Desalination and Water Treatment
www.deswater.com
1 (2009) 42–57
1944-3994/1944-3986

Alternative primary energy for power desalting plants in Kuwait: the nuclear option II — The steam cycle and its combination with desalting units

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Received 25 March 2008; Accepted 22 July 2008

ABSTRACT

In the first part of this study, it was shown that the use of nuclear option to fuel the cogeneration power desalting plants (N-CPDP) in Kuwait is more economical than the most efficient gas/steam turbines combined cycle GSCC using oil or natural gas. The power cost produced by N-CPDP was found to be at least 35% less than that of the GSCC. Furthermore, the use of fossil fuel in Kuwait would consume all of its oil reserves in less than 30 years if its present rate of fuel consumption prevails. The very high cost of oil fuel and the emission of greenhouse gases due to its burning (with its negative environmental effects) favor the use of nuclear energy. It was found that Kuwait, Saudi Arabia, Egypt, and United Arab Emirates satisfy the conditions required to consider the nuclear option in terms of: (1) needed additional power capacity, (2) needed seawater desalting capacity, (3) size of the electricity grid, and (4) the basic infrastructure required to build the N-CPDP. The use of a light water pressurized water reactor, the AP-600 (600 MW nominal power output), in N-CPDP was anticipated for Kuwait. This paper gives the details of the AP-600 steam cycle and its combination with thermal desalting plants with multi-effect distillation (MED), multi-stage flash (MSF), and thermal vapor compression (TVC) desalting systems. The water costs due to the coupling of MED, MSF, or TVC to the AP-600 nuclear power plant (NPP) were also calculated. Based on the required water-to-power ratio, either a back pressure steam turbine (BPST) or an extraction condensing steam turbine (ECST) was chosen. For the BPST, a maximum water-to-power ratio of 97 MIGD to 451 MW was obtained. Then, the use of ECST was chosen with a seawater desalting capacity of 50 MIGD. The results show that the cost of desalinating water with nuclear power is cheaper than that produced by fossil-fired plants, given the high cost of fossil fuel. Further, the estimated costs of producing electricity and water with MED+NCPP are lower than MSF+NCPP and TVC+NCPP. The unit product cost of the desalted water was calculated to be in the range of \$0.87–1.4 per m^3 of product water based on a plant capacity of 227.3×10^3 m^3/d. The presented techno-economic results for the different desalination scenarios can help decision makers in choosing the best option that is suitable for the Kuwaiti conditions.

Keywords: Nuclear cogeneration power desalting plants; Nuclear plants coupled with desalination systems; Economics of nuclear desalination.

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1. Introduction

Kuwait and other Arabian Gulf Cooperation Countries (GCC) depend mainly of desalted seawater to satisfy their potable water needs (more than 90% in the case of Kuwait and Qatar). The consumption of desalted water and electric power is on the rise in all of these countries which result in the consumption of huge amounts of fuel (mainly natural gases and fuel oil). In most cases the MSF desalting system is used and is combined with steam turbines to satisfy its need of relatively low pressure steam extracted from these turbines. The MSF system is known for its high energy consumption (about 250 kJ/kg thermal energy and 4 kWh/m^3 pumping energy). Fossil fuel is finite, very expensive, and cannot be supplied indefinitely. It seems that nuclear energy is the only economical viable large-scale alternative to fossil fuels for the generation of electric power as well as desalted water [1].

The steam conditions used in nuclear power plants (NPP) are different than those used in fossil fuel plants. As compared to the NPP, the steam in fossil fuel plants is generated at higher temperature (highly superheated), enthalpy and pressure, and lower spesicfic volume. In the NPP, the steam supplied to the high pressure (HP) turbine is almost saturated vapor. Thus, a wet expansion occurs in both HP and low pressure (LP) turbines. This exposes the turbine blades to water drop erosion (WDE), which necessitates the use of a moisture removal. Hence, the steam cycle in the NPP generally has (1) high steam mass and volumetric flow rates at the turbine inlet and (2) larger size turbines as compared to those used in case of fossil fuel power plants. Table 1 gives typical steam conditions and mass flow rates in these two types of plants for a 600 MW nominal plant capacity.

Due to the intensive energy consumed by the desalting plants, they are generally combined with power plants to satisfy their needs of thermal energy and/or mechanical energy. The thermal desalting methods of MED, MSF, and TVC are usually combined with steam turbines to extract steam from these turbines. This steam is at relatively low pressure and is extracted after its expansion in the high pressure part of the turbine is being done. Thus, it produces work before it enters the desalting units. These thermal desalting plants also consume pumping (mechanical) energy to run the pumps moving the streams in the desalting systems. In addition to the thermal desalting plants, there are mechanically driven desalting systems which include reverse osmosis (RO) and the mechanical vapor compression (MVC) system. These desalting units are usually driven by electric motors. When both power output and desalted water are produced from a single plant, it is called a cogeneration power desalting plant (CPDP) and it can use nuclear or fossil fuel.

Table 1
Typical exit steam conditions in various types of power plants

Parameters	Power plant type	
	Fossil fuel	PWR nuclear
Temperature, ºC	535	272.5
Pressure, bar	150	57.2
Enthalpy, kJ/kg	3419	2787
Specific volume	0.0225	0.0342
Mass flow rates, kg/s	562	1063

Desalination plants can be operated as a single-purpose plant or as a cogeneration plant. For single-purpose plants, energy is needed only for desalting and is produced on site (e.g., fuel fired boilers supply steam to thermal desalting plants and small power plants to supply the pumping energy needed; or large power plants to supply the mechanical energy needed to the RO and MVC units). Recently in China a nuclear reactor was fully dedicated to supplying thermal energy to a MED desalting system. For co-generation plants, only part of the energy is utilized for desalting. A co-generation plant produces both electricity and water simultaneously.

This paper describes the technical as well as the economic aspects of coupling of an AP-600 NPP, when it operates as CPDP, with thermal desalting process of MED, MSF, or TVC. This study includes the following cases:

- producing power only,
- producing both power and water with high water-to-power ratio by using BPST, and
- producing both power and 50 MIGD desalted water by using ECST coupled to MED, MSF, or TVC desalting systems.

Each one of these desalting systems requires specific supply steam conditions (e.g. 0.5 bar for LT-MED, 1.5 bar for MSF, 3.5 bar for TVC); and these supply conditions affect the power output of the plant as well as the enegy cost of producing the desalted water.

2. Base cycle (nuclear steam power cycle with the AP-600 unit)

A sketch of the NPP cycle using the AP-600 is shown in Fig. 1. If the cooling water inlet to the condenser is at 30.5ºC, the AP-600 gives 619 MWe gross power output (600 MWe net output and 1933 MW core thermal output) and 35% net power plant efficiency. The used PWR in this cycle has the advantage that if fuel leaks in the core, no radioactive contaminants pass to the turbine and condenser loop.

In this cycle, the generated steam is supplied to a steam turbine driving an electric generator. The exhausted steam

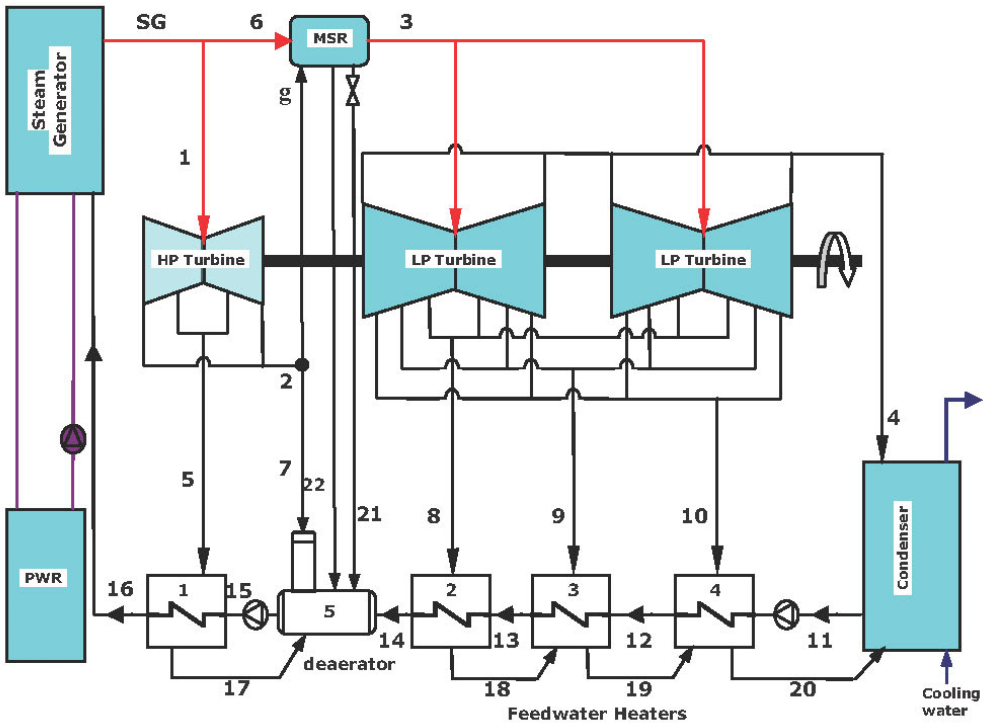

Fig. 1. AP-600 nuclear power plant steam cycle.

from the turbine condenses in the condenser by cooling water (once through seawater here). The steam condensate is pumped from the condenser to the steam generator through regenerative feedwater heaters by a series of pumps.

The available data for the AP-600 NPP steam cycle are very limited, but it can be constructed from the main data given by the AP-600 nuclear reactor and the rules of the steam power plant industry, namely:

- The generated steam is almost saturated vapor at a pressure of 57.2 bar (condition 1 in Fig. 1) and is at a flow rate of 1063 kg/s.
- The steam turbine consists of one dual-flow HP cylinder and two dual-flow LP cylinders.

