한국에너지공학회지(1999), 제8권 제4호
*Energy Engg. J*(1999), Vol. 8, No 4, pp. 546~552

# Development of the Fresh Water Generator

**Jun Seop Choi**

*Department of Technology Education, Korea National University of Education*

**Abstract** — In order to obtain the highly effective thermal energy from jacket cooling water of propulsion diesel engines, a development of the Fresh Water Generator (FWG) with a capacity of 30 ton/day was implemented. Newly developed experimental devices and data acquisition system were used to evaluate the performance of the FWG. In this study experiments were performed for various driving pressures by varying the mass flowrate of cooling seawater with or without a heat source instead of jacket cooling water.

## 1. Introduction

Owing to a high level of economic growth and increase of population growth, we have to cope with the severe problems in water pollution and shortage in the world[1]. One of the ways to solve the shortage of water is to develop technologies for abundant seawater desalination.

Drinking and general service water are very important for the crews and passengers to keep their lives in the ships which are far from the shore.

In seawater desalination technologies, ion exchanger (IE), reverse osmosis (RO), ultraviolet (UV), micro filteration (MF), and evaporative method to utilize flash evaporation[2] and high vacuum have been studied in the water production industry.

Furthermore, recently solar photovoltaic electrodialysis desalination for brackish water in rem-ote area has been developed[3].

Generally, evaporative type FWG (Fresh Water Generator) using high vacuum condition at chamber is relatively simple structure, simple operation system, trouble-free, and less energy required to operate rather than other desalination systems.

The recent trend of research and development on diesel engines is to obtain higher thermal efficiency with lower air pollution, lighter weight and better performance. As a result, the combustion chamber temperature and pressure on the diesel engines have become higher and eventually the outlet temperature of jacket cooling water has become 85°C.

In this study, basic experiments of the evaporative type FWG, which utilizes heat from waste jacket cooling water of the diesel engines, were performed for many operating conditions with or without steam from boiler instead of jacket cooling water.

## 2. How to Work the Evaporative Type FWG

Figure 1 shows the system concept of the marine FWG. The cooling seawater used in the condenser, upper part of FWG, becomes warmed up as the vapour gives up its heat of condensation. Most of this cooling seawater goes to overboard, and some part of this warm seawater is used as the feed water for the evaporator.

The heated seawater, which circulates to the outside of the heater tubes, evaporates as it enters the main chamber, lower part of FWG, due to the high vacuum condition. Water droplets are removed from the vapour by the mesh separator. The separated droplets fall back into brine, which is extracted from the

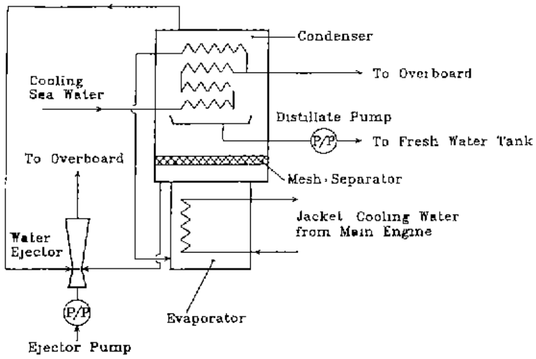



**Fig. 1. System Concept of Marine Fresh Water Generator.**

chamber and discharged to overboard by the water ejector.

The vapour passes to the condenser tube bundle which is cooled by seawater flowing inside tubes. The condensed vapour is collected and pumped to the fresh water tank by the distillate pump. A key role of the water ejector is to keep the evaporation chamber under high vacuum pressure, driven by seawater from the water ejector pump.

## 3. Design of the Developed FWG

The data of Table 1 are design conditions in the main part of the developed FWG. Heat balance at FWG is as follows:

3-1. The Heat Transferred from Steam Vapour to Cooling Seawater, $Q_1$ is

$$Q_1=m * c_p * (T_o-T_{in}) \tag{1}$$

where m is the mass flowrate [kg/h], $c_p$ is the specific heat at constant pressure [kJ/kg. °C]. $T_o$ is the outlet temperature of cooling seawater at the outlet condenser [°C], and $T_{in}$ is the inlet temperature of cooling seawater at inlet condenser [°C].

