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Efficiency Improvement of Neka Gas and Steam Power Plant Units with a Joint Project Involving Thermoelectric and Absorption Cooling Technology

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Abstract: In this study, a specific arrangement of thermoelectric modules is presented for energy recovery in the main steam condenser unit of Neka power plant. Thermoelectric modules convert a section of losing energy to electricity with a combination of semiconductor materials. In this study as a result of variation of thermal resistance of condenser pipes, a definite amount of steam entering the condenser is excess that has been checked with design of one steam split line to two systems. A part of steam transferred to absorption chiller and improved the efficiency and also the other sent to desalinate system. It has been shown that with the implementation of this plan, the output of a power plant will increase to 450 MW and power losses of warm days will be saved as much as 11257 MWh.

Key words: Steam unit, surface condenser, thermoelectric, absorption cooling, water desalination

INTRODUCTION

The steam power plants that works with Rankine cycle, convert approximately 37% of energy to work. Maximum Loss of energy occurs in the condenser that is above 50%. During this study, a part of the energy loss in condenser has been saved by using thermoelectric technology. Gas turbines at ISO conditions have approximately 33% of efficiency. By reduce the inlet air temperature, the efficiency of the turbine will be decreased. Hence by design of air inlet cooling and getting air temperature to 15°C, the losses will be reduced (Nguyen *et al*., 2001).

By design and installation of circular semi-conductors on the condenser tubes, according to the Seebeck effect, the significant amount of energy can be converted to electricity (Omer *et al*., 2001) and (Cheng and Lin, 2005). Modules installed on the pipe surface, the thickness of the pipe will increase and the pipe distances and so the pressure drop will be changed. With the change of the coefficient of thermal conductivity of pipes, heat flux will be directly changed that affects the temperature of cooling water (Glatz *et al*., 2006).

Selecting proper materials and low thermal conductivity tube new rate of 2.5%, the condenser thermal load will be less. The reason is that the load is directly affected by the change in the thermal conductance of condenser heat pipes. Consequently, to reduce condenser load and fix the shell pressure or should reduce the level of heat exchange namely the quantity of pipes or should decrease the steam flow rate entering the condenser. By applying these conditions and decrease the condenser thermal load, cooling water temperature leaving the condenser is less (Luan *et al*., 2005).

With the survey conducted, the remaining steam can be used in the hot months of the year in absorption chiller to cooling the turbine air inlet. Therefore, the increase the efficiency of gas units has been studied. The second application of surplus steam is for use in the evaporative desalination system that can be explored for the remaining months of the year.

CHANGES MADE TO STEAM CONDENSER UNIT AND THERMOELECTRIC ENERGY RECOVERY

In order to apply the necessary changes in the current system and increase efficiency, we must examine the system thoroughly.

Present status of the steam cycle: Examining existing situation of the steam cycle at design load, dimensional and operational specification is given in Table 1.

With surveys conducted and study the status of condensation and cycle in design load that present by manufacturer, the calculations are as follows:

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Table 1: Power plant steam condenser unit features

Title	Unit	Value	Title	Unit	Value
Impure production (GDP)	$m^3 h^{-1}$	44831	Cooling water flow rate	MW	440
Pipe heat conduction	$Wm^{-1} K^{-1}$	100	Fuel consumption	$m^3 h^{-1}$	110000
Number of condenser tubes	-	15600	Condenser heat load	MW	523.6
Tube length	m	10	Condenser pressure	kPa	6.8
Tube inside diameter	mm	22	Steam flow rate of condenser	ton h^{-1}	855.88
Tube outside diameter	mm	24	Inlet cooling water temperature	°C	21
Inside cooling surface	m^2	10782	Outlet cooling water temperature	°C	31.1
General heat transfer coefficient of condenser	W $m^{-2} K^{-1}$	4201	Steam temperature in condenser	°C	38.2
Perpendicular to flow distance between the pipes	mm	32	Logarithmic mean temperature	°C	11.56
Parallel to flow distance between the pipes	mm	27.71	Steam quality input to the exchanger	%	91.3

$$Q = m_{cw}.cp.\Delta T_{cw} = 44831\ m^3\ h^{-1} \times 995.8\ kg\ m^{-3} \times 4.179 \frac{kj}{kg°C} \times (31.1-21)°C \frac{1}{3600} = 523409.7\ kW \quad (1)$$

