Analysis of. Shell and Tube Heat Exchangers for Use in Ocean Thermal Energy Conversion Systems

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Considerations in the design of shell and tube heat exchangers specific- ally intended for use in Ocean Thermal Energy Conversion (OTEC) systems are presented. Factors such as selection of working fluid variations in thermodynamic and transport properties, seawater velocities, excess temperature differentials, and approach temperature of the evaporator are discussed as to their effect on the overall performance and economic return.

A minimum wall shear stress in the seawater tubes which has been demonstrated as an effective deterrent to the attachment of biofoulers is shown to be a necessary constraint on the system.

Uncertainty in the system design as to the actual length and diameter of the tubes is shown to have possible adverse effects on the economic return. These adverse effects are seen as geometry configurations, which do not permit productive operation of the system subject to the design constraints. Such economic infeasibility of certain geometries is not predicted .by analyses, which neglect the effects of variations in the thermal, and trans- port properties and heat transfer coefficients.

Description of the Problem

i. combined heat transfer modes:

- conduction
- convection
- phase change

ii. integral constraint:

- integral of heat flux along any single tube is equal to the change in enthalpy of the seawater

iii. wall shear stress constraint:

- the minimum tolerable shear stress to 'discourage' the attachment of biofoulers must be maintained

iv. a boundary value problem:

- conditions of the working fluid at the at the inlet and exit of the condenser and evaporator are defined by the Rankine cycle

v. an initial value problem:

the inlet temperature of the seawater is assumed to be constant

vi. a coupled system:

seawater pumping effects both demand and available temper- - a re difference thus altering the Rankine cycle as well as output and economic return

vii. only small temperature differences available:

many so called 'small effects' may not be negligible such as slight subcooling or small temperature rises through the recirculating pump

Degrees of .System Freedom

i. longitudinal:

change in temperature of seawater and working fluid along each tube

ii. radial:

a. 3-mode thermal circuit
b. phase change heat transfer coefficient depends on available temperature difference and local property values

iii. depth:

a. non-isothermal evaporator
b. condenser is assumed to be isothermal
c. inlet temperature of seawater is assumed to be constant while exit temperature will most definitely change with depth
d. inlet velocity and tube diameter is assumed to be constant, however further optimization might be possible through varying tube diameter and inlet velocity in the evaporator to take advantage of the subcooling

iv. steady state is assumed

A Combined Finite Element / Finite Difference Method

L longitudinal elements (IVP)
M depth elements (BVP)
1 governing equation (conduction)
1 integral constraint (the first law)
The heat transfer coefficients are unknown until the temperature distribution is found, which in turn is a function of the heat transfer coefficients.

Initialization:

The heat transfer process is originally assumed to be totally conductive and the temperature profiles are assumed to be linear

Iteration:

1. at each step in the iteration process l m by m square matrices must be solved for the temperature distribution
2. for each new temperature distribution a new set of heat transfer coefficients are calculated
3. a new temperature distribution is calculated and the process is repeated

Further reduction of effort is accomplished by using unequally spaced elements (based on the abscissas of Gaussian quadrature) thus acquiring greater accuracy with fewer elements and less inversion work (even though the matrix is not banded)

Solution is possible; but still too expensive!

Development of an approximate technique based on finite element i finite difference calculations

- Arithmetic average of heat flux shows typical errors of +33%
- Use of LMTD to predict the average heat flux shows typical errors of -25%
- What about some integral mean average based on a non-linear distribution?
- Heat transfer coefficients are assumed to be proportional to some ΔT^n
- n is a function of fluid properties as well as the heat transfer correlation (mode of phase change) and is numerically calculated to show the least square error in the overall heat transfer coefficient between the temperatures experienced in each case
- Weighting the heat flux at the inlet and exit by ΔTn and integrating results in less than 1.5% disagreement with the finite element i finite - difference solution for each case investigated
- Coupling of the system r effects in the heat exchangers are felt throughout the entire system
- The result? Drastic changes in the economic return factor (watts per dollar of original investment in the heat exchangers) may result from only small changes in the system
- Why? Typical efficiencies are less than 1% the net power output is so small compared to the total power 'handled' by the system that even small changes in the operating point of the heat exchangers can result in large changes in the power output even though the investment cost does not undergo drastic changes

Convergence?

- Successive substitution does not converge due to the nonlinear nature of the problem

- Overshoot may be in excess of 100%
- Relaxation reduces oscillation; but is slow to converge
- Method of nearest neighbor averaging in itself becomes a matrix problem in 2-dimensions

Numerical Prediction of an Asymptotic Limit?