In contrast with steam turbines in fossil fuel power plants, the steam turbines in NPP are operating under wet steam conditions in the turbine high and low pressure sections. However, the quality of steam expanded in the turbine sections should be greater than 0.88 to limit the effect of water drop errosion (WDE) on the turbine blades.

If the steam leaving the HP turbine, at condition 2 in Fig. 1 continues directly to expand in the LP turbine, its quality χ will be lower than the minimum accepted limit of 0.88. This necessites the use of a moisture separator after the HP turbine where water droplets are mechanically separated (see Fig. 2a). When this steam becomes almost saturated vapor, it is reheated to the superheated condition of point (3) (i.e., $T_3 = 240°C$ and 11 bar). Sometimes the separator and the reheater are combined together in one shell called moisture separator–reheater (MSR) [2] as shown in Fig. 2b. The value for the temperature T_3 is chosen to give the following steam conditions:

- χ_4 (at the LP turbine outlet) ≥0.88 with typical turbine isentropic efficiency, $\eta_{is} = 0.8$,
- reasonable terminal temperature difference (TTD) in the reheater, $T_1-T_3 = 32.5°C$.

3. Feedwater heaters arrangement

The feedwater heaters layout in the nuclear steam cycle greatly affects the performance of the plant. One layout using seven regenerative feedwater heaters (six closed and one de-aerator) was suggested by Famiani [3]. Another layout in a study conducted at the Oak Ridge National Laboratory by Williams [4] recommended the use of fewer regenerative feedwater heaters, namely four. The study concluded that only 1% loss in net output is obtained when using less number of feedwater heaters. This drawback can be outweighed by the reduction in the cycle complexity.

Therefore, in the present study, to reduce the cycle complexity, five feedwater heaters are proposed (four closed and one de-aerator). The feedwater heaters are located such that the temperature (and thus the enthalpy) difference between the feedwater heaters are almost

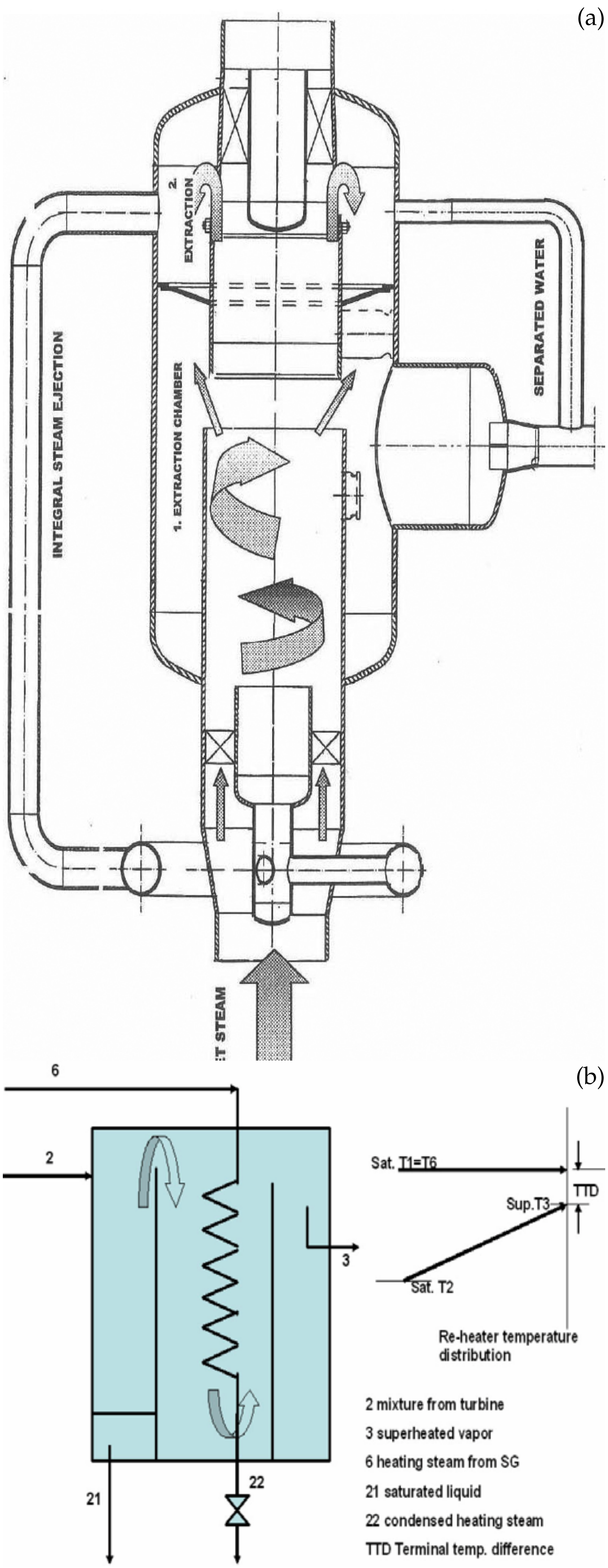

Fig. 2. Types of moisture removal and reheaters. (a) moisture separator; (b) moisture separator and reheater (MSR).

equal. Unfortunately, this cannot be done for the HP feedwater heater as P_2 is chosen to give χ_2 >0.88. The terminal temperature difference (TTD) and the drain cooler in each feedwater heaters are taken the same for all of them.

4. Steam turbine size

The steam turbine of the AP-600 is a single shaft type, and the rotating blades of its HP and LP sections are exposed to WDE. The WDE rate is proportional to the blade circumferential speed. The erosion effects of wet steam become more pronounced as the LP stage blades become longer with high tip circumferential speed. For the longest full speed, the tip circumferential speed is limited to about 750 m/s, although it could reach 830 m/s in newly developed machines. In the low-speed turbines (1500 or 1800 rpm), the circumferential speed does not exceed 500–530 m/s. To reduce the velocity, the exhaust area of the turbine should be enlarged. Therefore, the turbine is designed to operate at half speed and two exits. Low-speed turbines have the disadvantages of being heavier than full speed ones, need more advanced technologies, and have a higher capital cost. The extraction of steam to desalting plant reduces the volumetric flow rates at the end part of the turbine, and thus, reduces its size and the difficulties related to high circumferential velocities. Evidently, this is one of the advantages of combining this cycle with desalting plants. In this study, steam is extracted from the LP turbine at pressures of 0.5 bar, 1.5 bar, and 3.5 bar, for the feedwater heaters 2, 3, and 4, respectively (Fig. 1).

Currently, the largest high-speed wet steam turbine to enter service is rated at 1032 MW. This is the 3000 rpm Siemens turbine at the Trillo nuclear plant in Spain. It consists of one dual-flow HP cylinder and three dual-flow LP cylinders with a 1118 mm long blade at the end [TC-6F44].

In October 2006, Doosan Heavy Industries and Construction Company announced that under an agreement with GE, they will provide two 1455 MW 1800 rpm steam turbines for the two units of Shin Kori plant in South Korea. These will be the largest 60 Hz steam turbines in the world when they enter service in 2013.

5. Thermodynamic analysis of the base cycle

In order to calculate the cycle power output, to size the cycle equipment, and to know the conditions of the steam to be extracted from the turbine to the thermal desalting units (suggested to be combined with this cycle), it is essential to know (or to construct) the steam cycle flow sheet for the PWR power plant using the AP-600 reactor.

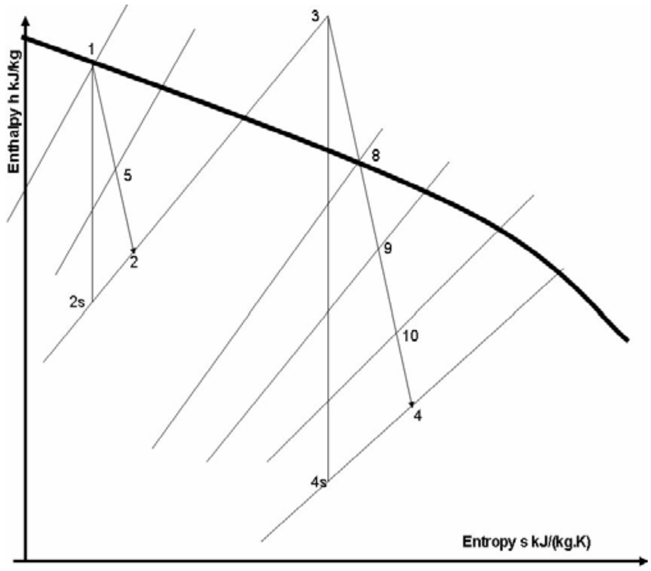

Fig. 3. Steam turbine expansion line on the enthalpy–entropy (h-s) diagram.

The suggested steam cycle has steam turbine expansion lines on the enthalpy–entropy chart similar to those shown in Fig. 3, and its components are given in Fig. 1.

The steam mass flow rate m_1 enters the HP turbine (point 1) as almost saturated vapor at 57.2 bar, 272.5°C, and entropy (s) of 5.912 kJ/kg.K. This steam expands to an intermediate pressure (IP) P_2. The pressure P_2 is chosen such that the steam quality χ at the exits of both HP and LP turbines is greater than 0.88, with typical isentropic efficiency η_{is} equal to, say, 0.84 and 0.8 in the HP and LP turbines, respectively. This gives steam conditions at the HP turbine exit as follows: P_2 =12 bar, T_2 = 187.99°C, χ_2 = 0.881, and h_2 (enthalpy) = 2548.7 kJ/kg.