The heat that is repelled by cooling water, through the turbine outlet is transferred to condensed water:

$$Q = \dot{m}_s.\Delta h \rightarrow 523409.7\ kW = 855.88 \frac{ton}{h} \times \frac{1000}{3600} \times \Delta h \rightarrow \Delta h = 2201.5\ kj\ kg^{-1} \quad (2)$$

If the saturated steam has pressure of 6.8 kPa and condensing temperature 38.3°C, the latent heat of vaporization will be 2410.66 kJ kg^{-1}:

$$x = \frac{\Delta h}{h_{fg}} = \frac{2201.5}{2410.66} = 91.3\% \quad (3)$$

Energy recovery with thermoelectric modules' installation on the condenser tubes: General heat transfer coefficient is calculated based on the internal surface of the tube condenser that is composed of three parts: Thermal convection of internal film of tube, thermal conduction of tube thickness and thermal convection of outer condensate film (Fig. 1):

$$U = \frac{1}{\frac{1}{h_i} + \frac{A_i \ln\left(\frac{r_o + t_{subs} + t_{TE} + t_{overc}}{r_i}\right)}{2\pi k_{eq} L} + \frac{A_i}{A_o}\frac{1}{h_o}} \quad (4)$$

Semiconductor rings and ring side and the middle insulators, include a single thermoelectric module with the parallel arrangement of the heat flux, have a coefficient of conduction as follows (Goncalves *et al*., 2008):

$$k_{TE} = \frac{k_p l_p + k_n l_n + k_{li} l_{li} + k_{mi} l_{mi}}{l_{TE}} \quad (5)$$

But with an underlying and the upper ceramic coated of thermoelectric, the total tube coefficient of thermal conduction will be:

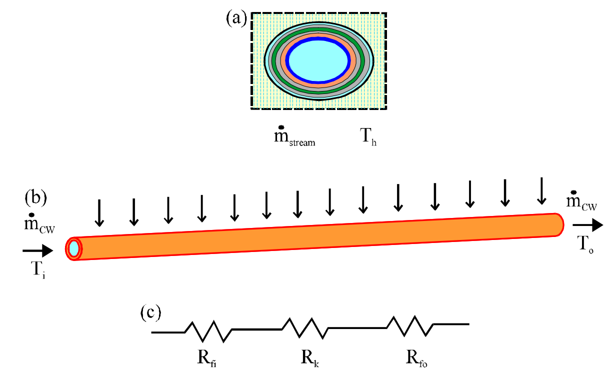

Fig. 1(a-c): (a) Internal film of cooling water, (b) Outer condensate film and (c) Series thermal resistance

$$k_{eq} = \left(\frac{\frac{\text{In}\ (r_o + t_{subs} + t_{TE} + t_{overc})}{r_i}}{\frac{\text{In}\ (\frac{r_o}{r_i})}{k_{cu}} + \text{In}\ (r_o)} \right) \quad (6)$$

Tube heat transfer coefficient can be calculated using equation petkov. The arrangement of a single module is presented in Fig. 2.

