423
11-26-79

MASTER

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

by

Lloyd G. Lewis and Norman F. Sather

Ocean Thermal Energy Conversion
Program

Argonne National Laboratory

Sponsored by
U. S. Department of Energy
under Contract W-31-109-Eng-38

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

The facilities of Argonne National Laboratory are owned by the United States Government. Under the terms of a contract (W-31-109-Eng-38) among the U. S. Department of Energy, Argonne Universities Association and The University of Chicago, the University employs the staff and operates the Laboratory in accordance with policies and programs formulated, approved and reviewed by the Association.

MEMBERS OF ARGONNE UNIVERSITIES ASSOCIATION

The University of Arizona	The University of Kansas	The Ohio State University
Carnegie-Mellon University	Kansas State University	Ohio University
Case Western Reserve University	Loyola University of Chicago	The Pennsylvania State University
The University of Chicago	Marquette University	Purdue University
University of Cincinnati	The University of Michigan	Saint Louis University
Illinois Institute of Technology	Michigan State University	Southern Illinois University
University of Illinois	University of Minnesota	The University of Texas at Austin
Indiana University	University of Missouri	Washington University
The University of Iowa	Northwestern University	Wayne State University
Iowa State University	University of Notre Dame	The University of Wisconsin-Madison

NOTICE

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States nor any agency thereof, nor any of their employees, makes any warranty, expressed or implied, or assumes any legal liability or responsibility for any third party's use or the results of such use of any information, apparatus, product or process disclosed in this report, or represents that its use by such third party would not infringe privately owned rights. Mention of commercial products, their manufacturers, or their suppliers in this publication does not imply or connote approval or disapproval of the product by Argonne National Laboratory or the United States Government.

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-4

ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

OTEC PERFORMANCE TESTS OF THE
CARNEGIE-MELLON UNIVERSITY VERTICAL FLUTED-TUBE CONDENSER

by

Lloyd G. Lewis and Norman F. Sather
Energy and Environmental Systems Division

May 1979

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

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

TABLE OF CONTENTS

	<u>Page</u>
NOMENCLATURE.	vi
ABSTRACT.	1
1 INTRODUCTION	1
2 C-MU FLUTED-TUBE CONDENSER	3
3 HEAT EXCHANGER TEST FACILITY	4
3.1 Piping Layout	4
3.2 Instrumentation	7
3.3 Data Handling System.	7
4 OPERATING PROCEDURE.	7
5 METHODS OF CALCULATION	10
5.1 Equations for the Overall Heat Transfer Coefficient	10
5.2 Error Analysis.	12
6 TEST RESULTS	15
6.1 Effect of Quality of Vapor Feed	15
6.2 Performance at Nominal Operating Conditions	16
6.3 Effect of Heat Duty	16
6.4 Individual Heat Transfer Coefficients	17
6.5 Water-Side Pressure Drop.	22
7 SUMMARY AND CONCLUSIONS.	22
APPENDIX: RESULTS FOR DIFFERENT HEAT DUTIES AND COLD WATER FLOW RATES.	25
REFERENCES.	33
ACKNOWLEDGMENTS	34

LIST OF FIGURES

	<u>Page</u>
1 C-MU Vertical Fluted-Tube Condenser.	3
2 Cross Section of a Condenser Tube.	4
3 OTEC Heat Exchanger Test Facility.	6
4 Sample Data Printout	9
5 Wilson Plot at 2.3 Million Btu/hr.	20
6 Wilson Plot at 3.2 Million Btu/hr.	20
7 Wilson Plot at 4.0 Million Btu/hr.	20

LIST OF TABLES

1 Nominal Design Parameters for the Fluted-Tube Condenser.	5
2 Instruments and Measurement Accuracy	8
3 Tube Surface Areas Relevant to Calculation of the Overall Heat Transfer Coefficient	12
4 Condenser Performance at Nominal Operating Conditions.	16
5 Effect of Heat Duty on Condenser Performance	16
6 Overall Heat Transfer Coefficient as a Function of Heat Duty and Water Flow Rate.	19
7 Ammonia-Side Heat Transfer Coefficient as a Function of Heat Duty. .	21
8 Water-Side Heat Transfer Coefficients	21
9 Water-Side Pressure Drop	22
A.1 Results for 3.2 Million Btu/hr and 1200 gpm.	26
A.2 Results for 3.2 Million Btu/hr and 2400 gpm.	26
A.3 Results for 3.2 Million Btu/hr and 3200 gpm.	27
A.4 Results for 3.2 Million Btu/hr and 4000 gpm.	27
A.5 Results for 3.2 Million Btu/hr and 6000 gpm.	28
A.6 Results for 4.0 Million Btu/hr and 6000 gpm.	28
A.7 Results for 4.0 Million Btu/hr and 4000 gpm.	29
A.8 Results for 4.0 Million Btu/hr and 3200 gpm.	29
A.9 Results for 4.0 Million Btu/hr and 2400 gpm.	29
A.10 Results for 4.0 Million Btu/hr and 1200 gpm.	30
A.11 Results for 2.3 Million Btu/hr and 1200 gpm.	30
A.12 Results for 2.3 Million Btu/hr and 2400 gpm.	31

LIST OF TABLES (Cont'd)

	<u>Page</u>
A.13 Results for 2.3 Million Btu/hr and 3200 gpm.	31
A.14 Results for 2.3 Million Btu/hr and 4000 gpm.	32
A.15 Results for 2.3 Million Btu/hr and 6000 gpm.	32

