OPTIMUM THERMAL COUPLING SYSTEM FOR COGENERATION NUCLEAR DESALINATION PLANTS

Mahmoud Samy Saadawy

Reactors Department, Nuclear Research Center, Atomic Energy Authority
P.O. 13759, Cairo, Egypt
E-mail: Saadawymahs@hotmail.com

ABSTRACT

Due to the little established nuclear desalination plants worldwide, there is lack of technical data and no engough practical experience in coupling field. Therefore, in this study different coupling technologies concerning cogeneration systems for joining nuclear power reactors with large thermal seawater desalination units (*MED & MSF*) have been reviewed and optimized. A new heat pipes heat exchangers are first proposed for the isolating intermediate loop (*IIL*), in addition to the previosly reviewed types: flash steam chamber and pressurized water loops. All studied thermal coupling methods are modeled mathimatically as heat exchanger loop transfers safely heat energy from nuclear power plant to desalination plant. A methodology for selecting the optimum coupling system is derived considering several preference indices that mainly depend on *IIL* characteristics: heat transfer surface area, thermal performance and consumed pumping power.

As an exploratory case study, these factors are analyzed and discussed for the *SMART* reactor type. The results of different thermal coupling schemes for wide range of top brine temperature *TBT* demonstrate that the heat pipe loop (*HPL*) is the optimum *IIL* type and becames the unique inevitable option, due to its attractive and advataguous characteristics for both *MED* and *MSF*. Furthermore, from the safety analysis view, the HPL is considered more reliable because the heat pipe itself can represent an additional barrier against product water cotamination. Also, technical guide values of design parameters are concluded from the current model for the design and construction purposes with high accuracy, where the maximum variation not exceed ± 5 % for small and medium nuclear power reactors (*SMRs*), delivering various power from 50-2000 MW_{th} from backpressure turbine as condenser rating to distillation plants. The present predictions showed also a reasonable agreement when compared with the other literature results for various heat exchangers.

Key Words: Cogeneration plants; Desalination processes; Nuclear reactor types; Coupling systems; Heat exchanger design; Heat pipes; Heat transfer; Two-phase flow.

INTRODUCTION

Water production of acceptable cost is of great interest in the Middle East region as most conservative predictions indicated a severe shortage of water in this area and other worldwide locations [1]. Seawater desalination is still one of the developed promising solutions for increasing fresh water resources. For different occasions, nuclear energy became the most important competitor to the fossil and renewable energy for use in seawater desalination. Although, for the meantime no real nuclear power plants exist in the Middle East region, but the regional concern in utilizing the dual-purpose plants of nuclear desalination is of future interest. People of these countries are quite convinced of the economic, environmental and safety incentives of implementing large nuclear cogeneration plants for generating electricity and production of potable water.

In nuclear cogeneration plants, the primary product has usually been electricity production, but some of the generated energy can additionally drive a desalination unit for producing fresh water from sea as a byproduct. Coupling of Nuclear Power Plants (NPP) with commercially large desalination plants is mainly classified into two different groups, based on the kind of supplied energy :

a. Electrical energy for Reverse Osmosis (RO) and Vapor Compression (VC) processes.
b. Heat energy for distillation processes; Multi-Stage Flash (MSF) and Multiple-Effect Distillation (MED).

Coupling of RO process with NPP is relatively simple through a direct electric connection. While, the thermal coupling of distillation processes, the subject of the current study, is relatively complicated. Where, the required heat energy for thermal processes are supplied by the nuclear rectors, as steam taken directly from steam turbine cycle's of NPP. But, steam is normally generated in reactors at very elevated temperatures, in the range of 250-285 ^{o}C (40-70 bar), due to the great temperature dependence of plant efficiency. While, the normal reject steam temperature, from a turbine to the relevant condenser is approximately 40 ^{o}C (18 kPa). The maximum allowable Top Brine Temperature (TBT) of thermal distillation processes is about 120 C, due to scaling limitations. Consequently, the TBT of distillation plant seems to be far higher than the normal exhaust condenser temperature, and in the same time lower than the output steam generator temperature. Therefore, in cogeneration plants, steam is permitted to be sent at moderate temperature to a distiller. Consequently, this actually causes a slight loss of output power, which is partially compensated by improving heat utilization. Provision of a desalination plant with low grade heat energy can be performed by various schemes, via back pressure turbine or extraction turbine. Back pressure steam turbine are usually utilized for the provision of steam to distillation plants at relatively low and medium TBT, in the range of 50-76 ^{o}C. While, steam is mainly extracted from the low and medium pressure turbines and supplied for thermal plants of higher TBT, up to about 110 ^{o}C [2].

Most studies demonstrate that, the water cooled reactors (PWR & PHWR) are the appropriate candidated types to be connected in cogeneration plants, particularly the most advanced inherent and passive safe nuclear reactors. Whereas, the first cogeneration plant started up in 1973 at Kasachstan used a fast breeder reactor (BN-350). Thermal coupling system requires a more striengent safe intermediate loop (*edioator*) connecting between the nuclear reactor plant *energy source*) and the seawater desalination plant (*distination*). This loop acts as an isolation system, to eliminate the possible risk of radioactive contamination of the product water or preventing salination of the secondarry coolant of NPP. Accordingly, each isolating loop consists of a main heat exchanger unit and recirculating pumps. Of the available coupling systems that was issued by IAEA [3], two types of intermediate loop can be utilized :

a. Flashed steam loop. b. Pressurized loop.

Recently, another coupling system was ultimately proposed at first in reference [4], which includied the advanced Heat Pipe Heat Exchanger (HPHE) for generating either heating steam or hot water for distillers. Therefore, M.S.Saadawy [5] developed a design study of heat pipe and thermosyphon heat exchangers (HPTHHE), and thermally analyzed the two main heat exchanger problem specifications, sizing and rating problems.

Development of nuclear dual-purpose plant performance's requires an intensive study and analysis of all units and cycles, involving the coupling schemes. Therefore, the present study mainly focuses on the analysis of different thermal coupling options for cogeneration nuclear/desalination plant to basically select an economic optimum option. Because the feasibility of such cogeneration system could substantially change, if an optimum option of coupling is readly achieved.

SYSTEM ANALYSIS

A dual purpose plant consists of three interconnected systems: *nuclear power plant* involving conversion cycle for the steam power generation and turbogenerator connection; *coupling system* (*intermediate isolating loop IIL*) and *thermal seawater desalination plant.* The current section presents comprehensively a detailed computational algorithm describing the steady state behavior for different coupling options.