Since the flow in the tube is turbulent and fully developed, the coefficient of thermal convection can be calculated by Nusselt number:

$$Nu_d = \frac{\left(\frac{f}{8}\right).Re_d.Pr}{1.07 + 12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(pr^{\frac{2}{3}} - 1\right)} \left(\frac{\mu b}{\mu w}\right)^n \quad (7)$$

Since, the wall temperature is more than the temperature of cooling water, consider n equal to 0.11. The petkov equation will be solved at film temperature (Glatz *et al*., 2006). By computation of the Nusselt number, the convective heat transfer coefficient inside the tube will be found:

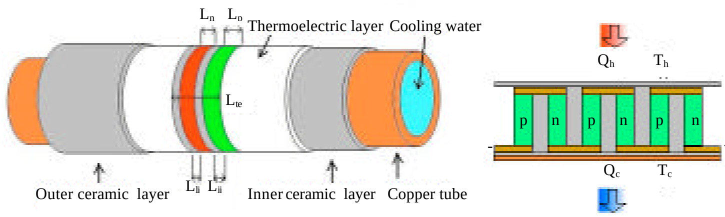

Fig. 2: A simple schematic arrangement of a single module

$$Nu_d = \frac{\bar{h}.d_i}{k_f} \tag{8}$$

Input heat flux to the cooling water control volume equals the heat flux through the material thickness.

In addition, the convective heat flux transfer from film, inner side of the copper pipe, lower ceramic layer, the thermoelectric module, outer layer of ceramic film and reaches to the condensate steam conductive (Pramanick and Das, 2006). If we assume the around of pipe as a control volume, the heat fluxes will be as follows:

$$\begin{aligned}
&Nu_d = \frac{\bar{h}.d_i}{k_f}\\
&Q = \dot{m}_{cw}C_{cw}(T_o - T_i)\\
&Q = h_c A_i (T_{wi} - T_{fi})\\
&Q = \frac{2\pi k_{eq} L}{\frac{In(r_o + t)}{r_i}}(T_{wo} - T_{wi})\\
&Q = h_h A_o (T_{fo} - T_{wi})
\end{aligned} \tag{9}$$

Therefore, the inner and outer wall temperature and then the average convective coefficient of outer condensate film will be obtained. Using these values, the amount of U will be 4201.14 $Wm^{-1}K^{-1}$.

Utilization of the ring thermoelectric modules on the condenser tubes, converts some of the energy exchange in the cooling water to electricity. In fact, the temperature difference between condensed steam and cooling water down the pipe is converted to potential difference according to the Eq. 1:

$$P = \left(\left(\frac{\Delta a \Delta T_{in} C_{cw} \dot{m}_{cw} r_i}{k_p l_p + k_n l_n + k_{ins} l_{ins}}\right).\left(1 - \exp\left(\frac{UL}{C_{cw}\dot{m}_{cw}}\right)\right)\right)^2 . \frac{\pi(l_{TE} - l_{ins})\ln\left(\frac{r_o}{r_i}\right)}{4n\left(\rho_p + \rho_n\right)} \tag{10}$$

Installation of annular modules reduces the distance perpendicular to and parallel with the steam flow. Also, by increasing the pressure inside the exchanger, the volume

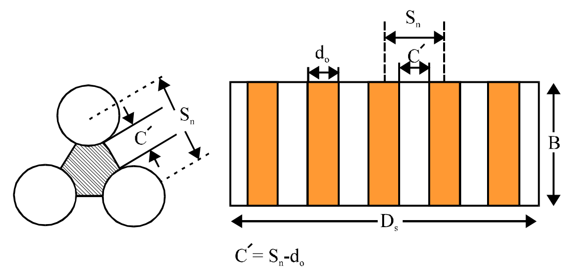

Fig. 3: Pipe arrangement, dimensions and distances between the tube and baffles

of free space within the exchanger will be reduced. There are two solutions to maintain pressure within the condenser shell (Omer and Infield, 1998). First, reduce the number of condenser tubes and the second, reducing the inlet steam flow to the condenser.

Reduce the number of tubes, cooling water flow rate will change and this approach is unacceptable for minimal changes. By optimizing the thickness and the distance between the condenser tubes, thermal load will reduce and thus the amount of low pressure steam will remain in turbine output. The arrangement of pipes and distances between the tube and baffles are shown in Fig. 3.

