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WITH FALLING JET EVAPORATOR
AND CONDENSER

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EVAPORATOR AND CONDENSER

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ABSTRACT

A configuration for the open cycle (OC) Ocean Thermal Energy Conversion (OTEC) system is presented incorporating a countercurrent falling jet evaporator and a concurrent falling jet condenser. The parameters governing performance of the proposed configuration are discussed and the sizing of equipment for a 100-MWe net power output OC OTEC plant is performed, based on recent experimental falling jet heat and mass transfer results.

The performance of an OC OTEC plant with falling jet evaporator-condenser is compared with the Westinghouse conceptual design that uses an open-channel evaporator and a surface condenser. Preliminary calculations indicate that falling jet heat and mass transfer, when applied in the proposed configuration, leads to a very simple and compact plant assembly resulting in substantial capital cost savings.

INTRODUCTION

Advantages of the open cycle Ocean Thermal Energy Conversion system, compared to the closed cycle (CC) were recognized in the early days of OTEC development. The OC OTEC system utilizes vapor flashed from a stream of warm seawater, used as a working fluid, that actuates a very low pressure turbine. By not using a secondary motive fluid, such as ammonia, requirements on construction materials become less stringent. Moreover, removal of the huge heat exchangers, which in the closed cycle transfer heat from warm seawater to a secondary fluid and from there to cold seawater with small temperature differences, reduces capital cost and eliminates the troublesome maintenance problems connected with the fouling of the heat exchanger tube surfaces.

The first successful demonstration of the technical feasibility of OTEC was performed by

Claude (1)** in 1930, with a 22-MWe open-cycle installation at Matanzas Bay, Cuba. More efforts on OTEC development were made by the French Government until 1958, but a large production plant was not constructed because OTEC could not compete with fossil fuel energy cost at that time. D'Arsonval's idea of harnessing the ocean thermal gradient (1882) had to wait almost a century before it would be considered economically feasible.

When renewed OTEC R&D efforts were undertaken in the United States in the wake of the Arab oil embargo of 1973, preference was given to the closed cycle concept for plants exceeding 25 MWe because the very large low pressure turbine required in an open cycle plant appeared to be beyond manufacturing capability. The current thrust of the DOE-sponsored program is devoted to implementing the closed cycle system.

THE WESTINGHOUSE OC OTEC CONCEPTUAL DESIGN

The open cycle concept, however, has not been abandoned. A recent Westinghouse study sponsored by DOE (2-4) developed an integrated 100-MWe OC OTEC plant concept that seems to compete with similar capacity contemporary closed cycle designs.

Sciubba (2) offered possible solutions in turbo-machinery, packaging, and housing of the OC OTEC plant that result in ease of fabrication and substantial cost savings. Westinghouse investigated turbines with up to 140-ft tip diameters that deliver 225 rpm providing up to 100-MWe net power under OC OTEC conditions. These turbines would be fitted with fiber reinforced composite blades and a disc of hollow plate or "spider-like" construction. The integrated design concept (shown in Fig. 1) centers around the vertical axis turbine (3). The open channel evaporator is disposed axisymmetrically around the turbine. Warm seawater enters the evaporator through a peripheral submerged

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**Numbers in parentheses refer to References listed at the end of the paper.

Figure 1. 100-MWe Open Cycle OTEC Plant Cross Section

sluice. Vapor flashed from the warm water flows along a bent path towards the turbine blades. The spent vapor leaving the turbine rotor is condensed in a surface condenser centrally located underneath the turbine. The warm seawater leaving the evaporator and the cold seawater leaving the condenser are rejected to the ocean through a common pipe.

This circular arrangement of the evaporator and condenser interfaces nicely with the turbine. Moreover, using the evaporator, turbine, and condenser housing walls as structural elements of the whole platform results in great savings.

The Westinghouse study showed a strong beneficial scale-up effect on plant cost. A comparison of estimated costs of closed cycle plants with the Westinghouse integrated design open cycle plant indicates a break-even point for a plant capacity of about 37 MWe. After that point the open cycle plant becomes more economical. The calculated cost of the power module, including power system and platform, for the 100-MWe CC plant fitted with 90-10 Cu-Ni condenser tubes is \$1825/kWe*; while the corresponding cost for the 100-MWe OC Westinghouse power module is \$1526/kWe.*

The cost breakdown of the 100-MWe Westinghouse design (4) is rather interesting. Two system components account for about three-fourths of the power module cost: the evaporation-condensation system (flash evaporator, surface condenser, and deaeration equipment, 38.7%) and

the concrete hull with accessories (34.1%). The remainder is distributed between the cost of turbine and generator (17%) and seawater pumps (10.2%).

