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OTEC Performance Tests of the Union Carbide Sprayed-Bundle Evaporator

by

David L. Hillis, James J. Lorenz,
David T. Yung, and Norman F. Sather

Ocean Thermal Energy Conversion
Program

Argonne National Laboratory

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9700 South Cass Avenue
Argonne, Illinois 60439

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UNION CARBIDE SPRAYED-BUNDLE EVAPORATOR

by

David L. Hillis,* James J. Lorenz,†
David T. Yung,† and Norman F. Sather‡

*Engineering Division
†Components Technology Division
‡Energy and Environmental Systems Division

May 1979

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NOMENCLATURE

A_i	Inside surface area of heat exchanger tubes, ft^2	ΔT_{lme}	Log-mean temperature difference in evaporator, $^{\circ}\text{F}$
A_{lm}	Log-mean area of heat exchanger tubes, ft^2	U_o	Overall heat transfer coefficient based on outside tube surface area, $\text{Btu/hr}\cdot\text{ft}^2\cdot^{\circ}\text{F}$
A_o	Outside surface area of heat exchanger tubes, ft^2	w_e	Water flow rate through evaporator, gpm
C_{pe}	Heat capacity of water in evaporator, $\text{Btu/lb}\cdot^{\circ}\text{F}$		
C'_{pe}	Heat capacity of liquid ammonia entering evaporator, $\text{Btu/lb}\cdot^{\circ}\text{F}$		
C_{ph}	Heat capacity of water in heater, $\text{Btu/lb}\cdot^{\circ}\text{F}$		
$\Delta H'_e$	Heat of vaporization of ammonia in evaporator, Btu/lb		
h_{fi}	Fouling coefficient for inside of tubes, $\text{Btu/hr}\cdot\text{ft}^2\cdot^{\circ}\text{F}$		
h_{fo}	Fouling coefficient for outside of tubes, $\text{Btu/hr}\cdot\text{ft}^2\cdot^{\circ}\text{F}$		
h_i	Heat transfer coefficient for inside surface (water), $\text{Btu/hr}\cdot\text{ft}^2\cdot^{\circ}\text{F}$		
h_o	Heat transfer coefficient for outside surface (ammonia), $\text{Btu/hr}\cdot\text{ft}^2\cdot^{\circ}\text{F}$		
k	Conductivity of tube wall material, $\text{Btu/hr}\cdot\text{ft}\cdot^{\circ}\text{F}$		
m_e	Mass flow rate of water through evaporator, lb/min		
m_h	Mass flow rate of water through heater, lb/min		
m'_h	Mass flow rate of ammonia around loop, lb/min		
Δp_e	Water-side pressure drop through evaporator, psi		
Δp_{pw}	Pressure increase across warm water pump, psi		
q_{e1}	Rate of heat transferred from warm water in evaporator calculated from water temperature change, Btu/hr		
q_{e2}	Rate of heat transferred from warm water in evaporator calculated from warm-water loop energy balance, Btu/hr		
q'_e	Rate of heat transferred to ammonia stream during passage through the evaporator, Btu/hr		
r_i	Inside radius of heat exchanger tubes, ft		
r_o	Outside radius of heat exchanger tubes, ft		
Δr	Wall thickness ($r_o - r_i$), ft		
T_a	Ambient temperature, $^{\circ}\text{F}$		
T_{ei}	Temperature of warm water entering evaporator, $^{\circ}\text{F}$		
T_{eo}	Temperature of warm water leaving evaporator, $^{\circ}\text{F}$		
T'_{eo}	Temperature of ammonia vapor leaving evaporator, $^{\circ}\text{F}$		
ΔT_e	Water-side temperature drop through evaporator, $^{\circ}\text{F}$		
$\Delta T'_e$	Temperature increase of liquid ammonia entering evaporator, $^{\circ}\text{F}$		
ΔT_h	Water-side temperature increase through heater, $^{\circ}\text{F}$		

NOMENCLATURE

OTEC PERFORMANCE TESTS OF THE UNION CARBIDE SPRAYED-BUNDLE EVAPORATOR

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ABSTRACT

Results of performance tests on a Union Carbide sprayed-bundle evaporator with High Flux titanium tubes are reported. This unit performed satisfactorily and was free of operational problems. Under nominal operating conditions (3200 gpm water flow rate, 3.2 million Btu/hr heat duty, and 40 gpm ammonia feed rate), the overall heat transfer coefficient (U_o) was found to be 760 Btu/hr·ft²·°F and the ammonia-side and water-side coefficients were 4600 and 1300 Btu/hr·ft²·°F, respectively. The overall water-side pressure drop in the heat exchanger was 4.0 psi, and the vapor quality measured at various heat duties and ammonia feed rates was 99.2 ± 0.25%. Variations in heat duty (from 2.4 million-4.0 million Btu/hr) and ammonia inlet temperature (from 52-72°F) did not affect the value of U_o , nor did changes in ammonia feed rate in the range of 40-100 gpm. However, a minimum ammonia feed rate below which U_o decreased sharply was observed; this corresponded to a local tube loading of 20 lb/hr·ft and was essentially independent of heat duty. Finally, contact between the liquid ammonia and the High Flux surface under nonboiling conditions appeared to deactivate nucleation sites, reducing U_o to <700 Btu/hr·ft²·°F. The deactivated surface could be reactivated by drying out the tubes while maintaining warm water flow.

1 INTRODUCTION

Energy stored in the warm surface waters of the tropical oceans can be converted to electrical energy in a suitably designed power plant. Although Ocean Thermal Energy Conversion (OTEC) plants can utilize the warm ocean water directly as the working fluid, the primary emphasis in the current federal development program is on closed-cycle plants that use ammonia or some other refrigerant in a Rankine power cycle. In such a plant the warm water is pumped through a heat exchanger where thermal energy of the water is transferred to the ammonia, causing it to vaporize. The vaporized ammonia, under pressure, flows from the exchanger through a turbine-generator that converts the pressure energy into electrical energy. The vapor then flows to a second heat exchanger

where the ammonia is condensed at low pressure. Here the heat of condensation is transferred to a cold water stream that is pumped up to the condenser from the ocean depths. Finally, the liquid ammonia is pumped back to the evaporator to complete the cycle.

