T'_o$ = the ammonia temperature,

$T_i$ = the inlet water temperature, and

$T_o$ = the outlet water temperature.

The instruments used to measure the water-side pressure drop and the quantities required for calculating the overall heat transfer coefficient are shown in Table 2, along with their respective accuracies. With the present instrumentation, the

Table 2 Instruments and Measurement Accuracy

| Quantity Measured | Instrument | Accuracy of Measurement |
|---|---|---|
| Temperature of: | | |
| Water into and out of heat exchanger<br>Ammonia vapor in evaporator and after expansion valve | Quartz crystal thermometer | ±0.010°F |
| Water into and out of heater and chiller | Copper-constantan thermocouple | ±0.1°F |
| Pressure of: | | |
| Ammonia vapor in heat exchanger and after expansion valve | Quartz crystal transducer | ±0.005 psia |
| Pressure difference of: | | |
| Water into and out of heat exchanger<br>Water into and out of pump<br>Static head of liquid ammonia in evaporator (liquid level) | Strain gauge transducer | ±0.5% of reading |
| Flow rate of: | | |
| Water through heat exchanger<br>Water through heater and chiller | Turbine meter | ±0.15% of reading |
| Liquid ammonia feed to evaporator | Turbine meter | ±0.5% of reading |

probable errors in both the overall heat transfer coefficient and the water-side pressure drop are within ±1%, well within the accuracy goals of the test program (see Test Facility section).

## Individual Heat Transfer Coefficients

Individual ammonia-side and water-side heat transfer coefficients can be determined by the standard Wilson procedure. This procedure makes use of the relationship between $U_o$ and the individual coefficients that expresses the total heat transfer resistance as the sum of its separate components. This can be expressed as:

$$\frac{1}{U_o} = \frac{A_o}{h_i A_i} + \frac{1}{h_o} + \frac{\Delta r A_o}{k A_{lm}} + \frac{1}{h_{fo}} + \frac{A_o}{h_{fi} A_i} \qquad (3)$$

where $A_o$ and $A_i$ are selected as the outside and inside areas, respectively. (For the fluted tubes, the mean inside and outside areas are employed.) The third term on the right-hand side of Eq. 3 is the wall resistance. In general, the water-side heat transfer coefficient, $h_i$, is proportional to some power, n, of the water flow rate, w. For the Union Carbide enhanced-tube condenser, n = 0.66 in accordance with the empirical correlation of Noranda.[3] For all other heat exchangers tested the standard value of n = 0.8 is employed. If a series of runs are made over a range of water flow rates under conditions where $h_o$ is constant, a plot of the values of $U_o^{-1}$ versus $w^{-n}$ should fall on a straight line. Extrapolation of the line to the intercept at $w^{-n} = 0$ will yield a value of $U_o^{-1}$ that is equal to the sum of the last four terms on the right side of Eq. 3. The value of the ammonia-side coefficient, $h_o$, can then be determined by subtracting the values of the tube wall and fouling resistances, if they are known. Finally, the values of the water-side coefficient, $h_i$, can be obtained for each value of w. It should be pointed out that the ammonia-side heat transfer coefficient is generally a function of location within the heat exchanger; consequently, the $h_o$ value in Eq. 3 represents an average value.

Clearly, the use of this procedure for determining $h_o$ and $h_i$ (w) requires that the data on $U_o$ (w) be obtained under conditions where the average value of $h_o$ is constant and the fouling coefficients, $h_{fo}$ and $h_{fi}$, are known. In order to ensure that $h_o$ remained the same during the runs at different water velocities, the heat duty, q, was held constant at the nominal design value of 3.2 million Btu/hr, and the ammonia feed rate (for test evaporators) was held constant. Although the inlet water temperature remained unchanged, the ammonia-side temperature had to vary slightly from run to run in order to accommodate the different values of $U_o$. Consequently, the profiles of the local difference between the water and ammonia temperature along the length of the tubes were not identical for the different flow rates. Nevertheless, the average ammonia-side coefficient, $h_o$, remained approximately the same because the heat duty was held constant.

The requirement that the fouling resistances be known is a somewhat less certain matter, particularly in the case of the water-side fouling coefficient, $h_{fi}$. Because of precautions taken to maintain the cleanliness of the ammonia system, it is unlikely that foreign materials from this system could foul the ammonia-side surfaces. The water system has been maintained under a nitrogen blanket since the initial filling. The water is continuously filtered during operation by high-capacity filters located in bypass lines around the water pumps. The cleanliness of the system and the effectiveness of the corrosion inhibitor in scavenging oxygen and preventing the deposit of hard calcium lead us to expect that the magnitude of the water-side fouling resistance approaches zero. Thus it is assumed that the last two terms in Eq. 3 are small enough to be neglected in determining $h_o$ and $h_i$ by the Wilson procedure.

