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OTEC Performance Tests of the Carnegie-Mellon University Vertical Fluted-Tube Evaporator

by

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

Ocean Thermal Energy Conversion
Program

Argonne National Laboratory

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Printed in the United States of America
Available from
National Technical Information Service
U. S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

NTIS price codes
Printed copy: A03
Microfiche copy: A01

ANL/OTEC-PS-5

ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
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OTEC PERFORMANCE TESTS OF
THE CARNEGIE-MELLON UNIVERSITY
VERTICAL FLUTED-TUBE EVAPORATOR

by

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July 1979

The Argonne Ocean Thermal Energy Conversion Program
is a joint effort of the
Energy and Environmental Systems Division
and Components Technology Division

sponsored by

U.S. Department of Energy
Assistant Secretary for Energy Technology
Division of Central Solar Technology

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NOMENCLATURE

A_i	Inside surface area of heat exchanger tubes, ft^2	T'_{eo}	Temperature of ammonia vapor leaving evaporator, $^{\circ}\text{F}$
A_{lm}	Log-mean area of heat exchanger tubes, ft^2	ΔT_e	Water-side temperature drop through evaporator, $^{\circ}\text{F}$
A_o	Outside surface area of heat exchanger tubes, ft^2	$\Delta T'_e$	Temperature increase of liquid ammonia entering evaporator, $^{\circ}\text{F}$
C_{pe}	Heat capacity of water in evaporator, $\text{Btu/lb}\cdot^{\circ}\text{F}$	ΔT_h	Water-side temperature increase through heater, $^{\circ}\text{F}$
C'_{pe}	Heat capacity of liquid ammonia entering evaporator, $\text{Btu/lb}\cdot^{\circ}\text{F}$	ΔT_{lme}	Log-mean temperature difference in evaporator, $^{\circ}\text{F}$
C_{ph}	Heat capacity of water in heater, $\text{Btu/lb}\cdot^{\circ}\text{F}$	U_o	Overall heat transfer coefficient based on mean outside tube surface area, $\text{Btu/hr}\cdot\text{ft}^2\cdot^{\circ}\text{F}$
$\Delta H'_e$	Heat of vaporization of ammonia in evaporator, Btu/lb	w_e	Water flow rate through evaporator, gpm
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_l	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 measured to midpoint of flutes, ft		
r_o	Outside radius of heat exchanger tubes measured to midpoint of flutes, 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, $^{\circ}\text{F}$		

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ABSTRACT

Performance tests were conducted with the Carnegie-Mellon University (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 (U_o) was 825 Btu/hr·ft²·°F and the ammonia-side and water-side heat transfer coefficients were 1800 and 2610 Btu/hr·ft²·°F, respectively. The water-side pressure drop was 3.2 psi. Tests were conducted at heat duties ranging from 2.4 million to 4.0 million Btu/hr and ammonia feed rates from 30 to 110 gpm. Over these ranges, U_o increased with ammonia feed rate and decreased with heat duty. For all runs the quality of the ammonia vapor leaving the evaporator exceeded 99.9%. The measured U_o values were very stable and repeatable for any continuous run at fixed operating conditions; however, they were not always repeatable following operating state changes or shutdown of the ammonia feed. These repeatability problems were attributed to variations in tube wetting and to the poor design of the ammonia applicator. Test results suggested 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.

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 larger 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 a vertical fluted-tube evaporator designed by Carnegie-Mellon University (C-MU). The tests were run at the OTEC heat exchanger test facility at Argonne. This evaporator is one of a series of five high-performance shell-and-tube heat exchangers that have been tested. The other exchangers tested to date are the Union Carbide flooded-bundle evaporator, the C-MU vertical fluted-tube condenser, the Union Carbide enhanced-tube condenser, and the Union Carbide sprayed-bundle evaporator. 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 C-MU VERTICAL FLUTED-TUBE EVAPORATOR

A fluted-tube evaporator was designed by the staff at Carnegie-Mellon University and fabricated by the Foster Wheeler Corporation as part of the DOE OTEC heat exchanger development program. Details of the unit are shown in Figs. 1-3,* and design parameters are given in Table 1. The evaporator

*The tube shown in Fig. 2 was selected for the present design on the basis of its superior performance in early single-tube tests at C-MU.