**Table 1. Detailed Design Conditions of FWG.**

| Item | Specification |
| --- | --- |
| Capacity | 30 ton/day |
| Working Press | 631 mmHg, Vacuum |
| Temp. in Condenser | 56.8°C |
| Salinity of Distillate | Below 10 ppm |
| Cooling Seawater | Inlet Temp.30.5°C, 70 $m^3$/h |
| | Outlet Temp 40.5°C |
| Jacket Cooling Water | Inlet Temp. 85°C |
| | Outlet Temp. 71.4°C |
| Feed Water for Evap | Inlet Temp. 50.6°C. 4.2 $m^3$/h |
| Heating Steam | 24 ton/day, 16 kg/$cm^2$ |
| Distillate Pump | 2.5 $m^3$/h×30 m×3450 rpm ×1.5 kW |
| Ejector Pump | 75 $m^3$/h×48 m×3450 rpm ×18.5 kW |
| Power Source | Motor AC, 440 V, 60 Hz, 3-phase |

Most of cooling seawater is heated up to 40.5°C, and some quantity of them 4.2 $m^3$/h is heated up to 50.6°C. If substituting these data of Table 1 into equation (1) $Q_1$ can be obtained as 742,420 kcal/h

3-2. The heat transferred from jacket cooling water for feed water, $Q_2$ will be calculated by equation (1), and obtained as 748,710 kcal/h or about 640 kW.

3-3. The fresh water generated, $W_f$, can be calculated with

$$W_f=Q_2/rh \tag{2}$$

where rh is the latent heat of the seawater and the value is 565.6 kJ/kg from steam table[4]. Using equation (2), we can calculate $W_f$=30,643.3 kg/day. From the design concept calculation, $Q_1$ is nearly equal to $Q_2$ and quantity of the fresh water generated is about 30 ton/day

## 4. Heat Balance at the Condenser Part

The tube bank of the cross-flow condenser is staggered and horizontal rows, and the vapour heat is transferred to the cooling seawater through the tube wall. Energy is mainly transferred by combination of conduction and convection heat transfer mechanism since radiative heat transfer could be ignored when the temperature in the condenser is not comparatively high. The total energy flow, Q, is expressed as

$$Q=U * A * (T_v-T_w) \tag{3}$$

where $T_v$ is the vapour temperature, $T_w$ is the cooling seawater temperature, and A is surface area for heat transfer consistent with the definition of U. The overall heat transfer coefficient U is defined by the relation

$$U=\frac{1}{1/h_1+1/k_1+1/h_2+f} \tag{4}$$

where $h_1$ is the convection heat transfer coefficient on the inside of tube, $k_1$ is the conduction heat transfer coefficient, $h_2$ is the convection heat transfer coefficient, for condensation on the outside of tube, and f is the fouling factor. And $h_1$ and $k_1$ are defined as

$$h_1=k_f*N_u*\frac{1}{2r_i},\ k_1=\frac{2*\pi*k_c}{\ln(r_o/r_i)} \tag{5}$$

where the subscripts i and o pertain to the inside and outside of the tube, r is the radius of the tube. And $k_f$ and $k_c$ are conductivity of fluid and tube materials, respectively. The convection heat transfer $h_2$ is able to be determined by Nusselt number obtained from the equation (6) for laminar film condensation on horizontal tubes. Where ρ is the density of the liquid, $\rho_v$ is the vapour density, g is the acceleration of gravity, $h_{fg}$ is the heat transfer coefficient evaluated at film conditions and saturated vapour conditions, d is the diameter of the tube, and $\mu_f$ is the dynamic viscosity of fluid.