The feasibility of the anticipated systems is readily discussed according to thermal hydraulic analysis of each system. An optimum selection for the appropriate system is substantially performed applying a comparison between the flash loop FSCL and heat pipe exchanger HPHEL for MED, and on the other hand between the pressurized loop PWL and HPHEL for MSF.

It seems impossible to establish a generally applicable selection for the IIL, since numerous factors influence the ultimate decision. Some of the pertinent considerations are economic: development cost, initial cost, operating cost, maintenance cost, ..etc. In addition, other factors such as reliability and safety must be considered. However, in assessment, it is useful to consider the thermal and hydraulic performance of isolating loop, particularly in the dominant practical case.

1. Preference Indices of Coupling Systems

The methodology for selecting the optimum coupling system is based on some previously studied indices [8] including three factors. But, these factors are not quite enough for a comprehensive comparison, thus additional four criteria are mainly proposed for the current preference process. The comparative factors concern the heat exchanger characteristics of heat transfer area, unit thermal performance and consumed pumping power. The following indices are defined for several commercial heat exchanger alternatives of coupling.

1. **Specific *heat transfer surface area* (*SP*A)** as a measure of the capability of heat exchanger to transfer power with minimal heat transfer area, in m^2/kW, where SPA = Q_{th} / A .
2. **Compactness (COMP)** is a measure of heat exchanger performance accounting for size, defined as the ratio of thermal conductance to the volume of heat exchanger, in W/m^3.C., where COMP = 1/ (R_T . V).
3. ***Equivalent overall heat transfer coefficient* (U)** is a measure of the thermal performance of heat exchanger, in W/m^2.C, where U = Q_{th} / (A.ΔT_{LMTD}) .
4. ***Effectiveness* (*E*)** is a measure of the heat exchanger efficiency for the given operating condition, defined as the ratio of the heat transferred by the real heat exchanger to the ideal heat transferred by a heat exchanger of infinite surface area.
5. ***Specific pumping power* (*SPW*)** as a measure of the pressure drop in both condenser and desalination branches of IIL, defined as the pumping power of the two streams per unit heat transferred power, in kW/MW, where SPW = W_p/Q_{th}.
6. ***Heat transfer- pressure drop ratio (HTPDR)*** is a measure of the power required to force the two fluid streams through the heat exchanger, in W/m^2.C / N/ m^2.
7. ***Area density* (*AVR*)** is a measure of the heat transfer area per unit volume of heat exchanger, in m^2/m^3, where AVR = A/V.

2. Output Thermal Condenser Rating

The isolating intermediate loop (IIL) for the thermally driven processes acts as barrier to avoid leakage of radioactive material to fresh water. Therefore, its main function is to indirectly transfer safe heat energy from nuclear power plant to desalination plant. Then, steam is bled from energy conversion loop of nuclear plant, from backpressure turbine or extraction turbine. The algorithm of the backpressure turbine is represented, and algorithm of extraction turbine is of a similar route. Figure 1 shows the details of steam power station's condenser in dual purpose plants. The steam exits from turbine at relatively higher pressure than the condensation condition in case of single purpose only. In the present study, steam condition (P_{st}, T_{st}, m_{st}) is held constant for all the four coupling options. Then, the circulated water condition of the intermediate circuit (T_o, $\dot{m}_{cw}$) remains also constant for a certain known T_i. According to the following heat balance for water and steam sides, the thermal condenser rating Q_{th} can be found:

$$Q_{th} = \dot{m}_{cw} \; c_p \; (T_o - T_i) \tag{1}$$

$$Q_{th} = m_{st} * \; h_{fg \, @ \, P_{st}} \tag{2}$$

where T_i and T_o are the cooling water temperatures of steam condenser. In the meantime, that cooling water is considered as an energy carrier medium to intermediate loop, and its temperatures are calculated as depicted in the Figure:

$$T_o = T_{st} - \Delta T_{ca} \tag{3}$$

$$T_i = T_o - \Delta T_{cr} \tag{4}$$

ΔT_{ca} and ΔT_{cr} are the condenser approach and condenser range, of assumed values 5 and 10^o C.

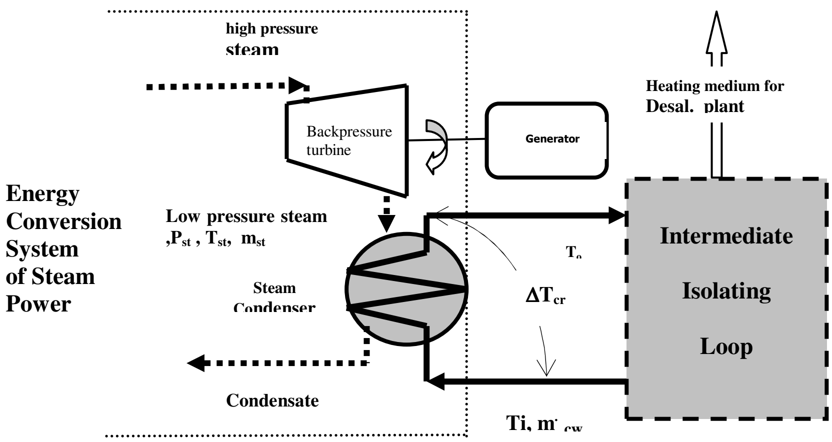


Fig. 1 Backpressure Turbine Arrangement for Thermal Coupling

COUPLING SCENARIOS MODELING

Three commercial coupling schemes are mainly candidated for distillation processes in this study:

a. Spray Flash Steam Chamber Loop (FSCL), for provision MED plant with heating steam.
b. Pressurized Water Loop (PWL), for supplying hot water to MSF plant.

c. Heat Pipe Heat Exchanger Loop (HPHEL), for generation steam or hot water to MED or MSF.

Detailed descriptions of the three models of coupling are enclosed in the following sections.