The size of the two major systems components is determined by an engineering compromise (Fig. 1). The 100-m hull diameter of the 100-MWe plant is needed in order to accommodate a 50-m O.D. turbine and a 25-m wide torus to house the open channel flash evaporator. The enormous warm seawater flowrate proceeds in a radial inward direction along the open channel in a 9-in. thick layer at a rate of about $1.5 \text{ m}^3/\text{s}$ per meter channel width.

Open channel flash evaporation can be quite effective at elevated temperatures when a small flashdown temperature difference is accompanied by a substantial pressure drop, which easily overcomes hydrostatic suppression of boiling in the liquid layer and enables vapor to disengage from the bulk of the liquid. Therefore, it has been applied extensively in multistage flash evaporation desalination technology where it has been studied in detail.

At the relatively low temperatures characteristic of OTEC, however, flash evaporation in channel flow is hindered by hydrostatic suppression. Under such conditions a channel flow length of about 20 m was estimated as necessary to reach a reasonable approach to thermal equilibrium by the combined action of turbulent convection and surface evaporation. Thus, the large dimension of the open channel evaporator mainly was dictated by the method for achieving flash evaporation, which relies on turbulent mixing to bring up eventually all the warm water close enough to the free surface to enable vapor disengagement.

JET EVAPORATION AND CONDENSATION

Flash evaporation is much more effective at low temperatures by distributing the warm seawater in falling jets, since the hydrostatic suppression of boiling is eliminated.

Heat and mass transfer to laminar liquid jets has been adequately treated in the literature by theory and experiment (5-8). At low noncondensable gas concentration, a condensation heat transfer coefficient exceeding $200,000 \text{ kcal/h-m}^2\text{-}^\circ\text{C}$ has been measured in condensing steam on a thin water sheet (8). Although the specific mass flowrates and jet dimensions in the experiments reported in Refs. 5, 6, and 8 are not directly applicable to OTEC jet evaporator and condenser design, the results obtained in these works are important to the effectiveness of jet heat and mass transfer at low temperatures. They helped dispel the long-held

*1977 dollars.

belief (9,10) that an important contribution to heat transfer resistance during condensation is a phase interface resistance caused by the reflection of most of the molecules hitting the surface from the vapor phase side.

By kinetic theory the phase interface conductance in condensation is expressed by

$$h_{ev} = f[\lambda^2/(T_v V_v)] [M/(2\pi RT_v)]^{1/2} \quad (1)$$

where

h_{ev} = interface conductance

f = condensation coefficient, i.e., fraction of molecules hitting the interface from the vapor side that are actually condensed

λ = heat of condensation

T_v = vapor temperature

V_v = vapor specific volume

M = molecular weight

R = gas constant

A decrease in vapor temperature has a strong influence on h_{ev} because of the rapid increase in V_v associated with a temperature decrease. The variation of h_{ev} , according to Eq. 1, is shown in Table 1 for two values of f . If the value of f were close to 0.04, as it was generally believed to be (9,10), Table 1 shows that at temperatures relevant to OTEC the interface resistance would have been the controlling factor in interphase heat and mass transfer. The expected improvements to heat and mass transfer performance using a jet evaporator or condenser would have been very limited.

Table 1. VARIATION OF CONDENSATION COEFFICIENT WITH TEMPERATURE

T (°F)	T (°C)	h_{if} (kcal/h-m ² -°C)	
		$f = 1.0$	$f = 0.04$
110	43	2,202,00	38,500
90	32	1,382,00	24,200
70	21	742,000	13,000
50	10	400,000	7,000
30	-1	210,000	3,700

The results of Refs. 5 and 8-10 have shown conclusively that the values of the coefficients of evaporation and condensation are close to 1. The interface resistance to heat and mass

transfer, therefore, is relatively unimportant even at the low OTEC temperatures. At low noncondensable gas concentrations in the vapor phase the controlling resistance resides in the liquid phase and very high heat transfer rates can be expected under turbulent jet conditions.

A theoretical treatment of turbulent jet heat and mass transfer by Kutateladze (12) and Isachenko et al. (13) assumes the vapor phase has no resistance to heat transfer and there is a constant eddy diffusivity ϵ . This has limited practical value since it presently is difficult to obtain a good estimate for ϵ to use in such calculations. Besides, ϵ is far from being constant along the jet.