$$h_2 = 0.725*\left[\frac{\rho*(\rho-\rho_v)*g*h_{fg}*k_f^3}{\mu_f*d*(T_g+T_w)}\right]^{1/4} \quad (6)$$

$k_2$ is 10,000 m²/kW from Handbook[5]. Here, by using the above equations (1)-(6) and design conditions of Table 1, the geometric shape of condenser was designed with 4 tube passes of O.D 19 mm * *l* 3,600 mm * 59 staggered rows, and baffles placed in the shell to induce high heat transfer. Finally, the capacity of 30 ton/day of FWG was confirmed by the developed simulation program.

## 5. Heat Balance at the Evaporator Part

Thermal design method in evaporator is similiar to the condenser part, and equations (3) and (4) can be used in the evaporator. In case of evaporator, $h_2$ is boiling heat transfer coefficient in equation (4). $k_1$ and $h_2$ are negligible small because $k_1$ is very larger than $k_2$, and $h_2$ is very larger compared to $h_1$. And correlations of the experimental data of Hilpert[6] for gases and Knudsen and Katz[7] for liquids indicate that the average heat transfer coefficients may be calculated with

$$N_u = C * R_e^n * P_r^{1/3} \quad (7)$$

where the constants C and n are 0.575 and 0.556, respectively, from the correlation of

Grimson[8]. Reynolds number could be expressed by

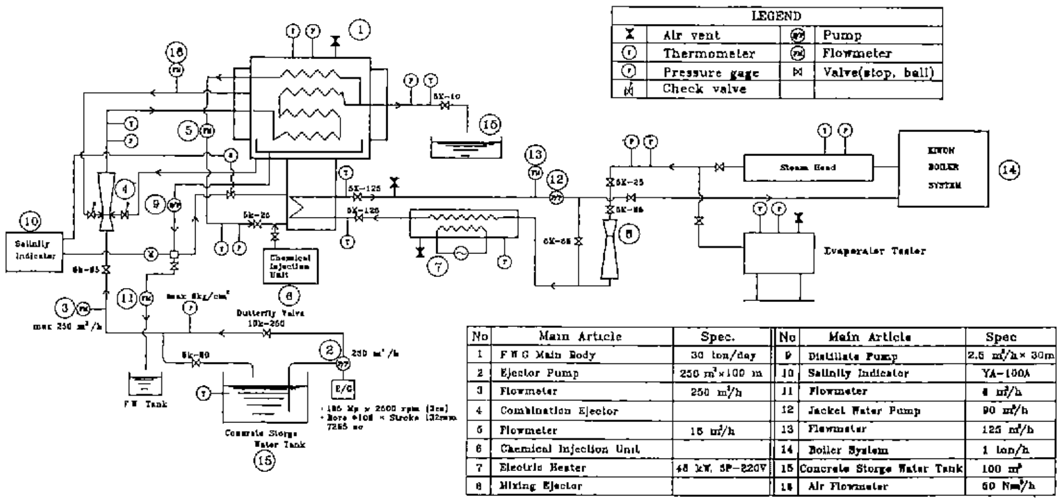
$$R_e = U_{max} * \frac{d}{v} \quad (8)$$

where $U_{max}$ is $U_\infty * S_n/(S_n - d)$. Here, Re can be obtained as 31,832 from equation (8) and Pr is 3.2 from the JSME Data Book[1]. Nu is obtained from the values of C, n, and Pr. And the overall heat transfer coefficient, U, could be determined by using equation (4). Therefore, the area of the evaporator part and the capacity of the fresh water generated could be calculated, and energy recovery from the waste cooling jacket water was estimated about 640 kW. With the consideration of the pressure drop, manufacturing and economic problems, the piping design of FWG was determined.

