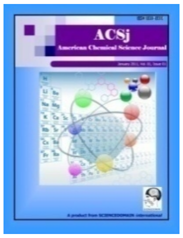

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Economic Design of OTEC Power Plant with Concurrent Production of Desalinated Water – A Case Study

C. M. Narayanan[1*]

[1]*Department of Chemical Engineering, National Institute of Technology, Durgapur 713209, India.*

Author's contribution

This whole work was carried out by the author CMN.

Original Research Article

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ABSTRACT

Ocean Thermal Energy Conversion (OTEC) power plants offer a green source of renewable energy. Since India is a tropical country and a peninsula, the prospects of OTEC power generation are extremely bright in India. Among the three modes of operation (open cycle, closed cycle and hybrid cycle) of OTEC system, the hybrid mode is most promising. However, one of the chief technical obstacles in OTEC power plant design is that since the temperature difference driving force available is of the order of 10-15ºC only, the size of the heat exchanger (evaporator / condenser) required becomes exorbitantly large. The use of variable area design, developed by the author and his co-workers, has been recommended in this connection. Such a design provides substantial increase in heat transfer coefficient (350 to 450% increase) with insignificant increase in the associated pressure drop penalty (118 to 120% increase). The required size of the heat exchangers thus gets reduced tremendously, while the operating cost does not increase materially, thereby making design and operation of OTEC power plants economical and cost- effective. The performance characteristics of such heat exchangers (Variable Area Heat Exchangers or VAEs) are discussed in detail in this paper. Further, in the hybrid mode of operation of OTEC system, low pressure steam is produced by the flash evaporation of sea water and this steam is used as heating fluid in the evaporator (of variable area design) to evaporate the working fluid (ammonia, freon). The condensate from this exchanger thus forms desalinated water, which constitutes a valuable by-product

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**Corresponding author: Email: cmn_recd@yahoo.co.in;*

of the process. Apart from generating clean electric power around the clock (without consuming any valuable raw material), this power plant thus produces several gallons of desalinated water also per day.

Keywords: OTEC power plants; desalinated water; variable area heat exchangers; hybrid cycle.

1. INTRODUCTION

OTEC power plants constitute one of the most promising sources of green energy. They generate electric power by utilising the temperature difference between the surface and the depths of tropical oceans and the interconnected seas. The power generation is fully renewable and environmentally clean. There is no fuel consumption, no disposal of solid residues, no emission of harmful gases, no adverse effect on ecology and environment. Above all, these power plants permit simultaneous production of desalinated water as a by product (when operated in the open cycle or hybrid cycle mode).

OTEC power plants may be operated in three modes, such as the open cycle, closed cycle and the hybrid cycle modes of operation. Among these, the hybrid cycle is most promising. This cycle (sketched in Fig. 1) involves the following components:

a) Flash drum (flash distillation still)
b) Evaporator (Shell and tube HE)
c) Turbo-generator
d) Condenser

A systems study on the operation of such hybrid cycle OTEC power plants has been reported by Panchal and Bell [1]. The warm sea water from the ocean surface (at 30-35ºC) is pumped into the flash drum in which the pressure is maintained as low as 25-26 torr. As a result, the sea water flashes, thereby generating low pressure steam. This steam is used as the heating fluid in the evaporator to vaporise the working fluid (for example, ammonia). The high pressure ammonia vapours so produced are used to drive the turbo-generator and thereby generate electric power. The steam condenses in the evaporator transferring the latent heat to liquid ammonia. The condensate discharged from the evaporator is thus potable water (desalinated water). Since the amount of sea water flash-vaporised in the flash drum is quite large (to attain a high rate of power generation), the rate of production of desalinated water is also quite large. The spent ammonia vapours are condensed to liquid ammonia in the shell and tube condenser, where sea water from the depth (which is at a temperature of 10-15ºC) is used as the cooling fluid. The liquid ammonia is then recycled back to the evaporator. Cogeneration of desalinated water in this way has the following distinct advantages:

1) Desalinated water is obtained as a by product and hence has no separate manufacturing cost.
2) Clean electric power is generated side by side and this provides improved cost efficiency.
3) Since large amount of sea water is handled in the power plant, the rate of production of desalinated water is substantially large.
4) The need for separate process equipment(s) for desalination is fully eliminated.