Some qualitative statements can confidently be made about evaporation and condensation at turbulent jet flow. In the case of a pure vapor these transport phenomena are governed by turbulent mixing in the liquid phase at the temperatures in our application. The evaporation and condensation heat transfer coefficients h_e and h_c , respectively, averaged over the jet length therefore increase with jet velocity at the nozzle exit section. A strong variation of the local coefficients h_e and h_c with distance from nozzle exit section can be expected because of turbulence decay downstream along the jet.

The condensation heat transfer coefficient is much reduced in the presence of noncondensable gas in the vapor. This is because a buildup of a noncondensable gas concentration gradient at the condensation interface. No such effect is expected at the evaporation interface.

The primary author recently performed exploratory experiments at the Department of Aeronautical Engineering, Technion, Haifa, on heat and mass transfer between adjacent coplanar warm and cold falling water jets under temperatures and specific flowrates similar to those expected in OTEC. The vapor flowed from the evaporation interface to the condensation interface crosscurrent in the water flow direction.

For planar jets falling a distance of 0.85 m from 4-mm wide slots at a specific flowrate of 37.2 ton/h-m, a total mean evaporation-condensation heat transfer coefficient close to $\bar{h}_{e-c} = 60,000$ kcal/h-m²-°C was obtained at a warm inlet temperature of 26°C and a flash-off temperature drop of 1.5°C. Within the accuracy of the temperature measurements the heat transfer resistance was equally distributed between the evaporation and the condensation sides. The concentration of noncondensable gas in the bulk of the vapor phase was maintained at about 200 ppm. When the experiment was repeated with jet lengths of 1.7 m, all other parameters being kept constant, $\bar{h}_{e-c}$ fell to approximately half its previous value. This demonstrated

vividly the fast decay of heat and mass transfer effectiveness downstream of the jet. The downstream half of jet length in the second experiment apparently was not contributing significantly to the transfer process.

Jet heat and mass transfer has found some successful large-scale application in recent years in the power generating industry. Bakay and Jaszay (13) outline the calculations in designing turbulent flat jet condensers with vapor crossflow used as part of an indirect air cooling system in a number of power plants in eastern Europe.

FALLING JET EVAPORATOR CONFIGURATION

Despite the very high heat transfer coefficients attainable with liquid jets, a crosscurrent jet configuration, geometrically similar to the one described in Ref. 13, does not lead to an acceptable evaporator or condenser design for large open cycle application. The vapor flow velocity required to prevent excessive liquid drop entrainment limits this.

For comparison consider the open channel evaporator in the 100-MWe Westinghouse conceptual plant using a crosscurrent falling jet evaporator of comparable capacity in which water evaporates from jets falling from radially disposed slots and flows across the evaporator periphery towards passages that lead to the turbine.

In the case of the Westinghouse open channel evaporator the cross section A_1 available to the vapor flow leaving the warm water layer is

$$A_1 = (\pi/4)[1 - (D_1/D_o)^2] D_o^2 \quad (2)$$

where D_1 and D_o are the inner and outer diameter of the evaporator, respectively.

Figure 2. Jets Falling from Slotted Tubes of Circular Cross Section

For a crosscurrent falling jet evaporator with the same outside diameter the cross section area A_2 of the vapor leaving at the evaporator periphery is given by

$$A_2 = \pi D_o H_j \quad (3)$$

where H_j is the length of the falling jets.

For $A_1 = A_2$ the required jet length increases with plant diameter:

$$H_j = (1/4)D_o[1 - (D_1/D_o)^2] \quad (4)$$

and for the 100-MWe plant size this leads to $H_j = 19$ m. This jet length represents an excessive loss of hydraulic head, since a rough calculation shows that the energy required to raise the warm water flow by 1 m equals about 5% of plant output.

This difficulty is circumvented in a countercurrent falling film evaporator layout as illustrated in Figs. 2-4. In this layout warm water is distributed through a number of radial manifolds to a set of circular coaxial slotted tubes disposed in a horizontal plane. Vapor flashed from the circular falling jets flows a short distance countercurrent to the falling jets and escapes into the plenum chamber above the slotted tube plane. The cross-sectional area available to the vapor in this configuration is given by Eq. 2, just as for the open channel evaporator. The circular cross section of the slotted tubes in Fig. 2 can be replaced by an elliptic cross section (shown in Fig. 3) to reduce vapor pressure loss associated with slotted tube cross-section form drag.