# Test Results

## Union Carbide Flooded-Bundle Evaporator

A summary of test results for the Union Carbide flooded-bundle evaporator, as well as for the other heat exchangers, is given in Table 3. Under nominal operating conditions (3200 gpm water flow* and 3.2 million Btu/hr heat duty), the overall heat transfer coefficient ($U_o$) is 785 Btu/hr·ft²·°F and the ammonia-side and water-side coefficients are 4800 Btu/hr·ft²·°F and 1400 Btu/hr·ft²·°F, respectively. The water-side pressure drop is 2.7 psi.

The tests on the Linde flooded-bundle evaporator show that the heat transfer performance depends in part on the exchanger's immediate past history of operation. In particular, contact between the liquid ammonia and the High Flux tube surface under nonboiling conditions appears to deactivate nucleate boiling sites, thus reducing the heat transfer coefficient. The observed rate of deactivation is fairly rapid at first: a boiling stoppage of only one hour caused a drop in $U_o$ from 750 to 690 Btu/hr·ft²·°F. The rate slows to zero over a shutdown period of several hours at the end of which the value of $U_o$ has reached its ultimate minimum of some 600 Btu/hr·ft²·°F. Operation of the evaporator under boiling conditions reverses this effect, but at a much slower rate. About 100 hours of continuous boiling at nominal design operating conditions are required to fully reactivate the High Flux surface to a $U_o$ value of 785 Btu/hr·ft²·°F from an initial value of 600 Btu/hr·ft²·°F. This behavior is illustrated in Fig. 4. Preliminary tests indicated that reactivation could be accelerated by circulating warm water with no liquid ammonia in the evaporator. This causes a rapid vaporization of liquid ammonia occluded in the pores of the High Flux surface and, hence, a more rapid reactivation than that which occurs during normal ammonia boiling.

The value of 4800 Btu/hr·ft²·°F obtained for the ammonia-side coefficient can be compared to a value of 5500 Btu/hr·ft²·°F measured for clean High Flux tubes in a bench-scale test at Union Carbide Corporation. The small difference between these two values may be due to the presence of fouling on the tube surfaces in the exchanger tested at Argonne; a fouling film with a resistance of only 0.000027 hr·ft²·°F/Btu, which would account for the difference between the two values of $h_o$, could have formed on the tube walls from residues left during cleaning and hydrotesting.

*This corresponds to the tube-side velocity of 6.5 ft/sec, except in the case of the Linde condenser for which the velocity is 5.0 ft/sec.

Table 3 Summary of Test Results for 1 MWt OTEC Heat Exchangers

| Heat Exchanger Type | Type of Enhancement | Tube Material Diameter (in.) Length (in.) | $U_o$ [1,2] | $h_{NH_3}$ [1,3] | $h_{H_2O}$ [1,3] | $H_2O$-side $\Delta p$ (psi) | Comments |
|---|---|---|---|---|---|---|---|
| Linde Flooded Evaporator (horizontal 3-pass) | High Flux on $NH_3$ side; no enhancement on $H_2O$ side | Ti OD = 1.5 ID = 1.43 L = 75 | 785 | 4800 | 1400 (8% higher than predicted by Sieder-Tate) | 2.7 | Approx. 100 hr of continuous running required to reach steady state after full deactivation |
| Linde Spray Evaporator (horizontal 4-pass) | High Flux on $NH_3$ side; no enhancement on $H_2O$ side | Ti OD = 1.5 ID = 1.43 L = 55 | 760 | 4590 | 1295 (5% higher than predicted by Sieder-Tate) | 4.0 | Performance was insensitive to recirculation ratio if greater than 1.27[4] |
| Linde Condenser (horizontal single-pass) | Proprietary enhancement on $NH_3$ side; fins on $H_2O$ side | Al 3003 OD = 1.5 ID = 1.37 L = 155 | 820 | 4860 | 1125 (enhancement ratio = 1.6; area ratio = 1.8)[6] | 1.8[5] | Within ranges tested $U_o$ was independent of heat duty, and hence liquid loading |
| C-MU Evaporator (vertical single-pass) | Fluted on both sides (40/40 flutes) | Al 6061 OD = 1.21 ID = 0.91 L = 172 | 825[4,7] | 1730 | 2610 (enhancement ratio = 1.95; area ratio = 2.07)[6] | 3.2 | $U_o$ repeatable during any continuous run, but not from one run to the next |
| C-MU Condenser (vertical single-pass) | Fluted on both sides (40/60 flutes) | Al 6061 OD = 1.21 ID = 0.91 L = 172 | 1045 | 3320 | 2470 (enhancement ratio = 2.5; area ratio = 2.07)[6] | 3.3 | Performance insensitive to inlet vapor states ranging from 90% quality to superheated |