NOMENCLATURE

A_i	Inside surface area of heat exchanger tubes, ft
A_{lm}	Log-mean area of heat exchanger tubes, ft ²
A_o	Outside surface area of heat exchanger tubes, ft ²
C_{pc}	Heat capacity of water in condenser, Btu/lb·°F
C_{pr}	Heat capacity of water in chiller, Btu/lb·°F
$\Delta H'_c$	Heat of condensation of ammonia in condenser, Btu/lb
h_{fi}	Fouling coefficient for inside of tubes, Btu/hr·ft ² ·°F
h_{fo}	Fouling coefficient for outside of tubes, Btu/hr·ft ² ·°F
h_i	Heat transfer coefficient for inside surface (water), Btu/hr·ft ² ·°F
$h_{i,st}$	Heat transfer coefficient for inside surface (water) calculated from Sieder-Tate equation, Btu/hr·ft ² ·°F
h_o	Heat transfer coefficient for outside surface (ammonia), Btu/hr·ft ² ·°F
k	Conductivity of tube wall material, Btu/hr·ft·°F
m_c	Mass flow rate of water through condenser, lb/min
m_r	Mass flow rate of water through chiller, lb/min
m'_l	Mass flow rate of ammonia around loop, lb/min
Δp_c	Water-side pressure drop through condenser, psi
Δp_{pc}	Pressure increase across cold water pump, psi
q_{c1}	Rate of heat transferred to cold water in condenser calculated from water temperature change, Btu/hr
q_{c2}	Rate of heat transferred to cold water in condenser calculated from cold-water loop energy balance, Btu/hr
q'_c	Rate of heat transferred from ammonia stream during passage through the condenser, 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_{ci}	Temperature of cold water entering condenser, °F
T_{co}	Temperature of cold water leaving condenser, °F
T'_c	Saturation temperature of ammonia in condenser, °F
ΔT_c	Water-side temperature increase through condenser ($T_{co} - T_{ci}$), °F
ΔT_r	Water-side temperature decrease through chiller, °F
ΔT_{lmc}	Log-mean temperature difference in condenser, °F
U_o	Overall heat transfer coefficient based on mean outside tube surface area, Btu/hr·ft ² ·°F
w_c	Cold water flow rate through condenser, gpm

OTEC PERFORMANCE TESTS OF THE
CARNEGIE-MELLON UNIVERSITY VERTICAL FLUTED-TUBE CONDENSER

by

Lloyd G. Lewis and Norman F. Sather

ABSTRACT

Results of performance tests on a vertical fluted-tube condenser designed by Carnegie-Mellon University are reported. The condenser was free of operational problems and performed in a stable and repeatable manner. At nominal design conditions (a heat duty of 3.2 million Btu/hr and a water flow rate of 3200 gpm), the overall heat transfer coefficient (U_o) was 1040 Btu/hr·ft²·°F and the water-side pressure drop was 3.3 psi. Varying the quality of the ammonia vapor feed to the condenser (in the range of 90-100%) did not affect performance. On the other hand, when the heat duty was increased from 2.3 million to 4.0 million Btu/hr, U_o rose from 1024 to 1058 Btu/hr·ft²·°F. This trend resulted from an increase in the ammonia-side coefficient (from 3270 to 3730 Btu/hr·ft²·°F) over this range of heat duties. Finally, the water-side coefficient was about 2.45 times greater than that calculated from the Sieder-Tate equation for smooth circular tubes of the same cross-sectional area. This enhancement ratio of 2.45 was some 20% greater than the enhancement factor of 2.07 expected from the increase in surface area alone.

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 a vertical fluted-tube condenser designed at Carnegie-Mellon University (C-MU). The tests were run at the OTEC heat exchanger test facility at Argonne. This condenser 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 evaporator, the Union Carbide sprayed-bundle evaporator, 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 C-MU FLUTED-TUBE CONDENSER

Fig. 1. C-MU Vertical Fluted-Tube Condenser

A fluted-tube condenser was designed by the staff of Carnegie-Mellon University and fabricated by Foster Wheeler Corporation as part of the DOE OTEC heat exchanger development program. As shown in Fig. 1, the condenser is a vertical shell-and-tube exchanger containing 240 fluted tubes of 6061 aluminum arranged in a 1.75-in. square pitch. The water side is single pass through the tubes, which have longitudinal straight flutes on the insides to enhance the water-side heat transfer. Ammonia vapor is supplied through a 12-in. nozzle located midway along the length of the shell side of the exchanger. From this point vapor flows up and down through vapor lanes, condensing along the length of the water tubes. Straight flutes on the ammonia side run the length of the shell, and surface tension forces draw the condensate into the grooves of the fluting. This leaves a very thin film on the crests of the flutes to enhance the ammonia-side heat transfer. The liquid ammonia then flows down the tubes to a reservoir at the bottom of the condenser.

Figure 2 shows the cross section and flutes of a condenser tube.* Table 1 gives the nominal design/operating parameters for this 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 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 fea-

Fig. 2. Cross Section of a Condenser Tube (all dimensions in inches)

ture 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.

*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.

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

Parameter	Value
Heat duty	3.2 million Btu/hr
Ammonia condensing temperature	48°F
Water inlet temperature	40°F
Water outlet temperature	42°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	2.95 psi
Shell inside diameter	42.25 in.
Shell-side design pressure	215 psig
Tube-side design pressure	100 psig
"Clean" overall heat transfer coefficient ^b	1045 Btu/hr·ft ² ·°F

^aBased on outside tube diameter to midpoint of flutes.