Since the temperature difference between the warm and cold ocean water is at most 35-40°F, the net energy conversion efficiency of the plant is low ($\approx 3\%$) and the amount of thermal energy that can be obtained from a pound of water is small (corresponding to 2-4°F sensible heat). Consequently, the heat exchangers must handle very large heat duties and water flows, on the order of 100 million Btu/hr and 100 thousand gpm per megawatt of generating capacity. This means that the heat exchangers will be much larger and more costly than those used in conventional coal and nuclear power plants.

However, improvements in the performance of the evaporator and condenser can significantly reduce both capital and operating costs. The performance of the exchangers is characterized by two quantities, the overall heat transfer coefficient and the water-side and ammonia-side pressure drops. The latter are important because the parasitic power required for pumping is proportional to them. Clearly, cost trade-offs between pumping power and heat-transfer performance are possible. For example, increasing the water velocity through the exchanger will improve the heat transfer coefficient and thereby reduce the amortized equipment cost, but at the expense of a greater pumping power cost. Water-side and ammonia-side heat transfer enhancement techniques offer similar trade-off possibilities and, hence, opportunities for maximizing the overall cost-effectiveness of the heat exchangers. The most cost-effective design would take into account costs of biofouling control, pumping, and the amortized cost of materials and fabrication.

The purpose of this report is to present and discuss the results of performance tests on an enhanced sprayed-bundle evaporator designed by Union Carbide Corporation. The tests were run at the OTEC heat exchanger test facility at Argonne. This evaporator is one of a series of five high-performance heat exchangers that have been tested. The other exchangers tested to date are the Union Carbide flooded-bundle evaporator, the Carnegie-Mellon University (C-MU) vertical fluted-tube evaporator and condenser, and the Union Carbide enhanced-tube condenser. Test results on the first of these have been published,¹ and reports on the others are forthcoming. In addition, the results of performance

tests on several shell-less types of heat exchangers will be issued later this year. All of the heat exchangers involved are sized to handle 3.2 million Btu/hr, corresponding to 1/40 MWe capacity. Consequently, through these tests it should be possible to determine whether the performance anticipated on the basis of single-tube experiments can be achieved in large tube-bundle configurations. Such information is necessary before full-size commercial OTEC plants can be designed.

2 SPRAYED-BUNDLE EVAPORATOR

The sprayed-bundle evaporator used in the testing program reported here was supplied by the Linde Division of Union Carbide Corporation. The evaporator, a horizontal shell-and-tube exchanger, has titanium tubes with the Linde High Flux coating on the shell side (i.e., the ammonia side). This surface coating is a thin layer of porous metal that promotes nucleate boiling. Ammonia is introduced into the shell side through a row of 19 spray-header tubes arrayed across the top of the tube bundle, as illustrated in Fig. 1. (An additional 62 spray tubes are included within the tube bundle; these, however, were plugged and not used during the tests described in this report.) The water side has four passes with the warm water feed at the bottom. The tube arrangement is triangular with a pitch of 1.875 in.; the number of functional tubes in each pass is shown in Fig. 1. The sprayed-bundle evaporator is shown in Fig. 2, and Table 1 gives the nominal design/operating parameters for the exchanger.

3 HEAT EXCHANGER TEST FACILITY

3.1 PIPING LAYOUT

A piping schematic of the OTEC heat exchanger test facility at Argonne is shown in Fig. 3. The facility consists of a warm water loop, a test evaporator, an ammonia loop, a test condenser, and a cold water loop.

The nominal heat duty of the warm water loop is 3.2 million Btu/hr; however, the heater and steam supply line are sized to handle up to 6 million Btu/hr. The heater can be set for a constant heat duty over the range of 0.5 million-6 million Btu/hr. The warm water flow rate is a nominal 3200 gpm, but

*1.5" O.D. TI TUBE EXTERNALLY COATED WITH UCC/LINDE'S HIGH FLUX SURFACE

Fig. 1. Diagram of Union Carbide Sprayed-Bundle Evaporator

Fig. 2. Union Carbide Sprayed-Bundle Evaporator

Table 1. Nominal Design Parameters for the Sprayed-Bundle Evaporator

Parameter	Value
Heat duty	3.2 million Btu/hr
Ammonia inlet temperature	72°F
Water inlet temperature	80°F
Water outlet temperature	78°F
Number of tubes	388
High Flux coated tube length	55 in.
Tube outside diameter	1.50 in.
Tube inside diameter	1.43 in.
Effective outside heat transfer area ^a	698 ft ²
Mean water velocity in tubes	6.6 ft/sec
Water-side pressure drop	2.1 psi ^b
Shell inside diameter	71 in.
Shell-side design pressure	215 psig
Tube-side design pressure	100 psig
"Clean" overall heat transfer coefficient ^c	740 Btu/hr·ft ² ·°F

^aBased on actual coated tube length of 55 in.; the total tube length between tube sheets is 6 in. longer.

^bDoes not include tube and nozzle inlet and exit losses.

^cBased on outside tube surface area; assumes no tube-side or shell-side fouling.

Fig. 3. OTEC Heat Exchanger Test Facility

the pipe and pump are sized for flows up to 8000 gpm. This large range of flows improves the accuracy of calculation of the individual water-side and ammonia-side heat transfer coefficients. An important feature of the flow loop is a bypass around the warm water heater. Typically, about 10% of the water flows through the heater and 90% flows through the bypass line. The temperature of the water passing through the heater is raised by about 20°F; when the heated water is mixed with the water from the bypass line, the desired water temperature is achieved. This arrangement greatly improves the accuracy of measurement of the heat input from the steam heater. Static mixers are used to eliminate radial temperature variations at locations where measurements of bulk mean temperatures are needed.

The ammonia loop is provided with a demister that removes entrained liquid droplets from the vapor stream. The demister contains a 6-in.-thick pad woven from polypropylene filaments. The ammonia droplets are caught on the pad and returned to the evaporator as a liquid stream from the bottom of the demister vessel. The manufacturer's rating indicates that all but sub-micron-sized particles are caught at the vapor velocities of the tests. A turbine flowmeter rated at 0.5-5 gpm is installed in the liquid line from the demister to measure the flow rate of the returned liquid. For very high quality vapor (>99%) an alternate means of measurement must be used. This consists of a shutoff valve located in the vertical return pipe from the demister and a differential pressure cell to measure the rate of accumulation of liquid in the line. The pressure cell is capable of detecting the accumulation of less than 1 in.³ of liquid in the running time of several hours. The ammonia loop also contains an expansion valve for adjusting the pressure of the vapor supplied to the test condenser. The ammonia liquid from the condenser is returned to the evaporator by a pump that can handle flows corresponding to heat duties up to 6 million Btu/hr.