1. Spray Flash Steam Chamber Loop (FSCL)

The intermediate isolating loop (IIL) used a flash tank to generate vapor to MED plant by flashing technique. As shown in Fig. 2, cooling water exits from the steam condenser with the condition of P_o and T_o, where T_o is usually lower than the saturation temperature corresponding to P_o. Then, the cooling water enters the flash tank which is maintained at pressure P_{fch} less than P_o, causing the vapor to flash at a temperature T_{fch} (lower than T_o) corresponding to the saturation pressure P_{fch}. The vapor passes directly to MED plant through a suitable demister, to separate the entrained sea water droplets, with a condition of m_v, P_{TBT} and T_{TBT}. Thus, the flashed vapor goes to the first effect of MED unit with the maximum allowable temperature (top brine temperature TBT). Neglecting all non-equilibrium terms at thermal equilibrium, the following equations govern all these relations and variables at steady state:

$$P_o = \mathrm{P_{fch}} + \Delta \mathrm{P_{nozz}} \tag{5}$$

$$P_{ii} = \mathrm{P_{fch}} + \rho\ g\ B_{lp} \tag{6}$$

$$P_{fch} = \mathrm{P_{TBT}} + \Delta \mathrm{P_{dem}} \tag{7}$$

$$\dot{\mathrm{m}}_{\mathrm{v}} = \mathrm{Q_{th}} / \mathrm{h_{fg\ @\ P_{fch}}} \tag{8}$$

Energy conservation equation can be applied to flash tank as:

$$\dot{m}_{cw}\ \mathrm{c_p}\ \mathrm{T_o} = \dot{\mathrm{m}}_{\mathrm{v}}\ \mathrm{h}''_{\mathrm{g\ @\ P_{TBT}}} + (\dot{\mathrm{m}}_{\mathrm{cr}} + \dot{\mathrm{m}}_{\mathrm{bd}})\ \ \mathrm{c_p}\ \ \mathrm{T_{fch}} \tag{9}$$

Mass conservation equation is:

$$\dot{m}_{cw} = \dot{m}_v + \dot{m}_{rc} + \dot{m}_{bd} \tag{10}$$

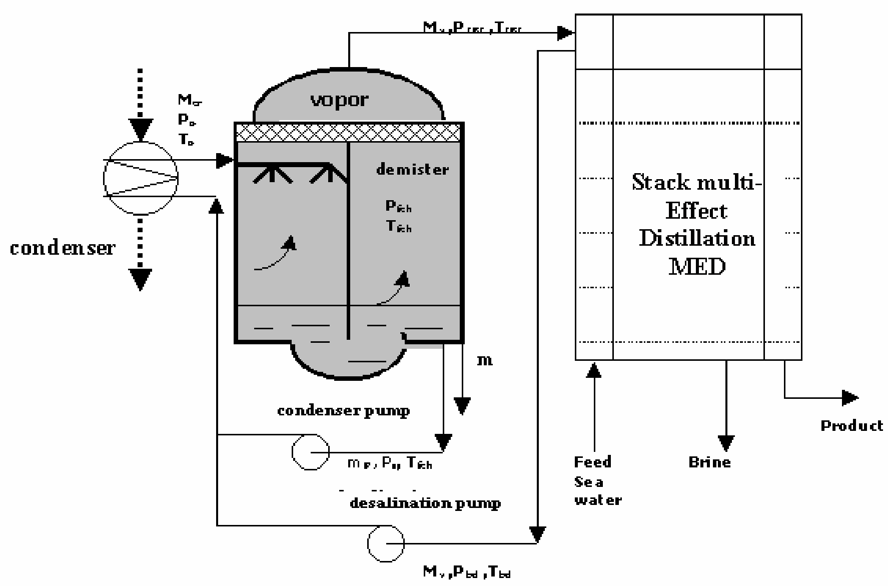


Fig. 2 Spray Flash Steam Chamber Loop (FSTL) for Coupling MED

Balance of point b is expressed, neglecting the change of c_p due to small temperature change:

$$m^{\cdot}_{cw}\ (T_i - T_{bd}) = m^{\cdot}_{rc}\ (T_{fch} - T_{bd}) \tag{11}$$

From Eq. (9) and Eq. (10), the following equation is derived:

$$m^{\cdot}_{cw}\ (T_o + T_{fch}) = m^{\cdot}_{v}\left(\frac{h_{fg\ @\ P_{TBT}}}{c_p} - T_{fch}\right) \tag{12}$$

The pressure drop in the demister is found as [9]:

$$\Delta P_{dem} = 3.296 * 10^{-13}\ \ \mathrm{Ln}\left(\frac{\rho_l}{\rho_v}\right)\ v_v^2 \tag{13}$$

Mean vapor velocity is calculated as [10]:

$$v_v = 1.4\left[\frac{\sigma\, g\, (\rho_l - \rho_v)}{\rho_l^2}\right]^{0.25} \tag{14}$$

Also, the outlet liquid velocity from water nozzle is:

$$v_l = \sqrt{\frac{2\ \Delta P_{nozz}}{\rho_l}} \tag{15}$$

Flash chamber length L and height B can be found as:

$$L = \frac{\dot{m}_v}{\rho\, v_v\, W} \tag{16}$$

$$B = B_{vs} + B_{lp} + B_{spr\&dem} \tag{17}$$

where the chamber width W is practically assumed in the range of 2.0 m, liquid pool height B_{lp} is nearly 1.0 m and the height necessary for spray and demister space is taken as 2.0 m.

Pumping power involves condenser pump, which circulates the cooling water between the condenser and IIL, in addition to desalination pump that circulates the makeup water between IIL and MED.

$$\begin{aligned} W_P &= W_{p.con} + W_{p.des} \\ &= \dot{m}_{IP} \frac{(P_o - P_{ii})}{\rho_{@\,Tfch} \cdot \eta_{ppc}} + (\dot{m}_v + \dot{m}_{pd}) \frac{(P_o - P_{TBT})}{\rho_{@\,Tma} \cdot \eta_{ppd}} \end{aligned} \tag{18}$$

Flow ratio FR of cooling water to vapor mass flow rate is expressed as:

$$FR = \frac{\dot{m}_{cw}}{\dot{m}_v} \tag{19}$$

The liquid directly converts to vapor, therefore, the flash process can be achieved without heat transfer area. But, to thermally compare this process with ordinary nucleate boiling, it is important to derive an equivalent overall heat transfer coefficient as:

$$U = \frac{Q_{th}}{A.\,\Delta T_{LMTD}} \quad : \quad \Delta T_{LMTD} = \frac{(T_o - T_{TBT}) - (T_{fch} - T_{in})}{Log\left(\dfrac{T_o - T_{TBT}}{T_{fch} - T}\right)} \tag{20}$$

The flash tank length is doubled to give sea water long time to flash and achieve thermal equilibrium. Therefore, the surface area is calculated as:

$$A = 2\,(2\,BL + BW + 2\,LW) \tag{21}$$

The flash process temperature difference ΔT_{fp} is the driving force by which heat is transferred from source T_o to reciever T_{TBT}. The intensity of heat transfer area (area per volume) AVR is calculated as:

$$AVR = \frac{1}{t_w} \tag{22}$$

where t_w is the wall thickness of flash tank, and taken as 0.015 m.