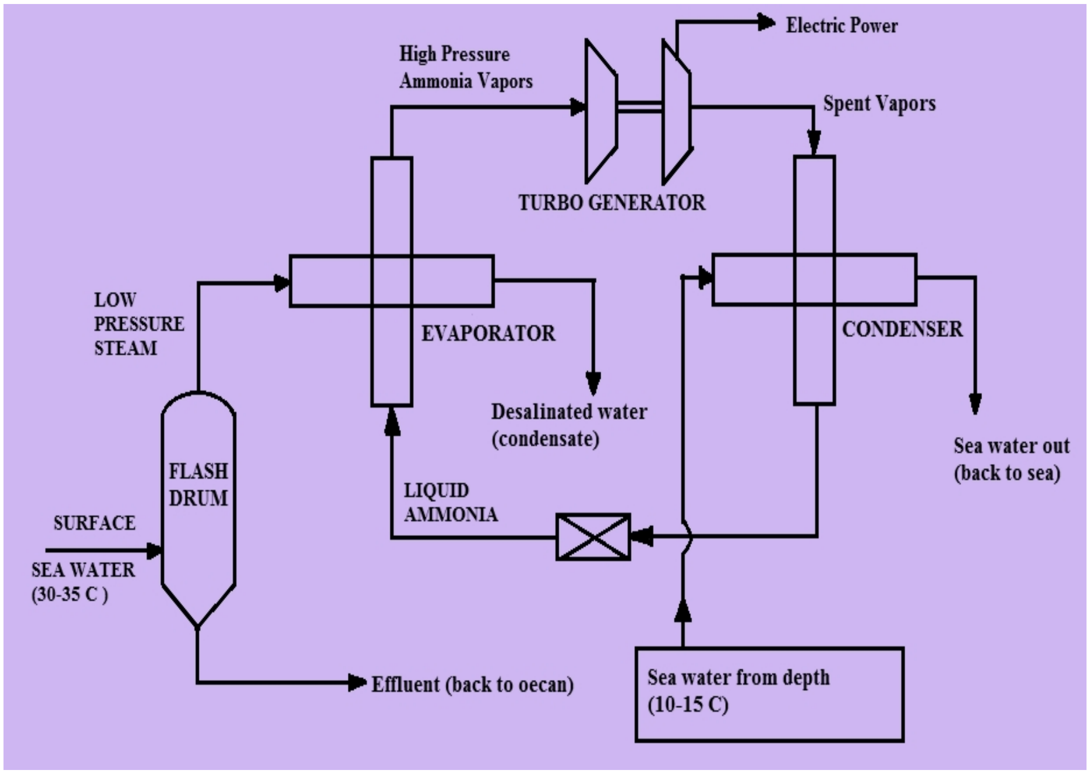

Fig. 1. Schematic of Hybrid Cycle OTEC System

2. TECHNICAL PROBLEMS

In the above power plant, both evaporator (where the working fluid is vaporised) and the condenser (in which the spent vapours of the working fluid are condensed) are of shell and tube design. For a large power output (for running a sufficiently large capacity power plant), the heat transfer surface requirement of these two heat exchangers become substantially large. The problem is more acute with the condenser. This is because the temperature change of the coolant employed (sea water from a depth of around 1.0 km) is not more than 20ºC (sea water enters at around 10-15ºC and leaves at 30-35ºC). Much higher outlet temperature of sea water is not advisable as it leads to significant corrosion problems (sea water is corrosive both chemically as well as biologically). The sea water flow rate cannot be increased beyond a limit, since that would make pumping cost prohibitive. In the evaporator, the heating fluid being low pressure steam, the heat transfer effectiveness is low. Since surface sea water enters the flash drum at around 30-35ºC, unless the operating pressure of the drum (distillation still) is kept fairly low, the rate of steam generation cannot be maintained satisfactorily high. Consequently, steam generated is at low saturation pressure and it condenses at comparatively low temperature on the surfaces of exchanger tubes, leading to decreased heat exchange. In order to obtain large quantities of high pressure ammonia vapour therefore, the heat transfer surface is to be augmented substantially. Typically, these heat exchangers demand as many as 5000 tubes of 25mm OD (outer diameter) and 5.0 m length, even when the capacity of the power plant is as low as 50 MW. Construction, installation and maintenance of such giant size heat exchangers become a formidable task both technically as well as economically. For the economical operation of an