The cold water loop contains a chiller capable of removing up to 6.4 million Btu/hr from the recirculating cold water flow. Condenser inlet water temperatures as low as 40°F are possible over most of the range of heat duties. However, at duties above 5 million Btu/hr, a 40°F cold water temperature cannot be attained because the required refrigerant temperature is below the capability of the chiller. The nominal cold water flow rate is 3200 gpm, but the pump and pipe are sized to handle as much as 8000 gpm if necessary. The chiller is rated

at 4000 gpm maximum; the bypass around the chiller can be used to obtain the higher flows needed for determining the film heat transfer coefficients. Static mixers are also used in the cold water loop to facilitate measurement of bulk mean temperatures.

3.2 INSTRUMENTATION

Pressures, temperatures, and flow rates are measured in each of the three test facility loops. The instruments used were chosen on the basis of measurement accuracy and compatibility with the computer data handling system. The original accuracy goals for measurements characterizing overall performance of the heat exchangers were $\pm 3\%$ for the overall heat transfer coefficient and $\pm 5\%$ for the water-side pressure drop. In all instances these goals were met or surpassed during testing. Table 2 lists the more important instruments used and the accuracy of measurement obtained routinely during test operations.

Table 2. Instruments and Measurement Accuracy

Quantity measured	Instrument	Accuracy of measurement
Temperature of:		
Water into and out of evaporator Ammonia vapor in evaporator and after expansion valve	Quartz crystal thermometer	$\pm 0.010^{\circ}\text{F}$
Water into and out of heater Liquid ammonia feed to evaporator	Copper-constantan thermocouple	$\pm 0.1^{\circ}\text{F}$
Pressure of:		
Ammonia vapor in evaporator and after expansion valve	Quartz crystal transducer	± 0.005 psia
Pressure difference of:		
Water into and out of evaporator Water into and out of pump	Strain gauge transducer	$\pm 0.5\%$ of reading
Flow rate of:		
Water through evaporator Water through heater	Turbine meter	$\pm 0.15\%$ of reading
Liquid ammonia feed to evaporator Liquid ammonia from demister	Turbine meter	$\pm 0.5\%$ of reading

3.3 DATA HANDLING SYSTEM

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. Figure 4 shows a typical hard-copy output of the data calculated for a particular test run. Most of the entries in Fig. 4 are self-explanatory; those that are not will be discussed in Sec. 5.

```

11/10/1978
11:43
LINDE SPRAY EVAP
RUN #      156

WATER TEMP HEATER IN=    77.809DEG F
WATER TEMP HEATER OUT=   107.600DEG F

WATER TEMP EVAP IN=     79.933DEG F
WATER TEMP EVAP OUT=     77.995DEG F

NH3 TEMP EVAP IN=       71.700DEG F
NH3 TEMP EVAP OUT=      72.867DEG F
NH3 TEMP AFTER R VALVE   55.103DEG F

HEATER WATER FLOW=      160.387GPM
EVAP WATER FLOW=       3203.570GPM

NH3 LOOP FLOW=          2.605GPM
NH3 EVAP FEED FLOW=     40.386GPM

NH3 DEMISTER FLOW=      0.000GPM
H3 SECONDARY FLOW=      40.386GPM

WATER TEMP CHANGE IN HEATER= 29.791DEG F
WATER TEMP CHANGE IN EVAP, CALCD, MEASD -1.942DEG F -1.944DEG F

QUARTZ P AFTER R VALVE=  85.942PSIA
QUARTZ T AFTER R VALVE=  55.103DEG F

LOGMEAN TEMP DIFF=      6.053DEG F
HEAT INPUT TO HEATER= 0.2330E 7BTU/HR

NET HEAT IN TO EVAP= 0.2802E 7BTU/HR
HEAT IN DUE TO PUMP= 0.4512E 6BTU/HR

HEAT INTO EVAP BASED ON H2O DELTA T= 0.3116E 7BTU/HR
HEAT INTO EVAP BASED ON NH3 EVAP RATE= 0.4017E 6BTU/HR
HT XCHGR AREA= 698.000FT2

HT XFER COEF BASED ON H2O DELTA T= 0.7374E 3BTU/HR/FT2/DEGF
HT XFER COEF BASED ON HT BALANCE= 0.6631E 3BTU/HR/FT2/DEGF

WATER PRESSURE CHANGE IN EVAP= 4.143PSI

WARM WATER PUMP DELTA P= 94.256PSI
H3 EVAP PRES= 135.074PSIA

NH3 LOOP MASS FLOW RATE= 13.202LBS/MIN

```

Fig. 4. Sample Data Printout

4 OPERATING PROCEDURE

The Union Carbide sprayed-bundle evaporator was installed in the test loop replacing the flooded-bundle evaporator which had been tested earlier. It was mounted on the test stand in the condition in which it had been received, without cleaning, since the heat transfer surfaces had been maintained in a protected, clean condition during shipment. Cleanliness was also maintained during installation. After all piping and instrumentation connections were made, the system was pressure-checked for leaks.