A one bar drop in pressure of the heated steam is assumed in the MSR, and accordingly, the steam leaves the MSR at 11 bar. The moisture removed from the steam leaving the HP turbine is directed to the de-aerator, point 22 as a saturated liquid (see Fig. 1). The heating steam used by the reheater is supplied directly from the steam generator SG (point 6). This steam leaves the MSR as a saturated liquid at 57.2 bar (point 21). It is then throttled and directed to the open feedwater heater (de-aerator) operating at 11 bar as shown in Fig. 1. Flow rate calculations of the heating steam, removed moisture, and steam flowing to the LP turbine and to the feedwater heaters are given in Appendix A. The steam conditions and parameters at the cycle different points are given in Table 2.

The HP turbine work output (W_{HP}) for this cycle is given by:

$$W_{HP} = m_1(h_1-h_2)-m_5(h_5-h_2) = 232.513 \text{ MW}$$

The LP turbine work output (W_{LP}) is given by:

Table 2
Steam conditions at different points in the AP-600 NPP cycle

Point	Mass (m), kg/s	Pressure (P), bar	Temperature (T), °C	Enthalpy (h) kJ/kg	Remarks
SG	1063.00	57.20	272.50	2787.45	Total SG output
1	1041.85	57.20	272.50	2787.45	HP turbine inlet, saturated vapor
2	921.01	12.00	187.99	2548.70	χ_2 = 0.881
3	755.10	11.00	240.00	2915.50	LP turbine inlet, superheated
4	623.23	0.10	45.81	2297.50	χ_4 = 0.884
5	120.84	30.00	233.90	2683.00	To first feedwater heater
6	21.14	57.20	272.50	2787.45	To MSR from SG
7	63.97	12.00	187.99	2548.70	To de-aerator from HP turbine
8	39.81	3.50	139.00	2732.40	To second feedwater heater
9	41.53	1.50	110.00	2650.00	To third feedwater heater
10	50.53	0.50	81.33	2480.00	To fourth feedwater heater
11	755.10	11.00	45.81	191.83	Condenser exit, saturated liquid
12	755.10	11.00	75.00	313.90	
13	755.10	11.00	105.00	440.15	
14	755.10	11.00	134.00	584.20	
15	1063.00	62.00	184.09	781.38	
16	1063.00	62.00	227.00	971.70	
17	120.84	30.00	233.40	1004.80	
18	39.81	3.50	138.88	584.33	
19	81.34	1.50	111.37	467.11	
20	131.87	0.50	81.33	340.49	
21	21.14	57.20	272.50	1197.30	
22	101.94	11.00	184.09	781.38	

$W_{LP} = m_3(h_3-h_4)-m_8(h_8-h_4)-m_9(h_9-h_4)-m_5(h_{10}-h_4) =$
425.476 MW

This gives the total cycle work ($W_{cyc} = W_{HP}+W_{LP}$) of 657.99 MW. The net turbine output (W_{net}) is less than the total cycle output due to (1) power consumed by auxiliary (L_{aux}) losses, (2) power end losses (L_{end}) by steam kinetic energy at the condenser inlet, and (3) mechanical (bearing) and generator losses. This can be expressed by efficiencies denoted by η which are equal to:

- $\eta_g = 0.99$ for the generator,
- $\eta_m = 0.99$ for mechanical,
- $\eta_{end} = 0.975$ ($=1-L_{end}$) when end loss is 2.5%, and
- $\eta_{aux} = 0.965$ by assuming 3.5% of the power ouput is used to operate the pumps, fans, and other auxiliaries of the plant.

Accordingly, the net output work (W_{net}) is given by:

$$W_{net} = W_{cyc} \times (0.99\times0.99\times0.975\times0.965) = 657.99 \times 0.922 = 607 \text{ MW}$$

This gives a value of 31.44% for the cycle thermal efficiency.

6. Desalination and energy

As mentioned before, all seawater desalting systems consume either thermal, mechanical energy, or both. In case of large seawater quantities, the desalting systems are closely combined to power plants to secure the intensive thermal and/or mechanical energy needed for the desalting process. The commonly used desalting systems are classified into:

1. Mechanically driven desalters that consume mechanical (or electric) energy as the main energy source. These include seawater reverse osmosis (SWRO) and mechanical vapor compression MVC systems. Auxiliary heat is sometimes added to the MVC system to compensate for the heat loss from the system and avoids the inceases of mechanical energy input. The RO permeate output is a function of the feed temperature. This feed can be slightly heated to raise the SWRO production rate.

2. Thermally driven desalters that consume heat as the main energy source. These include the desalting systems of MSF, conventional MED, and TVC. Pumping (mechanical) energy is consumed also in these desalting systems to move their streams. Recirculation MSF is the most used method in the GCC with unit capacity up to 12.5–16 MIGD. One MIGD is 4550 m^3/d (52.62 kg/s).

For mechanically driven desalters, the desalting process can be realized in a separate desalting plant (SDP). The SDP can use diesel engines or steam or gas turbines to drive the desalters directly or to produce power for driving the motors, or use motors with power from the electric grid.

For thermally driven desalters, the thermal energy required in the SDP can be supplied from on-site fossil fuel fired boilers or specially designed heating nuclear reactors (as shown in Fig. 4) to generate the steam (or hot water) required to the thermally driven desalting systems such as MSF, MED, and TVC. Alternatively, the desalting plant can be combined with a power plant to form a CPDP. The CPDP can be operated with fossil or nuclear fuel and can have steam (or gas) turbine plants, a combined gas–steam turbine plant, or diesel engines.

As mentioned above, the thermally operated desalting units in the CPDP are combined with power plants (such as NPP) to secure their needed thermal energy. It should be noticed that the saturation temperautres of the heating steam required for the MSF and low temperature MED (LT-MED) is about 120°C and 70°C, respectively. These are relatively low temperatures when compared with those of steam used in power plants. Therefore, in CPDP, steam generators (or nuclear reactors) generate steam at high temperatures and pressures. This steam is supplied first to the steam turbines to produce work before being extracted (or discharged) to the MSF or MED desalting plants (Fig. 5). The turbine in this case is called extraction–condensing steam turbine (ECST). When all the steam is discharged from the turbine at a specific pressure suitable to the desalting units, the turbine is called a back-pressure steam turbine (BPST).

Another method to supply steam to LT-MED units is to increase the saturation pressure in the power plant condenser from the usual 7–10 kPa to 40–50 kPa to have a reasonably high water temperature as a heat source for the distillation plant. The cooling water leaving the condenser, say at 70°C, is directed to a flashing tank where vapor is flashed and is used as a heating vapor for the LT-MED system (see Fig. 6). The increase of the condenser pressure (and temperature) significantly decreases the turbine power output.

7. Choice of desalination technology

The AP-600 NPP can be combined with MED, MSF, or TVC desalting units. Typical equivalent energy (which counts for the used thermal and pumping energies) consumed by these systems in kWh/m^3 are 20, 10, and 22, for MSF, MED, and TVC, respectively. As such, the heat required for the MED process per mass of desalted water produced is, on average, lower than the heat required for the less efficient MSF process. This is the reason behind the choice of MED by most of the studies conducted to combine thermally operated desalting systems with NPP.

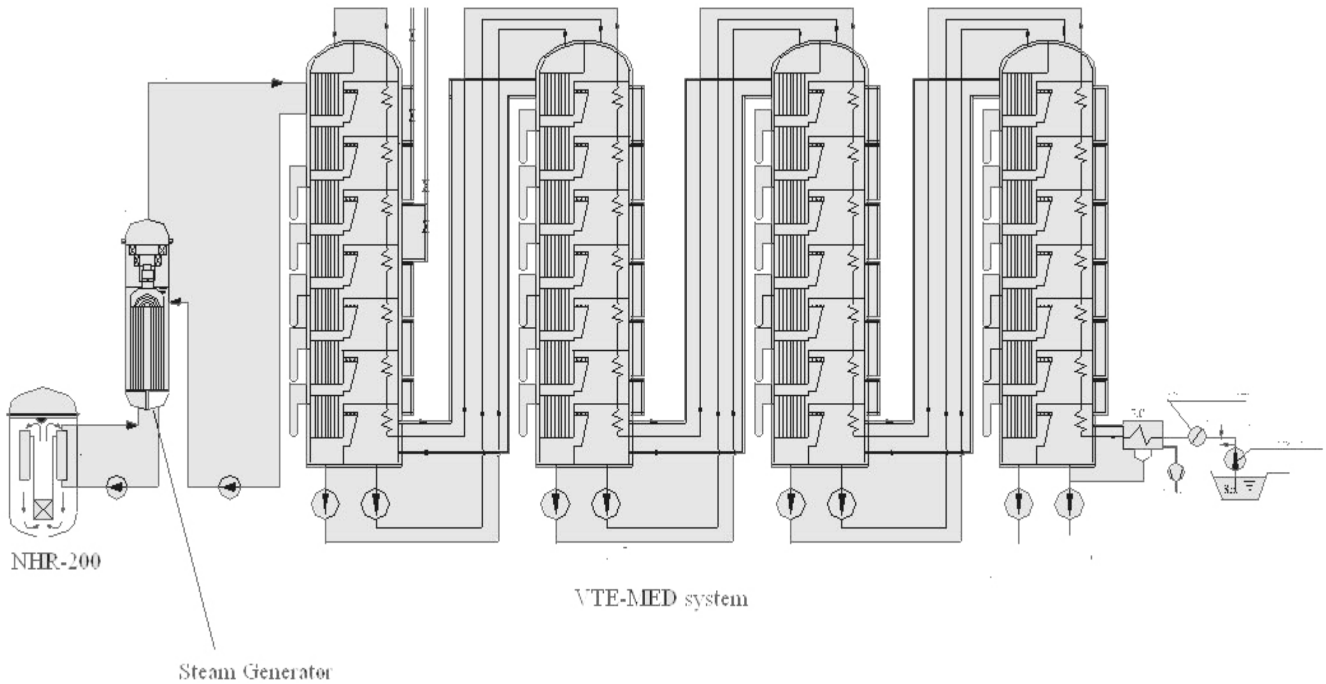

Fig. 4. Single-purpose desalting plant using a specially designed heating reactor.