## 6. Performance Test of the Developed FWG

### 6-1. Outline of the Experimental System

The FWG is mainly consist of the evaporation part,




| No | Main Article | Spec. | No | Main Article | Spec |
|---|---|---|---|---|---|
| 1 | F W G Main Body | 30 ton/day | 9 | Distillate Pump | 2.5 m³/h× 30m |
| 2 | Ejector Pump | 250 m³×100 m | 10 | Salinity Indicator | YA-100A |
| 3 | Flowmeter | 250 m³/h | 11 | Flowmeter | 4 m³/h |
| 4 | Combination Ejector | | 12 | Jacket Water Pump | 90 m³/h |
| 5 | Flowmeter | 15 m³/h | 13 | Flowmeter | 125 m³/h |
| 6 | Chemical Injection Unit | | 14 | Boiler System | 1 ton/h |
| 7 | Electric Heater | 48 kW, 3P-220V | 15 | Concrete Storge Water Tank | 100 m³ |
| 8 | Mixing Ejector | | 16 | Air Flowmeter | 60 Nm³/h |

**Fig. 2. Schematic Diagram for Performance Test of the Present FWG.**

the condensing part, the vapour-liquid separating part, and the driving part of ejector.

Figure 2 shows the schematic diagram for the performance test of the present FWG. In this figure, experimental system could be divided into four parts The first part is heating system including boiler to heat the feed water from the condenser. The second part is cooling system to cool vapour from evaporator. The third part is data acquisition system to obtain the data of temperatures, pressures, flowrates, and salinity. The fourth part is control system to manage total system.

Even though jacket cooling water from the diesel engine is unavailable when the vessel being at anchor, the FWG can be operated by a steam injector using heating steam. In this respect, the preformance test is mainly operated to use steam from the boiler instead of jacket cooling water.

In the heating system for the first part of performance test apparatus, the steam generated in boiler system ⑭ goes to mixing ejector ⑧, and steam and outlet jacket water are well mixed there. The mixed steam with jacket cooling water changed to liquid of water, and then enters the electric heater ⑦. The mixed water was heated up to the desired temperature by electric heater, and flowed into the evaporator of lower part FWG ①. The mixed water supplies heat to the feed water from the condenser, and goes back to boiler system.

In the cooling system for the second part of performance test apparatus, the ejector pump ② pumping water from the heat sink of concreate storage tank ⑮, and high pressurized water moved to the combination ejector ④ The key roles of ejector ④ are to keep the inside FWG high vacuum pressure of 631 mmHg in order to evaporate at 56.8℃, and removal from noncondensable gases in the condenser and droplets of brine in the upper part of evaporator. And then water passed ejector enters the condenser part of the FWG, and cools the vapour from evaporator to 56.8℃, and most of the cooling seawater goes back to the storage tank ⑮. The produced fresh water in the condenser is transferred to the fresh water tank by the distillate pump ⑨.

In the third part of performance test, data acquisition system acquired the 10 data of temperatures by using the model DT707-T & DT-2805 board (Data Translation Co.) and 3 data of pressures by using the model DPI 420 & PDCR 922 (Druck Co.).

The flowrates are measured by the reading scale through the naked eye, and the salinity is checked with the salinity indicator of the model YA-100A (Dae Sung Electric Electronic Co.)

In the fourth part of performance test, control system operates to adjust the designed value of temperatures, pressures, flowrates, and salinity. The salinity of the fresh water is examined by the salinity indicator ⑩ with a conductivity detector. If the salinity exceeds the specified level, the solenoid valve in the discharge line of the distillate pump ⑨ is automatically activated and the faulty distillate is returned to the brine side of the evaporative chamber The chemical injection unit ⑥ is supplied in order to keep scale free from fouling matter. Now total control system is carried out semi-automatically.