^bBased on mean outside tube surface area (1097 ft²). Assumes no tube-side or shell-side fouling.

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. The ammonia loop also contains an expansion valve and a spray desuperheater for adjusting the pressure and quality of the vapor supplied to the test condenser. The desuperheater can provide a vapor stream containing liquid ammonia droplets. Its 48 spray nozzles use high-velocity ammonia vapor jets to atomize small liquid ammonia jets. The liquid ammonia is obtained from the receiver at the bottom of the condenser, and the high-velocity vapor jets are supplied with ammonia vapor at the evaporator pressure. These spray heads are rated to produce a stream of droplets at least 90% of which are less than 10 microns in diameter.

Fig. 3. OTEC Heat Exchanger Test Facility

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 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.

4 OPERATING PROCEDURE

The C-MU fluted-tube condenser was mounted on the test stand in the condition in which it was received, without cleaning, since the heat transfer sur-

Table 2. Instruments and Measurement Accuracy

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

faces 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 ammonia side of the evaporator, from the ammonia side of the condenser, and from the ammonia piping. The system was then filled with ammonia.

The warm and cold water loops were cleaned and passivated by the same procedure used for the ammonia system. The loops were 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 to produce a concentration of 3 oz per gal.

OTEC TEST FACILITY
 ARGONNE NATIONAL LAB
 ARGONNE, ILL 60439 (312) 972-2000

6/15/1978
 21:17
 CMU CONDENSER
 RUN # 19

WATER TEMP CHILLER IN= 44.267DEG F
 WATER TEMP CHILLER OUT= 41.767DEG F

WATER TEMP COND IN= 42.064DEG F
 WATER TEMP COND OUT= 44.080DEG F

NH3 TEMP COND IN= 45.976DEG F
 NH3 TEMP COND OUT= 46.200DEG F

QUARTZ P AFTER R VALVE 82.571PSIA
 QUARTZ T AFTER R VALVE 53.665DEG F

CHILLER WATER FLOW= 3207.720CPM
 COND WATER FLOW= 3207.720CPM

NH3 LOOP FLOW= 18.564CPM

NH3 DEMISTER FLOW= 0.000CPM

WATER TEMP CHANGE IN CHILLER= -2.500DEG F
 WATER TEMP CHANGE IN COND, CALCD, MEASD IN DEG F 2.016 2.009

LOGMEAN TEMP DIFF= 2.774DEG F
 HEAT INPUT TO CHILLER= 0.4027E 7BTU/HR

NET HEAT IN TO COND= 0.3392E 7BTU/HR
 HEAT IN DUE TO PUMP= 0.6250E 6BTU/HR

HEAT INTO COND BASED ON H2O DELTA T= 0.3220E 7BTU/HR
 HEAT INTO COND BASED ON NH3 COND RATE= 0.3097E 7BTU/HR
 HT XCHGR AREA= 824.888FT2

HT XFER COEF BASED ON H2O DELTA T= 0.1409E 4BTU/HR/FT2/DEGF
 HT XFER COEF BASED ON HT BALANCE= 0.1484E 4BTU/HR/FT2/DEGF

WATER PRESSURE CHANGE IN COND= 3.113PSI

COLD WATER PUMP DELTA P= 97.229PSI
 H3 COND PRES= 82.446PSIA

NH3 LOOP MASS FLOW RATE= 97.260LBS/MIN

Fig. 4. Sample Data Printout

At the beginning of an operating period, the water and ammonia pumps were started, the steam flow to the heater was turned on, and the chiller was started. Flows and temperatures were adjusted to the desired values and, when steady operation had been attained (usually 40-60 minutes after startup), the data for the run were taken. Several sets of data were obtained at intervals of 15-30 minutes before changing to a new set of operating conditions. The procedure was then repeated.

At the end of the day the liquid ammonia was pumped from the system back to the storage tank. During those times that the system was not operating, such as nights and weekends, ammonia vapor was left under pressure in the ammonia loop to prevent air from leaking into the system. At the beginning of the next operating period, liquid ammonia was pumped back into the evaporator and condenser. The pumps, heater, and chiller were started, and data taking was resumed.

The normal flow of vapor causes noncondensable gases such as air to collect in the condenser. This tends to blanket the condenser surface and reduce the heat transfer rate. Periodically, noncondensables were bled off through purge nozzles located at the ends of the condenser (shown in Fig. 1). Three purgings were sufficient; additional purgings had no measurable effect on the overall heat transfer coefficient.

The temperature of the cold water flowing into the C-MU condenser was set at 42°F for these tests rather than at the nominal 40°F to prevent the refrigerant temperature in the chiller from going below 32°F at the high heat duties.

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 condenser. These are:

1. The rate at which heat is taken up by the cold water as it passes through the condenser. This can be expressed as follows:*

$$q_{c1} = 60m_c C_{pc} \Delta T_c - 1.484w_c \Delta p_c \quad (1)$$

where the first term is the sensible heat gained by the water as it flows through the condenser, and the second corrects for the frictional heat added. As might be expected, the frictional heat addition is quite small, amounting to less than 0.5% of q_{c1} for all runs with the condenser.

*See list of nomenclature for explanation of symbols.