Prior to the initial start-up, all piping, valves, and vessels of the ammonia system (excluding the test heat exchangers) were cleaned with hot caustic to remove grease. Acid washing (pickling) was then performed to remove rust, mill scale, and particles of iron left from the welding performed during fabrication. The surfaces were then passivated, flushed with water, and put under a nitrogen blanket. This process was not repeated before testing the sprayed-bundle evaporator, but the ammonia system was evacuated at this time to remove noncondensables and water from the shell side of the evaporator, from the ammonia side of the condenser, and from the ammonia piping. The warm water loop had also been cleaned and passivated by the same procedure used for the ammonia system. The loop was then filled with potable water that had first been through two water-softening treatments. A corrosion and fouling inhibitor (Nalco 8330) was added to the water in the approximate ratio of 3 oz per gal.,

The first step in the start-up procedure was to fill the condenser and evaporator receivers one-half to three-quarters full of ammonia. Then the evaporator ammonia head was filled until ammonia started to trickle through the openings of the spray tubes (this could be observed through a viewing port). The ammonia pumps were turned on and the flow recirculated through the receivers. Next, the cold water pump, the chiller, and the warm water pump were started; water flows were initially adjusted to the nominal rate of 3200 gpm, and the chiller temperature was set at the desired value. The ammonia flow to the evaporator was then started and the expansion valve adjusted. Finally, the steam valve was opened. Following start-up, a steady-state condition was usually reached in two to four hours, after which data were taken for two hours or more before changing operating parameters. Generally, hard-copy data were obtained every 10 to 20 minutes, with intermediate points often read and plotted

but not printed. Under ideal conditions, the facility is started up on Monday morning and run continuously until Friday afternoon, giving a total of about 100 hours of operation. In this series of tests (conducted between 10/19/78 and 11/17/78), the durations of the data-taking runs were 30, 18, 80, and 64 hours. After completion of an operating period, the facility was shut down and the ammonia transferred to storage tanks located outside the building. For reasons of safety, ammonia was not left in the facility while it was unattended.

When the sprayed-bundle evaporator was first filled with ammonia in August of 1978, it was discovered that some of the liquid was entering the shell side through the lower 62 spray-feed tubes. Although Union Carbide had requested that these be plugged at the factory, inadvertently this was not done. Since this condition made normal operation impossible, neoprene plugs were installed at Argonne. However, these proved unsatisfactory because of a tendency to deteriorate and shrink, so they were replaced by teflon plugs. There were no further problems associated with the optional spray-feed tubes.

5 METHODS OF CALCULATION

5.1 EQUATIONS FOR THE OVERALL HEAT TRANSFER COEFFICIENT

It is evident from the foregoing discussion of the measurements made during testing that there are three independent ways of determining the heat duty of the evaporator. These are:

1. The rate at which heat is given up by the warm water as it passes through the evaporator. This can be expressed as follows:*

$$q_{e1} = 60m_e C_{pe} \Delta T_e + 1.484 w_e \Delta p_e \quad (1)$$

where the first term is the sensible heat lost by the water as it flows through the evaporator, and the second is the frictional heat added. As might be expected, the frictional heat addition is quite small, amounting to less than 0.5% of q_{e1} for all runs with the sprayed-bundle evaporator.

2. The rate at which heat is added to the warm water as it flows around the loop from the evaporator discharge noz-

*See list of nomenclature for explanation of symbols.

zle back to the evaporator inlet. This includes the net effect of the heat added by the steam heater and the pump, and lost from the pipe surfaces by convection. Thus:

$$q_{e2} = 60m_h C_{ph} \Delta T_h + 113.4 w_e^{1/2} \Delta p_{pw} - 1500(T_{ei} - T_a) \quad (2)$$

The second term on the right side of Eq. 2 represents the rate of energy addition to the water by the pump. This term was obtained by making an empirical curve fit to the manufacturer's efficiency data for this pump for flows between 2400 and 4000 gpm.

3. The rate at which heat is added to the ammonia as it passes through the evaporator. This includes both the sensible heat gain of the subcooled liquid as it is warmed to the saturation temperature and the latent heat of vaporization added as the ammonia boils. In equation form, this is:

$$q'_e = 60m'_l (\Delta H'_e + C'_{pe} \Delta T'_e) \quad (3)$$

During steady-state operation these three values of the evaporator heat duty (q_{e1} , q_{e2} , and q'_e) must be equal; consequently, the accuracy of the measurements can be checked by comparing the values obtained from the experimental data. However, it must be emphasized that the three methods are not equally accurate. Therefore, it cannot be assumed that the difference between the results obtained by any two methods represents the error in either method.

Once a value of the evaporator heat duty, q_e , is obtained, the overall heat transfer coefficient, U_o , is calculated from the conventional definition:

$$U_o = \frac{q_e}{A_o \Delta T_{lme}} \quad (4)$$

where:

A_o = the outside surface area of the tubes, and

ΔT_{lme} = the log-mean temperature difference between the water and ammonia streams.

ΔT_{lme} can be expressed in equation form as follows:

$$\Delta T_{lme} = \frac{\Delta T_e}{\ln \left[\frac{T_{ei} - T'_{eo}}{T_{eo} - T'_{eo}} \right]} \quad (5)$$

5.2 ERROR ANALYSIS

Before the test data on the sprayed-bundle evaporator and the resulting heat transfer coefficients calculated from these equations are presented, a brief discussion of the magnitude of the expected uncertainties in q_e and U_o that result from inaccuracies in the directly measured quantities is perhaps in order. We can determine these uncertainties by analyzing the propagation of the potential measurement errors (discussed in Sec. 3.2) in calculations with Eqs. 1 to 5 above.

The first method of calculating the heat duty (Eq. 1) involves the ΔT_e measurement (with a measurement accuracy of $\pm 0.3\%$) and the flow rate measurement ($\pm 0.15\%$ accuracy); the expected accuracy in q_{e1} is therefore $\pm 0.4\%$ (note that the error contribution of the frictional term is negligible). This is in agreement with the observed scatter in the values of q_{e1} calculated from successive measurements during steady operation at fixed run conditions; the data generally fell well within a band that was 1% wide.

The uncertainty in U_o is determined by examining the following equation, which results from combining Eqs. 1, 4, and 5 and neglecting the frictional heat addition:

$$U_o \approx \frac{60m_e C_{pe}}{A_o} \ln \left[\frac{T_{ei} - T'_{eo}}{T_{eo} - T'_{eo}} \right] \quad (6)$$

The uncertainty in U_o is due in part to inaccuracies in measurements of T_{ei} and T_{eo} obtained with quartz crystal thermometers. These inaccuracies arise from the inherent precision limits of the thermometers and from small errors introduced during calibration. The standard calibration procedure is to place both measuring probes in a slush of ice made from distilled water. The readings are allowed to stabilize, and each probe is adjusted to read 32°F . Further improvement in the accuracy of the measurement of $T_{ei} - T_{eo}$ can be made by adjusting the instrument at the operating temperature of 80°F . After the standard calibration was completed, simultaneous readings of the warm water temperature at the same point in the loop were made with the two probes. Both were then adjusted to indicate the average of the original readings, and ΔT was adjusted to read zero. Values for T_{ei} , T_{eo} , and ΔT_e are given in Fig. 4; "water temp evap in" is T_{ei} , "water temp evap out" is T_{eo} , and "water temp change in evap" is ΔT_e . It can be noted that $T_{ei} - T_{eo}$ differs from ΔT_e by 0.002°F . This is

typical of the discrepancy in the two readings of the difference between T_{ei} and T_{eo} . Hence, we will use $\pm 0.010^\circ\text{F}$ as the maximum measurement error in readings of ΔT_e , T_{ei} , T_{eo} , and T'_{eo} .