### 6-2. Measurements

The temperature and pressure measurements were extensively conducted to study the performance of the Fresh Water Generator. The T-type (copper-constantan) thermocouples are used in this study. Ten thermocouples were mounted on jacket cooling water line, cooling seawater line, combination ejector line, steamline, and so on to measure inlet and outlet temperature in the experimental FWG. Pressures are monitored at 3 locations of condensing part, evaporating part, and combination ejector as shown in Fig. 2, and the pressures in the other places are measured by pressure gauges. And also the mercury manometer was installed in the condenser part as a reference pressure to be compared with the output of pressure digital indicator. In order to measure flowrate, a float type measuring device (Japan Flow Cell Co.) was installed in the liquid line as shown in Fig 2. Figure 3 shows the experimental flow chart to measure and calculate the data. Atmospheric pressure, room temperature, pressures and temperature (gauge type), number of channels, and maximum measuring time are initially given. The quasi-steady state for the experimental conditions was decided when the value of (T'-T)/T is below 1/100, where T is the current temperature, and T' is the former temperature. After

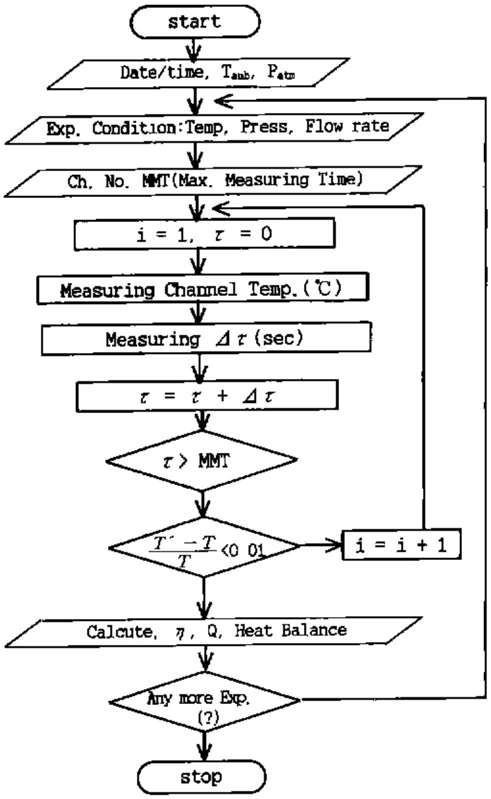



Fig. 3. Experimental Flow Chart.

finishing the data acquisition, calculation of the recovery energy, the heat balance, and the quantity of fresh water generated are mainly performed. Figure 4 shows the calibration results of thermometer. Temperature calibration range limits to 100℃ with consideration of the working temperature in FWG. In this figure, it is found that the correlation between voltage

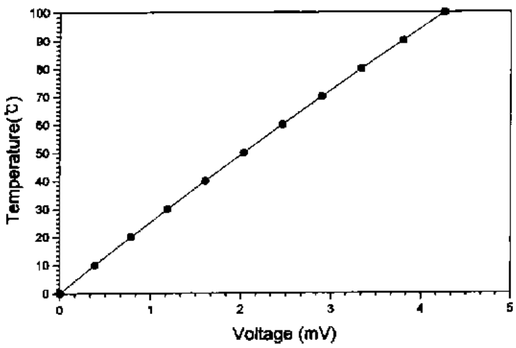



Fig. 4. Calibration Results of Thermometer.

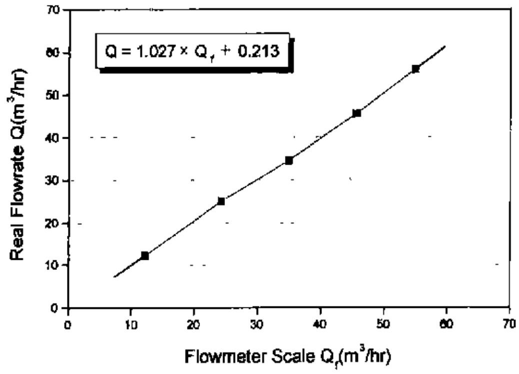



Fig. 5. Calibration Results of Flowmeter.

and temperature has a linearity. The calibration of temperature was carried out by the Korea Standards Research Institute. Figure 5 shows the correlation between flow meter scale and real flowrate for the flowmeter ⑤, and it is found that float-type flowmeter has a good linearity. The correlation equation is $Q_r$=1.027 * $Q_f$+0.213. where $Q_r$ is real flowrate, $Q_f$ is flowmeter scale. In this performance test, 5 flowmeters are used.