- All previous heat transfer coefficients are stored in an auxiliary matrix and used to calculate successive approximations to the Cezaro limit
- Convergence is achieved—and very rapidly
- The process consists of taking appropriately weighted sums of previous calculated values

$$\Lambda^{\infty} \sim \sum_{1}^{N} \Omega(N,n)\lambda_n + R_n \tag{1}$$

Four Level Optimization Scheme

1. Adjustment of the velocities of the seawater in the evaporator and condenser which were assumed to be different
2. The ratio of the temperature difference in the condenser to the approach temperature difference in the evaporator
3. The ratio of the temperature difference available to the Rankine cycle to that available to the heat transfer process
4. The ratio of the cost of the evaporator to that of the condenser

Suggestions for Further Investigation

- Choice of working fluid
- Experimental determination of actual growth rates and attachment rates of biofoulers over long periods of time
- Effects of turbulence on biofouler attachment different diameters and/or length of tubes in the heat exchangers
- Use of different diameters in the same heat exchanger to take the full advantage of the liquid subcooling
- Experimental determination of falling film heat transfer coefficients

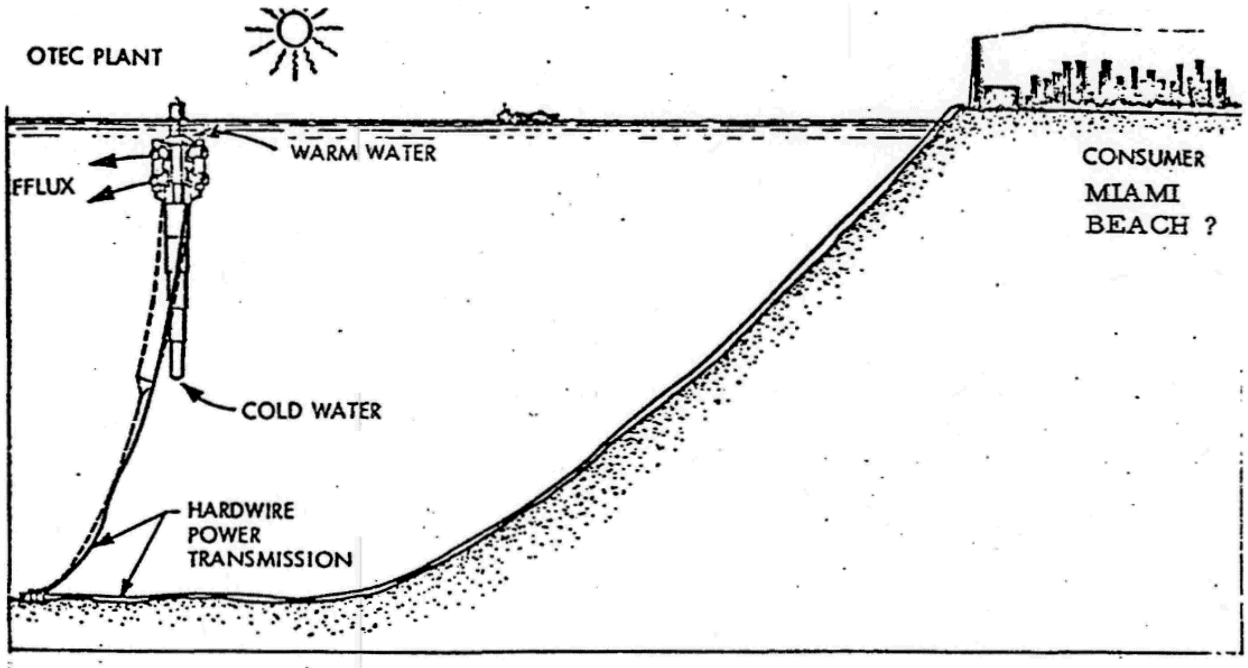

Figure 1. Typical Installation with Submerged Power Line to Shore

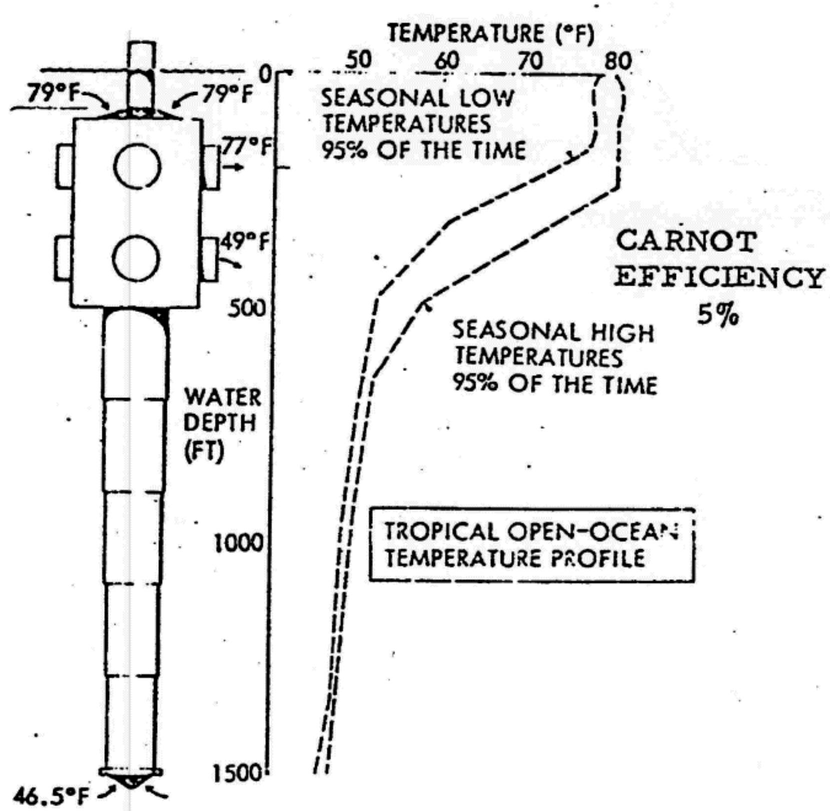

Figure 2. Illustration of Scale and Temperature Gradient

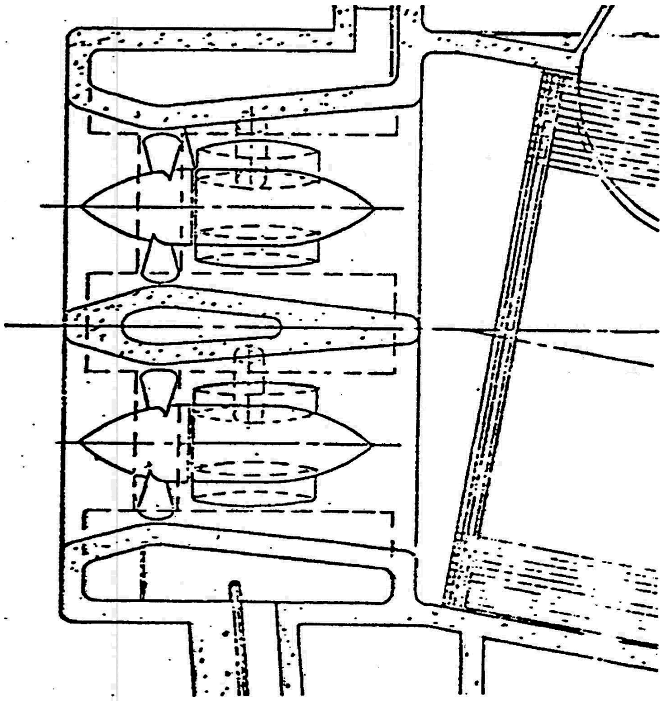

