Development of natural gas fired combined cycle plant for tri-generation of power, cooling and clean water using waste heat recovery: Techno-economic analysis

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Abstract: Tri-generation is one of the most efficient ways for maximizing the utilization of available energy. Utilization of waste heat (flue gases) liberated by the Al-Hamra gas turbine power plant, UAE is analyzed in this research work for simultaneous production of a) electricity by combining steam rankine cycle using heat recovery steam generator (HRSG), b) clean water by air gap membrane distillation (AGMD) plant and c) cooling by single stage vapor absorption chiller (VAC). The flue gases liberated from the gas turbine power cycle is the prime source of energy for the tri-generation system. The heat recovered from condenser of steam cycle and excess heat available at the flue gases are utilized to drive cooling and desalination cycles which are optimized based on the cooling energy demands of the villas. Economic and environmental benefits of the tri-generation system in terms of cost savings and reduction in carbon emissions were analyzed. Energy efficiency of about 82-85% is achieved by the tri-generation system compared to 50-52% for combined cycles. Normalized carbon dioxide emission per MWh is reduced by 51.5% by implementation of waste heat recovery tri-generation system. The tri-generation system has a payback period of 1.38 years with cumulative net present value of $66 million over the project life time.

Keywords: Tri-generation; waste heat; steam cycle; air gap membrane distillation; absorption chillers; flue gases; techno-economic;
1. Introduction

Tri-generation is one of most promising technology integration practice for production of three different outputs with a common primary energy source. Combined heat and power cycle (CHP) would be transformed to poly-generation system by integrating heat driven cooling and desalination cycles in the system. Poly-generation cycles utilize the energy efficiently and reduce the green house gas emissions effectively [1-5]. Combined cooling, heat and power (CCHP) is an extended version of CHP designed for centralized production of cooling, heating and power generation as large scale system with higher efficiency [6]. In general, power plants in Middle East and North Africa (MENA) region are combined with desalination plants for recovering the heat and to produce clean water for meeting domestic needs. But since decentralized air conditioning systems in the region accounts for maximum power consumption, tri-generation models with combined production of cooling, clean water and power (CCCWP) would be more beneficial in terms of energy savings and economics.

Previously, several researchers investigated CCHP configurations with gas turbine, steam turbine and organic rankine cycles by combining with absorption, adsorption or desiccant chillers. Ahmadi et al. conducted a detailed thermodynamic modeling of CCHP for providing cooling, hot water and electricity. The system is modeled with combined cycles of gas turbine and organic rankine cycle for power generation, single stage absorption chiller for cooling and hot water through heat recovery. The system is analyzed in terms of energy efficiency and environmental impact, exergy efficiency of tri-generation system is improved by 20% compared to conventional CHP [7]. Khalique simulated a gas turbine based tri-generation system for combined production of power, cooling and heat (process steam). Performance of the system is analyzed for different pressure ratios, turbine inlet temperatures, process heat pressures and evaporator temperatures of absorption chiller [8]. Ahmadi et al. conducted a detailed thermodynamic modeling and simulation of tri-generation system producing power through combined gas turbine and steam turbine cycles. The steam driven absorption chiller utilized to provide cooling, low pressure steam from the HWSG is utilized as heat source for absorption chiller and space heating [9].

Several concepts of tri-generation system involving reciprocating IC engines for CCHP purposes were evolved as its financially beneficial. Temir and Bilge studied the performance of tri-generation system for production of electricity by reciprocating engine, absorption cooling using saturated steam from the boiler and process heat recovery from exhaust outlet [10]. Huangfu et al. analyzed the performance of micro-scale CCHP for domestic and light commercial applications using reciprocating internal combustion engines, absorption chiller and heat recovery devices. Energy and economic analysis reported a short payback period of 2.97 years [5]. Sun proposed a combined production cycle of electricity by gas engine and cooling by absorption chiller, which provides primary energy savings of 37% compared to separate conventional power and cooling systems with payback return in 4.52 years [11].