2. Pressurized Water Loop (PWL)

On the contrary to FSCL, pressurized water loop (PWL) provides the multi-stage flash plant (MSF) with hot water instead of vapor. Therefore, a water heat exchanger is used in PWL as an isolating intermediate loop (IIL), for the transfer process of heat energy from nuclear reactor to MSF plant. A shell-tube heat exchanger type is selected in this study due to its high efficiency. As illustrated in Fig. 3, the cooling water stream is perfectly separated from the recirculated stream (which joins between the IIL and MSF). The desalination side branch (recirculated makeup flow is two-passes and its flow rate m_{rc} can be calculated as:

$$\dot{m}_{rc} = \frac{Q_{th}}{c_p \ (T_{TBT} - T_{ci})} \tag{23}$$

$$T_{TBT} = T_o - \Delta T_{ha} \tag{24}$$

$$T_{ci} = T_{TBT} - \Delta T_{bh} \tag{25}$$

Where, the heat exchanger approach ΔT_{ha} is assumed 4 ^oC, and the temperature difference of brine heater ΔT_{bh} is about 7-5 ^oC for large temperature range of MSF (high TBT), and 4.5-3 ^oC for medium and low TBT. The condenser approach ΔT_{ca} is taken in the range of 7 ^oC for steam condenser.

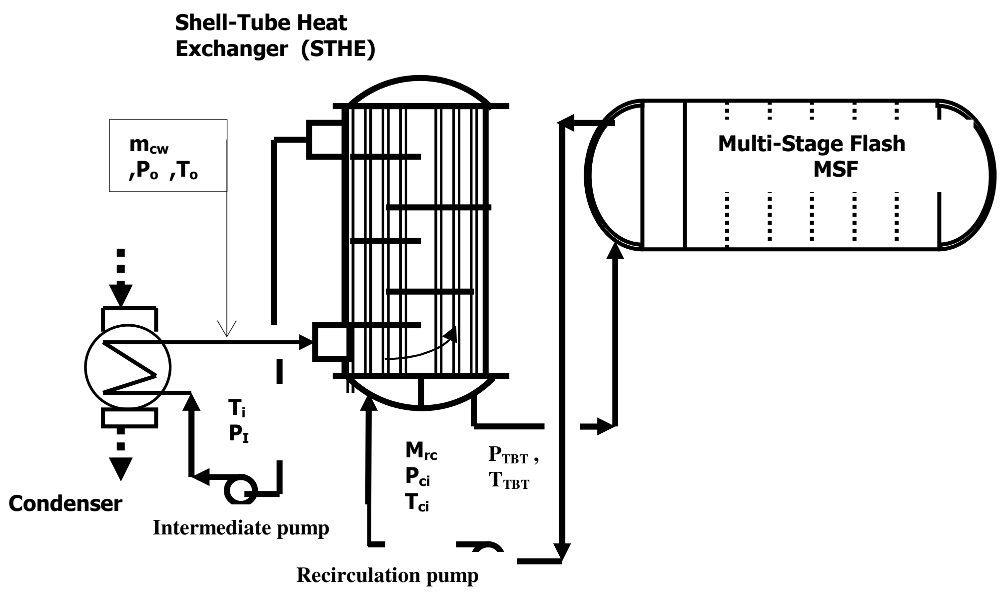


Fig. 3 Pressurized Water Loop (PWL) for Coupling MSF

Pumping power required for PWL consists of intermediate pump (condenser side) and makeup pump (desalination unit side).

$$W_P = W_{Ip.con} + W_{p.des}$$
$$= \dot{m}_{cw} \frac{\Delta P_{cs}}{\rho \, \eta_{pc}} + \dot{m}_{rc} \frac{\Delta P_{ds}}{\rho \, \eta_{pd}} \tag{26}$$

The pressure drop in condenser side branch ΔP_{cs} and desalination side branch ΔP_{ds} is found as [6]:

$$\Delta P_{cs} = 2f_s \frac{G_{cw}^2 \; D_{i\,sh}}{\rho \; d_h} (n_{b\,cs} + 1) (\frac{\mu}{\mu_l})^{.14} \quad : \quad \Delta P_{ds} = 2f_t \frac{G_{rc}^2 \; L}{\rho \; d_i} (n_{p\,ts} + 1) (\frac{\mu}{\mu_l})^{.14} \tag{27}$$

N_{bcs} and n_{pts} are the number of baffle on shield side and number of passes on tube side.

Overall heat transfer coefficient for PWL is:

$$FR = \frac{T_{TBT} - T_{ci}}{T_o - T_i} \tag{28}$$

$$U = \frac{Q_{th}}{A_{sht} \,.\, \Delta T_{LMTD}} \tag{29}$$

Heat exchanger Effectiveness E is:

$$E = \frac{c_h \; (T_o - T_i)}{c_{min} \; (T_o - T_{ci})} = \frac{C_h \,.\, \Delta T_{cr}}{C_{min\,.} (\Delta T_{ha} + \Delta T_{bh})} \tag{30}$$

$$\Delta T_{LMTD} = \frac{(T_o - T_{TBT}) - (T_i - T_{ci})}{\text{Log}\;(\dfrac{T_o - T_{TBT}}{T_i - T_{ci}})} = \frac{\Delta T_{bh} - \Delta T_{cr}}{Log\;(\dfrac{\Delta T_{ha}}{\Delta T_{ha} + \Delta T_{bh} + \Delta T_{cr}})} \tag{31}$$

$$A_{sht} = \pi \; d_o \; L \; n \tag{32}$$

The density of heat transfer area (area per unit volume) AVR is:

$$AVR = \frac{d_o}{d_o^2 - d_i^2} \tag{33}$$

3. Heat Pipe Loop (HPL)

The heat pipe as an advanced effective device for transporting thermal energy with small temperature difference is proposed in this study. The heat pipe heat exchanger (*HPHE*) is considered the main appliance and the milestone in designing HPL. The HPHE acts as a steam generator for MED, or heater for the provision MSF with hot water. According to the area of concern, wicking heat pipe or wickless heat pipe (*thermosyphon*) are possibly applied. As in PWL, the two heat carrier streams flow separately. Obviously from Fig. 4, the condenser side stream represents the evaporator section of heat pipe, while the desalination side streams represents the condenser stream of heat pipe. The mass rate for distillation plant is expressed as:

$$
\begin{aligned}
\dot{m}_{rc} &= \frac{Q_{th}}{h_{fg\ @\ T_{TBT}}} && \text{for MED} \\
&= \frac{Q_{th}}{c_p\,(T_{TBT} - T_{ci})} && \text{for MSF}
\end{aligned}
\qquad (34)
$$