OTEC power plant therefore, improved design of these heat exchangers (both evaporator and condenser) would have to be employed to bring down the heat transfer surface requirement. Heat exchangers with tubes coated with high heat flux porous material have been recommended by McGowan and Connell [2,3]. However, this technique provides heat transfer enhancement by 70% (1.7 times) only and the initial investment and overall maintenance cost get increased significantly. The variable area design developed by the author and co-workers [4,5,6] is a recommended option in this connection. Variable area heat exchangers (VAEs) are of shell and tube configuration, but employ diverging-converging (periodically constricted) tubes instead of straight, cylindrical tubes. The geometry of such a heat exchanger is sketched in Fig. 2. From the geometry shown, it is clear that the angle of convergence/divergence is related to the geometrical dimensions (D_1,D_2 and L_S) of the D-C tube (diverging – converging tube) as given below:

$$\tan(\theta) = (D_2 - D_1)/L_S \tag{1}$$

where D_1, D_2 = minimum diameter and maximum diameter of D-C tube respectively.

L_S = segment length

The values of D_1, D_2 and L_S are so chosen that

$$\tan(\theta) = 1/12$$

$$\text{or,} \qquad \theta = 5^0 \tag{2}$$

The above value of θ has been found to be the optimum angle of convergence/divergence [4,5].

Heat exchangers of this configuration have been installed in many process industries all over the world. Their performance characteristics have been investigated extensively by the author both mathematically as well as experimentally. Well-tested software packages have been developed to predict the performance of these heat exchanges for liquid-liquid heat transfer considering both Newtonian as well as non-Newtonian flow [7], film condensation of single vapours/ mixed vapours [8,9] and also for thin film evaporation. It has been observed that these VAEs provide significant enhancement (350-470%) in heat transfer coefficient, with relatively negligible increase in the pressure drop penalty (118-120%). The performance features of these exchangers are thus truly spectacular. They provide augmented heat transfer efficiency, with little increase in the operating cost. A typical plot of heat transfer coefficient (h_i) versus Reynolds number (Re_S) for an exchanger of this configuration is shown in Fig. 3. This figure also demonstrates the variation of percentage enhancement (in heat transfer coefficient) with Reynolds number (Re_S). The % enhancement (E) is defined as,

$$E\,(\%) = [h_{D-C} \,/\, h_{St}](100) \tag{3}$$

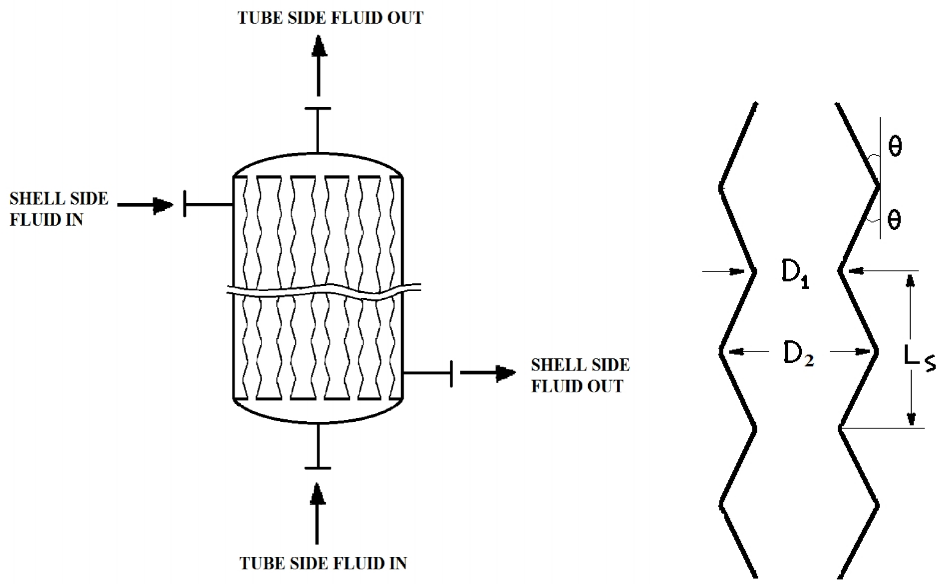

Fig. 2. Schematic of a variable area shell and tube heat exchanger (1-1 construction)