Tri-generation (or) poly-generation systems integrating the desalination technologies for clean water production is investigated by few researchers. Hussain developed a tri-generation system for simultaneous production of cooling, clean water and electricity and analyzed with different technologies to provide cooling and clean water. The system is optimized based on its fuel saving potential. Out of different combinations analyzed, combination of reverse osmosis, absorption chiller...
with gas turbine power cycle provides highest economic benefits [12]. Calise et al. modeled a solar energy based tri-generation for CCCWP applications. Multi-effect desalination unit and vapor absorption chiller are integrated together with PVT collectors for combined production of cooling, desalination and power. The transient simulations for different operational and design parameters were conducted and optimized in terms of energetic efficiency and economic viability [13].

In this research work, combined cooling, clean water and power cycle (CCCWP) incorporated with membrane distillation technology for clean water production as a part of tri-generation system is investigated. Membrane distillation is a promising thermal driven desalination technology, utilizing low grade heat energy for production of clean water [14]. Previously research was conducted on integration of membrane distillation system for cogeneration applications. Liu analyzed the performance of two different cogeneration systems by integrating membrane distillation modules with gas engine as combined power and desalination cycles and with vapor compression chiller as combined cooling and distillation unit [15]. Kullab analyzed the performance of air gap membrane distillation (AGMD) modules with different integration layouts [16]. Burrieza et al. conducted several parametric studies on air gap membrane distillation module to optimize the performance of the system that produces a maximum distillate flux of 20 l/h per module [17].

2. System description

A tri-generation system is proposed for effective utilization of the waste flue gases liberated by the Al-Hamra gas turbine power plant based in Ras-Al-Khaimah, UAE as shown in figure 1(a) and 1(b). Reheat brayton cycle configuration is utilized in Al-Hamra gas turbine power plant. Air at ambient temperature is compressed and passed to the combustion chamber (CC) for production of super heated gas. This super heated gas is further expanded in a series of gas turbines which are connected through reheat chamber (RH) for power production.

Figure 1. (a) Al-Hamra gas power plant. (b) Gas turbine in Al-Hamra power plant

The proposed Tri-generation system comprises of steam turbine rankine cycle, single stage LiBr/H₂O absorption chiller plant and air gap membrane distillation units. The schematic layout of tri-generation system is shown in figure 2. The hot flue gases liberated after the expansion in the gas
turbine is utilized in the HRSG for production of process steam at higher pressure and temperature, which is further expanded in a steam turbine for production of electricity. The steam is then condensed in a heat recovery system using sea-water. The heat liberated in the heat recovery system is used to drive both the absorption chiller and membrane distillation systems. The condensed steam (feed water) is circulated back to the HRSG using the feed water pump. The stack flue gases leaving the HRSG is utilized to produce additional hot sea water, which is integrated with the supply line of cooling and desalination systems.