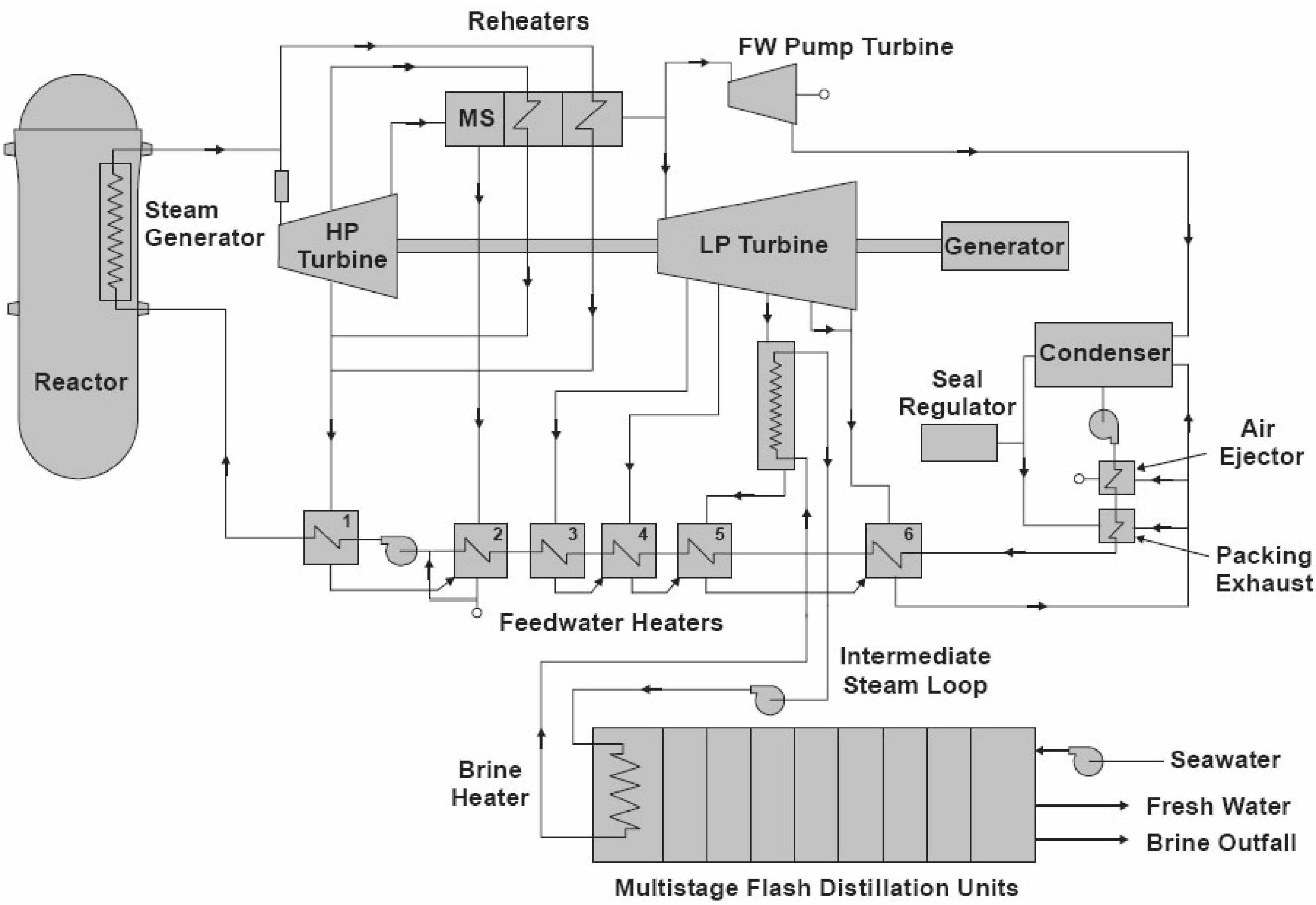

Fig. 5. CPDP using boiling water reactor and ECST with extracted steam to desalting units before the condenser.

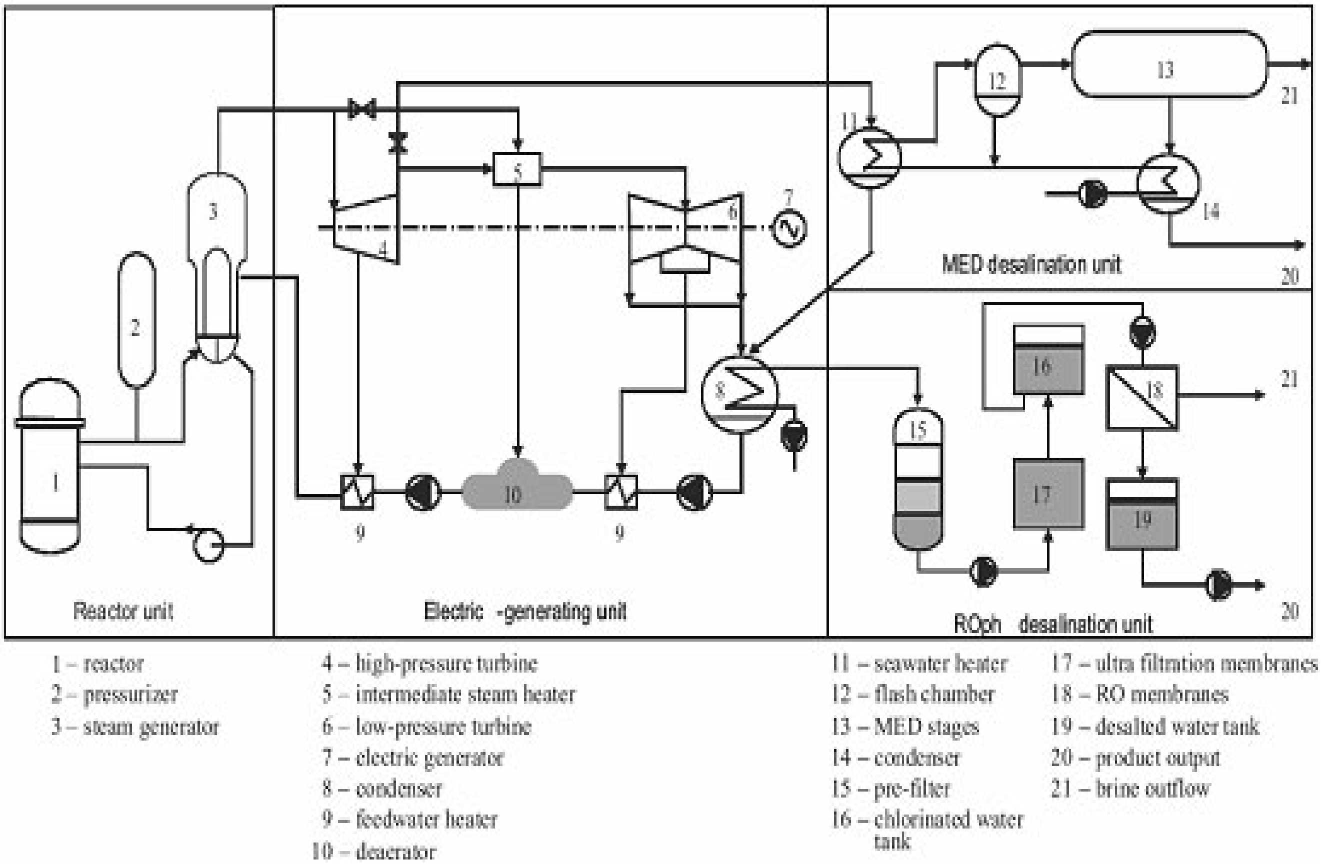

Fig. 6. CPDP using a boiling water reactor and elevated condenser conditions with a flash tank.

Also, the SWRO consumes typical equivalent mechanical energy of 4-6 kWh/m^3.

In this paper, different desalting seawater systems are combined with the NPP using the PWR to examine their economical aspects. These systems are assumed to consume the same amounts of energy but at different heating steam supply conditions. Four scenarios are considered to combine the desalting plant with the PWR nuclear power plant. In the first scenario, the steam turbine operates as a BPST with all of its steam discharged to the desalting units at the pressure required by these units. These units could be MED, MSF, or TVC units. In the three other scenarios, the steam turbine operates as an ECST where only part of its expanded steam is supplied to the desalting units. Apparently, the use of BPST gives more water output, but less power output, i.e. high water-to-power ratio, when compared with the ECST case. Also, it is assumed that, due to the valves existing on the steam entry to the desalting units, there is a difference in pressure between the exit of the turbine and the inlet to the desalting plant.

7.1. Scenario 1: BPST and LT-MED desalting units

The maximum desalted seawater can be obtained from the *AP*-600 NPP if the steam turbine operates as BPST (see Fig. 7). In this case, all the steam leaving the LP turbine is at the pressure required by the the LT-MED units, say at 0.5 bar pressure and saturation temperature (point 10 in Fig. 7) = 81.33°C (possible values in Kuwait).