## 7. Experimental Results

Experiments are carried out with or without a heat source, and consist of preliminary test and real test. Real test is conducted by a heat source, which is called hot test method.

Preliminary test is required to check if the FWG keep high vacuum pressure in order to evaporate under low temperature. and conducted without heating steam, which is called cold test method.

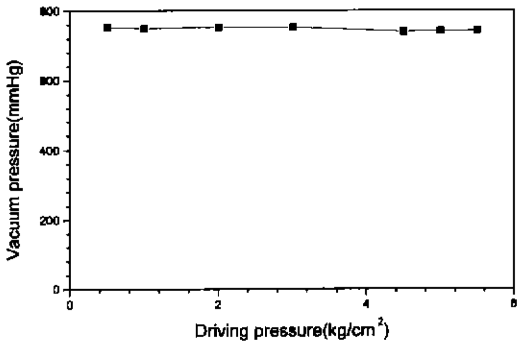



Fig. 6. Correlation between Driving Pressure and Vacuum Pressure.

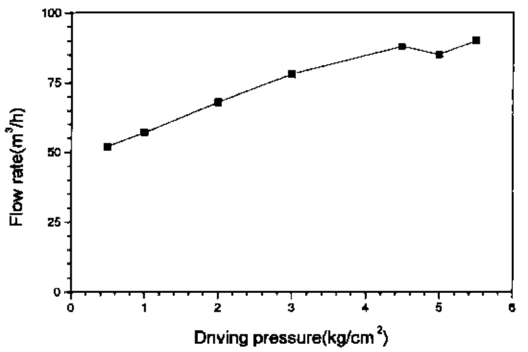



**Fig. 7. Correlation betweem Driving Pressure and Flowrate.**

Figure 6 shows the correlation between driving pressure and vacuum pressure. In this figure. the high vacuum of 745 mmHg is obtained sufficiently in the evaporating and condensing chambers through the all of the driving pressure ranges, where driving pressure means the pressure at the combination ejector ④ and flowrates are measured by flowmeter ③ in Fig. 2. All the produced vacuum pressure is nearly equal to about 745 mmHg. As a result, the performance of the combination ejector ④ in Fig. 2 is suitable for the developing FWG.

Figure 7 shows the correlation between driving pressure and flowrate. The driving pressure is proportional to flowrate. When the driving pressure is 5 kg/cm$^2$. the reduction of the flowrate is caused by the fluctuation of flowmeter level.

After finishing the preliminary test, real test for generating fresh water has been conducted to adjust the design specifications. The capacity of 30 ton/day is required for 1.5 ton/h boiler. but the capacity of existed experimental boiler in Fig. 2 is 1 ton/h. And thus electric heaters should be installed in order to supply the lack of 0.5 ton/h steam. The capacity of electric heater is required for 48 kW (elemental heater, 2 kW/ea*24ea), and this capacity is too large capacity to manage the electric heater in the experiment. For the experimental safety, some of the elemental heater are used to supply heat to feed water. From above reasons, real test is aimed to the capacity of 20 ton/day, and the maximum capacity of the FWG is calculated by the extrapolation method.

The data of Table 2 represent the temperature difference and generated fresh water, where temperature difference means the subtraction value of outlet temperature of jacket cooling water from inlet temperature of jacket cooling water The number of temperature difference is 6. and experiment is mainly conducted near to the temperature of 10°C. The range of vacuum pressure is 625 mmHg to 680 mmHg, and the ratio of calculated value to experimental one nearly equals to 1 within +/-23% error. And the obtained salinity is below 10 ppm. In real performance experiment, the quantity and quality of distillate were sufficiently accomplished.