By applying this measurement error to Eq. 6, the uncertainty in U_o can now be calculated. A 0.010°F error in T'_{eo} causes a relatively small ($\pm 0.145\%$) error in the natural logarithm factor. However, errors of 0.010°F in T_{ei} and T_{eo} cause errors in this factor of $\pm 0.435\%$ and $\pm 0.580\%$, respectively, at our operating conditions. Hence, the estimated uncertainty in U_o is the combined effect of the flowmeter uncertainty ($\pm 0.15\%$) and the three sources of uncertainty because of possible errors in T_{ei} , T_{eo} , and T'_{eo} . Since these error components are independent of one another, the total probable error is $\pm(0.15^2 + 0.435^2 + 0.580^2 + 0.145^2)^{\frac{1}{2}} = \pm 0.75\%$.

The second method of calculating the heat duty (q_{e2}) is subject to a different set of errors. The first term in Eq. 2 is the rate at which heat is added to the warm water in the heater. The flow through the heater, m_h , is measured by a turbine flowmeter to $\pm 0.15\%$ of the actual flow. The value of ΔT_h is measured to $\pm 0.2^\circ\text{F}$ in about 20°F , or to $\pm 1\%$. Thus, the total uncertainty in q_{e2} from this term is $\pm 1\%$. The second term in Eq. 2 was derived from pump efficiency data supplied by the pump manufacturer. It is a reasonable fit to the data up to 4000 gpm and contributes an uncertainty to q_{e2} of $\pm 1.3\%$. The third term in the equation represents the ambient heat loss from the piping, tanks, and other components of the warm water loop. It is less than 1% of the total heat, so the rough estimate used for T_a produces an error in q_{e2} from this term of only $\pm 0.2\%$. Hence, the total estimated uncertainty in q_{e2} is about 2.5%.

It should be noted that the second term in Eq. 2, which accounts for the frictional conversion of mechanical energy to heat, carries the assumption that the flow through the pump is the same as that through the evaporator. In fact, this is not the case because of a small bypass flow around the pump through a water filter (not shown in Fig. 3). The effect of this additional pump flow, which was neglected in the calculations of q_{e2} , was found by direct measurement of the water temperature increase across the pump to amount to about 200,000 Btu/hr at 3200 gpm (evaporator flow), or about 6% of q_{e2} . Spot checks of several runs comparing q_{e2} corrected for this additional heat input with q_{e1} showed agreement in all cases to within $\pm 20,000$ Btu/hr, well within the estimated combined uncertainties of q_{e1} and q_{e2} .

The third method for computing the heat duty is based on the rate at which energy is gained by the ammonia passing through the evaporator. As indicated above, q'_e is the sum of latent and sensible heat contributions. Although measured by a turbine meter that is accurately calibrated ($\pm 0.5\%$), the ammonia flow (m'_g) is rather unsteady for the following reason. The flow is controlled by a valve operated by Δp cells that indicate the liquid levels in the evaporator and condenser sumps. The valve controller is set up to keep the liquid in the receivers at a constant level so as to ensure a positive supply of liquid ammonia to the pump inlets at all times. Hence, although the instruments hold the liquid in the sumps at an almost constant level, the ammonia flow varies widely between the maximum and minimum. Attempts to improve this did not result in a steady flow. Manual control of the flow valve steadied the flowmeter reading, but not at the correct value, since invariably the liquid levels drifted higher or lower over time. Furthermore, the flowmeter apparently was not functioning properly during tests of the sprayed-bundle evaporator, and recorded flow rates were consistently below the calculated average value based on heat duty. For these reasons, no quantitative use can be made of the values of q'_e other than as a rough check on q_{e1} and q_{e2} . No calculations of U_o based on q'_e were made.

Of the three possible methods described above, the first is clearly the most accurate and is used exclusively in calculations reported here.

6 RESULTS²

6.1 U_o AT NOMINAL CONDITIONS

The first performance tests of the sprayed-bundle evaporator commenced on October 19, 1978 with a 30-hour run at nominal conditions (a water flow rate of 3200 gpm, a heat duty of 3.2 million Btu/hr, and an ammonia feed rate of 40 gpm). After some 20 hours of operation, U_o leveled out at about 790 Btu/hr·ft²·°F. Similar results were obtained in an 18-hour run on October 24. The next extended operating period was an 80-hour run beginning on November 7. For the entire week of operation, U_o at nominal conditions was consistently near 730 Btu/hr·ft²·°F ($\pm 1\%$). As shown in Fig. 5, during the 64-hour run of November 15 to 17, U_o recovered to a very consistent 760 Btu/hr·ft²·°F ($\pm 1\%$). The initial value of 790 Btu/hr·ft²·°F was never again attained, although the November 15 to 17 value of 760 Btu/hr·ft²·°F came very close. There are several possible explanations for this variance:

1. Aging -- the earlier value was attained with a new, as-received surface; the later value was achieved with a well-conditioned surface.
2. Activation/deactivation state of the surface -- refer to the discussion of ammonia-side wetting in Sec. 6.2.
3. Fouling -- a fouling resistance of only $0.00004 \text{ hr} \cdot \text{ft}^2 \cdot ^\circ\text{F} / \text{Btu}$ would account for the difference, but we consider fouling to be unlikely for reasons given in Sec. 6.3.
4. Contamination of ammonia by water -- post-test analysis indicated a relatively high water content of 1500 ppm; however, no analyses were conducted during testing.

For design purposes, we consider a U_0 value of $760 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$ to be the most representative of spray evaporator performance at nominal conditions. Although midway between the previously observed values of $730 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$ and $790 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$, it is the figure consistently achieved during the final week of testing. It was at this time that the tube surfaces had most likely reached a stable condition and the operators were most familiar with the characteristics of the evaporator and the test loop.