To improve the pressure drop, we can slightly reduce the distance between the baffles:

$$\begin{aligned}
&\Delta P = \frac{f.G^2.D_s(n_b + 1)}{5.22 \times 10^{10} D_e S.\varphi_s}, \quad G = \frac{W}{a_c}\\
&a_c = \frac{D_s(S_n - d_o)B}{144 S_n}
\end{aligned} \tag{11}$$

Condenser pressure drop compared with the current design, showed that the pressure drop rate has been increased 8.1%. So the pressure inside the shell is reduced slightly. Constant heat flux of cooling water to condensing steam decreases the coefficient of heat flux from 100 to 97.47 $Wm^{-1} K^{-1}$. So, with the loss of about 2.5% of the inlet steam flow, we applied the thermal balance of the new system to stabilize the pressure of exchanger.

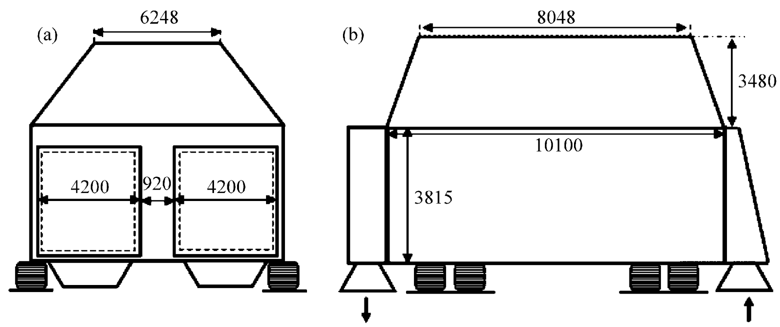

Fig. 4(a-b): (a) Longitudinal and (b) Transverse view of the condenser shell and its overall dimensions (mm)

Therefore, the pressure of exchanger will rise if the ratio of cooling capacity to a total volume of space is not preserved within the exchanger. Therefore, the space between the left and right of the pines should be used. Longitudinal and transverse view of the condenser shell and its overall dimensions are shown in Fig. 4. Considering that the 92 cm distance between the two halves of the exchanger is unused, it is possible to reduce the size of 65 cm:

$$\Delta w = \frac{\Delta V}{A_z} = \frac{(95.37 - 70.6)}{3.815 \times 10} = 0.65 \text{ m} \quad (12)$$

For the 2.5% reduction of the condenser inlet steam, the steam is removed from upstream. The superheated steam at Q = 2236 ton h^{-1} is split from intermediate turbine at 8.3 bar pressure (h_{ex} = 3035 kJ kg^{-1}) and will be utilized in absorption cooling or water desalination system.

Then, this condensate steam at 77.5c (h_o = 324.4 kJ kg^{-1}) will be sent to outlet line of low pressure heater and return to the cycle. The output energy of steam cycle, is consists of two terms: the first term is a value that has the ability to convert electricity and the second is the amount that will be reduced from condenser losses (Choi *et al*., 2007).

According to performed calculation, 26005200 ceramic coating modules are needed. The cost of installation plus the cost of power electronics to be collected, on average, is equal to 5000500$. As a result, the return on investment yearly, assuming the price of a kilowatt hour of energy equal to 450 riyals would be:

$$\text{ROR} = \frac{5000500 \times 11000}{7.2 \text{ MWh} \times 1000 \times 9320 \times 450} = 2.3 \quad (13)$$

The total power increase of four steam units would be equal to 28.8 MW.

GAS UNIT POWERED BY ABSORPTION COOLING

Here, the absorption system used and the amount of power that can be received from this system will be investigated.