FALLING JET CONDENSER CONFIGURATION

Circular slotted tubes arranged in a coplanar concentric array, as used in the evaporator conceptual design (Fig. 4), can also be used for a falling jet condenser. In this case, radial manifolds distribute cold seawater, supplied by the cold water pipe, to the circular slotted

Figure 3. Jets Falling from Slotted Tubes of Elliptic Cross Section

tubes. The spent vapor flows downward from the turbine through the space between the slotted tubes concurrent with the jets and condenses by direct contact with the cold seawater.

The effectiveness of the direct contact condenser is very sensitive to the concentration of noncondensable gas (n.c.g.) in the condensing vapor. The n.c.g. is entrained by the vapor towards the condensation surface. In steady state an equilibrium sets in between the convective transportation of n.c.g. towards the liquid-vapor interface and the diffusion of n.c.g. away from the interface driven by the n.c.g. concentration gradient in the vapor phase.

In a study on vapor condensation by direct contact with a flowing cold water film (14) samples of vapor were withdrawn from positions at different distances from the film surface and analyzed for their n.c.g. content. To maintain constant water exit temperatures and constant

Figure 4. Falling Jet Evaporator Configuration

Figure 5. Non-Condensable Gas Distribution Near Vapor Condensing Liquid Surface

condensation rates it was necessary to maintain vacuum pump suction from the test enclosure to remove traces of n.c.g. that leaked into the system. Under steady state conditions a strong variation of n.c.g. concentration w with distance z was found in the vicinity of the condensation surface. The experimental results (Fig. 5) followed the relation

$$w = a \exp(-bz) \quad (5)$$

In an extended experiment, suction was switched from a port situated in the water exit plane 100 mm from the water surface to a location close to the surface without changing the gas mixture suction rate. The rate of n.c.g. extraction was temporarily augmented by the increased n.c.g. concentration at the new suction port, and the heat and mass transfer steady state was thereby disturbed. The system drifted slowly towards a new steady state at an increased condensation rate. The n.c.g. concentration decreased at a fixed location in the vapor phase. Suction was then switched to a location close to the surface of the inflowing cold water, again, without changing the gas mixture suction rate further improving the condensation rate.

Such experiments demonstrated the importance of locating the n.c.g. extraction port at a strategic position, near the entering cold water surface where n.c.g. concentration is highest.

Figure 6. Detail of Condenser n.c.g. Suction Arrangement

The application of this conclusion to the removal of n.c.g. from the falling jet condenser is illustrated in Fig. 6. Segments of bent tubes are slipped on the water carrying slotted circular tubes and positioned coaxially with them by appropriate end O-rings or grommets.

These riding tube segments are slotted on their lower side, and equidistant small diameter suction tubes lead from their upper side to manifolds connected to the evacuation pump. Vapor rich in noncondensibles thus is extracted from the condenser along strips near the cold water surface at the jet entrance section where the water temperature is lowest.

THE SERI EVAPORATION-CONDENSATION TEST LOOP

A test loop now under construction at the Solar Energy Research Institute (SERI) Heat and Mass Transfer Laboratory will enable experimentation with a module of a falling jet evaporator-condenser in which specific flowrates and temperatures will duplicate those in a large scale OC OTEC plant (Fig. 7).

In a vacuum enclosure a stream of warm water is distributed into four planar parallel falling jets by four slotted tubes fed through a manifold. Similar cold water jets are within the vacuum enclosure parallel with the warm water jets. Vapor flashed off the warm water jets flows from the evaporation region through the open intervals between adjacent slotted tubes towards the condensation region where it is condensed by direct contact with the cold water jets.

A pump circulates warm water through a heat exchanger where the water temperature is restored by adding heat; then it goes back to the warm water distribution manifold in the test cell.

Figure 7. Heat and Mass Transfer Lab Piping Schematic

The cold water passes through a chiller before returning to the cold water distribution manifold.

The warm and cold water pumps have a 750 gpm flowrate capacity. With heater and chiller capacities of 10^6 Btu/h, evaporation-condensation experiments can be performed when a temperature drop of 6°F is attained in warm water jets with a total flowrate of 330 gpm.

The test cell can easily be refitted to perform evaporation and condensation experiments with various other jet or film configurations. Figure 8 shows the evaporation-condensation test cell. A closeup of the evaporator slotted tubes is shown in Fig. 9.