is a vertical single-pass shell-and-tube heat exchanger with doubly fluted aluminum (6061) tubes arranged in a 1.75-in. square pitch. As shown in Fig. 3, liquid ammonia is supplied to the fluted tubes via a pressurized inlet plenum. The upper portion of each tube is truncated (flutes partially machined down) in the region of penetration through the metering plate, forming small passages for the ammonia feed. The term "applicator" is used to identify this region where ammonia is loaded onto the tube surface. With the fluted-tube configuration, relatively high evaporative heat transfer coefficients are possible as a result of intermittent wetting of the flute crests by roll waves and subsequent evaporation of the thin liquid film that is left behind. Unevaporated ammonia is collected at the bottom of the unit and recirculated.

Fig. 1. Diagram of C-MU Vertical Fluted-Tube Evaporator

Fig. 2. Cross Section of an Evaporator Tube (all dimensions in inches)

Fig. 3. Detail of Applicator

Table 1. Nominal Design Parameters for the Fluted-Tube Evaporator

Parameter	Value
Heat duty	3.2 million Btu/hr
Ammonia evaporating temperature	72°F
Water inlet temperature	80°F
Water outlet temperature	78°F
Number of tubes	240
Tube length	172 in.
Tube outside diameter (to midpoint of flutes)	1.218 in.
Tube inside diameter (to midpoint of flutes)	0.915 in.
Outside heat transfer area ^a	1097 ft ²
Mean water velocity in tubes	6.5 ft/sec
Overall water-side pressure drop	2.70 psi
Shell inside diameter	42.25 in.
Shell-side design pressure	215 psig
Tube-side design pressure	100 psig

^aBased on outside tube diameter to midpoint of flutes.

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. 4. 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 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.

Fig. 4. OTEC Heat Exchanger Test Facility

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.

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

Table 2. Instruments and Measurement Accuracy

Quantity measured	Instrument	Accuracy of measurement
Temperature of:		
Water into and out of evaporator Ammonia vapor 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 warm water 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

4 OPERATING PROCEDURE

The C-MU fluted-tube evaporator was installed in the test loop as part of the initial construction package. 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.

As far as the test facility itself is concerned, 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. Prior to the start of testing, the ammonia system was evacuated 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 was also 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

```

12/19/1978
20:20
CMU BOILER
RUN *      85

WATER TEMP HEATER IN=    78.267DEG F
WATER TEMP HEATER OUT=   109.533DEG F

WATER TEMP EVAP IN=     79.956DEG F
WATER TEMP EVAP OUT=    77.986DEG F

NH3 TEMP EVAP IN=      48.667DEG F
NH3 TEMP EVAP OUT=     75.378DEG F
NH3 TEMP AFTER R VALVE, 55.684DEG F

HEATER WATER FLOW=    160.852GPM
EVAP WATER FLOW=    3219.151GPM

NH3 LOOP FLOW=        2.377GPM
NH3 EVAP FEED FLOW=   98.886GPM

NH3 DEMISTER FLOW=    0.000GPM

WATER TEMP CHANGE IN HEATER= 31.267DEG F
WATER TEMP CHANGE IN EVAP,CALCD,MEASD -1.969DEG F  -1.967DEG F

QUARTZ P AFTER R VALVE= 85.307PSIA
QUARTZ T AFTER R VALVE= 55.684DEG F

LOGMEAN TEMP DIFF=    3.498DEG F
HEAT INPUT TO HEATER= 0.2505E 7BTU/HR

NET HEAT IN TO EVAP= 0.2926E 7BTU/HR
HEAT IN DUE TO PUMP= 0.4503E 6BTU/HR

HEAT INTO EVAP BASED ON H2O DELTA T= 0.3168E 7BTU/HR
HEAT INTO EVAP BASED ON NH3 EVAP RATE= 0.3972E 6BTU/HR
HT XCHGR AREA= 824.000FT2

HT XFER COEF BASED ON H2O DELTA T= 0.1099E 48TU/HR/FT2/DEGF
HT XFER COEF BASED ON HT BALANCE= 0.1015E 48TU/HR/FT2/DEGF

WATER PRESSURE CHANGE IN EVAP= 3.148PSI

WARM WATER PUMP DELTA P= 93.604PSI
H3 EVAP PRES= 140.911PSIA

NH3 LOOP MASS FLOW RATE= 12.414LBS/MIN