Where h_{D-C} is the forced convection heat transfer coefficient in VAE that employs D – C(diverging-converging) tubes and h_{St} is the value of heat transfer coefficient attained at the same Reynolds number in a conventional exchanger that employs straight, cylindrical tubes of same surface area (heat transfer area) per unit length as the D-C tubes. In other words, the conventional exchanger employs tubes of diameter D_S, which is the surface diameter of diverging-converging tube. By surface diameter, we mean the diameter of a straight, cylindrical tube having the same surface area per unit length as the D-C tube. From simple geometry (sketched in Fig. 2), it can be easily deduced that

$$D_S = (D_2^2 - D_1^2)/(2\, L_S\, \sin\theta) \tag{4}$$

Accordingly, the Reynolds number (Re_S) is defined as,

$$Re_S = 4\,\dot{m}/(\pi\, D_S\, \mu_f) \tag{5}$$

where $\dot{m}$ is the mass flow rate and μ_f is the viscosity of the process fluid handled.

Typical plots [5 and 8] of pressure drop/ friction factor versus fluid flow rate / Reynolds number (Re_v) are shown in Figs. 4 and 5. Here, the Reynolds number is defined based on the volumetric diameter (D_V) of the diverging-converging tube (that is, the diameter of a straight, cylindrical tube having the same volume per unit length as the diverging-converging tube). By virtue of the geometry of D-C tube sketched in Fig. 2.

$$D_V = [(D_2^3 - D_1^3)\cot\theta\ \ /\ 3\, L_S]^{\frac{1}{2}} \tag{6}$$

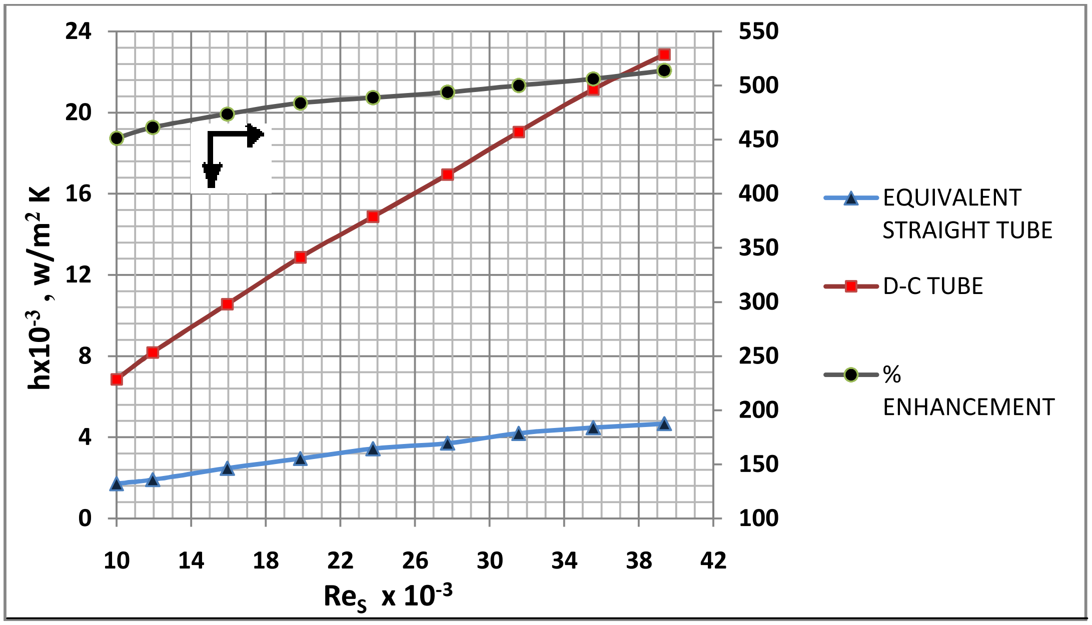

Fig. 3. Heat transfer coefficient versus reynolds number plots

The friction factor (f_V) and the Reynolds number (Re_V) are thus defined as

$$f_V = (-\Delta P)\, D_V\, \rho_f \,/\, (2\, L\, G^2) \tag{7}$$

$$Re_V = 4\,\dot{m}/(\pi\, D_V\, \mu_f) \tag{8}$$

G = mass velocity of process fluid

$$= 4\,\dot{m}/(\pi\, D_V^2) \tag{9}$$

L = total length of each tube

$$= n\, L_S \tag{10}$$

n = the total number of segments in each D-C tube

Incidentally, when the angle of convergence/divergence is close to 5°,

$$D_V \simeq\ D_S \simeq\ (D_1 + D_2)/2 \tag{11}$$

Fig. 4 illustrates f_V versus Re_V plots for both VAE and the equivalent conventional exchanger (that employ straight tubes of diameter, D_V). Fig. 5 presents plots of shell side pressure drop versus fluid flow rate for VAE (which uses an un-baffled shell) and for the equivalent conventional exchanger (that uses a baffled shell). Data presented are experimental values [5,6].