6.2 DEACTIVATION/REACTIVATION

Previous tests of the Union Carbide flooded-bundle evaporator had shown that conditioning of the High Flux tube surfaces can have a significant effect on performance.³ For example, contact between nonboiling liquid ammonia and the High Flux flooded-bundle tubes for a period of several hours caused a reduction in U_0 of as much as 25%. Apparently this occurred because nucleate boiling sites were deactivated by partial flooding of the surface pores with liquid ammonia. This deactivation was reversed by continuous operation of the flooded-bundle evaporator; a period of about 100 hours was required for full recovery of performance as measured by the value of U_0 .

If deactivation were to occur in the case of a sprayed-bundle evaporator, it should be possible to reactivate the surface by operating the exchanger in a dry condition, i.e., with no ammonia feed. Absorption of heat from the warm water flowing through the tubes would facilitate rapid evaporation of the liquid ammonia occluded in the surface pores. The evaporation rate would likely be increased by opening the valve in the ammonia vapor line to the condenser and by cooling the condenser with cold ocean water.

A series of runs were made with the sprayed-bundle evaporator to test these expectations. With the evaporator operating at nominal conditions, the warm water flow was stopped in order to deactivate the surface. This was followed by a dryout period with the warm water flow on and the ammonia feed off for the purpose of reactivation.

The first such test took place on October 24. With the evaporator running at a steady-state U_0 value of $785 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$, the warm water was shut off for 70 minutes while the spray continued. When the warm water was turned back on, the measured heat transfer coefficient had dropped to $660 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$; after two hours of further operation, it recovered only to about $665 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$. This was followed by a dryout period of some 70 minutes, during which the ammonia feed was shut off and the water flow was on. When the ammonia feed was reinstated, the value of U_0 was $830 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ at first, but this dropped to about $815 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ after two hours of operation.

The next dryout test was carried out on November 2. During start-up the tube bundle was accidentally wetted before the warm water was turned on, resulting in U_0 values of about $665 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ after four hours of operation. Since this was considerably below the normal value, a 30-minute dryout was undertaken. An hour after the ammonia feed was restarted, U_0 reached $740 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$.

On November 7, at the beginning of the first extended run, accidental wetting of the tubes again occurred, so a 75-minute dryout was conducted. The recovered U_0 value was $730 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$, which is consistent with values for long-term operation obtained throughout that week.

For the last extended run, which started on November 15 at 1200 hours, accidental wetting of the tubes was avoided, and a steady U_0 value of $760 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ was attained. Although this was higher than previous results obtained during November, a dryout was again performed in an effort to raise U_0 to the $785 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ level reached during the October runs. When normal operations were resumed, however, U_0 had actually decreased to $725 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$. Following this, a series of tests were conducted at various water-side flow rates as required for estimating the film coefficients by the Wilson method (see Sec. 6.3). By the end of the test period, U_0 at nominal conditions had recovered to the $760 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ value that was representative of data taken earlier that week.

Figure 5 shows the final 22 hours of the extended run that began on November 15. At 0200 hours on November 17, the last dryout was conducted. When the spray was reestablished at 1100 hours, U_0 dropped to an average value of 705 Btu/hr·ft²·°F, and there was no evidence of recovery during the remaining four hours of the run.

From the foregoing, it is evident that the past history of the High Flux surface can have a marked effect on heat transfer performance for evaporators of both the sprayed-bundle and flooded-bundle types. The behavior of the flooded-bundle and sprayed-bundle evaporators is similar in a number of ways. For both evaporators flooding of the High Flux surface pores with liquid ammonia under nonboiling conditions appears to deactivate nucleation sites, reducing the heat transfer coefficient to < 700 Btu/hr·ft²·°F. In both cases, a deactivated High Flux surface can generally be reactivated by evaporation of occluded liquid ammonia from the pores. This can be accomplished by operating the flooded-bundle evaporator under boiling conditions and by drying the surface of the sprayed-bundle evaporator while maintaining warm water flow. However, a unique result of the testing of the sprayed-bundle evaporator is that the heat transfer coefficient can actually be decreased by a dryout* if it is done when the surface is already fully activated. This decrease in U_0 is most likely attributable to incomplete wetting of the tubes following the dryout. Whereas the tubes within the flooded-bundle evaporator are totally immersed in liquid ammonia, the tubes within the sprayed-bundle evaporator are wetted by a thin falling film of ammonia. Rewetting a dry heated surface by a thin liquid film is known to be extremely difficult. The surface may become covered with dry patches that can be rewetted 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. Following the dryout shown in Fig. 5, the ammonia spray was reestablished at the previous value of 40 gpm. But, as evidenced by the low heat transfer coefficient of 705 Btu/hr·ft²·°F, for this particular spray system this flow appears to be insufficient to completely rewet the tubes. Time limitations on the Argonne test schedule have precluded a comprehensive parametric study to determine whether there is a

*This does not imply a progressive decrease in the heat transfer coefficient with time.

Fig. 5. Final 22 Hours of Extended Run Commencing on November 15

critical ammonia flow rate that would ensure complete wetting after the surface has been dried out. A better-designed spray system could likely eliminate this problem of nonuniform wetting.

6.3 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. For circular-tube heat exchangers this relationship is:

$$\frac{1}{U_o} = \frac{A_o}{h_i A_i} + \frac{1}{h_o} + \frac{\Delta r A_o}{k A_{lm}} + \frac{1}{h_{fo}} + \frac{A_o}{h_{fi} A_i} \quad (7)$$

For circular tubes the water-side coefficient, h_i , is generally found to be proportional to the 0.8 power of the water flow rate, so U_o^{-1} is linearly proportional to $w_e^{-0.8}$. Thus, 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 $w_e^{-0.8}$ should fall on a straight line. Extrapolation of the line to the intercept at $w_e^{-0.8} = 0$ 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. 7. 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_e .

Clearly, the use of this procedure for determining h_o and h_i (w_e) requires that the data on U_o (w_e) be obtained under conditions where 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 evaporator heat duty, q_e , was held constant at the nominal design value of 3.2 million Btu/hr. Although the inlet warm water temperature was held constant, 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 the same for the different flow rates. However, since the nucleate boiling coefficient for the High Flux surface is only very weakly dependent on the water-ammonia temperature difference,⁴ the value of h_o can

safely be regarded as constant during the runs for the Wilson determination of h_o .