2. The rate at which heat is removed from the cold water as it flows around the loop from the condenser discharge nozzle back to the condenser inlet. This includes the net effect of the heat removed in the chiller and added in the pump work. The cold water piping system is heavily insulated, so convective heat losses are negligible. Thus:

$$q_{c2} = 60m_r C_{pr} \Delta T_r - 113.4 w_c^{1/2} \Delta p_{pc} \quad (2)$$

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 removed from the ammonia as it passes through the condenser:

$$q'_c = 60m'_l \Delta H'_c \quad (3)$$

During steady-state operation, these three values of the condenser heat duty (q_{c1} , q_{c2} , and q'_c) 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 condenser heat duty, q_c , is obtained, the overall heat transfer coefficient, U_o , is calculated from the conventional definition:

$$U_o = \frac{q_c}{A_o \Delta T_{lmc}} \quad (4)$$

where:

A_o = the outside surface area of the tubes, and

ΔT_{lmc} = 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 midpoints 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.**

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 quantity ΔT_{lmc} in Eq. 4 can be expressed as follows:

$$\Delta T_{lmc} = \frac{\Delta T_c}{\ln \left[\frac{T'_c - T_{ci}}{T'_c - T_{co}} \right]} \quad (5)$$

Here, T'_c is the saturation temperature of the ammonia vapor fed to the condenser; for these tests this temperature was calculated from the measured inlet pressure rather than measured directly, because more accurate values could be obtained in that way.

5.2 ERROR ANALYSIS

Before the test data on the fluted-tube condenser and the resulting heat transfer coefficients calculated from these equations are presented, a brief discussion of the magnitude of the expected uncertainties in q_c and U_o that

*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 condenser.

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_c measurement (with a measurement accuracy of $\pm 0.3\%$) and the flow rate measurement ($\pm 0.15\%$ accuracy); the expected accuracy in q_{c1} 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_{c1} 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:

$$U_o \approx \frac{60m_c C_{pc}}{A_{om}} \ln \left[\frac{T'_c - T_{ci}}{T'_c - T_{co}} \right] \quad (6)$$

The uncertainty in U_o is due in part to inaccuracies in measurements of T_{ci} and T_{co} 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 $T_{co} - T_{ci}$ can be made by adjusting the instrument at the operating temperature of 42°F . After the standard calibration was completed, simultaneous readings of the cold 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_{ci} , T_{co} , and ΔT_c are given in Fig. 4; "water temp cond in" is T_{ci} , "water temp cond out" is T_{co} , and "water temp change in cond" is ΔT_c . It can be noted that $T_{co} - T_{ci}$ differs from ΔT_c by 0.007°F . This is typical of the discrepancy in the two readings of the difference between T_{ci} and T_{co} . Hence, we will use $\pm 0.010^\circ\text{F}$ as the maximum measurement error in readings of ΔT_c , T_{ci} , and T_{co} . The error in T'_c resulting from a ± 0.005 psi uncertainty in the ammonia pressure measurement is $\pm 0.003^\circ\text{F}$.

By applying these measurement errors to Eq. 6, the uncertainty in U_o can now be calculated. A 0.003°F error in T_c' causes a relatively small ($\pm 0.043\%$) error in the natural logarithm factor. However, errors of 0.010°F in T_{ci} and T_{co} 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_{ci} , T_{co} , and T_c' . Since these error components are independent of one another, the total probable error is $\pm(0.15^2 + 0.435^2 + 0.580^2 + 0.043^2)^{1/2} = \pm 0.74\%$.

The second method of calculating the heat duty (q_{c2}) is subject to a different set of errors. The first term in Eq. 2 is the rate at which heat is removed from the cold water in the chiller. The flow through the chiller is measured by a turbine flowmeter to $\pm 0.15\%$ of the actual flow. The value of ΔT_r is measured to $\pm 0.2^\circ\text{F}$ in about 2.5°F , or to $\pm 8\%$. Thus, the total uncertainty in q_{c2} from this term is $\pm 8\%$. The second term in Eq. 2 was derived from pump efficiency data supplied by the pump manufacturer. It contributes an uncertainty to q_{c2} of $\pm 1.3\%$. Hence, the total estimated uncertainty in q_{c2} is about 10% .

The third method for computing the heat duty is based on the rate at which energy is removed from the ammonia as it passes through the condenser. 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 condenser, 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_c' other than as a rough check on q_{c1} and q_{c2} . No calculations of U_o based on q_c' 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 TEST RESULTS²

The testing of the fluted-tube condenser was carried out over a four-week period during May and June, 1978. After a series of shakedown runs to check out the instrumentation and the operating and calculational procedures, the first test run was made on June 14. The purpose of this run was to determine the effect of inlet ammonia vapor quality on the performance of the condenser. This was followed by a two-day run on June 15-16, during which the effects of heat duty and cold water flow rate on the condenser performance were investigated. Runs were made at heat duties of 2.3 million, 3.2 million, and 4.0 million Btu/hr. At each heat duty, the cold water flow rate was varied from 1200 to 6000 gpm. The data obtained at the different flow rates were for the purpose of determining the individual ammonia-side and water-side heat transfer coefficients for each of the three heat duties.

6.1 EFFECT OF QUALITY OF VAPOR FEED

The vapor discharged from the expansion valve is always superheated and is, therefore, not characteristic of a turbine exhaust in an OTEC power system. As mentioned previously, the spray desuperheater can be used to provide a vapor stream that contains liquid ammonia droplets and corresponds to the quality of a typical turbine exhaust. The run on June 14 to determine the effect of vapor quality on heat transfer performance was made at nominal operating conditions. With the liquid spray on so as to produce a wet vapor containing 10% (by mass) entrained droplets, the measured value of U_o was slightly lower (about 1%) than that obtained with superheated vapor. Since this decrease is of the same magnitude as the scatter in the measured values of U_o at constant operating conditions, it must be concluded that for vapor qualities above 90% the quality of the vapor feed has essentially no effect on the performance of the fluted-tube condenser. Hence, all subsequent tests were made with the superheated vapor fed directly from the expansion valve to the condenser, with no liquid spray added.