```

Fig. 5. Sample Data Printout

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. The ammonia pumps were turned on and the flow recirculated through the receivers. The cold water pump, the chiller, and the warm water pump were then started; water flows were initially adjusted to the nominal rate of 3200 gpm, and the chiller was set at the desired temperature. Next, the ammonia flow to the evaporator was started and the expansion valve adjusted. Finally, the steam valve was opened. Following start-up, it generally took two to four hours to reach a steady-state condition, after which data were recorded for two hours or more before changing operating parameters. Generally, a set of hard-copy data was taken every 10 to 20 minutes, with intermediate points often read and plotted but not printed. Under ideal conditions, the system was started up on Monday morning and run continuously until Friday afternoon, giving a total of about 100 hours of operation. The C-MU evaporator was tested during three consecutive weeks in December 1978; testing was completed before the end of the third week. At the end of each week, the system was shut down in approximately the reverse order in which it was started up, and the ammonia inventory was transferred to storage tanks outside the building. For reasons of safety, ammonia was not left in the system during periods in which it was unattended.

Tests performed included Wilson-plot runs and parametric studies to determine the effects of varying the heat duty and the ammonia feed rate on heat transfer. With respect to the latter, it should be noted that the maximum ammonia feed rate attainable was about 110 gpm, which may not have been high enough to achieve peak performance.

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:*

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

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 fluted-tube evaporator.

*See list of nomenclature for explanation of symbols.

2. The rate at which heat is added to the warm water as it flows around the loop from the evaporator discharge nozzle 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 heat 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. This was checked by direct determination of the heat addition from measurements of the water temperature change through the pump at different flow rates.

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:

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

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.

In the case of a heat exchanger with noncircular tubes, the definition of the heat transfer area, A_o , is somewhat arbitrary, since several choices are possible. These include:

A_{oa} , the actual total area of the fluted outer surface of the tubes.

A_{oo} , the outer surface area of imaginary circular cylinders circumscribed about the tubes and passing through the crests of the flutes. This is the so-called "rubber-band" area.

A_{om} , the mean outside surface area equal to the area of imaginary circular cylinders passing through the mid-points between the flute crests and valleys.

Although any of these areas (and likely others, as well) could be chosen for A_o , a specific choice must be made, and ours is to use $A_o = A_{om}$. All values of U_o given in this report will be based on A_{om} ,* which is calculated as follows:

$$A_{om} = 240\pi \frac{(1.218)(172)}{144} = 1097 \text{ ft}^2$$

The values of U based on areas A_{oa} and A_{oo} , as well as on the corresponding inside tube surface areas A_{im} and A_{ia} , can be obtained from those reported below by taking ratios of the values of the relevant areas in Table 3.**

The quantity ΔT_{lme} in Eq. 4 can be expressed 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 fluted-tube 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.

Table 3. Tube Surface Areas Relevant to Calculation of the Overall Heat Transfer Coefficient

Area	Value (ft ²)
A_{im}	824
A_{ia}	1706
A_{om}	1097
A_{oo}	1133
A_{oa}	1723

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_{el} 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_{el} calculated from successive measurements

*The U values given in the sample data printout are based on the mean inside area.

**Note that A_{im} is also the inside surface area of a circular-tube heat exchanger having the same inside cross-sectional area, and hence the same tube-side water velocity, as the fluted-tube evaporator.

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:

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

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. 5; "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 $\pm(0.15^2 + 0.435^2 + 0.580^2 + 0.145^2)^{1/2} = \pm 0.75\%$.

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. 4). 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_a') 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 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 just described, the first is clearly the most accurate and is used exclusively in all calculations reported here.

6 TEST RESULTS²

6.1 REPEATABILITY OF RESULTS

The measured U_o values were generally found to be very stable and repeatable during any continuous run at a fixed set of operating conditions; however, they were not always repeatable after changes in the operating state or following even brief interruptions in the ammonia feed. This behavior is illustrated in Figs. 6 and 7 at heat duties of 3.2 million and 4.0 million Btu/hr. The expected trend of U_o increasing with ammonia feed rate is observed in the runs of both weeks, but the values of U_o for the runs on December 13 and 14 are from 20-60% higher than those of the previous week. (All subsequent values of U_o fell within these limits.) Runs at a heat duty of 2.4 million Btu/hr exhibited a similar lack of repeatability. Moreover, the U_o values at 2.4 million Btu/hr were found to be especially sensitive to brief ammonia flow shutdowns (see Fig. 8). In this test sequence, which commenced at 0020 hours on December 12, the average U_o value was initially