[1]All results at nominal heat duty (3.2 million Btu/hr) and water flow rate (3200 gpm); $U_o$ and h are in Btu/hr·ft$^2$·°F.

[2]$U_o$ is referred to tube outer area; for C-MU units $U_o$ is based on mean outer area.

[3]Calculated via Wilson plot with $V^{-0.8}$, except for Linde condenser where $V^{-0.66}$ was used in accordance with Noranda correlation.

[4]Recirculation ratio = mass rate of liquid feed/mass rate of evaporation.

[5]$\Delta p$ is relatively small because the velocity is only 5 ft/sec compared to ≈6.5 ft/sec for the other units.

[6]Enhancement ratio is the ratio of the actual $h_{H_2O}$ to the predicted $h_{H_2O}$ (using Sieder-Tate) for a smooth tube; area ratio is the ratio of actual inside area to smooth tube inside area.

[7]Recirculation ratio = 5.

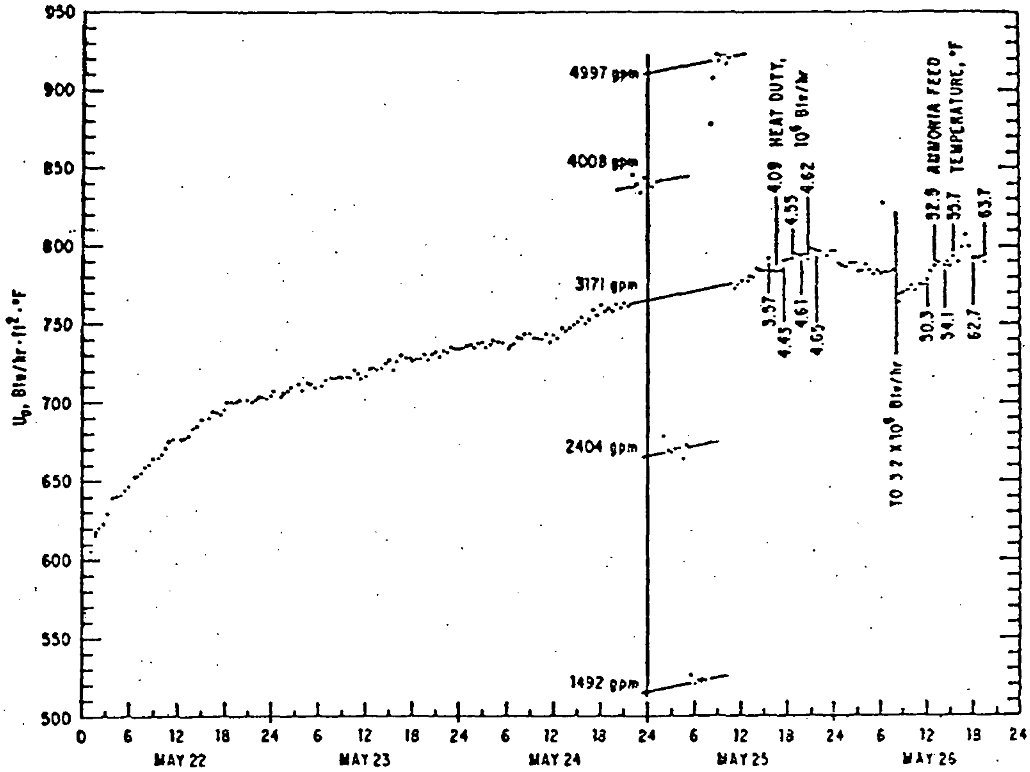


Fig. 4 Extended run results for the Linde flooded-bundle evaporator.

With respect to other matters pertaining to the performance of the flooded-bundle evaporator, it was found that the amount of liquid droplet entrainment is so small as to be undetectable; this indicates that the vapor quality is >99.9%. It was also found that varying the heat duty from 3.2 million to 4.6 million Btu/hr produced no measurable changes in $U_o$.