Specifications of required absorption chiller: Neka combined power plant consists of two Siemens V94.2 gas units with nominal capacity of 2×137.5 MW and recovery boilers with the capacity of 160 MW. Since, temperature in Neka coast is reached in warm years are over and the hottest day of the ISO temperature to 38′, The use of an inlet air cooling system by absorption chiller to maintain nominal efficiency is recommended. The capacity of a chiller is required energy of air cooling from 37′ to 50′ that consists of sensitive and latent terms. The air pressure is slightly more than an atmospheric air pressure in the hot days of summer (the Caspian Sea) equals 103 kPa and enters to a cooling system with 37′ temperature, 0.78 relative humidity and flow rate of 506 kg sec^{-1}. The outlet condition is atmospheric pressure 101 kPa, 15c temperature and 60% relative humidity (Fig. 5).

Moisture in the moist air through the cooling coils will condense and the amount of distilled water will produce.

Humidity partial pressure and specific humidity of air entering the cooling system (at 37°C with relative humidity 78%) are equal to:

$$\begin{aligned} P_{vi} &= \varphi_i P_{g@T_i} = 0.78 \times 6.3 \text{ kpa} = 4.714 \text{ kPa} \\ P_a &= P_i - P_v = 103 - 5.17 = 97.83 \text{ kPa} \\ \omega_i &= 0.622 \frac{P_{vi}}{P_{ai}} = 0.622 \times \frac{4.914}{97.83} = 0.03124 \end{aligned} \quad (14)$$

The output values of the cooling system (temperature 15 degrees Celsius with relative humidity 60%) will be equal to:

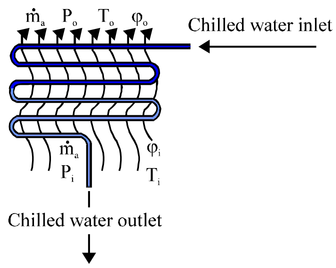

Fig. 5: Assumed coils containing cold water and air pass through it for cooling

$$
\begin{aligned}
P_{vo} &= \varphi_o P_{geTo} = 0.6\times 1.7051 \text{ kpa} = 1.023 \text{ kPa}\\
P_{ao} &= P_o - P_{vo} = 101.3 - 1.023 = 100.277 \text{ kPa}\\
\omega_o &= 0.622\frac{P_{vo}}{P_{ao}} = 0.622\times\frac{1.023}{100.277} = 0.00635
\end{aligned} \tag{15}
$$

The specific humidity reduction and air cooling from 37-15 degrees, simply implies that some of the steam is condensed in the air and its value is equal to:

$$\dot{m}_1 = \dot{m}_a\ (\omega_1 - \omega_2) = 500\times\left(0.03124 - 0.00635 = \frac{12.45 \text{ kg}}{\text{s}}\right) \tag{16}$$

The amount of distilled water, multiplied by 3.6 for the two units will be equal to 90 cubic meters of gas that can enter the water tank of the second plant. Cooling heat exchanger for heat exchange can be written:

$$
\begin{aligned}
&\frac{Q}{\dot{m}_a} = C_{pa}\ (T_o - T_i) + \omega_o h_o - \omega_i h_{vi} + (\omega_1 - \omega_2)h_1\\
&= 1.0057\ (15-37) + 0.00635\times 2528.9 - 0.03124\times 2568.2\\
&+(0.03124 - 0.00635)\times 63 = -84.74 \text{ kJ kg}^{-1} \text{ dry air}
\end{aligned} \tag{17}
$$

Due to the constant flow rate of air, the power exchange rate shows the cooling capacity of chiller will be:

$$Q = 500 \text{ kg sec}^{-1}\times\left(-84.74 \text{ kJ kg}^{-1}\right) = -42370 \text{ kW} \tag{18}$$

Divide the cooling capacity into the 3.5, the cooling rate is obtained of 12105 tons of refrigeration. As a result, two gas units with 24210 ton chiller are needed overall.

The coefficient of performance is the essential parameter in chiller design, which will be computed as following relationship:

$$COP = \frac{\text{Cooling}}{\text{Energy}} = \frac{42370\times 2}{67344} = 1.26 \tag{19}$$

The review of existing chillers, four 6, 000 tons chiller can be used. The temperature of inlet cooling water to the chiller in summer is assumed 34 degrees of Celsius.