The mass flow rate extracted from this point in the LP turbine ($m_3 - m_8 - m_9 = m_{10}$ = 601.71 kg/s) is discharged to the MED units. The MED units can operate with heating steam of 75°C saturation temperature (P = 38.58 kPa) due to the following: (1) the existing pressure drop between the turbine exit and the desalting units, (2) top brine temperature (TBT) of 72°C, (3) last effect temperature of 39°C, and (4) temperature difference across each effect ΔT_{effect} = 3°C. This gives the number of effects as 11, with a typical gain ratio (GR) of 8.5 (GR ≅ kg of desalted water per kg of steam) and a desalted water output (D) of 5114.52 kg/s (97.2 MIGD). The LP turbine power output for this scenario is:

$$W_{\text{LP}} = m_3(h_3 - h_{10}) - m_8 (h_8 - h_{10}) - m_9 (h_9 - \text{h}_{10}) = 298.597 \text{ MW}$$

The loss in power due to the discharging the steam at point 10 instead of the condenser is:

$$W_{\text{Loss}} = W_{\text{LP (base cycle)}} - W_{\text{LP}} = 425.476 - 298.58 = 126.879 \text{ MW}$$

This gives specific work loss from the turbine due to the

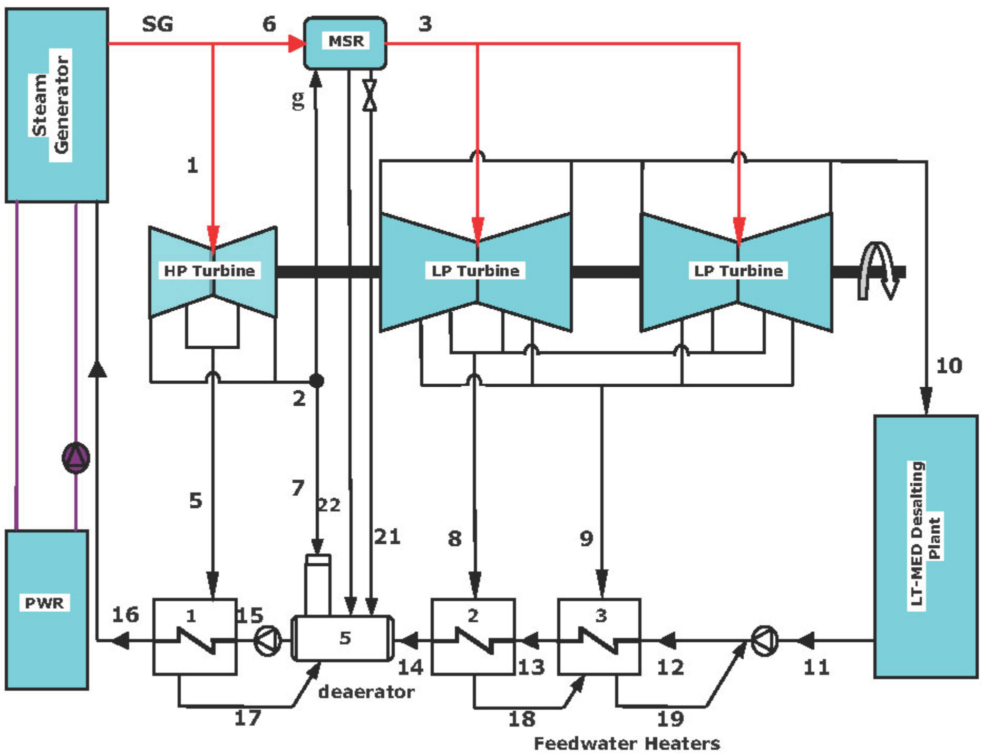

Fig. 7. AP-600 NPP with back pressure steam turbine and LT-MED desalting unit (Scenario 1).

steam extracted (as heat supply) to the deslater per unit desalted water as W_{ls} = 24.8 kJ/kg (= 6.9 kWh/m^3). When pumping energy of 2 kWh/m^3 is added to the work equivalent to heat, the specific total equivalent work (W_d) is 8.9 kWh/m^3. So, the cycle work (W_{cyc}) becomes:

$$W_{cyc} = 657.99 - 126.879 = 529.11 \text{ MW}$$

The net work (W_{net}) becomes:

$$Wnet = \eta * W_{cyc} = 0.922*529.11 = 487.8 \text{ MW}$$

The thermal energy supplied to the desalter (Q_d) if the heating steam condensate leaves the desalters at 75°C (and enthalpy h_{10} = 313.93 kJ/kg) is:

$$Q_d = 601.708 * (2480-313.93)/1000 = 1303 \text{ MW}$$

The specific heat consumption (Q_d/D) is:

$$Q_d/D = 1303*1000/5114.52 = 254.83 \text{ kJ/kg}$$

In the present scenario, the values of m_1, m_2, m_3, m_5, m_6, m_7, m_{21}, and m_{22} remain the same while m_8, m_9, m_{10} were recalculated as given in Appendix A and their respective values are: 50.637 kg/s, 102.755 kg/s, and 601.71 kg/s.

7.2. Scenario 2: ECST with LT-MED units producing 50 MIGD desalted water

When only 50 MIGD (2630.8 kg/s) desalted water (D) is required by the use of LT-MED units, part of the steam reaching point 10 on the turbine expansion line is extracted to the LT-MED units at a rate of m_d = 309.506 kg/s (based on GR = 8.5) and $m_{10'}$ is extracted to the fourth feedwater heater, while the balance $m_{4'} = (m_3 - m_8 - m_9 - m_d - m_{10'})$ kg/s continues its expansion to the condenser (Fig. 8).

The mass flow rate, m_8, m_9, $m_{10'}$ can be calculated as:

$$m_3 (h_{14} - h_{13}) = m_8 (h_8 - h_{18})$$

and hence, m_8 = 50.637 kg/s. Similarly,

$$m_9 = [m_3 (h_{13} - h_{12}) - m_8 (h_{18} - \mathrm{h}_{19})]/(h_9 - \mathrm{h}_{19}), \text{ and}$$

$$m_9 = 40.953 \text{ kg/s}$$

$$m_{10'} = [(m_3 - m_d)(h_{12} - h_{11}) - (m_8 + m_9)(h_{19} - h_{20})]/(h10 - h_{20}),$$
$$m_{10'} = 20 \text{ kg/s}$$

It is noticed here that the water leaving the fourth feedwater heater and the desalting units have the same temperature of T_{12} = 75°C and enthalpy h_{12} = 313.92 kJ/kg.

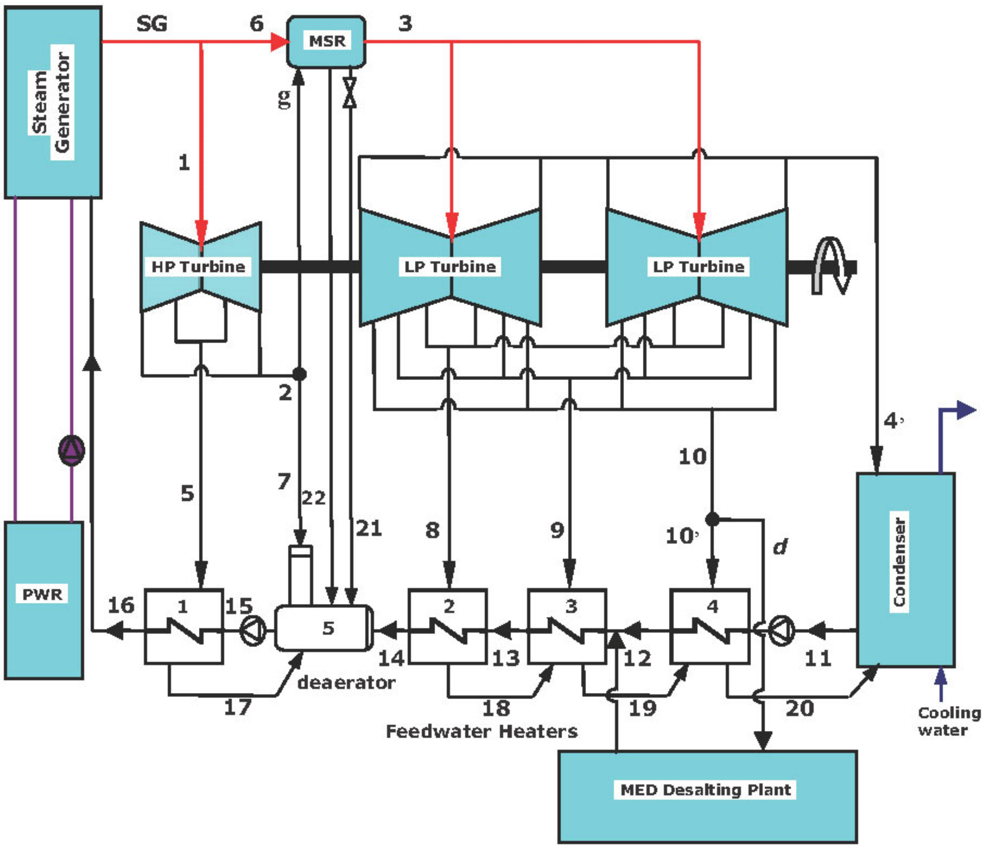

Fig. 8. AP-600 NPP with ECST and a LT-MED unit producing 50 MIGD desalted water (Scenario 2).