Figure 2. Schematic layout of tri-generation system

<table>
<thead>
<tr>
<th>No</th>
<th>Specification</th>
<th>No</th>
<th>Specification</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>Inlet air entering compressor</td>
<td>21</td>
<td>Condensed water exiting condenser</td>
</tr>
<tr>
<td>2</td>
<td>Outlet air from compressor</td>
<td>22</td>
<td>Water vapor entering evaporator (VAC)</td>
</tr>
<tr>
<td>3</td>
<td>Combustion gases leaving combustion chamber</td>
<td>23</td>
<td>Saturated water entering absorber</td>
</tr>
<tr>
<td>4</td>
<td>Hot gases leaving first gas turbine</td>
<td>24</td>
<td>LiBr solution leaving heat exchanger</td>
</tr>
<tr>
<td>5</td>
<td>Combustion gases leaving reheat chamber</td>
<td>25</td>
<td>LiBr solution entering absorber</td>
</tr>
<tr>
<td>6</td>
<td>Hot flue gases leaving second gas turbine</td>
<td>26</td>
<td>Low pressure LiBr–water mixture</td>
</tr>
<tr>
<td>7</td>
<td>Feed water entering HRSG</td>
<td>27</td>
<td>High pressure LiBr–water mixture</td>
</tr>
<tr>
<td>8</td>
<td>Hot water leaving economizer</td>
<td>28</td>
<td>Weak solution entering generator</td>
</tr>
<tr>
<td>9</td>
<td>Saturated steam leaving evaporator</td>
<td>29</td>
<td>Strong solution leaving generator</td>
</tr>
<tr>
<td>10</td>
<td>Superheated steam entering steam turbine</td>
<td>30</td>
<td>Cold sea water entering absorber</td>
</tr>
<tr>
<td>11</td>
<td>Expanded steam entering heat recovery unit</td>
<td>31</td>
<td>Sea water leaving absorber</td>
</tr>
<tr>
<td>12</td>
<td>Flue gases leaving HRSG</td>
<td>32</td>
<td>Cold sea water entering condenser</td>
</tr>
<tr>
<td>13</td>
<td>Flue gases exiting heat exchanger</td>
<td>33</td>
<td>Sea water leaving condenser</td>
</tr>
<tr>
<td>14</td>
<td>Sea water entering heat recovery unit</td>
<td>34</td>
<td>Circulation water entering evaporator (VAC)</td>
</tr>
<tr>
<td>15</td>
<td>Hot sea water leaving heat recovery unit</td>
<td>35</td>
<td>Chilled water supplied to building</td>
</tr>
<tr>
<td>16</td>
<td>Sea water entering heat exchanger</td>
<td>36</td>
<td>Hot sea water supplied to 2nd stage MD</td>
</tr>
<tr>
<td>17</td>
<td>Hot sea water exiting heat exchanger</td>
<td>37</td>
<td>Cold sea water supplied to 1st stage MD</td>
</tr>
<tr>
<td>18</td>
<td>Hot sea water entering generator</td>
<td>38</td>
<td>Cold sea water supplied to 2nd stage MD</td>
</tr>
<tr>
<td>19</td>
<td>Hot sea water to 1st stage membrane distillation plant</td>
<td>39</td>
<td>Sea water leaving cold side of MD</td>
</tr>
<tr>
<td>20</td>
<td>Water vapor entering condenser</td>
<td>40</td>
<td>Distillate produced in MD</td>
</tr>
</tbody>
</table>
3. System modeling

3.1. Brayton cycle

The reheat Brayton configuration is utilized in Al-Hamra gas turbine power plant as shown in figure 1. The energy balance of reheat brayton cycle is modeled as follows:

The air at ambient temperature enters the compressor at point 1, the temperature of air ($T_2$) leaving the compressor is calculated by,

$$\frac{T_2}{T_1} = \left(\frac{P_r_2}{P_r_1}\right)^{\gamma - 1}$$

(1)

Where $T_1$ and $T_2$ are the temperatures of air entering and leaving the compressor, $P_{r_1}$ and $P_{r_2}$ are the pressures of air before and after compression process, $\gamma$ is the specific heat ratio. Work done by the compressor is calculated by,

$$W_{\text{Compressor}} = m_a C_{p, \text{air}} (T_2 - T_1)$$

(2)

Where $W_{\text{Compressor}}$ is the work done by the compressor, $m_a$ is the mass flow rate of air and $C_{p, \text{air}}$ is the specific-heat capacity of air. The compressed air is supplied to the combustion chamber (CC), where the compressed air is combusted with addition of fuel. Energy balance of processes in CC is given by:

$$Q_{\text{CC}} = m_{g,1} C_{p, \text{gas,1}} T_3 - m_a C_{p, \text{air}} T_2$$

(3)

$$Q_{\text{CC}} = m_{f,1} LHV$$

(4)

Where $Q_{\text{CC}}$ is the heat energy supplied by the combustion chamber, $m_{g,1}$ is the mass of gas leaving the combustion chamber, $T_3$ is the temperature of gas leaving the combustion chamber, $m_{f,1}$ is the mass flow rate of fuel supplied to CC and $LHV$ is the lower heat-value of fuel. The gas at super-heated temperature ($T_3$) is expanded in first gas turbine,

$$Q_{gt,1} = m_{g,1} C_{p, \text{gas}} (T_3 - T_4)$$

(5)

Where $Q_{gt,1}$ is the energy extracted from the first gas turbine and $T_4$ is the temperature of flue gases leaving the gas turbine. The flue gases liberated from the first gas turbine is further combusted in reheat chamber, energy balance in the reheat chamber is,

$$Q_{\text{RH}} = m_{g,2} C_{p, \text{gas,2}} T_5 - m_{g,1} C_{p, \text{gas,1}} T_4$$

(6)

$$Q_{\text{RH}} = m_{f,2} LHV$$

(7)

$Q_{\text{RH}}$ is the heat energy supplied by the reheat chamber, $m_{g,2}$ is the mass of gas leaving the reheat section, $T_5$ is the temperature of gas leaving the reheat section and $m_{f,2}$ is the mass flow rate of fuel supplied to RH. The reheated gas is expanded in the second gas turbine,
Where $T_6$ is the temperature of flue gases leaving the system, which is utilized as the heat source for driving the tri-generation system.