Figure 3. Possible Arrangement of Sea Water Pumps

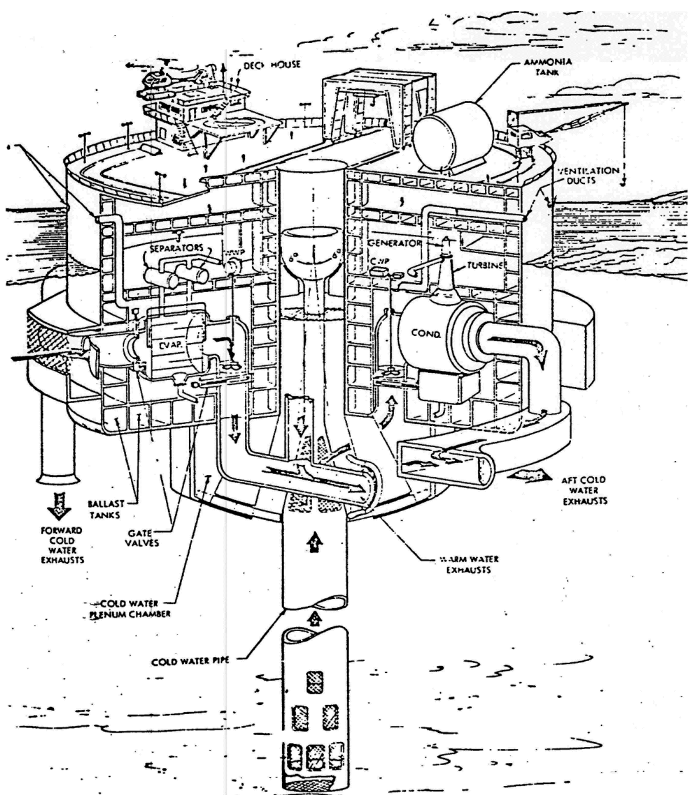

Figure 4. Total Power Station Concept

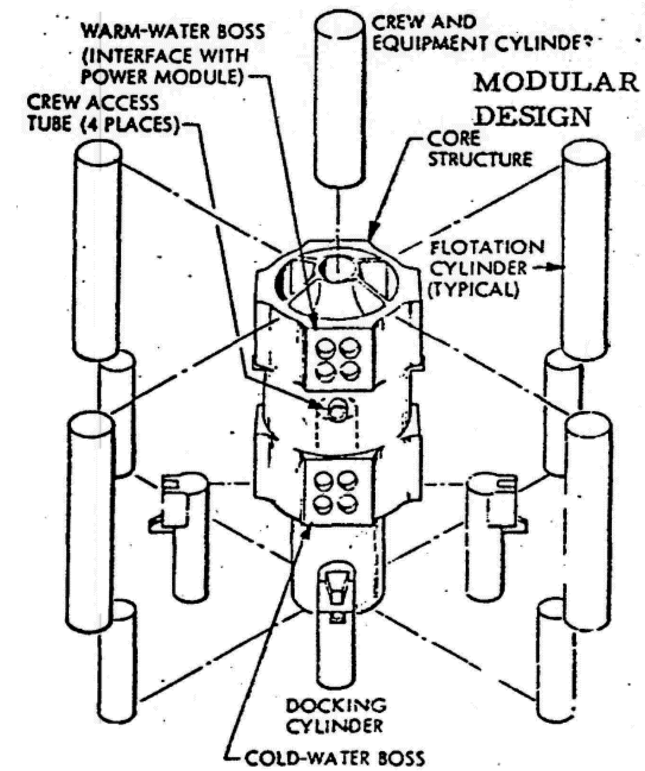

Figure 5. Possible Modular Design

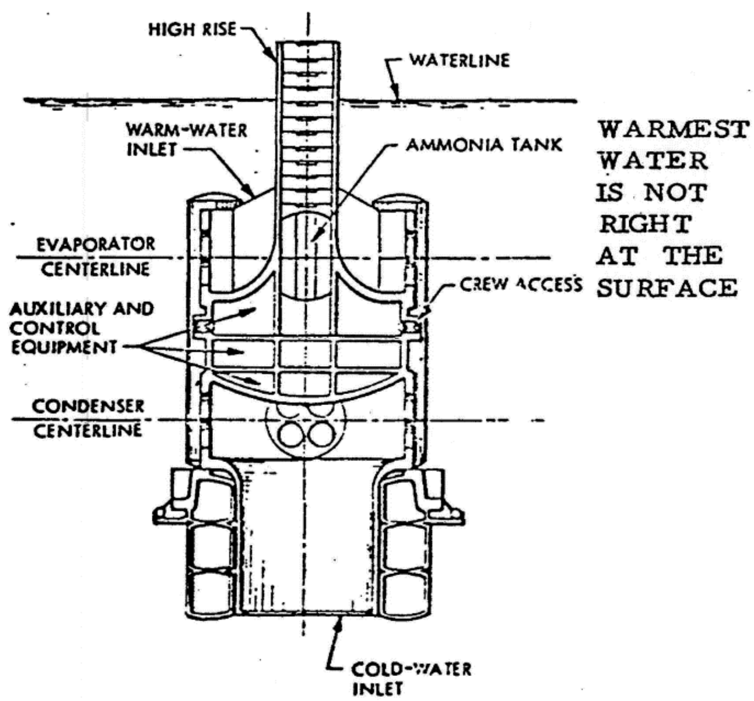

Figure 6. Cold Water Riser

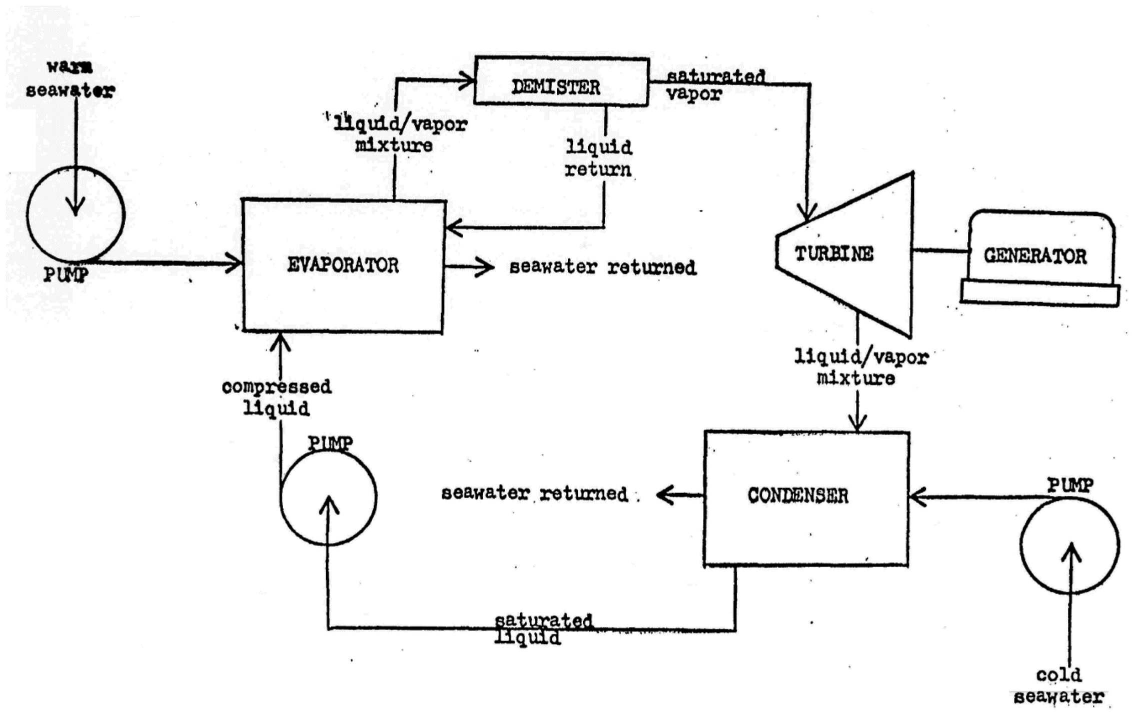

Figure 7. Schematic of Rankine Cycle

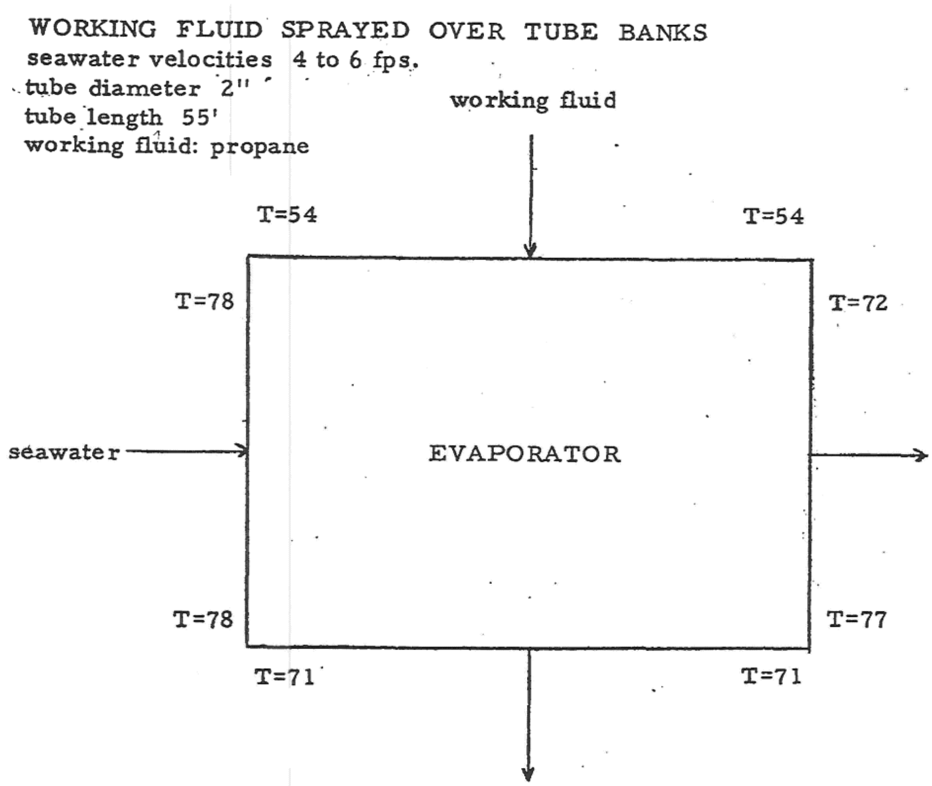