## 8. Conclusions

Based on the experimental performance test of FWG. the following conclusions can be drawn.

(1) Economic/compact type FWG and its test facilities were developed

(2) During cold test, evaporating and condensing chamber pressures were kept at 745 mmHg vacuum

**Table 2. Temperature Difference $\Delta T(T_{ji}-T_{jo})$ and Generated Fresh Water.**

| Item \ $\Delta T(T_{ji}-T_{jo})$ | 6.5 | 7.4 | 10.0 | 10.1 | 10.3 | 14 8 |
|---|---|---|---|---|---|---|
| Vacuum Pressure (mmHg, Vacuum) | 680 | 661 | 674 | 625 | 635 | 662 |
| Temp. at Condenser Evap. Temp (Steam Table) | 41.9 (44.25) | 51 4 (49 22) | 37 6 (46.11) | 58.6 (51.12) | 51 4 (54.42) | 49.2 (56.60) |
| Flowrate of Jacket Cooling Water (m$^3$/h) | 72 | 70 | 36 | 62 | 58 | 28 |
| Inlet Temp. of Jacket Cooling Water (°C), $T_{ji}$ | 81.5 | 78.7 | 66.8 | 88.7 | 89.4 | 85.0 |
| Outlet Temp. of Jacket Cooling Water (°C), $T_{jo}$ | 75.0 | 71.3 | 56.8 | 78.6 | 79.1 | 70.2 |
| Flowrate of Fresh Water. Experiment (ton/day) | 19.68 | 22.56 | 14.81 | 25.92 | 25.92 | 17.88 |
| Calculated Flowrate of Fresh Water (ton/day) | 19.24 | 21.86 | 15.18 | 26.8 | 25.3 | 17.49 |
| Experiment/Calculation | 1.02 | 1.03 | 0.98 | 0 97 | 1.02 | 1.02 |
| Driving Press. (kg/cm$^2$) | 5.1 | 5.1 | 5 0 | 5 0 | 5.0 | 5 0 |

by newly developed ejectors

(3) The thermal energy saving for a FWG was estimated about 640 kW

(4) The capacity of 30 ton/day for distillate is sufficiently accomplished by the extrapolation method.

(5) Salinity of distillate is below 10 ppm

## Acknowledgment

Support for this work by the Citizens National Bank is gratefully acknowledged. The author wishes to thank Mrs. Sin Dol Kang and Tae Soo Kim at Kiwon Industry Co. helped to implement the performance test. Also the author would like to express the sincere gratitude to Drs. Kook Young Ahn and Young Il Jeong of Korea Institute of Machinery & Metals (KIMM) the kind assistance and fruitful discussions. And finally the author thanks Mr. Min Woo Yu, employed at KIMM, helped drawing the figures and tables.

## References

1. Frederiksen H.D.: "Water Crisis in Developing World: Misconceptions about Solutions", J. of Water Resources Planning and Management, Vol. 123, No. 5, pp. 303-314 (1997).
2. Harrington, R.L.. "Marine Engineering", The Society of Naval Architects and Marine Engineers, pp. 530-563 (1971).
3. Water Re-Use Promotion Center: "Desalination and Water Re-Use Technologies in Japan" Annual Report (1991-1992)
4. JSME Data Book: "The Thermophysical Properties of Fluids" (1983).
5. JSME Mechanical Engineers' Handbook B. Applications/B8: Heat Exchangers, Air Conditioning and Refrigeration, p. B8-17 (1985)
6. Hilpert, R.: "Waermeabgabe von Geheizen Drahten und Rohren", Forsch. Geb. Ingenieurwes., Vol. 4, p. 220 (1933).
7. Knudsen, J.D and Katz. D.L.: "Fluid Dynamics and Heat Transfer", McGraw-Hill Book Company. New York (1958).
8. Holman, J.P: "Heat Transfer", Int. Student Edition, pp. 491-523 (1986).