Finally, reducing the subcooling of the liquid ammonia feed to the evaporator had only a slight effect on thermal performance. In particular, raising the liquid ammonia feed temperature from 48°F to 64°F resulted in an increase in $U_o$ of only 10 Btu/hr·ft$^2$·°F. Consequently, it is doubtful that the use of an ammonia preheater can be justified on a cost basis.

Union Carbide Sprayed-Bundle Evaporator

Under nominal operating conditions (3200 gpm water flow, 3.2 million Btu/hr heat duty, and 40 gpm ammonia feed rate), the overall heat transfer coefficient is 760 Btu/hr·ft$^2$·°F and the ammonia-side and water-side coefficients are 4590 Btu/hr·ft$^2$·°F and 1295 Btu/hr·ft$^2$·°F, respectively. The water-side pressure drop is 4.0 psi.

As with the flooded-bundle evaporator, the past history of the High Flux surface can have a marked effect on the heat transfer performance of this exchanger. Contact between the liquid ammonia and the High Flux surface under nonboiling conditions appears to deactivate the nucleation sites, reducing the heat transfer coefficient to ≤700 Btu/hr·ft$^2$·°F. It was found that a deactivated High Flux surface could generally be reactivated by "drying out" the surface while maintaining constant warm water flow. However, if the "dryout" was conducted when the surface was already activated, the heat transfer coefficient actually decreased; this was attributed to incomplete rewetting of tubes following the dryout. Whereas the tubes within the flooded bundle are totally immersed in ammonia, the tubes within the spray evaporator are wetted by a thin falling film of ammonia. Rewetting a dry heated surface by a thin liquid film is known to be very difficult. The surface becomes covered with dry patches that can be eliminated only by "flooding" the surface or by reducing the surface temperature. (Once the dry spots are eliminated, the liquid flow rate can be reduced to a much lower value before dry areas reappear.) These dry patches represent regions of poor heat transfer, and hence result in a lower overall heat transfer coefficient.

The overall heat transfer coefficient was found to be insensitive to ammonia feed rates above the nominal (i.e., up to 100 gpm). However, as shown in Table 4, at a given heat duty there exists a minimum ammonia feed rate below which $U_o$ drops off sharply. During the runs at 3.2 million Btu/hr, the "break" occurred between 26 and 24 gpm, implying a minimum recirculation ratio of about 1.27 to maintain full heat transfer effectiveness. In the tests at 4.0 million Btu/hr, the break was between 30 and 28 gpm, as shown in Fig. 5. The corresponding recirculation ratio was 1.17. Also in Fig. 5, note that the heat transfer coefficient recovered when the ammonia feed rate was increased to 52 gpm. At 2.4 million Btu/hr, the break occurred between 20 and 18 gpm and the recirculation ratio was 1.29.

The sharp drop in $U_o$ at the breakpoint in Fig. 5 most likely reflects the initiation of dry patches on the lower tubes. At the breakpoint for each heat duty, Table 4 gives the corresponding critical loading on the bottom row of tubes. These "local" values were readily estimated from appropriate heat

Table 4 Minimum Feed Rates and Critical Loadings for the Sprayed-Bundle Evaporator at Various Heat Duties

| Nominal Heat Duty (million Btu/hr) | Ammonia Evaporation Rate (gpm liquid) | Minimum Ammonia Feed Rate (gpm) | Minimum Recirculation Ratio[a] | Critical Loading on Bottom Row of Tubes (lb/hr·ft) |
|---|---|---|---|---|
| 2.4 | 14.7 | 19 | 1.29 | 18 |
| 3.2 | 19.7 | 25 | 1.27 | 22 |
| 4.0 | 24.7 | 29 | 1.17 | 20 |

[a]Recirculation ratio = mass rate of liquid feed/mass rate of evaporation.