Figure 8. Schematic of Evaporator

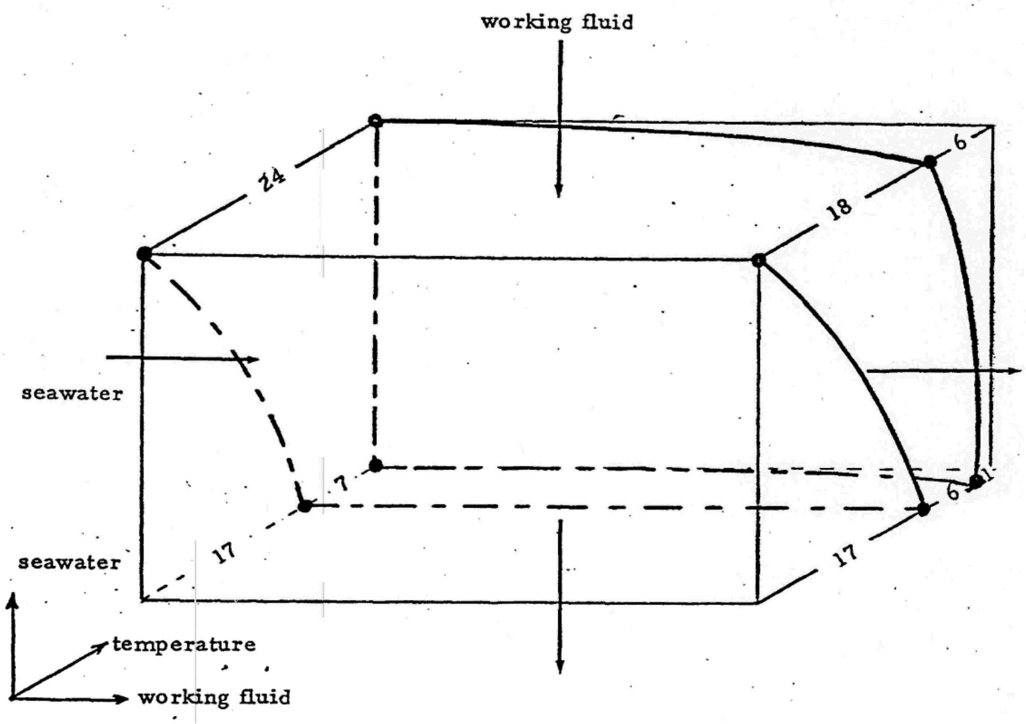

Figure 9. Computational Element

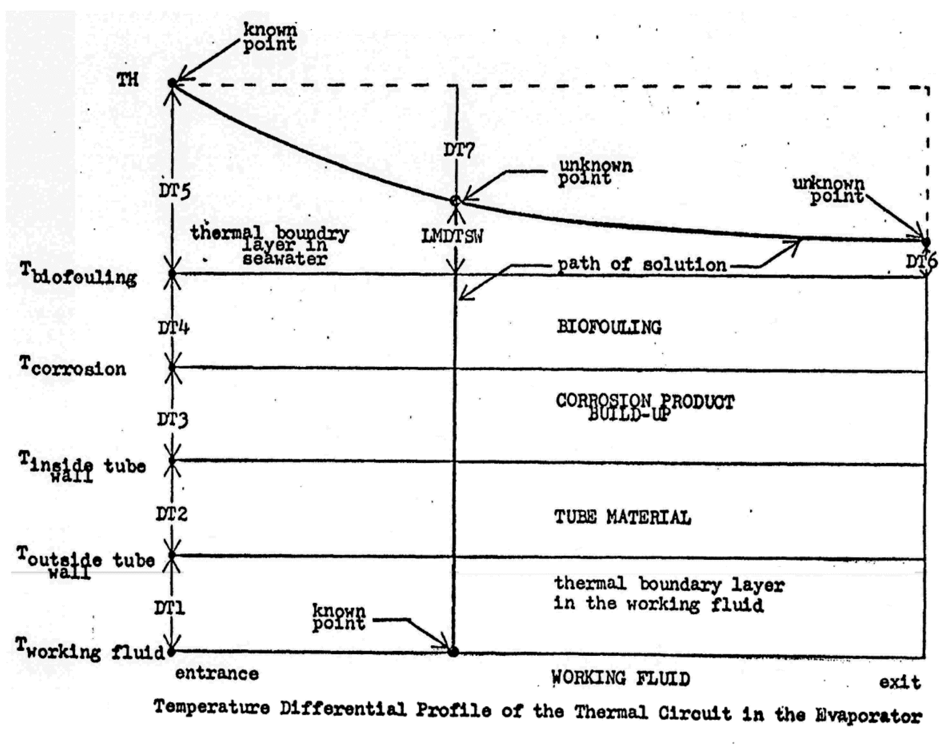

Figure 10. Thermal Circuit

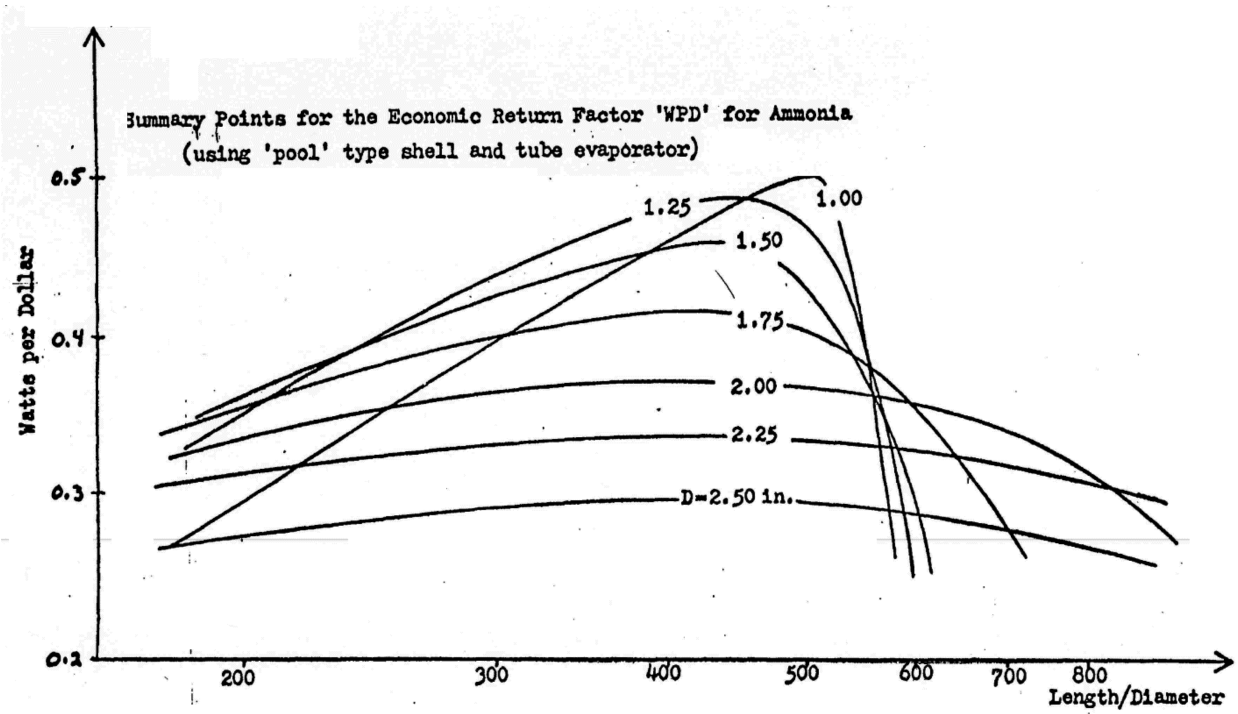

Figure 11. Results for Ammonia as the Working Fluid (pool)

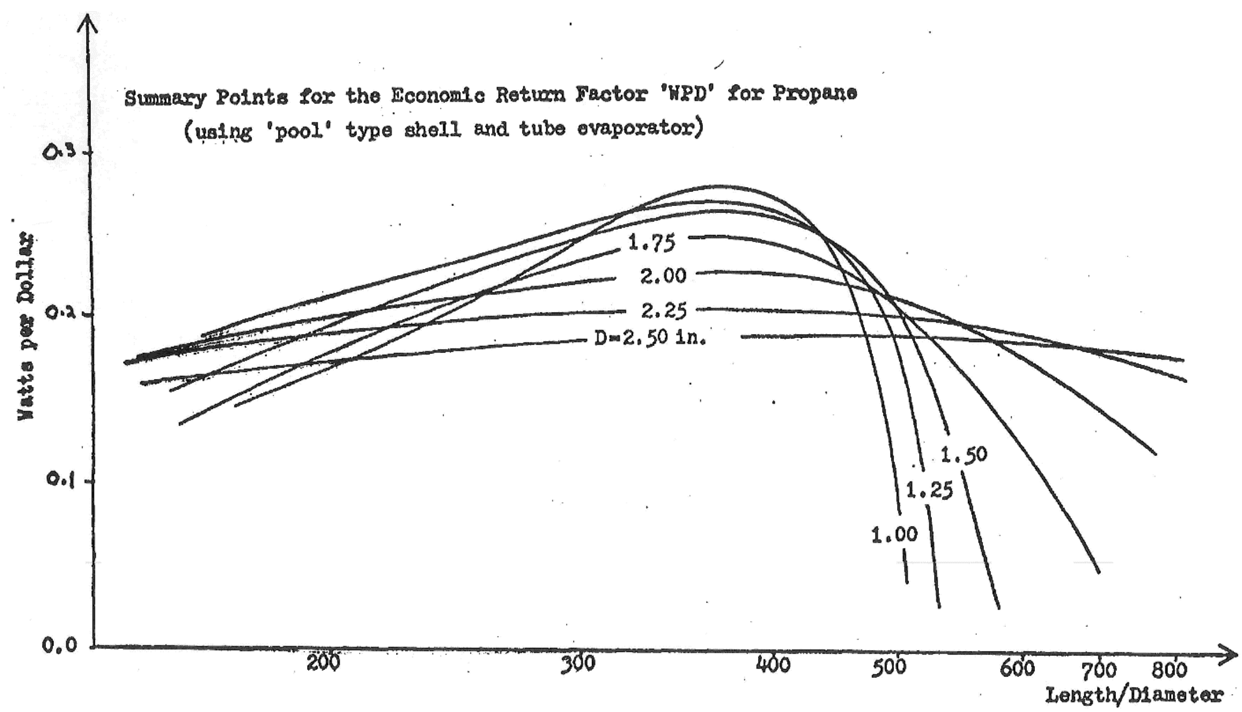

Figure 12. Results for Propane as the Working Fluid (pool)