Fig. 6. U_o versus Ammonia Feed Rate at 3.2 Million Btu/hr

Fig. 7. U_o versus Ammonia Feed Rate at 4.0 Million Btu/hr

Fig. 8. Effect of Flow Interruption on U_o

590 Btu/hr·ft²·°F. At 0240 hours the ammonia flow was shut off (while maintaining constant warm water flow); it was reestablished after six minutes. When steady-state conditions prevailed, U_o was found to be 694 Btu/hr·ft²·°F; this was 15% higher than the previous value. At 0540 hours a second ammonia feed shutdown was conducted, and a further excursion to 815 Btu/hr·ft²·°F was measured. After a third such shutdown, U_o dropped slightly to 807 Btu/hr·ft²·°F. Additional test sequences under different operating conditions were conducted during that week, yielding similar but less pronounced differences.

In a given run at any heat duty, the curve of U_o versus ammonia feed flow was generally repeatable regardless of whether measurements were taken as the ammonia flow was incrementally increased or decreased. However, in one sequence of runs at 2.4 million Btu/hr an interesting hysteresis effect was observed (see Fig. 9). Hysteresis did not occur in runs at heat duties of 3.2 million and 4.0 million Btu/hr, and occurred at 2.4 million Btu/hr only in the particular test sequence plotted in Fig. 9. The numbers in the figure indicate the order in which measurements were made. As shown, the U_o values measured for increasing feed rates were not retraced as the feed rates were subsequently reduced. Instead, the later values were higher.

The hysteresis in Fig. 9 is likely related, at least in part, to tube wetting. It is known that dry patches may form on an initially wetted surface if the flow is decreased below some critical value.^{3,4} Once established, these dry patches tend to persist, and flow rates higher than the critical value are often required to eliminate them. Based on this, a plausible explanation for the behavior illustrated in Fig. 9 can be advanced. During the early measurements, the tube surfaces were incompletely wetted, yielding relatively low initial values of U_o . As the ammonia feed rate was increased to 110 gpm, some, but not necessarily all, of the dry areas were eliminated. When the feed rate was subsequently reduced, the fraction of wetted area remained relatively high, resulting in relatively high U_o values.

Fig. 9. U_o versus Ammonia Feed Rate at 2.4 Million Btu/hr

As previously mentioned, hysteresis was not observed in any other runs. This suggests that within the ranges tested, changes in the ammonia feed rate were generally insufficient to alter existing flow patterns on the tubes. The initial extent of tube wetting tends to govern subsequent behavior. If the tubes are initially well wetted, they remain well wetted. If the tubes are initially incompletely wetted, they remain incompletely wetted, at least over moderate changes in feed rate. Apparently the key to good performance is to have initially well-wetted tube surfaces. It is reasonable to expect that the initial extent of tube wetting is influenced by the rate at which ammonia

is first 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. 6 and 7. In the runs on December 5 and 8 the system was started with a feed rate of 60 gpm, whereas 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 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. Moreover, the U_o value may be improved still further by repeatedly shutting off the ammonia feed flow and starting up again until no additional increase in U_o occurs.

Finally, the ammonia applicator design can affect heat transfer behavior. One measure of applicator performance is the uniformity of liquid feed around the periphery of each tube. It should be evident that a non-uniform feed can contribute to the problem of tube wetting. Another measure of applicator performance is the degree to which the 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. There was also 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 little rain (and U_o was relatively high). This inconsistent behavior reflects a poorly operating applicator. Rothfus⁵ has found that a collar-type applicator section with nontruncated flutes performs more effectively than the type used in the tests reported here.

6.2 U_o AT NOMINAL CONDITIONS

The nominal operating conditions for the C-MU evaporator are defined as 3200 gpm water flow, 3.2 million Btu/hr heat duty, and 100 gpm ammonia feed rate. The value of 100 gpm was selected as the nominal ammonia feed rate because U_o increases with feed rate, and 100 gpm was nearly the highest attainable. At nominal conditions, the recirculation ratio* was 5 and the mean temperature of the ammonia feed to the evaporator was 67°F.