### 3.2. Steam Cycle

#### 3.2.1. Heat recovery steam generator (HRSG)

In the tri-generation cycle, single pressure HRSG with economizer, evaporator and super heater sections are utilized for the generation of super-heated process steam. The temperature profile of HRSG is shown in figure 3. The pinch point is the temperature difference between the gas and water (at saturation temperature) at the point of entry into the evaporator. Pinch point plays a vital role in energy modeling of HRSG. The approach temperature between the economizer exit and entry of evaporator is considered as constant in the modeling. Approach temperature depends on the tube layout of economizer circuit.

Energy balance of individual sections of HRSG is modeled. Using pinch point temperature difference, energy balance of the evaporator is analyzed

$$Q_{gt,2} = \dot{m}_{g,2} C_{p,\text{gas,2}} (T_5 - T_6)$$

Where $T_5$ is the mass flow rate of steam generation line, $T_6$ is the mass flow rate of hot flue gases, $C_{p,\text{gas}}$ is the heat transfer coefficient of hot flue gases, $h_{\text{er}}$ and $h_9$ are the enthalpy of the saturated water and steam respectively, $T_a$ is the temperature of gas entering the evaporator and $T_b$ is the temperature of gas leaving the evaporator. Energy balance of the super heater is shown below,

$$\dot{m}_w (h_{9} - h_{\text{er}}) = \dot{m}_g C_{p,\text{g}} (T_a - T_b)$$

Where $\dot{m}_w$ is the mass flow rate of steam generation line, $\dot{m}_g$ is the mass flow rate of hot flue gases, $C_{p,\text{g}}$ is the heat transfer coefficient of hot flue gases, $h_{\text{er}}$ and $h_9$ are the enthalpy of the saturated water and steam respectively, $T_a$ is the temperature of gas entering the evaporator and $T_b$ is the temperature of gas leaving the evaporator. Energy balance of the economizer can be shown as:

$$\dot{m}_w (h_{10} - h_9) = \dot{m}_g C_{p,\text{g}} (T_6 - T_a)$$

Where $h_{10}$ is the enthalpy of the super-heated steam and $T_6$ is the temperature of hot flue gas entering the HRSG. Energy balance of the economizer can be shown as:

$$\dot{m}_w (h_8 - h_7) = \dot{m}_g C_{p,\text{g}} (T_b - T_{stack})$$

Where $h_7$ and $h_8$ are the enthalpies of the hot water at the entry and exit of economizer and $T_{stack}$ is the stack temperature of flue gases leaving the HRSG.

#### 3.2.2. Steam turbine

High pressure super-heated steam generated in the HRSG is expanded to ambient pressure using a steam turbine. Energy extracted ($W_{ST}$) in the turbine is calculated as follows,

$$W_{ST} = \dot{m}_w (h_9 - h_{10})$$

Where $h_{10}$ is the enthalpy of steam leaving the turbine.
3.2.3. Heat recovery system

The steam liberated from the steam turbine is condensed in a heat recovery system, where steam exchanges heat with sea water to drive both the absorption chiller and membrane distillation unit. Additionally a part of heat energy is recovered by sea water from the stack flue gases. It is integrated with the supply line of refrigeration and desalination modules. Energy balance of the heat recovery system is given by,

\[ Q_{\text{extract}} = \dot{m}_w (h_{11} - h_7) + \dot{m}_g C_{p,g} (T_{\text{stack}} - T_{ff}) \]  
\[ Q_{\text{extract}} = \dot{m}_{SW} C_{p,SW} (T_{\text{SW,out}} - T_{\text{SW,in}}) \]  