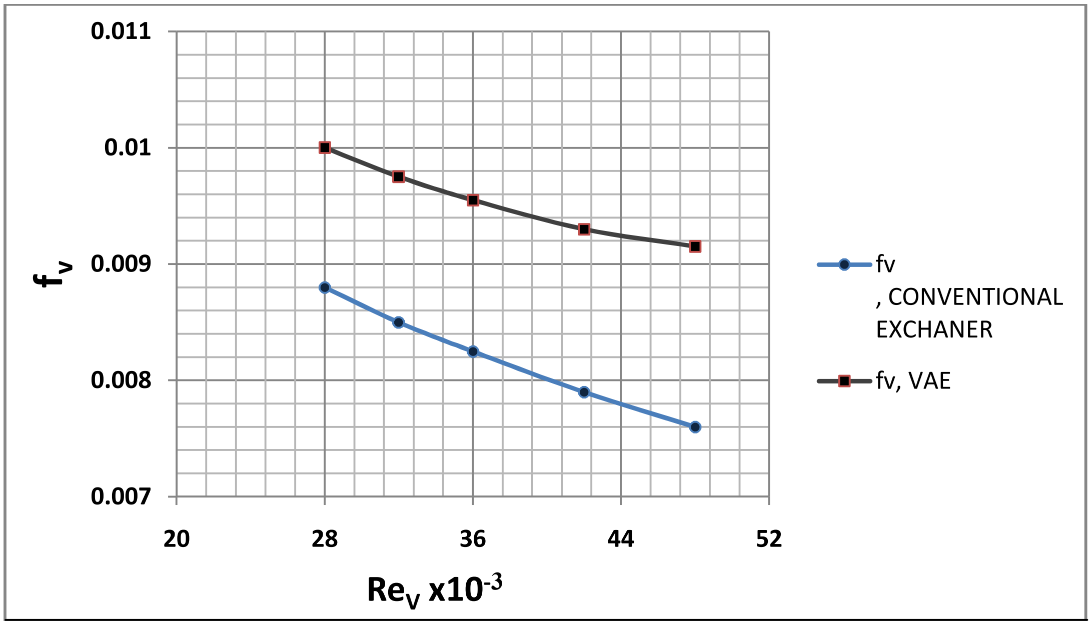

Fig. 4. Friction factor versus Reynolds number plots

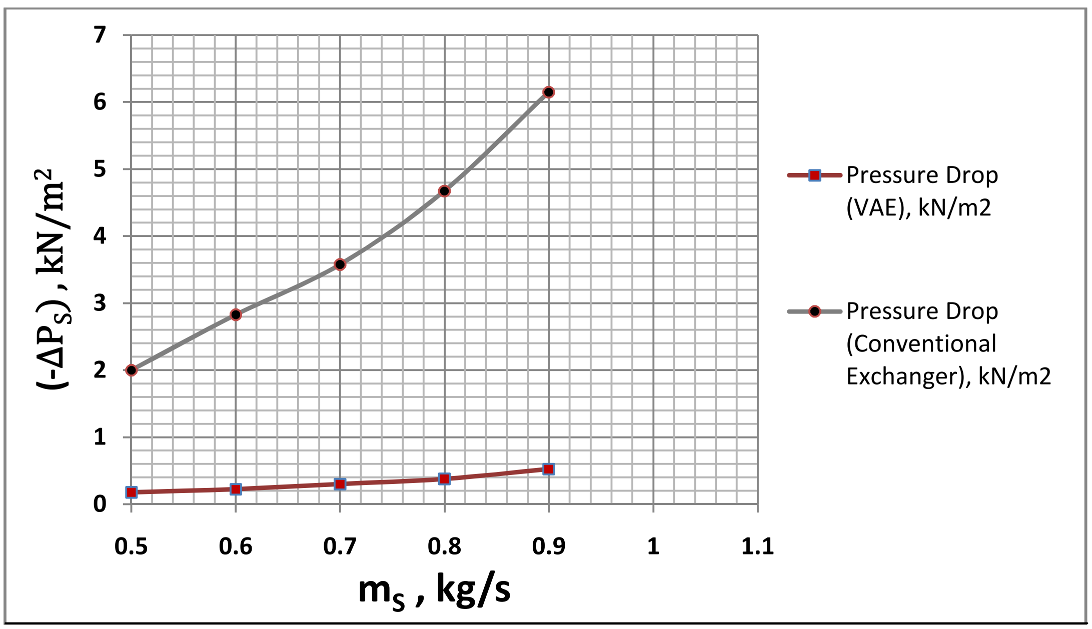

Fig. 5. Shell side pressure drop versus fluid flow rate plots

It can be seen from Figs. 3 to 4 that the VAE of proposed design provides substantially higher heat transfer coefficient (Fig. 3), but the pressure drop in the systems (which is proportional to the friction factor, f_V) is only marginally higher than that in the equivalent, conventional exchanger (Fig. 4). To keep in mind that both exchangers are of same heat transfer area per unit length. Fig. 5 further demonstrates that the shell side pressure drop in the VAE is conspicuously lower than that in the conventional exchanger. This must be viewed keeping in mind that the shell of the VAE is un-baffled, while that of the conventional exchanger is fitted with 25% cut segmental baffles. Since the VAE provides substantially enhanced heat transfer coefficient, its shell is not required to be baffled. The performance data of VAE presented are thus for un-baffled shell.

To elucidate the phenomena further, a plot of thermal enhancement factor (η) versus Reynolds number (Re_S) is illustrated in Fig. 6. Here, η is defined as

$$\eta = [Nu(DC)/Nu(St)]/[f_{DC}/f_{St}]^{1/3} \tag{12}$$

where Nu(DC) and Nu(St) represent Nusselt number for the variable area exchanger and that for the equivalent conventional exchanger respectively, while f_{DC} and f_{St} stand for friction factor in VAE and that in the equivalent conventional exchanger respectively. From Fig. 6, it can be observed that the value of thermal enhance factor (η) remains in the range of 2.8 to 3.6 within the entire range of Reynolds number (Re_S) considered. This further ascertains the distinct merits of variable area construction.