Pressure drop ΔP for the two streams is calculated as:

$$\Delta P_{cs} = P_o - P_i \qquad : \qquad \Delta P_{ds} = P_{ci} - P_{TBTi} \qquad (35)$$

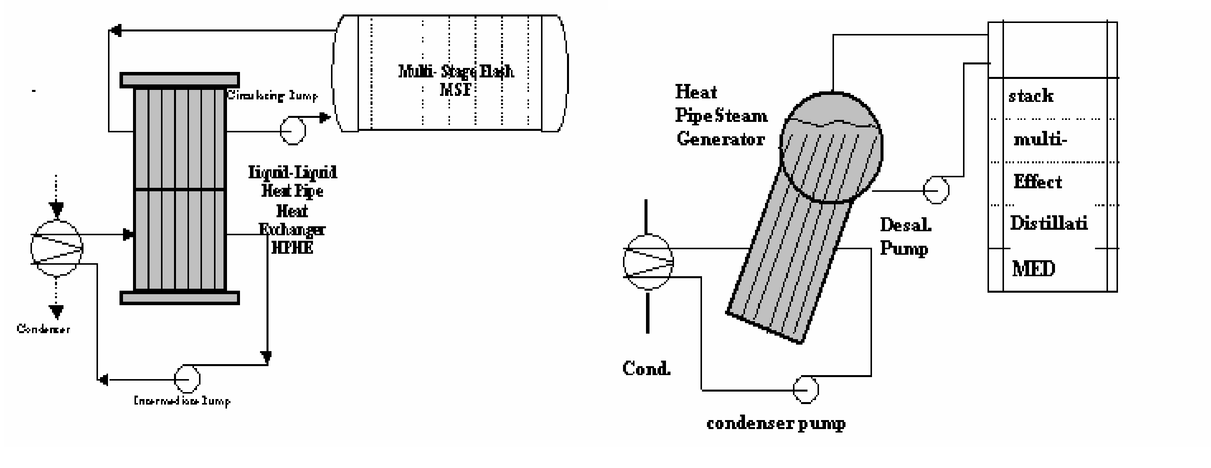
Pumping power required for HPL is:

$$
\begin{aligned}
W_P &= W_{IP} + W_{ma} \\
&= \dot{m}_{cw}\,\frac{\Delta P_{cs}}{\rho\,\eta_{pc}} + \dot{m}_{rc}\,\frac{\Delta P_{ds}}{\rho\,\eta_{pd}}
\end{aligned}
\qquad (36)
$$

Overall heat transfer coefficient is calculated as given by [5], based on total thermal resistance R_T and total heat transfer area:

$$U = \frac{1}{\pi\ d_o\,L\,N\ R_T} \qquad (37)$$

R_T is the total thermal resistance and the method of calculation is analyzed in details in reference [5].



(a) Liquid-Liquid HE for MSF **(b) Steam generator for MED**

Fig. 4 Heat Pipe Loop (HPL) for coupling MSF and MED

The boiling heat transfer coefficient, in case of using steam generator for MED, is calculated by Imura's correlation [11]:

$$h_v = 0.32 \; \frac{\rho^{.65} \; k^{.3} \; c_p^{.7} \; g^{.2} \; q^{.4}}{\rho_v^{.25} \; h_{fg}^{.4} \; \mu^{.1}} \; (\frac{P}{P_a})^{.3} \tag{38}$$

While, the hot water side heat transfer coefficient for the case of MSF is found in [7]:

$$h_{con} = 0.35 * J \; Re^{.6} \; Pr^{.36} \; (\frac{k}{d_o}) \; (\frac{P_t}{P_l})^{.2} \; (\frac{Pr}{Pr_w})^{.25} \; (\frac{\mu}{\mu_W})^{.14} \tag{39}$$

$$FR = \frac{h_{fg \; @ \; P_{TBT}}}{c_p \; (T_o - T_i)} = \frac{h_{fg \; @ \; P_{TBT}}}{c_p \; .\Delta T_{cr}} \qquad \text{for MED}$$

$$= \frac{T_{TBT} - T_{ci}}{T_o - T_i} = \frac{\Delta T_{bh}}{\Delta T_{cr}} \qquad \text{for MSF} \tag{40}$$

RESULTS AND DISCUSSION

All the above calculation procedures are readily programmed for the design conditions in **NDESCOP**. As an exploratory case study, the Korean nuclear reactor SMART is nominated as small reactor type for cogeneration system. The technical reactor data are based on IIL characteristics. The main results are presented individually for each MED and MSF options. An optimization study is substantially performed for selection between FSCL and HPL for MED option to be coupled with NPP, while the comparison is conducted between PWL and HPL for the case of selecting MSF to be coupled with NPP. The main comparison parameters considered for several IIL are: *the specific heat transfer surface area* ***SPA****, compactness* ***COMP,*** *equivalent overall heat transfer coefficient* ***U,*** *effectiveness* ***E,*** *specific consumed pumping power* ***SPW*** and *heat transfer-pressure drop ratio* ***HTPDR.***

Figures 5-9 show MSF results at various ranges of TBT. Figures 5 and 6 illustrate that HPL has lower values of heat transfer area as shown from the relative small values of SPA and high values of COMP. In addition, Figs. 8 and 9 result low pumping power of HPL than PWL. While, heat pipe has a relatively better loop performance as depicted in Fig. 7 for the higher overall heat transfer coefficient U, although the low heat exchanger thermal effectiveness was 0,526. Referred to the economic considerations, this really mean the preference of HPL rather than PWL, due to its smaller capital cost (*A and U effect*) and smaller operating cost (*W effect*). The relative improvement in heat transfer area and overall coefficient are ranging from 19-70 % over all TBT ranges, and 0.3 % for pumping power in the high TBT range.