Cooling water flow rate of large chillers is average of one cubic meter per hour per a ton of refrigeration. The total heat that must be disposed in the chiller condenser is equal to the total input energy and produced cooling, therefore:

$$
\begin{aligned}
&Q_c\ \dot{m}_{ccw}.C_p\ (T_{ccwo} - T_{ccwi})\\
&= 6000\times 1\frac{m^3}{h}\times\frac{1\text{ h}}{3600\text{s}}\times 993\frac{\text{kg}}{m^3}\times 4.174\frac{\text{kj}}{\text{kg°C}}\ (T_{ccwo} - 34) = 152084.4 \text{ kW}\\
&\Rightarrow T_{ccwo} = 39.51
\end{aligned} \tag{20}
$$

Internal consumption of four 6000 ton absorption chillers, including pumps, water pumps and condenser cooling pump is the average 1.5 MW.

Efficiency improvement of gas unit: According to Siemens V94.2 gas turbine documents reviewed and the parameters measured in gas unit functional test, the compressor and turbine design conditions (ISO) is calculated. Considering that the compressor outlet temperature of 319.5°C, the specific heat capacity of air in the compression process used by the compressor and the compressor consumed work is:

$$
\begin{aligned}
&\bar{c}_{pk} = \frac{h_2 - h_1}{T_2 - T_1} = \frac{336.34 - 15.084}{319.5 - 15} = 1.055 \text{ kJ/kg°C}\\
&Pv = \dot{m}_a\bar{c}_{pv}\ (T_2 - T_1)\\
&= 500\times 1.055\ (319.5 - 15) = 160623 \text{ kW}
\end{aligned} \tag{21}
$$

With turbine inlet temperature of 1000 degrees in base load and equal to 1040°C in the peak load, flow rate of combustion products identical to 508.2 kg $\sec^{-1}$ and the temperature of the exhaust fume is 512.3 degrees of Celsius. Therefore, the turbine power is:

$$
\begin{aligned}
&P_T = \dot{m}_T\bar{c}_{pT}\ (T_4 - T_3)\\
&= 508.52\times 1.20004\ (1000 - 512.3) = 297616 \text{ kW}
\end{aligned} \tag{22}
$$

515 kilowatts of Mechanical losses of the system can be reduced from the efficiency of the turbine and with 98.42% generator efficiency, net shaft power is:

$$P_k = (297616 - 156766 - 515)\times 0.9842 = 138118 \text{ kW} \tag{23}$$

After reduce the unit internal consumption, the output power and efficiency are 137.6 MW and 33%, respectively. According to Siemens's documentation, temperature increment of compressor inlet air from 15-37 c, the air flow rate will be decreased about 9%. Thus, output power of basic load will be decreased from 100-84% and from 107-94% in peak load. Consequently, recovery of lost power at peak load and base load times will be:

$$P^1 = \frac{107-94}{107} \times 137.6 = 16.75 \text{ MW} \quad (24)$$

As regards, the time operation in the summer at base load is 2076 h and at peak load is equal to 900 h, energy recovery of a gas unit in a year is equal to:

$$(900\times16.72+2076\times22.02-1.5\times2976) = 112575.432 \text{ MWh} \quad (25)$$

The cost of installation of four 6000 ton chiller unit is equal to an average of 200 billion riyals, As a result, the Return on Investment (ROR) equal, assuming the price of a kilowatt hour of energy equal to 450 Rials per year would equal to:

$$\text{ROR} = \frac{2\times10^9}{112578.432 \text{ MWh}\times1000\times450} = 3.95 \quad (26)$$

USE OF EVAPORATING WATER DESALINATION

Philosophy of water desalination in Neka power plant: Water needed is supplied by three wells that is located in 20 km in the east of the city. Given the low quality of well water in recent years, increasing need for agricultural water in the city of Neka and population, the supply of steam cycles reparative water and other power plant needs will be compromised. By the same token, the issue of sea water is in the hands of experts (Dai *et al*., 2003).