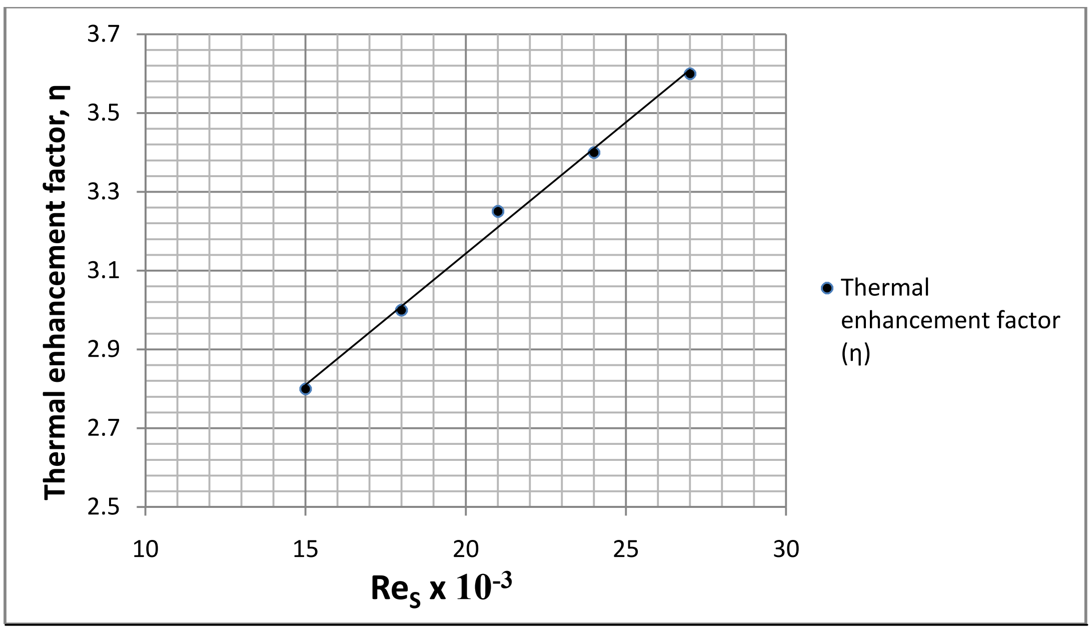

Fig. 6. Thermal enhancement factor (η) versus Reynolds number (Re_S) plot

An additional advantage of the proposed design is that exchangers of this kind exhibit much lower tendency to foul. Laboratory tests conducted by the author [5,6] have indicated that the rate of scale deposition and consequent precipitation fouling are much lower on diverging-converging surfaces as compared to straight cylindrical surfaces. This is due to the fact the tortuous wall geometry of the tube induces a degree of turbulence into the flowing liquid and as a result, the deposited dirt gets re-entrained into the flowing stream. This characteristic is of specific interest in the case of OTEC systems, since in OTEC power plants, both the working fluid (ammonia, freon) as well as the utility fluid (sea water) are corrosive in nature. When fouling does occur, mechanical cleaning of these tubes would be relatively more

troublesome. However, cleaning using suitable solvents (chemical cleaning) or by the use of high pressure liquid jets (hydraulic cleaning) can be conveniently employed.
A few constructional features of these exchangers are given below:

a) While installing the tube bundle, the tubes may be arranged in such a way that the maximum cross-section of the tubes is in contact with the tubesheet. In other words, the tube hole diameter be kept equal to (or slightly larger than) the maximum diameter (D_2) of the tube. This would demand a slightly larger size shell, but would facilitate pulling-out of tube in case of leakage or fouling.
b) As stated earlier, the shell of variable area exchangers is not required to be baffled. However, in case large scale tube vibrations are anticipated, baffles with baffle spacing (B_S) = B_S (max) = shell diameter may be installed to act as support plates for tubes.
c) It has also been stated earlier that the optimum value of angle of convergence/divergence (θ) is 5º. We have investigated the effect of θ on heat transfer enhancement [5,8] and it has been found that through the percentage enhancement in heat transfer coefficient does increase with increase in θ, the simultaneous increase in pressure drop penalty also becomes appreciable (due to possible boundary layer separation). A most reasonable choice of θ is, therefore, θ = 5^o, at which the heat transfer enhancement is substantial, but at the same time, the increase in pressure drop penalty is reasonably low.