6.2 PERFORMANCE AT NOMINAL OPERATING CONDITIONS

The results of the test at nominal operating conditions, i.e., at a heat duty of 3.2 million Btu/hr and a water flow rate of 3200 gpm (corresponding to a tube velocity of 6.5 ft/sec), are shown in Table 4. These data were obtained at approximately 30-minute intervals during a steady-state run of some three hours' duration. The average value of the overall heat transfer coefficient, U_o , based on the outside mean tube surface area was 1043 Btu/hr·ft²·°F. This is in very close agreement with the expected value of 1045 Btu/hr·ft²·°F predicted from single-tube heat transfer measurements at Carnegie-Mellon University. The average value of the water-side pressure drop for the condenser during this run was 3.3 psi.

Table 4. Condenser Performance at Nominal Operating Conditions

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	T_c' (°F)	U_o (Btu/hr·ft ² ·°F)	P_c (psi)
17	3.23	3206.4	41.93	45.85	1055	3.37
18	3.21	3205.2	42.04	45.99	1022	3.40
19	3.22	3207.7	42.06	45.98	1058	3.11
20	3.24	3207.1	42.06	45.98	1051	3.34
21	3.23	3209.5	42.06	45.99	1041	3.33
22	3.24	3211.4	42.08	46.05	1031	3.44
Average	3.23	3208.0	42.04	45.97	1043	3.33

6.3 EFFECT OF HEAT DUTY

The values of the overall heat transfer coefficient obtained at the three different heat duties are given in Table 5. For each heat duty the reported U_o value is an average of six values obtained during a three-hour steady-state run. During all of these runs, the water flow rate was 3200 gpm and the inlet water temperature was 42°F. It can be seen that the overall heat transfer coefficient increases slightly with heat duty. Since the water flow rate was the same for each heat duty, the water-side heat transfer coefficient should be constant. Hence, the increase in U_o with heat duty is evidently the result of an increasing ammonia-side coefficient. This is shown

Table 5. Effect of Heat Duty
on Condenser Performance

q_{cl} (million Btu/hr)	U_o (Btu/hr·ft ² ·°F)
2.3	1024
3.2	1043
4.0	1058

to be the case in the next section, which describes the results of determining the individual film coefficients by means of the Wilson method.

6.4 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:

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

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 $w_c^{-0.8}$. Rothfus³ has shown that his data on the water-side heat transfer coefficient for fluted tubes similar to those of the test condenser are well represented by the standard correlation with an 0.8 power dependence on flow rate for Reynolds numbers above 18,000. 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 $w_c^{-0.8}$ should fall on a straight line. Extrapolation of the line to the intercept at $w_c^{-0.8} = 0$ 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 noted that the ammonia-side coefficient depends on the local condensate loading, which may vary from tube to tube in the bundle. Consequently, h_o in Eq. 7 is the average value for the condenser as a whole. Finally, the values of the water-side coefficient, h_i , can be calculated from Eq. 7 for each value of w_c .

Clearly, the use of this procedure for determining h_o and h_i (w_c) requires that the data on U_o (w_c) be obtained under conditions where h_o is con-

stant and the fouling coefficients, h_{fo} and h_{fi} , are known. Since the heat duty (and, hence, the condensate loading) was the same during the runs at different water velocities, h_o is a constant for each set of runs at a given heat duty provided the temperature profiles along the length of the tubes are identical from run to run. Although the inlet 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 exactly the same for the different flow rates. However, the effect of this small difference in temperature profiles on the thermophysical properties of the condensate film is negligible, so the value of h_o determined from these data by the Wilson procedure should adequately represent the average performance of the tube bundle.

The requirement that the fouling resistances be known is a somewhat less certain matter. Because of precautions taken to maintain the cleanliness of the test facility's ammonia system (see Sec. 4), it seems unlikely that foreign materials from this system could foul the condenser tube 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 cold water pump. The extreme cleanliness of the system and the effectiveness of the corrosion inhibitor in scavenging oxygen and in preventing the deposition of hard calcium salts lead us to expect that the magnitude of the water-side fouling resistance is essentially zero. Thus, it is 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.

The results for U_o given in Table 6 for the three different heat duties are averages of five to eight values obtained during steady runs of several hours' duration. The complete data from which these averages were obtained are tabulated in the Appendix to this report. Wilson plots of these data are given in Figs. 5-7. The straight lines in these figures were determined by a least-squares fit of the data. From the values of the intercept at

Table 6. Overall Heat Transfer Coefficient
as a Function of Heat Duty and
Water Flow Rate

q_{cl} (million Btu/hr)	w_c^a (gpm)	U_o (Btu/hr·ft ² ·°F)
2.36	1204	608
2.32	2397	899
2.28	3192	1024
2.27	4009	1103
2.24	5992	1249
3.26	1203	618
3.25	2412	924
3.23	3208	1043
3.23	3995	1138
3.22	5984	1299
3.99	1205	628
4.00	2396	935
3.98	3198	1058
4.04	4000	1154
4.01	5989	1333

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

$w_c^{-0.8} = 0$ shown in the figures and the magnitude of the wall resistance term*

$$\frac{\Delta r A_o}{k A_{lm}} = \frac{r_o \ln(r_o/r_i)}{k} = 1.47 \times 10^{-4} \text{ hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}/\text{Btu}$$

we can calculate the ammonia-side coefficient, h_o , for each heat duty. The results of this calculation are given in Table 7. It can be seen that there is a small, but definite increase in h_o with heat duty and, therefore, with condensate loading. This indicates that the range of condensate loading studied is below the critical loading at which the resistance of the condensate film begins to dominate the heat transfer process.