The superheated steam is used in an absorption cooling system in hot seasons and in other months of the year the temperature reaches below the 15 degrees Celsius. Therefore, the unused steam is suitable for feeding system for evaporative desalination.

Evaporator of MED system consists of multiple tandem units that the surface temperature and pressure drop of the first unit (hot side) to the terminal unit (cold side) is required.

Each unit consists of a horizontal pipe that and due to gravity, inlet sea water sprayed from the top on them. The hot steam enters to the pipe.

Hot pipes are cooled by cold sea water and their hot steam is converted to distilled water.

The latent heat of the input superheated steam warms the input sea water and part of it will evaporate. Due to evaporation, the sea water is concentrated in a single floor which is called brine.

Steam temperatures rose in the upper part of unit is less than the steam temperature of the supply unit. However, the steam from evaporating sea water can still be used as the heating interface and the next unit as heating steam fed into the horizontal pipe of the second unit.

This process continues and steam is condensed in a conventional tube and shell exchangers in second unit. This exchanger is named last condenser, will be cooled by sea water. In the last condenser outlet, part of the sea water will be used as feed and the remaining puts out to sea (Huang *et al*., 2000; Yilbas and Sahin, 2010). Brine and distilled water will be collected and by centrifugal pumps. The generated steam pressure of first unit is absolute pressure of 0.3 bar.

CONCLUSION

Installation of Thermoelectric modules on the main condenser tubes of two steam units increased the net power about 14.4 MW. Because of the extra steam in the cycle, it was proposed to send the 21.65 tons of steam per cycle to the absorption chiller. With the cooling in the warm months, it is possible to cooling the air inlet to gas turbine. Therefore, the yearly power increase was about 6.3 MW. Since the surplus steam is used in the absorption chiller in the warm month, it can be used in the evaporative water desalination system in the other month of the year. It is possible to produce the 174 $m^3 h^{-1}$ water by excess steam.

Since, the water supply of Neka power plant will be threatened, water supply from sea by evaporating systems in cold month of the year has a better efficiency. The membrane systems can be used in the warm months of the year for the whole of water requires 360 $m^3 h^{-1}$ and can be start up in autumn with 50% of capacity. Select the appropriate configuration of thermoelectric systems, absorption cooling, evaporation and water desalination membrane depends on the amount invested. This review has been tried to introduce the amount of energy and resource efficiency, while using existing technology.

ACKNOWLEDGMENT

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NOMENCLATURE

P	=	Recovered power
$\Delta\alpha$	=	Seebeck difference of p and n semiconductors
ΔT_{in}	=	Temperature difference between steam and inlet cooling water
C_{CW}	=	Specific thermal capacity
$\dot{m}_{CW}$	=	Mass flow rate of cooling water
r_i, r_o	=	Inner and outer radii of the condenser tubes covered with a module

k_p, k_n = Conduction heat transfer coefficient of p and n semiconductor base
L_p, L_n = The semiconductor, lengths (in cylindrical coordinates)
k_{ins} = Conduction heat transfer coefficient of module's electrical insulation
l_{ins} = The length of module's electrical insulation
D_e = Hydraulic diameter
n_b = Number of baffles
f = Mean friction coefficient inside the shell
U = General heat transfer coefficient of condenser tubes
L = Condenser length
l_{TE} = The overall length of a module consists of p and n rings
n = The number of modules on a pipe
ρ_p, ρ_n = P and n semiconductor density
B = Baffles distance
D_s = Inside shell diameter
a_c = effective cross section of in passing Fluid
W = Mass flow rate
G = Mass flow rate per unit area
S = Cooling surface(outside the tubes)
ϕ_s = Pressure drop correction factor

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