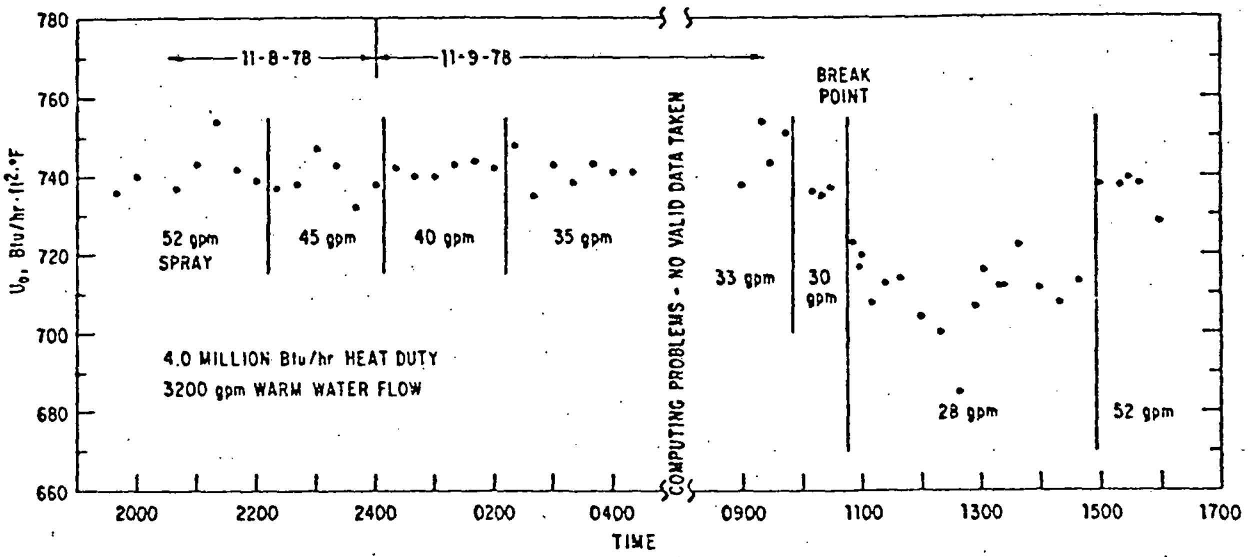


Fig. 5 Determination of minimum ammonia feed rate for the Linde sprayed-bundle evaporator.

balances on the tube bundle. It can be seen that the critical loadings are all in the neighborhood of 20 lb/hr·ft, with no apparent dependence on heat duty.* Since this critical loading is a local limiting value, it should apply to essentially any size spray evaporator having Union Carbide High Flux tubes. To ensure good heat transfer performance, it is necessary to design the feed system such that all tubes receive at least the critical loading of 20 lb/hr·ft.

It should be emphasized that this minimum loading is the limiting value approached from a completely wetted, well-activated state. No tests were conducted to determine the minimum loading when approached from an incompletely wetted condition, but the loading would probably be greater.

*These results agree favorably with the single-tube results of Czikk et al.,[4] who measured the minimum loading on High Flux tubes as a function of vapor crossflow velocity and heat flux. No discernable heat flux dependence was found, and at low vapor velocities the minimum loadings were scattered within a band ranging from 15 lb/hr·ft to 30 lb/hr·ft.

In addition to the foregoing, vapor quality, heat duty, and ammonia inlet subcooling were also studied. In all cases the measured vapor quality was of the order of 99%. The 1% liquid entrainment created sufficient fog to obscure one's view of the tube bundle. The overall heat transfer coefficient was not significantly affected by variations in heat duty from 2.4 million Btu/hr to 4.0 million Btu/hr nor by changes in ammonia inlet temperature in the range of 52°F to 70°F.

Union Carbide Enhanced-Tube Condenser

Under nominal operating conditions, the overall heat transfer coefficient is 820 Btu/hr·ft$^2$·°F and the ammonia-side and water-side heat transfer coefficients are 4860 and 1125 Btu/hr·ft$^2$·°F, respectively. The water-side velocity is 5.0 ft/sec and the pressure drop is 1.8 psi.

The values for the ammonia-side and water-side heat transfer coefficients were obtained using the Wilson Plot method, but with a velocity power of $V^{-0.66}$ conforming to the Noranda correlation for finned tubes (see Fig. 6). The ammonia-side heat transfer coefficient, 4860 Btu/hr·ft$^2$·°F, is about three times the value predicted by the Nusselt expression for condensation on a smooth tube.

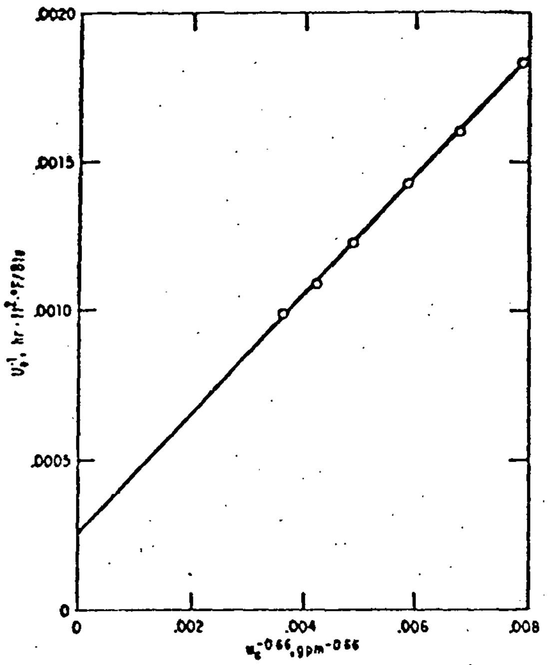


Fig. 6 Wilson plot for the Linde enhanced-tube condenser.