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 test facility's ammonia system (see Sec. 4), it is unlikely that foreign materials from this system could foul the High Flux surface. As far as water-side fouling is concerned, the water system has been maintained under a nitrogen blanket since the initial filling. The water is continuously filtered during operation by a high-capacity filter located in a bypass line around the warm water pump. The extreme cleanliness of the system and the effectiveness of the corrosion inhibitor in scavenging oxygen and in preventing the deposit of hard calcium lead us to expect that the magnitude of the water-side fouling resistance is essentially zero. Thus, it can be assumed that the last two terms in Eq. 7 are small enough to be neglected in determining h_o and h_i by the Wilson procedure.

On November 15 and 16 a series of runs were conducted at five different water-side flow rates ranging from 1500-5000 gpm; during these runs the heat duty was fixed at 3.2 million Btu/hr and the ammonia feed rate at 40 gpm. Results are summarized in Table 3, and a plot of U_o^{-1} versus $w_e^{-0.8}$ is given in Fig. 6. A least-squares algorithm was employed to fit a straight line through the data in Fig. 6. The value of U_o^{-1} at the intercept $w_e^{-0.8} = 0$ was calculated to be $0.000531 \text{ hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}$. The wall resistance term on the right side of Eq. 7 was determined using the following parameters for the spray evaporator:

$$\begin{aligned} r_o &= 0.06250 \text{ ft} \\ r_i &= 0.05958 \text{ ft} \\ k &= 9.5 \text{ Btu/hr} \cdot \text{ft} \cdot ^\circ\text{F}^* \end{aligned}$$

From this we obtain:

$$\frac{\Delta r A_o}{k A_{lm}} = 0.000313 \text{ hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}$$

Substituting these values into Eq. 7, h_o is calculated to be $4600 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$. This can be compared to the value of $4800 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$ measured for the flooded-bundle evaporator. Although the difference in the h_o values is very small, the

*This value was obtained from the supplier of the tubes to Union Carbide.

Table 3. Values of U_o as a Function of Flow Rate

w_e (gpm)	U_o (Btu/hr·ft ² ·°F)
1504	500
2384	637
3177	760
4012	835
5004	905

Fig. 6. Wilson Plot

gpm. On November 8 and 9, the heat duty was raised to 4.0 million Btu/hr and the feed rate was varied from 52-28 gpm (see Fig. 8). Testing at 2.4 million Btu/hr was conducted on November 9 and 10 with feed rates from 32-18 gpm.

Results are summarized in Table 5. During the runs at 3.2 million Btu/hr, the "break" occurred between 26 and 24 gpm, implying a minimum recirculation

slightly larger value may indicate that wetting of the High Flux tubes is more complete in the flooded bundle than in the sprayed bundle.

With the foregoing value of h_o and the values of U_o in Table 3, h_i can readily be calculated for each flow rate. Results are summarized in Table 4 and plotted in Fig. 7. It can be seen that the h_i values are well correlated by the assumed empirical form, $h_i \propto w_e^{0.8}$. The data lie 5% and 9%, respectively, above the widely accepted Sieder-Tate and Dittus-Boelter correlations.

6.4 EFFECT OF AMMONIA FEED RATE

It is expected that a minimum ammonia feed rate exists below which the lower tubes begin to dry out and hence the overall heat transfer coefficient drops off. A series of tests were conducted to determine this minimum feed rate. These tests all took place when the High Flux surface was fully activated. On October 19 and 20 and again on November 7 and 8, tests were run at 3.2 million Btu/hr and at feed rates varying from 40 down to 24

Table 4. Dependence of h_i on Flow Rate

w_e (gpm)	h_i (Btu/hr·ft ² ·°F)
1504	710
2384	1030
3177	1300
4012	1560
5004	1860

Fig. 7. Variation of Water-Side Coefficient with Water Flow Rate

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. 8. The corresponding recirculation ratio was 1.17. Also in Fig. 8, 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. 8 most likely reflects the initiation of dry patches on the lower tubes. Table 5 gives the critical loading on the bottom row of tubes corresponding to the breakpoint for each heat duty. These "local" values were readily estimated from appropriate heat 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

*Recirculation ratio = mass rate of liquid feed/mass rate of evaporation.

**These results agree favorably with the single-tube results of Czikk et al.,⁵ who measured the minimum loading on High Flux tubes as a function of vapor crossflow velocity and heat flux. No discernible 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.

Fig. 8. Determination of Minimum Ammonia Feed Rate

Table 5. Minimum Feed Rates and Critical Loadings 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

^aRecirculation ratio = mass rate of liquid feed/mass rate of evaporation.

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, fully activated state. No tests were performed to determine the minimum loading when approached from an incompletely wetted condition. However, based on the discussion of rewetting in Sec. 6.2, the loading would probably be greater.

On November 16 and 17, a test was conducted to examine the effect of higher feed rates on heat transfer performance. This test essentially simulated the operating conditions of the upper portion of a large spray evaporator. As shown in Fig. 5, the U_o value was insensitive to ammonia feed rates in the range of 40-100 gpm. This behavior was expected because flooding has no effect on the performance of a High Flux surface that is completely wetted and fully activated.

6.5 EFFECT OF HEAT DUTY

No effect on U_o was observed as a result of varying the heat duty. During the period from November 7 to 10, tests were run at heat duties of 3.2 million, 4.0 million, and 2.4 million Btu/hr (in that order). For nominal feed rates (recirculation ratio of 2.0), U_o values of 730-740 Btu/hr·ft²·°F were consistently obtained. The higher values tended to occur at the end of the test week; this was probably related to conditioning of the tube surfaces, rather than to changes in heat duty.

6.6 EFFECT OF AMMONIA FEED SUBCOOLING

At nominal conditions, ammonia returning from the condenser at 48°F mixes with an equal amount of recirculated ammonia at 72°F and then enters the spray header at a mixed mean temperature of about 60°F.

On November 17, a two-hour test was performed to determine the effect of ammonia subcooling on U_o . The chiller was turned off in order to increase the temperature of the ammonia returning from the condenser. During the test, the temperature of the ammonia feed to the evaporator rose steadily from about 50°F to 72°F. As shown in Fig. 5, varying the inlet temperature within this range had no measurable effect on U_o .