The LP turbine power output for this scenario is:

$$W_{LP} = m_3\,(h_3-h_4)-m_8\,(h_8-h_4)-m_9\,(h_9-h_4)-(m_d+m_{10'})\,(h_{10}-h_4) = 370.06 \text{ MW}$$

The loss in power due to extracting the steam to the desalting unit, instead of the condenser is:

$$W_{Loss} = W_{LP\,(base\ cycle)} - W_{LP} = 425.476-370.06 = 55.42 \text{ MW}$$

This gives specific work loss from the turbine due to heat supplied to the desalter per unit desalted water as W_{ls} =21.07 kJ/kg (= 5.852 kWh/m^3). When pumping energy of 2 kWh/m^3 is added to the work equivalent to heat, the specific total equivalent work (W_d) is 7.852 kWh/m^3. So, the cycle work (W_{cyc}) becomes:

$$W_{cyc} = 657.99-55.42 = 602.57 \text{ MW}$$

The net work (W_{net}) becomes:

$$Wnet = \eta * W_{cyc} = 0.922*602.57 = 555.57 \text{ MW}$$

The thermal energy supplied to the desalter (Q_d) if the heating steam condensate leaves the desalters at 75°C (and enthalpy h_{10} = 313.93 kJ/kg) is:

$$Q_d = 309.506\,(2480-313.93)/1000 = 670.42 \text{ MW}$$

The specific heat consumption (Q_d/D) is:

$$Q_d/D = 1303*1000/5114.52 = 254.83 \text{ kJ/kg}$$

In the present scenario, the values of m_1, m_2, m_3, m_5, m_6, m_7, m_{21}, and m_{22} remain the same while m_8, m_9, $m_{10'}$ were re-calulated as given above.

7.3. Scenario 3: ECST with MSF units producing 50 MIGD desalted water

This scenario is similar to the previous case, but the MSF desalting units are used here. The steam turbine in the AP600 cycle operates as a ECST as before, but the steam is extracted to MSF units at point 9 of 1.5 bar and 110°C saturation temperature (point 9 in Fig. 9) (possible values in Kuwait). Part of the steam at point 9 is extracted to the third feedwater heater $m_{9'}$, and another part m_d to the MSF plant. The MSF units can be operated with steam

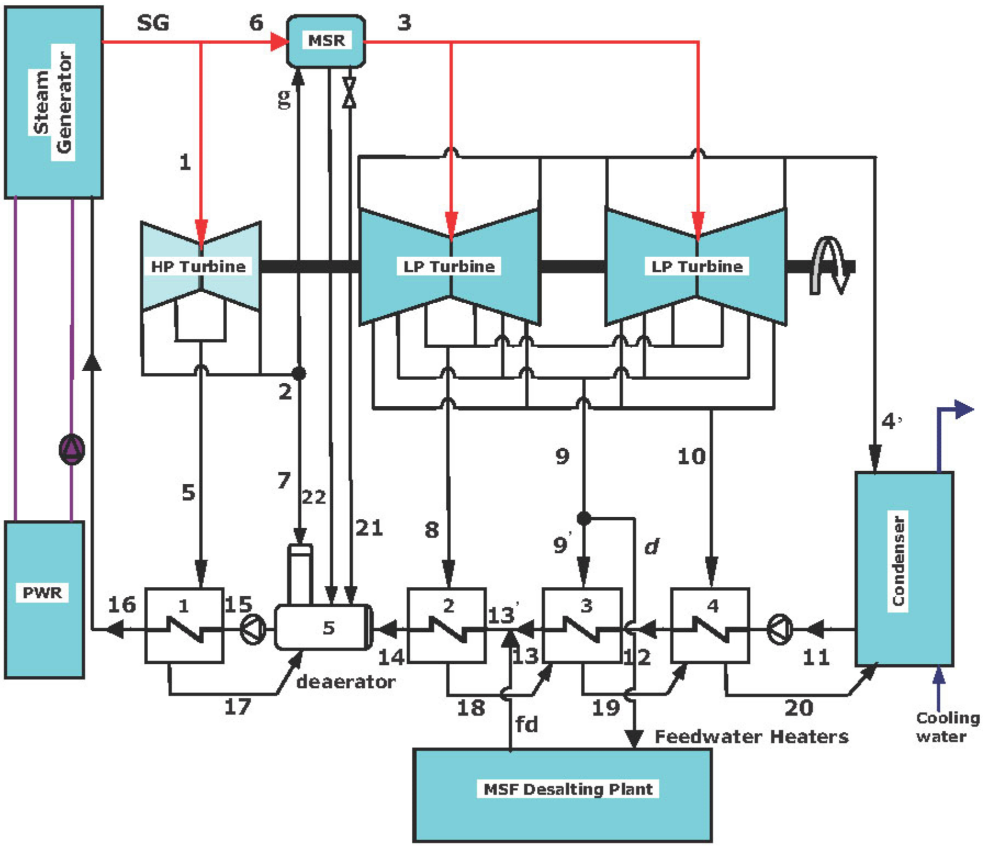

Fig. 9. AP-600 NPP with ECST and a MSF unit producing 50 MIGD desalted water (Scenario 3).

of 100°C saturation temperature ($P = 101.3$ kPa) due to the following: (1) pressure drop between the turbine exit and the desalting units, (2) TBT of 90°C, (3) last stage effect temperature of 40°C, and (4) 2°C temperature difference across each stage. This gives 25 stages, with a typical GR of 8. If the desalted water output (D) is 50 MIGD (2630.8 kg/s), the extracted steam to the MSF units (m_d) is equal to 328.85 kg/s (based on GR=8). The condensate from the MSF units (saturated liquid at 100°C and enthalpy $h_{fd} = 419.04$ kJ/kg) joins the feed water leaving the third feedwater heater at enthalpy h_{13} and the mixture enters the second feedwater heater at enthalpy $h_{13'} =$ 430.94 kJ/kg. So, the steam extracted to the second feedwater heater m_8 is calculated by:

$m_3\,(h_{14}-h_{13a}) = m_8\,(h_8-h_{18})$, and $m_8 = 53.875$ kg/s. Similarly,

$m_{9'} = [(m_3-m_d)\,(h_{13}-h_{12})-m_8\,(h_{18}-h_{19})]/(h_9-h_{19})$, and

$m_{9'} = 21.76$ kg/s

$m_{10} = [(m_3-m_d)\,(h_{12}-h_{11})-(m_8+m_{9'})\,(h_{18}-h_{19})/(h_{10}-h_{20})]$,
$m_{10} = 20.176$ kg/s

The LP turbine power output for this scenario is:

$W_{LP} = m_3\,(h_3-h_4)-m_8\,(h_8-h_4)-(m_{9'}+m_d)\,(h_9-h_4)-m_{10}\,(h_{10}-h_4)$
$= 315.95$ MW

The loss in power due to extracting the steam to the desalting unit, instead of the condenser is:

$W_{Loss} = W_{LP\,(base\,cycle)} - W_{LP} = 425.476-315.95 = 109.528$ MW

This gives specific work loss from the turbine due to heat supplied to the desalter per unit desalted water as $W_{ls} =$ 41.633 kJ/kg (= 11.565 kWh/m^3). When pumping energy of 4 kWh/m^3 is added to the work equivalent to heat, the specific total equivalent work is 15.565 kWh/m^3. So, the cycle work (W_{cyc}) becomes:

$W_{cyc} = 657.99-109.528 = 548.46$ MW

The net work (W_{net}) becomes:

$W_{net} = \eta * W_{cyc} = 0.922*548.46 = 505.68$ MW

The thermal energy supplied to the desalter (Q_d) if the heating steam condensate leaves the desalters at 100°C (and enthalpy $h_{fd} = 419.04$ kJ/kg) is:

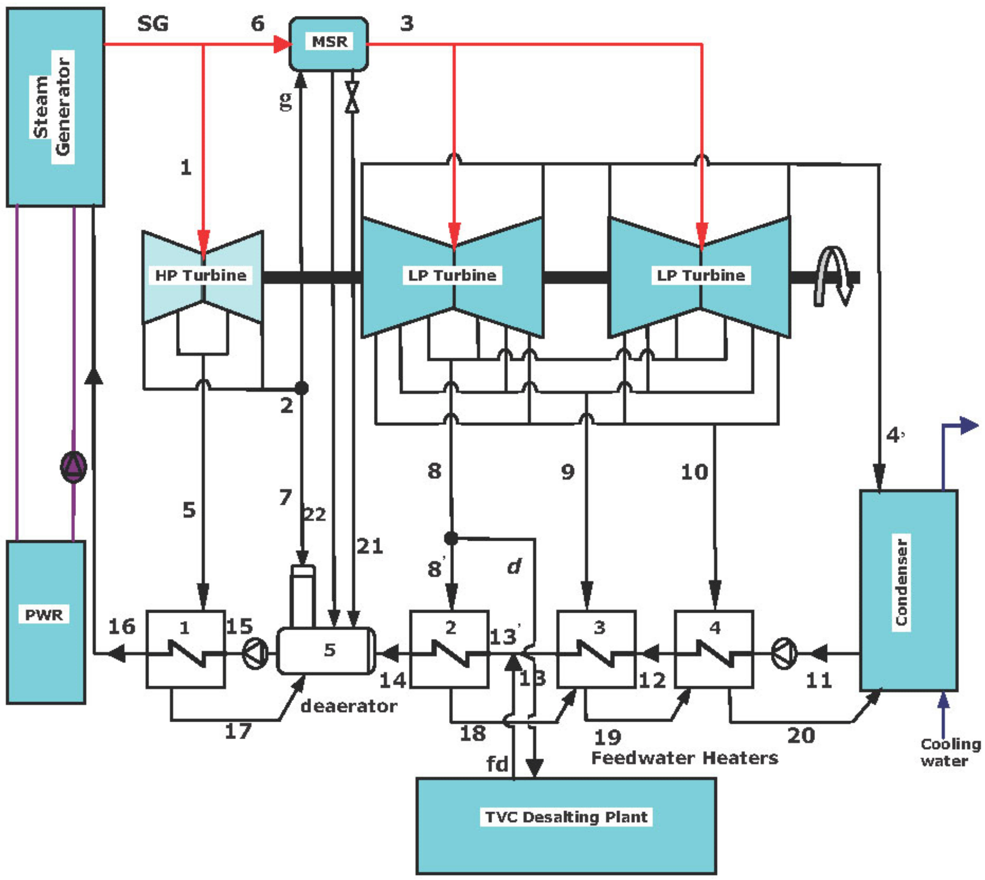

Fig. 10. AP-600 NPP with ECST and a TVC unit producing 50 MIGD desalted water (Scenario 4).