In Fig. 7 it can be seen that the values of the water-side coefficient are correlated well by the velocity power based on the Noranda correlation and that they are very close to those predicted by the Noranda correlation (about 3% higher). For comparison, the Sieder-Tate correlation for smooth unenhanced tubes is also plotted in Fig. 7. In the present range of water flow rates, the $h_i$ values are about 1.50 to 1.75 times larger than those predicted by the Sieder-Tate equation. This enhancement ratio, incidentally, is consistent with the 1.8 ratio of the actual wetted inside surface area to the nominal inside surface area.

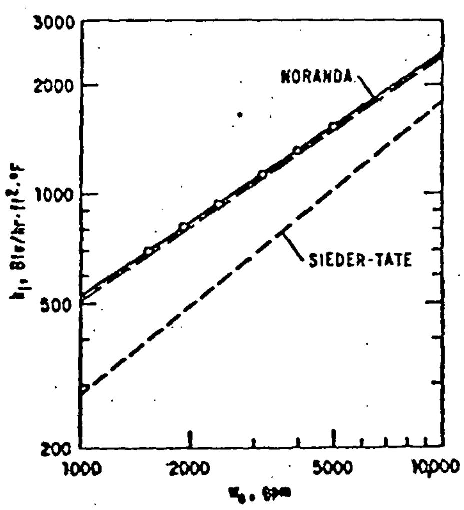


Fig. 7 Variation of water-side coefficient with water flow rate for the Linde condenser.

It was found that the individual coefficients as determined by the Wilson method are very sensitive to the choice of the velocity exponent. For example, using the conventional -0.8 power, rather than the -0.66 power as in the Noranda correlation, yields an ammonia-side coefficient of 2530 Btu/hr $\cdot ft^2 \cdot °F$ and a water-side coefficient of 1410 Btu /hr$\cdot ft^2 \cdot °F$. Consequently, the accuracy of the calculated individual coefficients is strongly dependent on the accuracy of the exponent in the Noranda correlation.

Additional tests were conducted to study the effects of heat duty and noncondensables. The overall heat transfer coefficient was not affected when the heat duty was varied from 2.4 million to 4.0 million Btu/hr. Since the condensate flow rate is proportional to the heat duty, this implies that the ammonia-side heat transfer coefficient is essentially independent of liquid loading within that range. Apparently, small amounts of noncondensable gases were present throughout the testing, as evidenced by a 3% increase in the $U_o$ value following a purge at the conclusion of the test period.

C-MU Vertical Fluted-Tube Evaporator

Under nominal operating conditions (3200 gpm water flow, 3.2 million Btu/hr heat duty, and 100 gpm ammonia feed rate), the overall heat transfer coefficient is 825 Btu/hr$\cdot ft^2 \cdot °F$ and the ammonia-side and water-side heat transfer coefficients are 1730 Btu/hr$\cdot ft^2 \cdot °F$ and 2610 Btu/hr$\cdot ft^2 \cdot °F$, respectively. The water-side heat transfer coefficient of 2610 Btu/hr$\cdot ft^2 \cdot °F$ represents an enhancement ratio of 1.95, which is nearly equal to the area ratio of 2.07. At nominal operating conditions, the water-side pressure drop is 3.2 psi, the recirculation ratio is 5, the ammonia quality exceeds 99%, and the ammonia temperature entering the evaporator is 67°F.

The measured $U_o$ values were found to be stable and repeatable during any continuous run; however, they were not always repeatable from run to run or even following a brief ammonia flow shutdown. This behavior is illustrated in Figs. 8 and 9 at heat duties of 3.2 million Btu/hr and 4.0 million Btu/hr. The expected parametric trend in the $U_o$ versus ammonia feed rate curve is observed (i.e., $U_o$ increases with feed flow), but the $U_o$ values for the runs on December 13 and 14 are 20-60% higher than those of the previous week. (All subsequent $U_o$ values fell within these limits.) Runs at a heat duty of 2.4 million Btu/hr exhibited a similar degree of nonrepeatability from one week to the next. These repeatability problems are attributed to variations in tube wetting and to the poor design of the feed channels by which the liquid ammonia film is applied to the tops of the tubes; for good evaporator performance well-wetted tube surfaces are essential. It is reasonable to expect that the extent of tube wetting is influenced by the initial rate at which ammonia is fed to the tubes, which are completely dry before starting up. This may explain the difference in performance from one week to the next that was observed in Figs. 8 and 9. In the runs on December 5 and 8 the system was started with a feed rate of 60 gpm, while on December 13 and 14 the initial feed rates were 100 and 110 gpm, respectively. It is likely that in the latter case "flooding" of the surfaces resulted in well-wetted tubes and high $U_o$ values. On the other hand, in