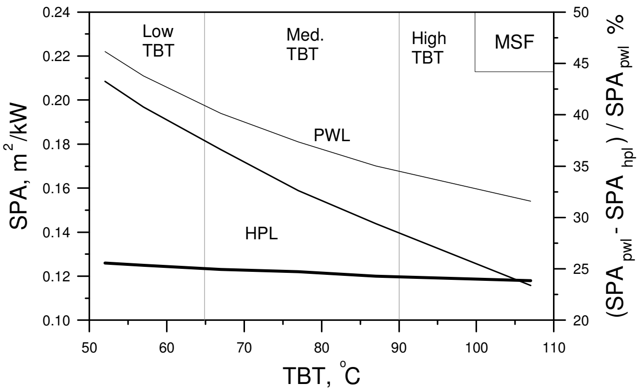


Fig. 5 Variation of SPA in Case of MSF

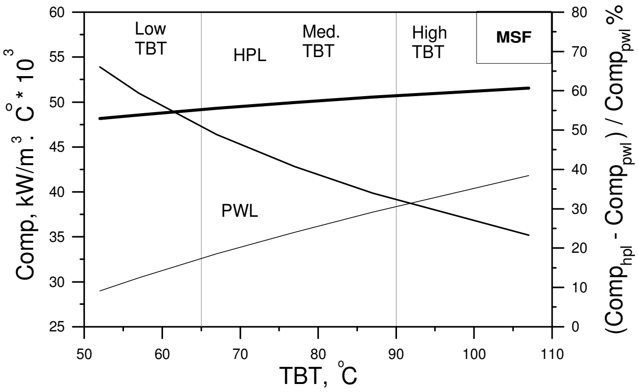


Fig. 6 Variation of Compactness in Case of MSF

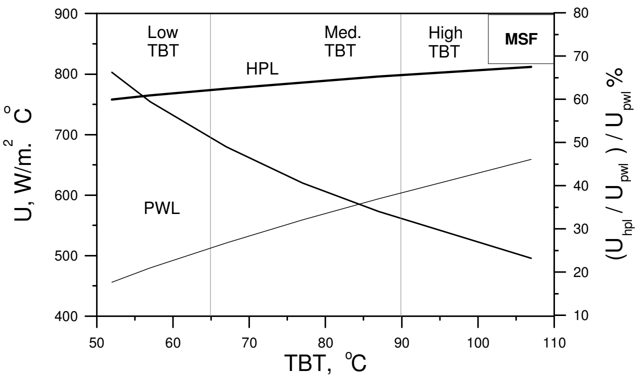


Fig. 7 Variation of U in Case of MSF

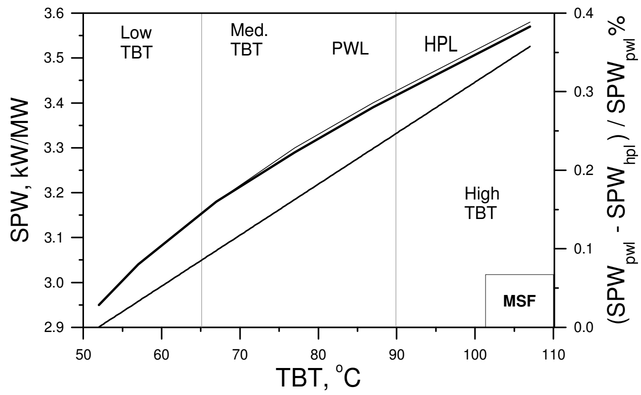


Fig. 8 Variation of SPW in Case of MSF

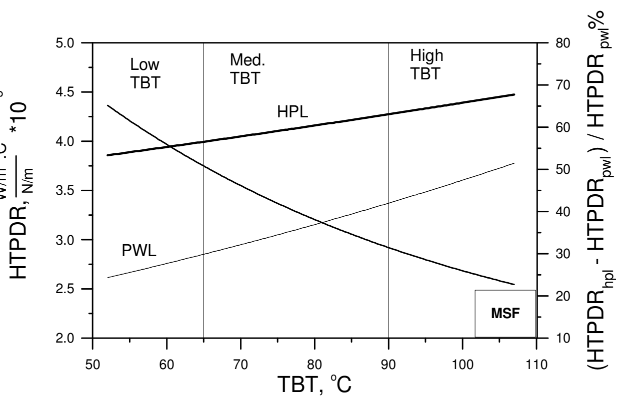


Fig. 9 Variation of HTPDR in Case of MSF

Results of MED are depicted in Figs. 10-14. The trend is not simple as in case of MSF, but it is somewhat different and complicated. The FSCL seems to be possibly more efficient, because of the relatively high overall heat transfer coefficient as evident in Fig. 12. On the other hand, as shown from Figs. 10 and 11, FSCL are basically non-compact exchangers. Thus, it required a considerable amount of space, support structure, foundations and capital and installation costs. In addition, from Figs. 13 and 14 it consumes more pumping power over HPL, as a result of the high differential pressure required for atomization process through water sprinkles. The studied parameters have not only different values, but contradicting actions. As heat transfer is proportional to surface area, the best geometry is actually at higher area/volume ratio. Therefore, the HPL becomes more efficient as it has higher values of area density AVR = 810 than the PWL (AVR = 67). Finally, it is possibly necessary to perform complete study for optimization purposes.