*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 so large ($k = 99 \text{ Btu/hr} \cdot \text{ft} \cdot ^\circ\text{F}$), the wall resistance is less than one-third of the total resistance and the error introduced by this assumption is very small.

Fig. 5. Wilson Plot at
2.3 Million Btu/hr

Fig. 6. Wilson Plot at
3.2 Million Btu/hr

Fig. 7. Wilson Plot at
4.0 Million Btu/hr

Table 7. Ammonia-Side Heat Transfer Coefficient as a Function of Heat Duty

q_c (million Btu/hr)	h_o (Btu/hr·ft ² ·°F)
2.3	3270
3.2	3590
4.0	3730

Using these values of h_o and the U_o values for the different flow rates, we can calculate h_i , the water-side heat transfer coefficient, based on the mean inside tube surface area, A_{im} . The results are given in Table 8 along with the values of h_i calculated from the Sieder-Tate equation for smooth circular tubes with the same inside diameter (0.915 in.) and, thus, the same flow cross section.

Also shown is the heat transfer enhancement ratio, $h_i/h_{i,st}$, which is a measure of the effect of the tube fluting on the heat transfer performance. Since the area of the fluted surface

Table 8. Water-Side Heat Transfer Coefficients

w_c^a (gpm)	U_o (Btu/hr·ft ² ·°F)	h_i (Btu/hr·ft ² ·°F)	$h_{i,st}$ (Btu/hr·ft ² ·°F)	Enhancement ratio
$q = 2.3$ million Btu/hr				
1204	608	1128	462	2.44
2397	899	1956	796	2.46
3192	1024	2460	999	2.46
4009	1103	2952	1197	2.47
5992	1249	4071	1650	2.47
$q = 3.2$ million Btu/hr				
1203	618	1126	463	2.43
2412	924	1964	802	2.45
3208	1043	2467	1005	2.45
3995	1138	2940	1196	2.46
5984	1299	4062	1650	2.46
$q = 4.0$ million Btu/hr				
1205	628	1139	466	2.44
2396	935	1973	799	2.47
3198	1058	2486	1003	2.48
4000	1154	2973	1198	2.48
5989	1333	4107	1652	2.49

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

is $1706/824 = 2.07$ times that of circular tubes with the same flow cross section, it is apparent that the fluting provides somewhat more heat transfer enhancement than would be expected from the area increase alone.

6.5 WATER-SIDE PRESSURE DROP

The pressure drop on the water side was measured by a differential pressure transmitter. The pressure taps are located in the supply pipe about six feet upstream of the condenser and in the return pipe about four feet downstream of the condenser discharge nozzle. Although these taps are at different elevations, the hydrostatic head in the instrument piping (which was purged of all air) just balances the hydrostatic head in the exchanger; thus the differential pressure reading is the pressure drop from fluid friction alone. Pressure drop readings, which were taken for each run, are tabulated in the Appendix. The average values for each flow rate at the nominal heat duty of 3.2 million Btu/hr are collected in Table 9.

Table 9. Water-Side Pressure Drop

w_c (gpm)	ΔP_c (psi)
1203	0.57
2412	2.05
3208	3.33
3995	4.59
5984	9.59

7 SUMMARY AND CONCLUSIONS

The heat transfer performance of the vertical fluted-tube condenser designed by Carnegie-Mellon University has been measured at typical OTEC operating conditions in the Argonne heat exchanger test facility. The condenser was found to perform in a stable and repeatable manner and was free of operational problems. For operation at nominal design conditions -- i.e., a heat duty of 3.2 million Btu/hr and a cold water flow rate of 3200 gpm (corresponding to a tube velocity of 6.5 ft/sec) -- the measured overall heat transfer coefficient was $1040 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ and the water-side pressure drop was 3.3 psi.

The heat transfer coefficient was found to be unaffected by the quality of the ammonia vapor feed to the condenser for the range (90-100%) used in the tests. However, changes in the heat duty (or, equivalently, the condensate loading on the condenser tubes) produced a measurable change in U_o . Specifically, at a water flow rate of 3200 gpm, U_o rose from 1024 to 1058 $\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ over a heat duty range of 2.3 million to 4.0 million Btu/hr. Examination

of the individual ammonia-side and water-side heat transfer coefficients by means of the Wilson procedure showed that this trend in U_o resulted from an increase in the ammonia-side coefficient from 3270 to 3730 Btu/hr·ft²·°F over this range of heat duties. The Wilson procedure based on an 0.8 power dependence of the water-side coefficient on flow rate was found to give a satisfactory correlation of the measured values of U_o . Finally, the water-side coefficient, which was also determined by the Wilson method, was some 2.45 times greater than that calculated from the Sieder-Tate equation for smooth circular tubes of the same cross-sectional area. This enhancement factor of 2.45 was essentially the same for all water flow rates from 1200 to 6000 gpm (tube velocities from 2.4 to 12.2 ft/sec) and was about 20% greater than the heat transfer enhancement factor of 2.07 expected from the increase in surface area alone.