$Q_d = m_d\,(h_9 - h_{fd}) = 328.85 * (2650.0 - 419.04)/1000$
$= 733.65$ MW

The specific heat consumption (Q_d/D) is:

$Q_d/D = 733.65{*}1000/2630.8 = 278.87$ kJ/kg

In the present scenario, the values of m_1, m_2, m_3, m_5, m_6, m_7, m_{21}, and m_{22} remain the same while m_8, $m_{9'}$, and m_{10} were re-calculated as given above.

7.4. Scenario 4: ECST with TVC units producing 50 MIGD desalted water

This scenario is similar to the previous case, but the TVC desalting units are used here. The steam turbine in the AP-600 cycle operates as a ECST as before, but the steam is extracted to TVC unit at point 8 of 3.5 bar and 138.88°C saturation temperature (point 8 in Fig. 10) (possible values in Kuwait).

The steam supplied to the TVC can be at much higher temperature than the TBT since this steam is not directly heating the seawater (or the brine). The TVC can be operated with motive steam of $P_{mot} = 3$ bar due to: (1) pressure drop between the turbine exit and the desalting units, (2) 70°C saturation temperature of the heating vapor discharged from the thermal compressor $P_d = 31.2$ kPa to the first effect, (3) TBT of 64°C, and (4) last effect temperature of 40°C (at $P_{suc} = 7.384$ kPa). This gives an expansion ratio $P_{mot}/P_{suc} = 300/7.384 = 40.63$, and compression ratio $P_d/P_{suc} = 31.2/7.384 = 4.22$, and the motive steam to the sucked vapor = 2.5. The use of six effects gives a 4°C temperature difference across each effect. The maximum reported GR of this TVC for the motive pressure range of 3 bar is 8. Therefore, the desalted water output (D) of 50 MIGD (2630.8 kg/s) requires a motive steam supply m_d of 328.85 kg/s.

In scenario 4, the steam is extracted at point 8 to the second feedwater heater, and the TVC desalting units. The condensate from the TVC units (saturated liquid at 70°C and enthalpy h_{fd} = 292.98 kJ/kg) joins the feed water leaving the third feed water heater at 105°C and h_{13} of 440.15 kJ/kg. The mixture enters then the second feedwater heater at enthalpy $h_{13'}$ of 376.06 kJ/kg. So, the steam extracted to the second feedwater heater $m_{8'}$ is calculated by:

$m_3\ (h_{14}-h_{13a}) = m_{8'}(h_8-h_{18})$, and $m_{8'} = 37.17$ kg/s

Similarly,

$m_9 = [(m_3-m_d)\ (h_{13}-h_{12})-m_{8'}\ (h_{18}-h_{19})]/(h_9-h_{19})$, and

$m_9 = 21.76$ kg/s

$m_{10} = [(m_3-m_d)\ (h_{12}-h_{11})-(m_{8'}+m_9)\ (h_{19}-h_{20})/(h_{10}-h_{20})]$,
$m_{10} = 18.763$ kg/s

The LP turbine power output for this scenario is:

$$W_{LP} = m_3(h_3-h_4)-(m_{8'}+m_d)(h_8-h_4)-m_9(h_9-h_4)-m_{10}\ (h_{10}-h_4) = 281.085 \text{ MW}$$

The loss in power due to extracting the steam to the desalting unit, instead of the condenser is:

$$W_{Loss} = W_{LP\ (base\ cycle)}-W_{LP} = 425.476-281.085 = 144.392 \text{ MW}$$

This gives specific work loss from the turbine due to heat supplied to the desalter per unit desalted water as W_{ls} = 54.88 kJ/kg (= 15.245 kWh/m^3). When pumping energy of 2 kWh/m^3 is added to the work equivalent to heat, the specific total equivalent work is 17.245 kWh/m^3. So, the cycle work (W_{cyc}) becomes:

$$W_{cyc} = 657.99-144.392 = 513.59 \text{ MW}$$

The net work (W_{net}) becomes:

$$W_{net} = \eta * W_{cyc} = 0.922*513.59 = 473.54 \text{ MW}$$

The thermal energy supplied to the desalter (Q_d) if the heating steam condensate leaves the desalters at 70°C (and enthalpy h_{fd} = 292.04 kJ/kg) is:

$$Q_d = m_d(h_8-h_{fd}) = 328.85* (2650.0-292.04)/1000 = 802.2 \text{ MW}$$

The specific heat consumption (Q_d/D) is:

$$Q_d/D = 802.2*1000/2630.8 = 304.93 \text{ kJ/kg}$$

In the present scenario, the values of m_1, m_2, m_3, m_5,m_6, m_7, m_{21}, and m_{22} remain the same while $m_{8'}$, m_9 and m_{10} were re-calculated as given above.

8. Comments on the proposed four scenarios

A summery for the results of the previous four scenarios is given in Table 3. Generally, as can be observed, the largest possible value for the D/W_{net} [912.25 (m^3/d)/MW] is obtained from the back pressure turbine. Also, both MED cases give the lowest specific equivalent energy (which counts for the used thermal and pumping energies). Finally, when comparing scenarios 2 and 3, the following can be also noticed:

- The specific equivalent work (SEW) (due to heat and pumping energy) in the case of MSF units is 15.565 kWh/m^3, which is 98.2% more than that of MED units of 7.852 kWh/m^3.
- The specific heat consumed by both MSF and MED are almost the same (278.87 kJ/kg for MSF and 254.84 kJ/kg for MED). This shows the advantage of supplying steam to desalters at lower specific availability. If

Table 3
Comparison of the proposed coupling scenarios

	Coupling scenarios				
	NPP, no desalting	Scenario 1 BPST+MED	Scenario 2 ECST+MED	Scenario 3 ECST+MSF	Scenario 4 ECST+TVC
GR	0	8.5	8.5	8	8
W_{HP}, MW	232.513	232.513	232.513	232.513	232.513
W_{LP}, MW	425.476	298.597	370.06	315.95	281.085
W_{cyc}, MW	657.99	529.11	602.57	548.46	513.59
W_{net}, MW	607	487.8	555.57	505.68	473.54
D output, MIGD	0	97.2	50	50	50
Q_d, MW	0	1303.0	670.42	733.65	802.2
$W_{net}-W_{pumping}$, MW	607	450.97	536.63	486.74	454.60
Q_d/D, kJ/kg	0	254.83	254.84	278.87	304.92
$W_{d''}$ (heat equivalent), MW	N/A	126.879	55.54	109.53	144.39
W_d (heat and pumps), MW	N/A	163.869	74.48	147.41	163.33
SEW (= W_d/D), kWh/m^3	N/A	8.9	7.852	15.565	17.246
D/W_{net}, (m^3/d)/MW	0	912.25	409.5	449.88	480.42

the steam is supplied to the MSF units at 3 bar (as in the case of CPDP in Kuwait), the SEW can easily reach 20 kWh/m^3.

9. Levelized desalted water costs

In calculating the water cost for the previous desalination options, it is required to find the capital and operation and maintenance (O&M) costs [5].

9.1. Capital costs

As given in the literature [6–11], the capital cost is in the range of \$750–1320 per (m^3/d) for MED and in the range of \$1000–1470 per (m^3/d) for MSF. The TVC is expected to be in the same range of the MED. Therefore, when calculating the water cost, the initial capital cost in this study will be taken as \$1200 per (m^3/d) for MED units, \$1300 per (m^3/d) for the MSF units, and \$1200 per (m^3/d) for the TVC units. According to a Korean study [10], there are additional capital costs such as the intermediate loop cost and the cost of water intake/outfall structures. In the present study, the intermediate loop cost is taken as \$85 per (m^3/d) for all desalters, or \$19.302 million for a required water capacity of 227.3×10^3 m^3/d. The water intake/outfall structures cost is taken as \$71 per (m^3/d) for the MED and TVC units, or \$16.14 million for the required capacity. For the MSF units, it is expected that cost of the water intake/outfall structures is double that of the MED units since the seawater intake for the MSF is at least double that of the MED. Hence, the water intake/outfall cost for the MSF is \$32.277 million for the required capacity.

Consequently, for the 50 MIGD desalting plant operating with the MED or TVC system (227.3×10^3 m^3/d), the total initial capital cost [based on \$1200 per (m^3/d)] is \$272.76 million plus \$35.44 million for the intake/outfall and intermediate loop costs. In other words, the total initial capital cost for MED or TVC plants is \$308.22 million. Similarly, for the 50 MIGD operating by MSF plant (227.3×10^3 m^3/d), the total initial capiatal cost [based on \$1300 per (m^3/d)] is \$295.49 million plus \$51.6 million for the intake/outfall and intermediate loop costs. Therefore, the total initial capital cost for MSF plant is \$347.09 million.

Real interest rates used in many industrialized and developing countries range from 5% to 10% according to IAEA studies. In the present economic assessment, 8% interest/discount rate is considered. As such, for a \$100 million loan to be paid back over an economic life of 30 years, the interest cost along the economic life is equal to \$120 million [(=100/2) × interest rate (0.08) × number of years (30)]. Hence, the total payment (principal and interest) over the economic life is \$220 million which is divided into a fixed payment of \$7.33 million each year (= \$220/30 million). The ratio of the fixed payment to principal, which is called the fixed charge rate (R), is equal to 7.33% (= 7.33/100) in the present case. The annual fixed payment (A) is equal to R×total initial capital cost. The annual fixed payment, after the first year, has a lower real value (compared to today or present money value) due to the inflation. The real value of this payment in terms of today value of money (dollars) is called the present levelized value P, and can be calculated by the following relation [9]:

$$P = \frac{A}{30}\frac{1-(1+i)^{-n}}{i}$$

where i is the inflation rate, n is the number of years, and A is the annual fixed payment.
For n = 30 and i = 3%, the term

$$\frac{1-(1+i)^{-n}}{i}$$

is equal to 19.6. The annual levelized present value of the capital for the three desalination plants are calculated and given in Table 4.