6.7 WATER-SIDE PRESSURE DROP

Water-side pressure drops across the evaporator were measured and recorded for all hard-copy data points. A sampling of about 60 points taken from November 15-17 (mostly during the Wilson-plot runs) gives the average results shown in Table 6. It should be noted that these are overall pressure drops

Table 6. Average Measured Pressure Drops as a Function of Flow Rate

Flow rate (gpm)	Measured pressure drop (psi)
1505	0.90
2385	2.28
3200	4.00
4012	6.25
5005	9.40

including entrance and exit losses in the water pipe connection nozzles. The pressure drops are proportional to the square of the flow rate to within a few percent.

6.8 VAPOR QUALITY

Estimates of vapor quality were made by trapping liquid ammonia removed in the demister in a section of 2-in. schedule 40 pipe and noting the time required to fill the pipe

to a known volume. Ten measurements were made on November 9 and 10 at heat duties of 2.4 million and 4.0 million Btu/hr and ammonia feed rates varying from 18-52 gpm. In all cases, the indicated quality was $99.2 \pm 0.25\%$. The amount of liquid entrainment created sufficient fog in the evaporator to obscure one's view of the tube bundle.

7 CONCLUSIONS

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 (U_o) is 760 Btu/hr·ft²·°F and the ammonia-side and water-side coefficients are 4600 and 1300 Btu/hr·ft²·°F, respectively. The water-side pressure drop (including entrance and exit losses) is 4.0 psi.

The operating history of the High Flux surface can have a marked effect on heat transfer performance. Contact between the liquid ammonia and the High Flux tubes under nonboiling conditions appears to deactivate nucleation sites, reducing the heat transfer coefficient to < 700 Btu/hr·ft²·°F. A deactivated High Flux surface can be reactivated by dry heating. This is done by shutting off the ammonia feed while maintaining constant warm water flow. However, if the dryout is conducted when the surface is already activated, the heat transfer coefficient may actually be reduced. This behavior is attributed to incomplete rewetting of tubes following the dryout.

The overall heat transfer coefficient is insensitive to ammonia feed rates in the range of 40-100 gpm. However, there exists a minimum ammonia feed rate below which U_o decreases sharply. For a warm water flow rate of 3200 gpm, the minimum ammonia feed rates are 19, 25, and 29 gpm at heat duties of 2.4 million, 3.2 million, and 4.0 million Btu/hr, respectively. (The corresponding recirculation ratios are 1.29, 1.27, and 1.17, respectively.) From these findings, a local minimum (or critical) loading of 20 lb/hr·ft was determined, and this value appears to be independent of 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.

Other aspects of evaporator performance that were studied included vapor quality, heat duty, and ammonia inlet subcooling. In all cases the measured vapor quality was $99.2 \pm 0.25\%$. The overall heat transfer coefficient was not affected by variations in heat duty from 2.4 million-4.0 million Btu/hr, nor by changes in ammonia inlet temperature in the range of 52-72°F.

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 C.H. Kruger, Jr., Stanford U.
 N.J. Palladino, Pennsylvania State U.
 N.C. Rasmussen, Massachusetts Inst. Technology
 M.A. Schultz, Pennsylvania State U.
 A. Sesonke, Purdue U.
 H. Thielsch, ITT Grinnell Corp.
 Y.C.L.S. Wu, U. Tennessee Space Inst.
 H. Abelson, Mitre Corp.
 Alfa-Laval Thermal, Inc.
 J. Connell
 J.E. Yaffo
 J.H. Anderson, Solar Sea Power, Inc.
 K.J. Bell, Oklahoma State U.
 A. Bergles, Iowa State U.
 Carnegie-Mellon U.
 J. Fetkovich
 A. Lavi
 R. Rothfus
 A. Westerberg
 F.H. Davis, Globe Engineering Co.

Department of Energy

C. Castellano
 R. Cohen
 H. Coleman
 S. Gronich (12)
 J. Hartman
 E.H. Kinelski
 L. Lewis
 H.H. Marvin
 D.E. Neely
 K. Read
 W. Richards
 W.G. Sherwood

Electrotechnical Laboratory, Japan

T. Homma
 T. Kajikawa

Energy Technology and Engineering Center

P. Archbold
 R. Glumace
 A. Klein
 F. Poucher

E.N. Ganic, U. Illinois

General Electric Co.

R. Hindle
 M.G. Olmsted

J.H. Gibbons, Office of Technology Assessment, U.S. Congress

G.K. Hart, DSS Engineers, Inc.

Johns Hopkins U./Applied Physics Laboratory

R. Cusick
 J. Funk
 J. George
 J. Keirsey
 R. Makofski
 D. Richards

H. Kamogawa, Toshiba Corp., Japan

D.E. Kash, USGS

J. Knudsen, Oregon State U.

Lockheed Missiles & Space Co.

R. Conti
 M. Leitner
 B. Messinger
 F. Naef
 W.L. Owens
 L. Trimble

P. Marto, Naval Post Graduate School

R.A. Meyer, The OTEC Liaison

Oak Ridge National Laboratory

R. Lyon
 J. Michel
 R. Murphy

L. Perrigo, Pacific Northwest Laboratories

Rand Corp.

C. Gazley
 R. Pei

Rockwell International

M.T. Constantine

W.R. Wagner

D. Wright

C.M. Sabin, Geoscience, Ltd.

J. Sangiovani, United Technologies Research Center

D. Sasscer, U. Puerto Rico

J. Sladky, Value Engineering

Solar Energy Research Inst.

F. Kreith

B. Shelpuk

Trane Co.

D. Ashworth

H. Foust

TRW

J. Denton

R. Douglass

J. Kaellis

E. Snyder

A. Sprouse

H. Uehara, Saga U., Japan

Union Carbide Corp.

A. Czikk

H. Fricke

P. O'Neill

U. Hawaii

E.K. Noda

J. Shupe

H.J. White

P. Yuen

U. Massachusetts

W. Heronemus

J. McGowan

R. Webb, Pennsylvania State U.

Westinghouse Electric Corp.

E. Barsness

J. Gertz

R.E. Williams, III, PRC Energy Analysis Co.

J.F. Yampolsky, General Atomic Co.