APPENDIX

Results for Different Heat Duties
and Cold Water Flow Rates

Table A.1. Results for 3.2 Million Btu/hr and 1200 gpm

Run number	q_{cl} (million Btu/hr)	w_c^a (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr · ft ² · °F)	Δp_c (psi)
3	3.24	1200.9	42.01	5.38	50.05	608	0.66
4	3.26	1202.2	42.07	5.40	50.04	621	0.56
5	3.26	1204.1	42.02	5.39	50.00	622	0.46
6	3.26	1203.2	41.99	5.40	49.98	620	0.59
7	3.26	1202.9	41.95	5.40	49.91	623	0.54
8	3.25	1202.1	41.92	5.39	49.88	624	0.59
9	3.26	1203.3	41.94	5.40	49.97	613	0.58
10	3.26	1202.9	41.92	5.40	49.96	613	0.56
Average	3.26	1202.7	41.98	5.40	49.97	618	0.57

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

Table A.2. Results for 3.2 Million Btu/hr and 2400 gpm

Run number	q_{cl} (million Btu/hr)	w_c^a (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr · ft ² · °F)	Δp_c (psi)
11	3.23	2409.9	42.07	2.68	46.79	924	2.03
12	3.25	2409.4	42.08	2.69	46.84	913	2.03
13	3.25	2409.6	42.04	2.70	46.78	917	2.01
14	3.25	2412.9	42.00	2.69	46.71	944	2.11
15	3.26	2415.2	41.98	2.69	46.72	922	2.16
16	3.25	2416.2	41.94	2.68	46.64	932	1.99
Average	3.25	2412.2	42.02	2.69	46.74	924	2.05

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

Table A.3. Results for 3.2 Million Btu/hr and 3200 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
17	3.23	3206.4	41.93	2.02	45.85	1055	3.37
18	3.21	3205.2	42.04	2.01	45.99	1022	3.40
19	3.22	3207.7	42.06	2.01	45.98	1058	3.11
20	3.24	3207.1	42.06	2.02	45.98	1051	3.34
21	3.23	3209.5	42.06	2.01	45.99	1041	3.33
22	3.24	3211.4	42.08	2.02	46.05	1031	3.44
Average	3.23	3208.0	42.04	2.02	45.97	1043	3.33

Table A.4. Results for 3.2 Million Btu/hr and 4000 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
23	3.23	3999.8	42.07	1.62	45.57	1116	4.60
24	3.22	3997.7	42.07	1.62	45.54	1143	4.40
25	3.24	3993.5	42.06	1.63	45.51	1146	4.60
26	3.22	3992.6	42.05	1.62	45.55	1123	4.68
27	3.24	3998.3	42.04	1.63	45.48	1156	4.86
28	3.21	3986.5	42.04	1.62	45.50	1142	4.44
Average	3.23	3994.8	42.06	1.62	45.53	1138	4.59

Table A.5. Results for 3.2 Million Btu/hr and 6000 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
29	3.20	5985.5	41.98	1.10	44.84	1287	9.76
30	3.23	5986.8	42.00	1.11	44.84	1299	9.39
31	3.22	5983.4	42.00	1.11	44.82	1312	9.49
32	3.21	5978.7	42.01	1.10	44.82	1315	9.77
33	3.22	5988.9	42.02	1.11	44.87	1286	9.80
34	3.22	5980.7	42.02	1.11	44.87	1294	9.32
Average	3.22	5984.0	42.00	1.11	44.84	1299	9.59

Table A.6. Results for 4.0 Million Btu/hr and 6000 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
35	4.00	5981.4	41.95	1.37	45.41	1336	9.52
36	4.00	5993.6	42.21	1.37	45.71	1329	9.60
37	4.03	5990.2	42.08	1.38	45.54	1345	9.91
38	4.02	5988.2	41.92	1.37	45.42	1323	10.17
39	4.01	5989.5	41.91	1.37	45.40	1331	9.89
Average	4.01	5988.6	42.01	1.37	45.49	1333	9.82

Table A.7. Results for 4.0 Million Btu/hr and 4000 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
40	4.03	3996.5	41.98	2.02	46.28	1152	4.98
41	4.05	4008.4	42.00	2.03	46.30	1150	4.61
42	4.02	3997.1	42.01	2.02	46.33	1140	4.61
43	4.04	3999.8	42.02	2.03	46.31	1158	5.23
44	4.04	3996.2	42.00	2.03	46.29	1170	4.88
Average	4.04	3999.6	42.00	2.02	46.30	1154	4.86

Table A.8. Results for 4.0 Million Btu/hr and 3200 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
45	3.99	3199.5	41.99	2.49	46.80	1061	3.70
46	3.97	3199.9	41.96	2.48	46.77	1058	3.38
47	3.97	3196.0	41.99	2.49	46.81	1058	3.60
48	3.98	3199.4	42.01	2.49	46.83	1056	3.48
49	3.98	3196.0	42.02	2.49	46.84	1057	3.37
Average	3.98	3198.2	41.99	2.49	46.81	1058	3.50

Table A.9. Results for 4.0 Million Btu/hr and 2400 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
50	4.00	2395.6	42.06	3.33	47.85	938	1.86
51	4.00	2394.8	42.01	3.34	47.80	935	1.98
52	4.00	2395.6	41.96	3.33	47.76	932	1.98
53	4.00	2397.2	41.96	3.33	47.75	934	2.02
Average	4.00	2395.8	42.00	3.33	47.79	935	1.96

Table A.10. Results for 4.0 Million Btu/hr and 1200 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
55	3.99	1204.6	42.00	6.60	51.70	630	0.58
56	3.99	1205.7	41.95	6.59	51.69	623	0.66
57	4.00	1204.6	42.02	6.61	51.72	627	0.59
58	4.00	1204.9	42.01	6.61	51.71	630	0.61
59	3.99	1204.2	42.02	6.61	51.72	630	0.46
Average	3.99	1204.8	42.00	6.60	51.71	628	0.58