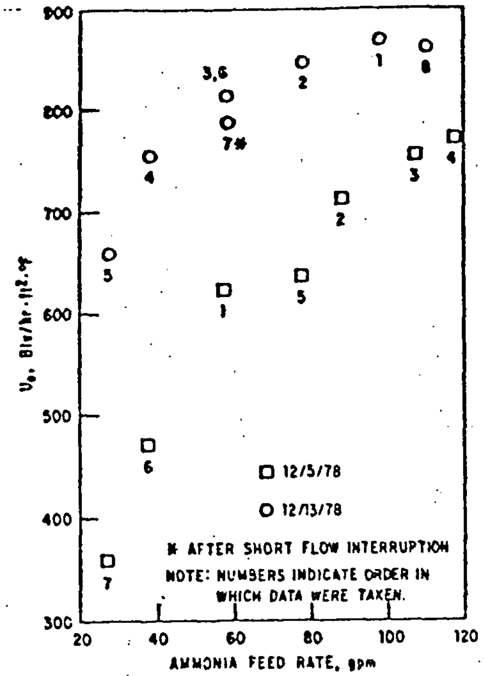


Fig. 8 $U_o$ versus ammonia feed rate at 3.2 million Btu/hr for the C-MU evaporator.

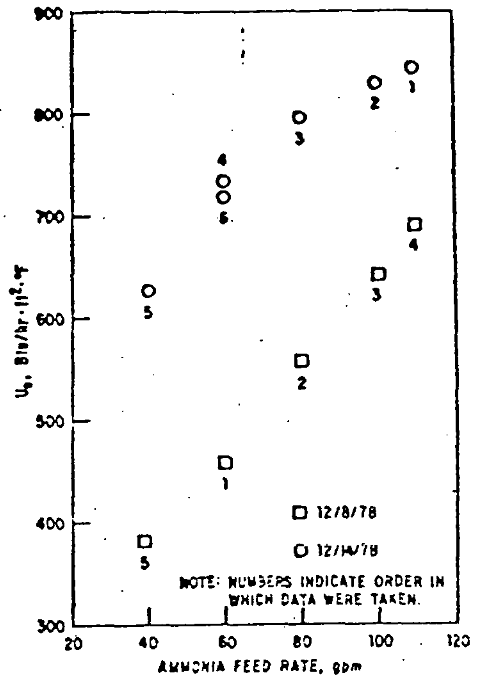


Fig. 9 $U_o$ versus ammonia feed rate at 4.0 million Btu/hr for the C-MU evaporator.

the former case the lower feed rate resulted in the formation of dry patches and lower $U_o$ values. The foregoing suggests that to ensure sufficient wetting and good thermal performance, the system should be started with a high ammonia feed rate and then cut back to the desired operating value.

Heat transfer performance is strongly dependent on proper distribution of liquid ammonia. One measure of applicator performance is the uniformity of the liquid feed around the periphery of each tube. It should be evident that a nonuniform feed can contribute to the problem of tube wetting. Another measure of applicator performance is the degree to which liquid ammonia remains on the tubes. With the present applicator design there was a tendency for the liquid ammonia to run along the underside of the metering plate and down the inside surface of the evaporator shell. Furthermore, there was a tendency for the liquid to spray from the tube surface in the region of application and then fall as "rain" within the heat exchanger. For identical operating conditions, the amount of rain observed varied significantly. During the runs on December 5 and 8 there was considerable rain (and $U_o$ was relatively low); in contrast, during the following week there was only a little rain (and $U_o$ was relatively high). This inconsistent behavior reflects a poorly operating applicator. In other tests, C-MU has found that an applicator section with a sleeve extending down over the top of the fluted section performs much more effectively than the uncovered applicator used in the evaporator tested at Argonne.