d) In conventional shell and tube heat exchangers, shell side heat transfer coefficient often gets diminished due to supplementary effects such as bundle bypassing and baffle leakage. Such additional effects could bring down the shell side transfer coefficient by as much as 40%. The bundle bypassing effect is due to the fact that a part of the shell side fluid tends to execute essentially parallel flow when flowing between the outer-most tube and the shell wall and thereby tends to bypass the tube bundle. To keep in mind that heat transfer coefficient in parallel flow is much lower than that in the crossflow. This effect would not be significant in the proposed design since this bypass stream shall also execute a tortuous flow due to the diverging-converging nature of the tube wall geometry. The baffle leakage effects (leakage of shell side fluid through shell to baffle and tube to baffle clearances) will also not be predominant in VAEs since the shell is to be fitted with minimum number of baffles only.

Incidentally, presence of non – condensable gases in the working fluid / process fluid has not been considered in the present analysis. This may be treated as a limitation of the study reported.

3. CONCLUSION

a) Hybrid OTEC power plants are most recommended for the generation of clean electric power as well as simultaneous production of desalinated water (as by product) on commercial scale.
b) Variable area heat exchangers that employ diverging-converging tube bundle are strongly recommended for the construction and installation of these OTEC systems.
c) These exchangers provide excellent augmentation in heat transfer efficiency with little increase in associated pressure drop penalty (operating cost). The overall economy of installation and operation of OTEC power plants thus gets enhanced significantly. These exchangers are also more resistant to fouling and scale deposition.

d) Fabrication cost of these exchangers shall be, no doubt, higher. However, it is not to be forgotten that the increase in fabrication cost shall cause increase in initial investment only. Since the exchanger operates with augmented heat transfer efficiency without any significant increase in operating cost, the additional initial investment required can be easily recovered within a short span of time. Successful pilot scale studies conducted by the author and co-workers have demonstrated that the payback period for this type of construction shall not exceed 16 to 18 months.
e) Once economic construction and operation of hybrid OTEC power plants become possible by the use of Variable Area Heat Exchangers, large scale production of desalinated water becomes feasible without any additional investment.

COMPETING INTERESTS

Author has declared that no competing interests exist.

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