EVAPORATOR AND CONDENSER SIZING FOR A 100-MWe OC OTEC PLANT

Correct sizing of a falling film evaporator and condenser requires exact data on the amount of n.c.g. to be extracted from the condenser region and on jet evaporation and condensation heat transfer coefficients within the range of specific flowrates and temperatures relevant to OTEC. Research programs have been initiated to determine such data.

A rough estimate of the falling jet evaporator and condenser dimensions for a 100-MWe OC OTEC plant was obtained assuming that in the range of specific flowrates of 30 to 60 $\text{m}^3/\text{h-m}$, jets of falling length $H_j \leq 0.85$ m produced by 1/4-in. width slots will exceed the following values of heat transfer coefficient:

$$\overline{h_E} = 90,000 \frac{\text{kcal}}{\text{h-m}^2\text{-}^\circ\text{C}}$$

$$\overline{h_C} = 80,000 \frac{\text{kcal}}{\text{h-m}^2\text{-}^\circ\text{C}}$$

The nominal values of warm and cold seawater flowrates and temperatures given in the Westinghouse report (3) were used as a data base:

Warm seawater flowrate	$2,947 \times 10^6 \text{ lb}_m/\text{h}$
Warm seawater inlet temperature	80°F
Warm seawater outlet temperature	75.20°F
Evaporator saturation temperature	74.20°F
Cold seawater flowrate	$2,475 \times 10^6 \text{ lb}_m/\text{h}$
Condenser saturation temperature*	46.69°F

Cold seawater outlet temperature 45.69°F

Cold seawater inlet temperature 40.00°F

A 2.5-in. nominal size was chosen for the circular slotted tubes. The evaporator inside diameter was equated to the turbine diameter (50 m), and the distance between consecutive

Figure 8. Evaporation-Condensation Test Cell

Figure 9. Evaporator Slotted Tubes

Table 2. FALLING JET EVAPORATOR

H_s		$\frac{\dot{w}}{h-m}$	Re_s	H_j (m)	$H_j + H_s$ (m)	n	D_o (m)	(D_o/D_{ow})
ft	m	$\left(\frac{m^3}{h-m}\right)$						
1	0.3	34.4	21,200	0.34	0.65	182	86.8	0.75
2	0.6	48.6	30,000	0.47	1.08	137	77.6	0.60
3	0.9	59.6	36,800	0.58	1.50	116	73.4	0.54

Table 3. FALLING JET CONDENSER

Δ		n	$\frac{\dot{w}}{h-m}$	Re_s	H_j (m)	H_s (m)	$H_j + H_s$ (m)
in.	mm						
4	102	295	34.7	12,800	0.82	0.31	1.13
3-3/4	95	315	32.5	12,000	0.77	0.27	1.04
3-1/2	89	337	30.4	11,200	0.72	0.23	0.95

evaporator tubes was fixed at $\Delta = 4$ in. The hydraulic head loss through the slots was varied in the range $1 \leq H_s \leq 3$ ft. The results are shown in Table 2 where $\dot{w}$ is the specific flowrate, Re_s is the Reynolds number at the slot, n is the number of tubes, D_o is the outer diameter of the falling jet evaporator, and D_{ow} is the outer diameter of the Westinghouse evaporator.

The outside diameter of the condenser is determined by the turbine size, while the inside diameter is limited by the diameter of the cold water pipe. Values of $D_o = 50$ m and $D_i = 20$ m were assumed.

The condenser tubes can be spaced more densely than the evaporator tubes since the distance between adjacent condensation jets is not limited by the droplet entrainment problem. The slot hydraulic head loss H_s and the jet falling distance H_j , therefore, were determined for tube spacings in the range $3.5 \leq \Delta \leq 4$ in. The results are shown in Table 3.

CONCLUSION

Replacement of an open channel evaporator by a falling jet evaporator leads to a very substantial reduction in the outside diameter of the hull in an OC OTEC plant. Since the hull of the

more than one-third of the plant cost, the use of a falling jet evaporator can save 8% to 15% of the capital cost. This estimate assumes that the cost of the hull grows roughly in proportion with its horizontal projected area.

A similar cost savings can be expected by replacing the expensive surface condenser (25% of the cost of the Westinghouse plant) with the falling jet condenser. In this case, however, the plant will not produce desalted water as a by-product.

An additional advantage obtained by introducing the falling jet condenser is the 18% increase in the temperature drop across the turbine with a corresponding increase in power output. The total loss of hydraulic head associated with jet formation and jet fall is 1 m.

The estimates do not represent an optimized plant design. Plant optimization could be confidently undertaken only when more reliable experimental data on falling jet heat transfer become available.

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