Table A.11. Results for 2.3 Million Btu/hr and 1200 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
60	2.41	1203.7	42.02	3.99	47.93	612	0.50
61	2.38	1203.5	41.97	3.93	47.82	615	0.50
62	2.36	1204.3	42.00	3.91	47.82	611	0.50
63	2.33	1203.0	41.98	3.86	47.77	604	0.59
64	2.33	1203.6	41.97	3.86	47.81	602	0.55
Average	2.36	1203.6	41.99	3.91	47.83	608	0.53

Table A.12. Results for 2.3 Million Btu/hr and 2400 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
65	2.32	2397.3	41.97	1.93	45.47	882	2.13
66	2.31	2386.4	42.08	1.94	45.51	909	2.07
67	2.34	2407.2	41.99	1.94	45.41	917	2.08
68	2.31	2390.6	41.99	1.93	45.44	889	2.00
69	2.30	2394.0	41.97	1.92	45.37	903	2.09
70	2.32	2403.7	41.97	1.93	45.41	896	1.94
Average	2.32	2396.6	42.00	1.93	45.44	899	2.05

Table A.13. Results for 2.3 Million Btu/hr and 3200 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr •ft ² •°F)	Δp_c (psi)
71	2.28	3188.1	42.01	1.44	44.82	1027	3.58
72	2.28	3193.9	42.04	1.44	44.84	1022	3.38
73	2.28	3195.7	42.04	1.43	44.87	1024	3.39
74	2.29	3192.2	42.03	1.44	44.87	1017	3.64
75	2.28	3193.9	41.98	1.44	44.80	1023	3.56
76	2.27	3187.2	41.98	1.43	44.79	1031	3.28
Average	2.28	3191.8	42.01	1.44	44.83	1024	3.47

Table A.14. Results for 2.3 Million Btu/hr and 4000 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr · ft ² · °F)	Δp_c (psi)
77	2.27	4010.8	41.98	1.14	44.53	1068	5.14
78	2.28	4013.2	42.03	1.15	44.52	1112	4.54
79	2.28	4009.9	41.98	1.15	44.48	1110	4.96
80	2.25	4000.5	41.99	1.14	44.48	1100	5.11
81	2.27	4008.7	41.98	1.14	44.44	1116	5.06
82	2.26	4009.3	41.97	1.14	44.44	1113	4.85
Average	2.27	4008.7	41.99	1.14	44.48	1103	4.94

Table A.15. Results for 2.3 Million Btu/hr and 6000 gpm

Run number	q_{cl} (million Btu/hr)	w_c (gpm)	T_{ci} (°F)	ΔT_c (°F)	T'_c (°F)	U_o (Btu/hr · ft ² · °F)	Δp_c (psi)
83	2.24	5983.4	41.98	0.78	44.04	1238	9.98
84	2.25	6002.5	42.01	0.78	44.04	1261	9.18
85	2.24	5998.4	41.96	0.78	43.98	1271	9.78
86	2.24	5997.0	41.99	0.78	44.03	1252	9.86
87	2.24	5978.7	42.04	0.78	44.08	1246	9.73
88	2.23	5993.0	42.04	0.78	44.09	1229	9.44
89	2.24	5993.0	42.02	0.78	44.04	1248	9.99
Average	2.24	5992.3	42.01	0.78	44.04	1249	9.71

REFERENCES

1. Lewis, L.G., and N.F. Sather, OTEC Performance Tests of the Union Carbide Flooded-Bundle Evaporator, Argonne National Laboratory Report ANL/OTEC-PS-1 (Dec. 1978).
2. Hard-copy volumes of the test results are available from N.F. Sather.
3. Rothfus, R.R., and C.P. Neuman, The OTEC Program at Carnegie-Mellon University, Proc. Fourth Annual Conf. on Ocean Thermal Energy Conversion, New Orleans, La., p. VI-55 (March 1977).

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; 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 Susan Janowski 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.

Distribution for ANL/OTEC-PS-4Internal:

F.W. Ahrens	J.J. Dzingel	J.J. Roberts
P. Bensen	D.L. Hillis	G.S. Rosenberg
C. Brzegowy	J.S. Horowitz	N.F. Sather (100)
T.R. Bump	A.B. Krisciunas	A. Thomas
P.S. Chopra	L.G. Lewis	D.T. Yung
J.M. Clinch	J.J. Lorenz	R.S. Zeno
E.J. Croke	K.S. Macal	ANL Contract File
J.B. Darby	W.E. Massey	ANL Libraries (5)
J.D. Ditmars	E.G. Pewitt	TIS Files (6)
J.E. Draley	G.F. Popper	

External:

DOE-TIC, for distribution per UC-64 (423)
 Manager, Chicago Operations and Regional Office, DOE
 Chief, Office of Patent Counsel, DOE-CORO
 President, Argonne Universities Association
 Energy and Environmental Systems Division Review Committee:

E.E. Angino, U. Kansas
 R.E. Gordon, U. Notre Dame
 W.W. Hogan, Harvard U.
 L.H. Roddis, Jr., Charleston, S.C.
 G.A. Rohlich, U. Texas at Austin
 R.A. Schmidt, Booz, Allen, & Hamilton

Components Technology Division Review Committee:

P.F. Cunniff, U. Maryland
 W.E. Kessler, Commonwealth Associates
 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. Sesonske, 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. Krcith

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.