In Fig. 6, it can be seen that the U_o value at nominal conditions ranged from about 751 Btu/hr·ft²·°F for the run on December 5 to about 864 Btu/hr·ft²·°F for the run on December 13. All subsequent U_o values at nominal conditions fell within these limits. Figure 10 shows a typical steady-state run at nominal conditions conducted on December 20. For this run the average U_o value was 825 Btu/hr·ft²·°F and the attendant deviation from the mean was less than ±1%.

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

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 can be expressed as:

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

In our calculations for the fluted-tube evaporator, $A_o = A_{om}$ and $A_i = A_{im}$ (see Sec. 5 for definition of areas). Accordingly, the overall heat transfer coefficient, U_o , is referenced to the mean outside area, and the individual coefficients, h_o and h_i , are referenced to the mean outside and inside areas, respectively. The fluted-tube studies of Rothfus⁵ indicated that the water-side coefficient, h_i , is nearly proportional to the 0.8 power of the water flow rate. 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. It should be pointed out that the ammonia-side evaporative heat transfer coefficient is a function of local ammonia film flow rate, which, due to evaporation, may vary with position along the tube. Consequently, the h_o value in Eq. 7 represents an average value over the entire tube length. Finally, the values of the water-side coefficient, h_i , can be calculated from Eq. 7 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,

Fig. 10. Typical Steady-State Run at Nominal Conditions

the evaporator heat duty, q_e , was held constant at the nominal design value of 3.2 million Btu/hr, and the ammonia feed rate was held constant at 100 gpm. Although the inlet water temperature remained unchanged, from run to run it was necessary to vary the ammonia-side temperature slightly in order to maintain constant heat duty. 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 the same because the heat duty and feed flow rate were 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 test facility's ammonia system (see Sec. 4), it is unlikely that foreign materials from this system could foul the ammonia-side 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 salts 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 December 19 and 20, a series of runs were conducted at five different water-side flow rates in the range of 1500-5000 gpm; for these tests the heat duty was fixed at 3.2 million Btu/hr and the ammonia feed flow rate was 100 gpm. Results are summarized in Table 4, and a plot of U_o^{-1} versus $w_e^{-0.8}$ is given in Fig. 11. A least-squares algorithm was employed to fit a straight line through the data in Fig. 11. The value of U_o^{-1} at the intercept $w_e^{-0.8} = 0$ was calculated to be 0.0007016 hr·ft²·°F/Btu. The wall resistance term on the right side of Eq. 7 was calculated using the following parameters for the C-MU evaporator:

$$r_i = 0.038125 \text{ ft}$$

$$r_o = 0.05075 \text{ ft}$$

$$k = 99 \text{ Btu/hr·ft·°F}$$

From this we obtain:

$$\frac{\Delta r A_o}{k A_{lm}} = 0.000147 \text{ hr·ft}^2\text{·°F/Btu*}$$

Substituting these values into Eq. 7, h_o is calculated to be 1800 Btu/hr·ft²·°F. This value of h_o represents an enhancement ratio (actual h_o to h_o predicted by Chun-Seban correlation for an equivalent smooth tube) of about 2.5. The outside area ratio (actual outside area of fluted tube/outside area of equivalent smooth tube) is 1.57; thus it is evident that the flutes provide more than an area-ratio enhancement.

From the foregoing, the h_i values are readily calculated at each flow rate. Results are summarized in Table 5 and plotted in Fig. 12. The Sieder-Tate correlation shown in Fig. 12 is based on an equivalent smooth tube having

*This assumes that the heat transfer resistance is the same as that for plain circular tubes with inner and outer diameters equal to flute midpoint diameters. Since the thermal conductivity of the tube wall is relatively large (yielding a wall resistance that is less than one-third of the total resistance), the error introduced by this assumption is very small.

Table 4. Values of U_o as a Function of Water Flow Rate

w_e^a (gpm)	U_o (Btu/hr·ft ² ·°F)
1510	610
2420	750
3200	825
4050	885
4940	945

^aThese values can be converted to tube-side mean flow velocity v (ft/sec) by the relationship $v = 0.002033w_e$.

Fig. 11. Wilson Plot

the same inside flow cross-sectional area. As shown, the enhancement ratio (ratio of actual h_i to h_i predicted by Sieder-Tate equation) is 1.95. It is interesting to note that this enhancement ratio is nearly equal to the area ratio (actual inside area of fluted tube/inside area of equivalent smooth tube) which is 2.07.