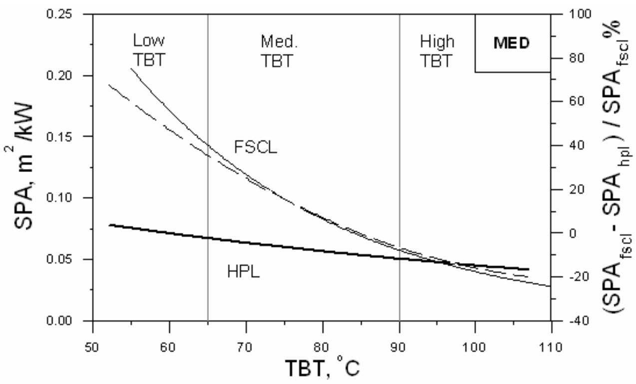


Fig. 10 Variation of SPA in Case of MED

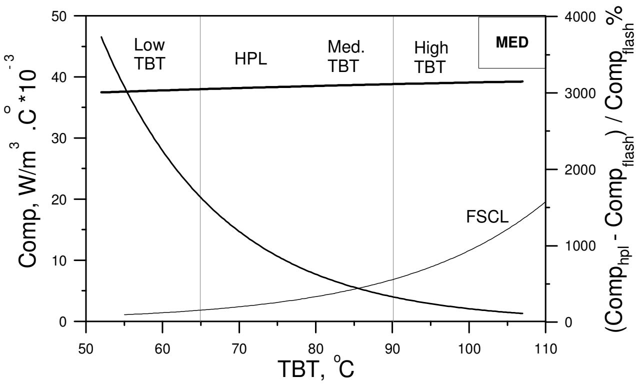


Fig. 11 Variation of Compactness in Case of MED

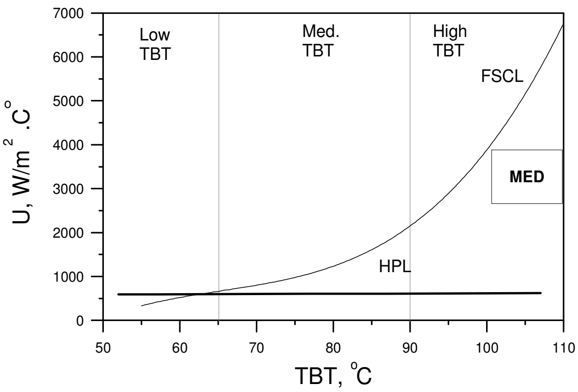


Fig. 12 Variation of U in case of MED

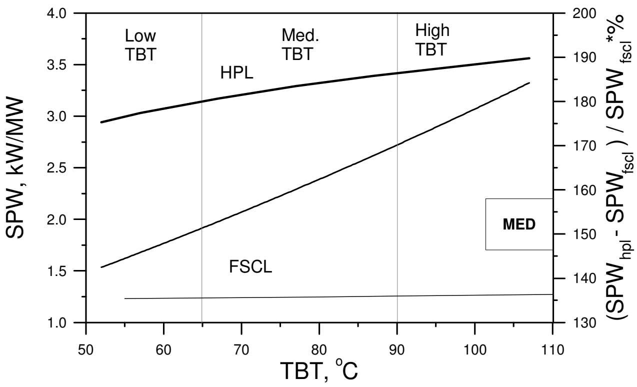


Fig. 13 Variation of SPW in case of MED

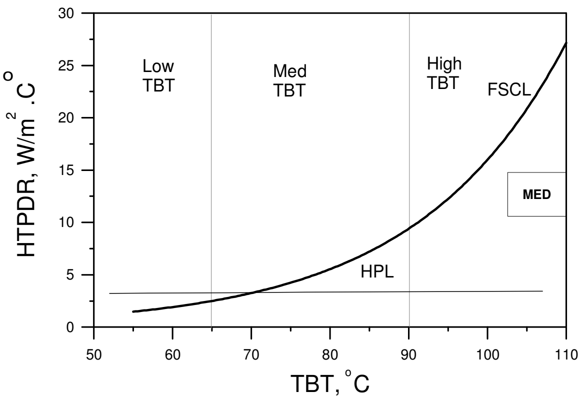


Fig. 14 Variation of HTPDR in case of MED

An optimization technique is applied to select and outweigh the best IIL between different coupling schemes based on the rank of varies criteria. Therefore, Table (1) gives a point - weight as well as performance characteristics for various combined options of MED and MSF. As evident from the table, the new proposed heat pipe loop HPL as intermediate isolating loop IIL has higher scores, thus it immediately offers the greatest advantageous features in terms of size, reduction in capital and running cost (as shown in improvement and reduction column). This makes the HPL an obvious option for coupling system of both MED and MSF with NPP or any other conventional heat source in cogeneration plants. However, HPL has additional encouraging characteristics over any other conventional type, such as:

- The high and low temperature fluid streams are completely isolated, which eliminats the cross – contamination.
- The rate of heat transferred can be controlled by adjusting the tilt angle of heat pipes.
- It is redundant in design. If one individual heat pipe fails, the heat exchanger still efficiently operate with no contamination between the two fluids streams, in addition, the deteriorated heat pipe can be easily replaced.
- HPTHHE has the capability to operate as thermal transformers, by altering the relative lengths of both evaporator and condenser sections at partial energy load.
- The higher reliability of HPTHHE as passive devices, because they have no moving parts or auxiliary power requirements, in addition to its nature to reversibly operate, i.e. heat can be transferred in either direction if required.

The current model program (NDESCOP) is tested for a wide range of output thermal rating varies from 50 to 2000 MW_{th} calculated at TBT = 75 ^{o}C. This thermal range concerns the small and medium nuclear power reactors (SMRs), which are suitable for the production of about 20-860 MW_e. Verification of the program results is applied for comparison with other published data [8] in Table (2). The table contains the average value of criteria, and the maximum difference in the studied power range for SMRs is about ± 5%. Reliability and feasibility is well clarified from the agreement of the table's data of the model predictions with that previously published.

Table (1) Point – weight optimization
(Data are estimated at wide range of TBT from 50-110 ^oC)

NO	Comparison criterion	Parameter	MED Options			MSF Options		
			FSCL	HPL	Relative improvement/ Reduction %	PWL	HP L	Relative improvement/ Reduction %
1	**Heat transfer Surface Area**	SPA	**L**	**E** at low & med. TBT	up to 65 %	**P**	**E**	30 – 50 %
		COMP	**P**	**E**	up to 20 folds	**P**	**E**	19 – 70 %
		AVR	**P**	**E**	12 folds	**E**	**E**	Having nearly equal values.
2	**Thermal Performance**	U	**E** at med. & high TBT	**L**		**P**	**E**	19 – 70 %
		E	**E**	**G**		**G**	**L**	
3	**Pumping Power**	SPW	**P**	**E**	100 %	**L**	**G**	0.3 %
		HTPDR	**L** at low TBT	**G** at med. & high TBT		**E**	**L**	
	Total Points		8	17		9	16	

"E" means Excellent = 3 marks "G" means Good = 2 marks
"L" means Low = 1 mark "P" means Poor = 0 mark

Table (2) Comparison of the current results with previous data
(Data cover a power range for SMRs producing 20-860 MW_e and TBT = 75 ^oC)