9.2. Operating and maintenance costs

The variable O&M costs include two parts. The first part is the cost of labor, management, and maintenance. The second part is the energy cost. According to a Korean study [10], the first part of O&M can be taken as:

- \$29,700 per year, average labor salary for a water capacity of 40,000×10^3 m^3/d. Hence, for the required water capacity of 227.3×10^3 m^3/d, the average labor salary becomes \$168,770 per year.
- \$66,000 per year, average management salary for 40,000×10^3 m^3/d or \$375,045 per year for water capacity of 227.3×10^3 m^3/d.
- \$1.96 million per year, annual water plant maintenance cost for 40,000 m^3/d. Hence, for the required water capacity of 227.3×10^3 m^3/d, the annual maintenance cost becomes \$11.14 million per year for each desalter.

Adding up the above three items, each desalting plant will have an annual water plant O&M cost of \$11.68 million/y for the required capacity of 227.3×10^3 m^3/d.

The second part of the O&M costs (the energy cost) can be easily obtained from the equivalent consumed mechanical work for the three desalting plants. As shown in Table 3, the equivalent consumed mechanical work is an average of 8.4 kWh/m^3 for the MED units, 15.6 kWh/m^3 for the MSF units, and 17.25 kWh/m^3 for the TVC units. When using a value of 0.9 as the capacity factor for the desalting water, the annual desalted water for a 50 MIGD plant becomes 74.67 million m^3/y. Hence, the annual energy consumed by each desalter is: 627.215 GWh for MED units, 1164.83 GWh for MSF units, and 1288.03 GWh for TVC units. The value of the specific energy cost for the *AP*-600 NCPP is calculated as 61.27 ($/MW.h) [1]. If this value is used in all scenarios, the annual energy cost becomes $38.43 million (=627,214.7×61.27) for the MED units, $71.37 million (=1,164.83×61.27) for the MSF units, and $78.92 million (=1,288.05×61.27) for the TVC units.

For comparison purposes, all the above costs (capital and O&M) of the selected options are summarized in Table 4. The levelized costs of water given in Table 4 are arrived at by dividing the sum of the levelized capital, operation and maintenance, and energy by the discounted value of the quantity of water produced. The resulting value is expressed in $/$m^3$ of produced water. As can be observed, the computed levelized costs of desalted water (based on plant operating availability of 90%) are 0.87 $/$m^3$ when using a MED plant, 1.334 $/$m^3$ when using a MSF plant, and 1.4 $/$m^3$ when using TVC units. Thus, nuclear MSF and TVC desalination plants have similar water production costs. As compared to a 600 MW combined co-generation gas turbine + MED plant (CC-600), the desalination costs of the proposed AP-600 + MED nuclear plant (0.87 $/$m^3$) is 47% lower than the corresponding cost of water produced by the CC-600 plant + MED (1.641 $/$m^3$) as previously calculated in the literature [7].

Table 4
Comparison of water cost[a] for the AP-600 NPP unit coupled with different desalination plants

Parameter	Coupling scenario		
	ECST+ MED	ECST+ MSF	ESCT+ TVC
SEW, kWh/m^3	8.4	15.6	17.25
Initial capital cost, M$	308.22	347.09	308.22
Annual costs, M$/y			
Fixed payment	22.603	25.453	22.603
Levelized present value	14.76	16.63	14.76
Water plant O&M cost	11.68	11.68	11.68
Energy cost	38.43	71.37	78.92
Total annual costs	64.87	99.68	105.36
Levelized cost of desalted water ($/$m^3$)	0.87	1.334	1.410

[a]Based on the following assumptions:
- Specific energy cost of 61.27 $/MW-h [1]
- Interest/discount rate of 8%.
- Plant operating availability of 90%.
- Plant capacity of 227×10^3 m^3/d.
- An economic life of 30 years.

In general, for the three proposed plants, the major contributing factor to the cost is the energy portion (60% for MED, 71% for MSF, and 75% for TVC). As can be seen, MSF and TVC consume approximately 100% more energy than MED. Thus, savings in energy costs can dramatically affect the water costs. This can be achieved by using a hybrid thermal/RO system similar to the one used by Faibish [10]. As can also be observed, the second largest component to the water cost is the levelized capital cost, which is greatly affected by the interest rate. As a final note, the O&M cost has the smallest impact on the water production cost.

Finally, the plant operating availability factor plays an important role in the economics of nuclear desalination [5]. In the present study, with a 6% increase in the availability factor (to become 96% instead of 90%), the water cost decreases 13% for the case of MED, 25% for the MSF, and 32% for the TVC.

10. Conclusions

This paper examines several choices for a desalination plant to be coupled to a nuclear power plant for the cogeneration of power and water. The techno-economic results associated with the various desalination plant options can help in choosing the best option that is suitable for the Kuwaiti conditions.

Several conclusions are reached by examining the presented results:
- Nuclear desalination costs from thermal systems are in the range of 0.87 $/$m^3$ to 1.4 $/$m^3$. MSF and TVC systems coupled to nuclear power plants give similar water production costs of 1.4 $/$m^3$, whereas MED coupled to nuclear power plants gives the lowest cost of 0.87 $/$m^3$.
- The cost of water produced from MED systems coupled to the AP-600 nuclear cogeneration power plant is 47% lower than the corresponding cost of the CC-600 plant.
- The energy cost represents 60% of the total water cost for MED and 71% and 75% for MSF and TVC, respectively. Thus, savings in energy costs are the main contributor to the lower overall product water costs of the nuclear desalination.

Finally, it appears that, in Kuwait, based on current prices charged for water, the use of nuclear energy for the production of water and energy is, from an economic point of view, a more competitive option than other energy sources using fossil fuel. Thus, nuclear desalination appears to be a possible option deserving serious analysis and investigation to solve Kuwait's ongoing water shortage problem.

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Appendix A — Heat Balance of the Steam Cycle

The data of PWR shows that steam generator SG produces M_{SG} = 1063 kg/s. If part of this steam, m_1, is supplied to the HP turbine, and the balance $m_6 = M_{SG} - m_1$ is directed to the reheater, part of the steam expanded in the HP turbine, m_5, is extracted to the HP feed heater between the de-aerator and the SG to heat the feedwater (of mass flow rate M_{SG}) from point 15 (at the deaerator outlet) to point 16 (at the the SG inlet). An energy balance for this steam generator gives

$$m_5^*(h_5 - h_{17}) = 1063^*(h_{16} - h_{15})$$

The use of enthalpy values h (in Table 2) gives m_5 of 120.55 kg/s. The mass leaving the HP turbine ($m_1 - m_5$) is divided to m_7 supplied to the open feedwater heater (de-aerator), and the balance ($m_1 - m_5 - m_7$) has T_2 = 187.99°C and χ_2 = 0.881, enthalpy h_2 = 2548.7 kJ/kg enters the moisture separator reheater and leaves as saturated vapor at a mass flow rate of 0.881 ($m_1 - m_5 - m_7$) = m_3, while the balance 0.135 m_3 leaves as liquid to the de-aerator (point 22). Notice that $m_1 = m_5 + m_7 + m_3/0.881$, where $m_3/0.881$ = 1.135 m_3 is the mass leaving the HP turbine minus m_7. The steam is heated from the saturation condition of 12 bar and 187.99°C (and h = 2784.8 kJ/kg) to become superheated steam at 240°C and 11 bar (due to pressure loss in the reheater). The heating steam of flow rate m_6 at pressure 57.2 bar loses its latent heat as it leaves the reheater as saturated liquid. An energy balance in the reheater gives

$$m_6(h_1 - h_{21}) = m_3\,(h_3 - h_g)$$

where h_1, h_3, h_{21}, and h_g are the enthalpies at SG the outlet (saturated vapor at 57.2 bar), saturated liquid at 57.2 bar, superheated vapor at 11 bar and 240°C, and saturated vapor at 12 bar, respectively. Substituting the values for h_1, h_3, h_{21}, and h_g in the previous equation gives m_6/m_3 = 0.028. An energy balance around the deaerator gives:

$$m_3(h_{15} - h_{14}) = m_7\,(h_2 - h_{15}) + m_5(h_{17} - h_{15}) + m_6\,(h_{21} - h_{15})$$

Substituting the values of h's, the value of m_5 and m_6/m_3 = 0.028 gives the following equation:

$$m_7 = 0.10495\,m_3 - 15.278$$

By using [$m_3 = 0.881\,(m_1 - m_5 - m_7)$] and ($m_1 + m_6 = m_5 + m_7 + 1.135\,m_3 + m_6 = M_{SG}$ = 1063 kg/s), the values of m_1, m_3, and m_7 were obtained, and are given in Table 2.

The energy balance in the other three feedwater heaters gives m_8, m_9 and m_{10} as follows:

$$m_8\,(h_8 - h_{18}) = m_3\,(h_{14} - h_{13}) \text{ gives } m_8$$

$$m_3\,(h_{13} - h_{12}) = m_9(h_9 - h_{19}) + m_8\,(h_{18} - h_{19}) \text{ gives } m_9$$

$$m_3\,(h_{12} - h_{13}) = m_{10}\,(h_{10} - h_{20}) + (m_8 + m_9)\,(h_{19} - h_{20}) \text{ gives } m_{10}$$