As mentioned above, for heat duties of 3.2 million and 4.0 million Btu/hr Figs. 8 and 9 exhibit the expected increase in $U_o$ with higher ammonia feed rates. No maximum is observed, which according to Rothfus[5] is characteristic of performance with an applicator having uncovered truncated flutes. Rothfus has found that with the sleeve configuration the curves exhibit a maximum in the vicinity of the laminar-turbulent transition. This occurs at a film Reynolds number of 1500-2000, which corresponds to an ammonia loading of 32-42 gpm.

Comparing Figs. 8 and 9, it is evident that the $U_o$ values increase with decreasing heat duty. This result was anticipated because at lower heat duties less ammonia is evaporated, so the amount of ammonia liquid on the tubes is greater.

## C-MU Vertical Fluted-Tube Condenser

Under nominal operating conditions, the overall heat transfer coefficient is 1045 Btu/hr·ft$^2$·°F and the ammonia-side and water-side heat transfer coefficients are 3320 Btu/hr·ft$^2$·°F and 2470 Btu/hr·ft$^2$·°F, respectively. The water-side heat transfer coefficient of 2470 Btu/hr·ft$^2$·°F represents an enhancement ratio of 2.5, which is about 20% higher than the area ratio of 2.07. At nominal operating conditions, the water-side pressure drop is 3.3 psi. The overall heat transfer coefficient increased slightly when the heat duty was varied from 2.4 million Btu/hr to 4.0 million Btu/hr.

The pressure of the vapor available to the Carnegie-Mellon condenser can be adjusted to correspond to the exhaust pressure expected from a turbine. The thermodynamic state of the superheated vapor discharged from the expansion valve in the test loop is different from that of a typical turbine exhaust, since the turbine exhaust is expected to be wet and to have micron-sized liquid droplets entrained in the saturated vapor. The mass flow rate of the droplets may be several percent of the total mass flow rate from the turbine.

The Argonne test facility is equipped with a spray type of desuperheater that provides a vapor stream containing liquid ammonia droplets. This desuperheater contains 48 spray nozzles that use high-velocity ammonia vapor jets to atomize small liquid ammonia jets. The liquid ammonia is obtained from the receiver at the bottom of the condenser, and the high-velocity vapor jets are supplied with ammonia vapor at the evaporator pressure. These spray heads are rated to produce 90% of the flow of droplets under 10 microns in diameter.

The effect of vapor quality on the Carnegie-Mellon condenser was measured during a series of runs at the design heat duty of 3.2 million Btu/hr. $U_o$ was first measured while using the superheated ammonia vapor available from the evaporator. Feed valves were then opened to permit a flow of 2.04 gpm of liquid ammonia to the desuperheater spray nozzles; this produced a "wet" ammonia feed (90% quality) to the condenser. Values of $U_o$ for this "wet" condition were determined. Finally, the spray desuperheater was shut off to repeat the superheated condition.

The observed value of $U_o$ with 90% vapor quality was slightly less (about 1%) than with superheated vapor. This decrease was of the same order of magnitude as the scatter in the measured values of $U_o$ at constant operating conditions. We may, therefore, conclude that wet vapor (90% quality) has no significant effect on the condenser performance.

## Concluding Remarks

The experimental data obtained at the Argonne OTEC test facility can be used to predict the performance of other units. Of course, the extension of the results to exchangers of a different size and/or geometry must take into account the range and limitations of the tests, in addition to other considerations. For example, with the Union Carbide flooded-bundle evaporator, the static head penalty must be considered. With the Union Carbide sprayed-bundle evaporator, the ammonia distribution system must be designed to provide the minimum of 20 lb/hr·ft to each tube. In the case of large sprayed bundles having high vapor velocity, the problem of vapor/liquid interaction and entrainment must also be considered.[6] For the Union Carbide and C-MU condensers, the influence of heat duty (and hence liquid loading) was examined over a limited range, and any extrapolation beyond that range must be made via a sound analytical model. Of all the units tested, the C-MU evaporator results are the most difficult to scale up because: 1) the dependence of $U_o$ on ammonia feed rate was not measured beyond 110 gpm; 2) uncertainty exists regarding the ammonia applicator performance; and 3) an analytical model describing the fundamental mechanism of evaporation on fluted tubes is lacking.

ANL is carrying on an active analytical modeling program aimed at establishing the required models and algorithms for scaling the test results. The procedures for scaling will be very useful as the program progresses into tests of the plate-and-fin heat exchangers as well as the PSD-I and PSD-II test articles.


## Acknowledgments

The authors wish to express their appreciation to O. Despe for his contribution to instrumentation, to R. Mandernack for supervision of facility operations, and to C. Lemberg for her editorial effort.


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