6.4 EFFECT OF AMMONIA FEED RATE

At heat duties of 3.2 million and 4.0 million Btu/hr, U_o increased with ammonia feed rate as expected (see Figs. 6 and 7).^{*} No maximum was observed, which according to Rothfus⁵ is characteristic of performance with an applicator having truncated flutes. Rothfus has found that with a collar-type applicator 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. 6 and 7, it can be seen that the U_o values increase with decreasing heat duty. This result was anticipated because at lower heat duties less ammonia is evaporated, and a greater amount of ammonia liquid remains on the tubes.

6.5 WATER-SIDE PRESSURE DROP

Water-side pressure drops across the evaporator were measured and recorded for all runs. Average pressure drops during a test sequence on December 14 are given in Table 6. Each value is an average of about seven data points. The pressure drops are proportional to the square of the flow rate to within a few percent.

^{*}It should be pointed out that the ammonia inlet subcooling decreases with increasing ammonia feed rate, but the effect of subcooling on U_o is expected to be negligibly small.

Table 5. Dependence of h_i on Water Flow Rate

w_e (gpm)	h_i (Btu/hr·ft ² ·°F)
1510	1431
2420	2087
3200	2610
4050	3150
4940	3692

Fig. 12. Variation of h_i with Water Flow Rate

6.6 VAPOR QUALITY

Estimates of vapor quality were made by trapping the run-off from the demister in a vertical section of pipe and noting the time required to fill the pipe to a certain level. Several runs of 15 minutes duration or longer were made, and in each instance no measurable amount was collected. From this we calculate that the vapor quality in all cases was >99.9%.

7 CONCLUSIONS

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·ft²·°F and the ammonia-side and water-side heat transfer coefficients are 1800 and 2610 Btu/hr·ft²·°F, respectively. The ammonia-side heat transfer coefficient of 1800 Btu/hr·ft²·°F represents an enhancement ratio of 2.5 which is greater than the outside area ratio of 1.57. The water-side heat transfer coefficient of 2610 Btu/hr·ft²·°F represents an enhancement ratio of 1.95 which is nearly equal to the inside area ratio of 2.07. At nominal operating conditions, the water-side pressure drop is 3.2 psi, the recirculation ratio is 5, and the ammonia temperature entering the evaporator is 67°F.

Tests were conducted at heat duties ranging from 2.4 million to 4.0 million Btu/hr and ammonia feed rates ranging from 30 to 110 gpm. Over these ranges, U_o was found to increase with ammonia feed rate and decrease with heat duty. No maximum in U_o was found at any ammonia feed rate between 30 and 110 gpm. For all runs the quality of the ammonia vapor leaving the evaporator exceeded 99.9%.

The measured U_o values were found to be very stable and repeatable for any continuous run at fixed operating conditions; however, they were not always repeatable following major operating state changes or shutdown of the

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

Flow rate (gpm)	Measured pressure drop (psi)
1467	0.75
2395	1.93
3200	3.21
4077	4.73
5033	7.07

ammonia feed. These repeatability problems were attributed to variations in tube wetting and to the poor design of the applicator that feeds the liquid ammonia onto the tubes. The results suggest that to ensure good wetting and thermal performance, the system should be started with a high ammonia feed rate and then cut back to the desired operating value. Moreover, the U_o value may be improved still further by repeatedly shutting off the ammonia flow and starting up again until no additional increase in U_o occurs.

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ACKNOWLEDGMENTS

We wish to acknowledge the valuable contributions made by several persons at Argonne and elsewhere to this work. We thank Tony Thomas, manager of the OTEC power system development program at Argonne; Lloyd Lewis, Floyd Davis (of Globe Engineering), and Chester Brzegowy for their work on the design and construction of the test facility; Oscar Despe for assistance with the data-handling equipment; Bob Mandernack for supervising test facility operations; Christine Lemberg for editorial assistance; and Letitia Kaatz for typing the report. We also appreciate the helpful comments and suggestions made by Bob Rothfus of Carnegie-Mellon University, E.N. Ganic of the University of Illinois, and John Michel of Oak Ridge National Laboratory, all of whom reviewed the report prior to publication. Our thanks to all for their fine efforts.

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