No.	Parameter	Published Data [8]			Current Model Results		
		Rotary Generator	Plate Heat Exchanger	Heat Pipe Heat Exchanger	Flash Chamber	Shell-Tube Heat Exchanger	Heat Pipe Heat Exchanger
1	**SPA** m^3/kW	N/A	N/A	N/A	0.085 with ± 0.4 % variation	0.182 with ± 1.3 % variation	0.047-0.129 with ± 1.1 - 5 % variation
2	**Comp** $W/m^3.C$	5400	4140	7200	3561	35000 with ± 1.5 % variation	37000-52000 with ± 1.1% variation
3	**AVR** m^2/m^3	N/A	N/A	N/A	67	810	810
4	**U** $W/m^2.C$	N/A	N/A	N/A	1170 with ± 0.4 % variation	550 with ± 1.6 % variation	600-820 with ± 1.1 - 5 % variation
5	**E** %	80	65	60	56	99	52-77
6	**SPW** kW/MW	N/A	N/A	N/A	1.247	1-21	1-21
7	**HTPDR** $W/m^3.C/ Pa$	115	20	20	4.87 E-5 with ± 0.4 % variation	3.05 E-5 with ± 1 % variation	[3.3-4.5]1E-5 with ± 1.6 % max. variation

CONCLUSION

Due to the limited existence of cogeneration plants using nuclear reactors, in addition to the lack of available data, analysis and feasibility of the coupling options is the aim of this study to develop an optimization procedure. Three options are modeled as intermediate isolating loop (IIL) for coupling systems in cogeneration plants. Additionally, the performance of the proposed heat pipe heat exchanger is favorably compared with the other two early options: flash chamber and pressurized water loops. Combination systems include the large thermal seawater desalination plants, MED and MSF with nuclear reactor plant. The concluded results involve the following:

- Variation of coupling options for combining cogeneration plants with thermal desalination plants, and the influence of several technical and economic parameters on performance, make the use of system modeling by computer programming is stringent.
- The results of wide analysis for different thermal coupling schemes demonstrated that the heat pipe loop (HPL) is the optimum IIL type and became the only inevitable option as a result of its advanced features. The relative improvement reached up to 70 % of its criteria for the case study of small reactor (SMART) at various ranges of top brine temperature 50-110 ^{o}C.
- Important technical design values of governing factors concerning the IIL are estimated for the comparison and design purposes, based on a wide range of output thermal rating varies from 50-2000 MW_e. The accuracy of these values is practically acceptable, where the maximum variation is not more than ± 5 % for the SMRs generating about 20-860 MW_e.
- The present predictions of the heat pipe loop (HPL) generally confirm well the results of previous literature studies on heat exchangers.

Finally, the data and results presented here are of importance to the nuclear desalination industry and other applications of dual purpose plants.

NOMENClATURE

A	heat transfer surface area, [m^2]
AVR	heat transfer area to unit volume ratio, [m^2/ m^3]
B	unit hight[m]
COMP	unit compactness, [E/ m^3]
C_P	specific heat, [J/kg.C]
d	diameter, [m]
E	unit effectiveness
FR	flow ratio of two streams
G	mass flux, [kg/ m^2 s]
h	heat transfer coeffiient,[W/ m^2 C]
h_{fg}	latent heat of vaporization, [kJ/kg]
HTPDR	heat transfer to pressure drop ratio,[W/ m^2 C/Pa]
k	fluid thermal conuctivity, [W/m C]
L	length, [m]

M	mass flow rate,[kg/s]
n	number of baffles or passes
N	number of tubes
P	pressure, [Pa]
q	heat flux, [W/ m^2]
Q_{th}	output thermal condenser rating, [MW_{th}]
R_T	total thermal resistance, [C/W]
SPA	specific heat transfer area, [m^2/kW]
SPW	specific pumping power, [kW/W]
T	temperature, [C]
U	overall heat transfer coefficient, [W/ m^2 C]
v	velocity, [m/s].
V	volume, [m^3].
W	unit width, [m].
W_P	pumping power, [W]
ρ	fluid density, [kg/m^3]
μ	fluid dynamic viscocity, [Pa.s]
ΔT	temperature difference, [C]

SUBSCRIPTS

bd	blowdown
cs	condenser side
cw	cooling water
dem	demister
ds	desalination side
fch	flash chamber
i	inlet stream
Ip	intermediate loop
l	liquid
ma	makeup
nozz	nozzle
o	outlet stream
rc	recirculating
st	steam
v	vapor
w	wall

REFERENCES

1. IAEA, *Potential for Nuclear Desalination as a Source of Low Cost Potable Water in North Africa*. IAEA-TECDOC-917, Vienna, 1996.
2. K.S. Spiegler and A.D.K. Laird, *Principles of Desalination*, Academic Press Inc., New York, 1980.
3. IAEA, *Use of Nuclear Reactors for Seawater Desalination*, IAEA-TECDOC-574, Vienna, 1990.

4. A.H. Mariy, "Economic Evaluation of Dual-Purpose Plants Using Nuclear Tecnology and the Comparison with Conventional Dual-Purpose Plants: General Concept for the Choice of the Inherently Safe Reactor and its Safety and Control Systems; Choice of the Appropriate Power and its Effect on the Unit Water Cost", Academy of sientific Research Technology, 2nd Progress Report, Vol. II, Cairo, Egypt, 1999.
5. M.S. Saadawy, "Design Study of Heat Pipe and Thermosyphon Heat Exchangers (HPTHHE)", *Al-Azhar Engineering 7th Int. Conference*, Cairo, Egypt, 2003.
6. D. Q. Kern, *Heat Transfer*, McGrow-Hill Kogakusha Ltd., Tokyo (Japan), 1950.
7. W. M. Rosenow, J. P. Hartnett and Y. I. Cho, *Hand Book of Heat Transfer*, 3rd Edition, McGrow-Hill New York (USA), 1998.
8. Amir Faghri, *Heat Pipe Science and Technology*, Taylor & Francis, Washington (USA), 1995.
9. H. El-Dessouky, H.I. Shaban and H. Al-Ramadan, "Steady-State Analysis of Multi-Stage Flash Desalination Process", *Desalination*, Vol. 103, pp. 271-287, 1995.
10. P.B. Whally, *Boiling, Condensation and Gas-liquid flow*, Clarendon Press, Oxford, 1987.
11. M. Shiraishi, Kikauchi and T. Yamanishi, "Investigation of Heat Transfer Characteristics of a Two Phase Closed Thermosyphon", *4th International Heat Pipe Conference*, pp. 95-104, London (UK), 1981.