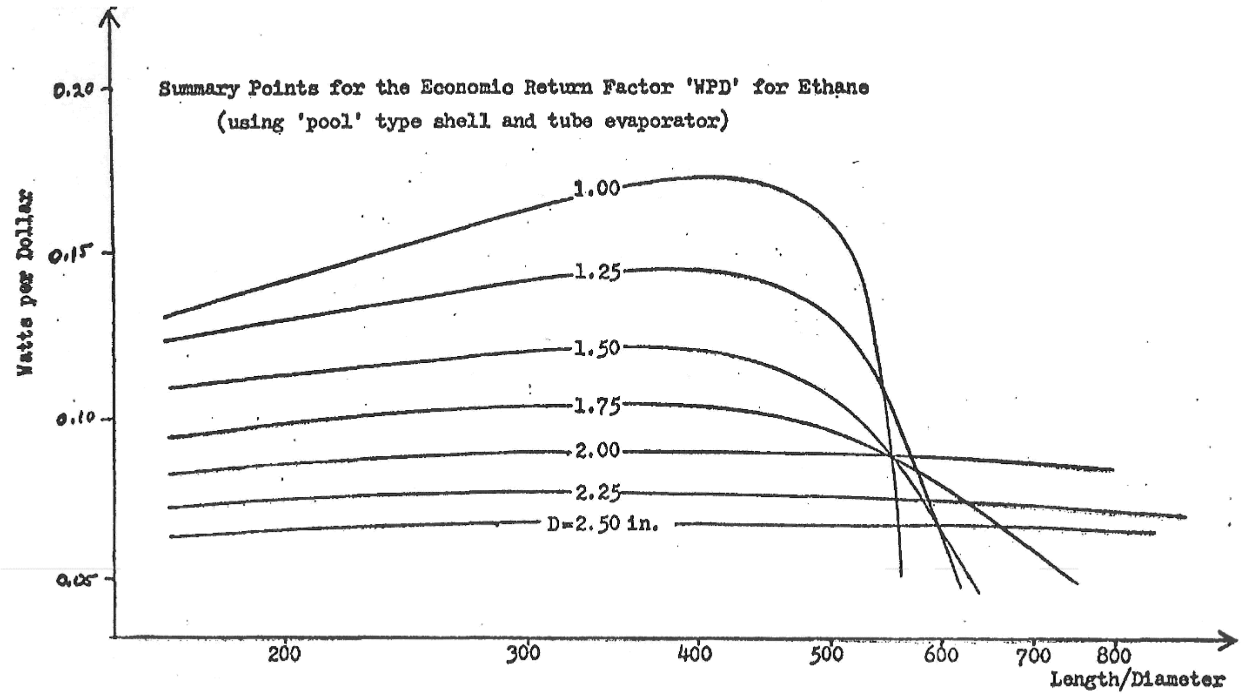

Figure 13. Results for Ethane as the Working Fluid (pool)

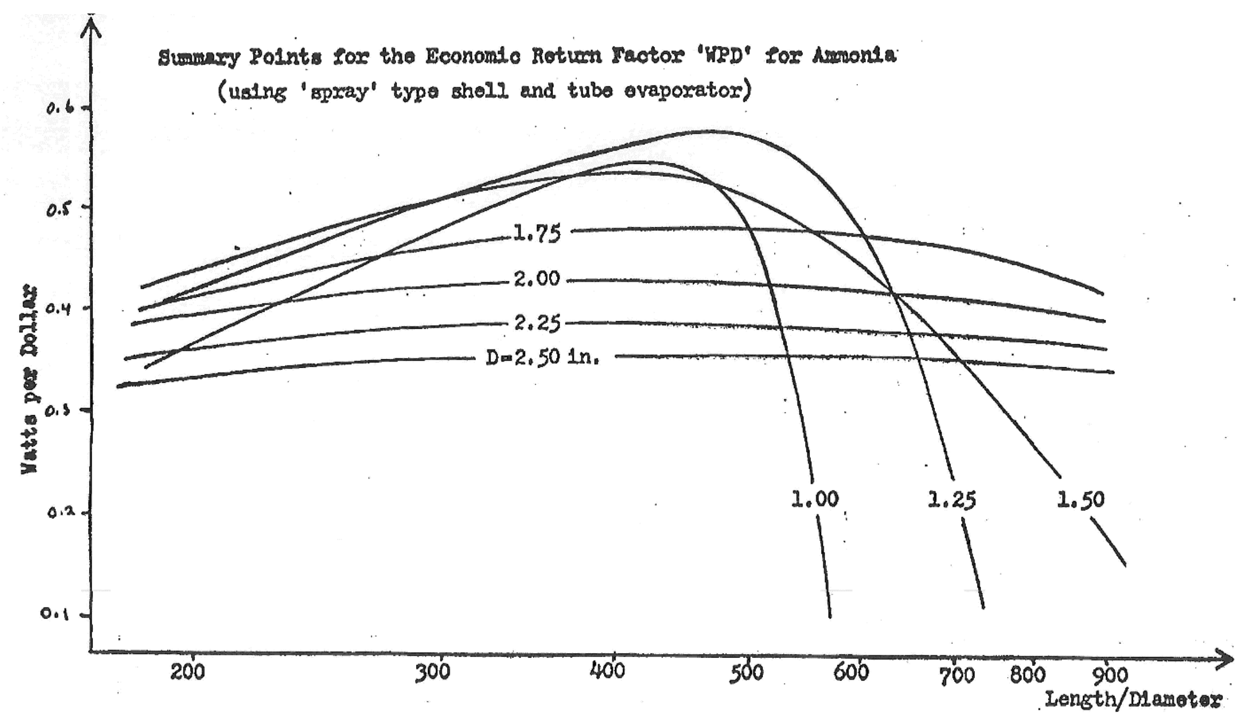

Figure 14. Results for Ammonia as the Working Fluid (spray)