Where \( Q_{\text{extract}} \) is the heat extracted from the steam, \( h_7 \) is the enthalpy of water leaving the condenser at 100°C, \( T_{ff} \) is the temperature of flue gas leaving the heat exchanger, \( \dot{m}_{SW} \) is the mass flow rate of sea water, \( C_{p,SW} \) is specific heat capacities, \( T_{\text{SW,in}} \) and \( T_{\text{SW,out}} \) are the temperatures of sea water entering and leaving the heat recovery system. The area of heat recovery system is determined by,

\[ A_{\text{HRS}} = \frac{Q_{\text{extract}}}{U_{\text{HRS}} \Delta T_{\text{LMTD}}} \]  

Where \( A_{\text{HRS}} \) is the area of the heat recovery system, \( U_{\text{HRS}} \) is the overall heat transfer coefficient of the heat recovery system and \( \Delta T_{\text{LMTD}} \) is the logarithmic mean temperature difference.

3.3. Absorption chiller

Absorption chiller considered in the tri-generation system is LiBr/H₂O vapor absorption chiller. The system is designed to provide district cooling to multiple duplex-villas in the region of Al-Hamra, UAE as shown in figure 4. The system is optimized based on cooling load requirements of villas. Energy balance of vapor absorption chiller is calculated as:

\[ Q_{\text{ch}} = \dot{m}_{ac,ch} C_p \Delta T_{ac,ch} \]
\[ Q_{gen} = \dot{m}_{SW} C_{P,SW} \Delta T_{ac,h} \]  

\[ COP_T = \frac{Q_{ch}}{Q_{gen}} \]  

Where \( Q_{ch} \) is the chilling capacity of absorption chiller, \( Q_{gen} \) is the heat supplied the generator of absorption chiller, \( \dot{m}_{ac,ch} \) and \( \dot{m}_{SW} \) are the mass flow rates of chilled water and hot water flowing through the absorption chiller, \( \Delta T_{ac,ch} \) is the temperature difference of inlet and outlet of chilled-water circuit and \( \Delta T_{ac,h} \) is the temperature difference between hot water inlet and outlet of the generator.

### 3.4. Membrane distillation

In the tri-generation system, membrane distillation module with air-gap configuration is considered for clean water production. The membrane modules considered for the analysis is manufactured by Scarab development AB, Sweden. The hot sea water leaving the generator of the absorption chiller is supplied to large number of multi-effect membrane distillation modules connected in parallel. Technical specifications of membrane distillation module are shown in table 1.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Membrane area</td>
<td>2.8m²</td>
</tr>
<tr>
<td>Porosity (( \phi ))</td>
<td>0.8</td>
</tr>
<tr>
<td>Membrane thickness (b)</td>
<td>0.2mm</td>
</tr>
<tr>
<td>Air gap Width (L)</td>
<td>1mm</td>
</tr>
<tr>
<td>Height of the module</td>
<td>730mm</td>
</tr>
<tr>
<td>Width of the module</td>
<td>630mm</td>
</tr>
<tr>
<td>Thickness of the module</td>
<td>175mm</td>
</tr>
</tbody>
</table>

The mass and energy balance of membrane modules are calculated based on experimental equations developed through series of experiments conducted in Nyköping, Sweden [15]. The mass flow rate of sea water supplied to hot and cold side is optimally selected as 1200 kg/h based on detailed experimental campaign [15]. The experimental equations for mass and energy flux are derived in function of porosity, air gap thickness, membrane thickness and inlet temperatures of hot and cold fluids. These equations provide better approximations [15, 16].

\[ M_{dis} = 4.1 \times 10^{-3} \times \frac{1}{[b/(\phi \sqrt{T_{MD,H,in}})] + (L/\sqrt{T_{MD,C,in}})} \times \ln \left( \frac{1 - X_c}{1 - X_h} \right) \]  

Where \( M_{dis} \) is the mass of distillate produced per hour for unit surface area of the membrane, \( b \) is the thickness of the membrane, \( L \) is the air gap distance, \( X_c \) and \( X_h \) are the mole fraction of water vapor on at the condensation and evaporation surfaces, \( T_{MD,H,in} \), \( T_{MD,H,out} \), \( T_{MD,C,in} \) and \( T_{MD,C,out} \) are the temperature of hot water inlet, hot water outlet, cold water inlet and cold water outlet of the AGMD.
Where $p_{i,c}$ and $p_{i,h}$ are the partial pressures of the vapor at hot and cold sides. $P$ is the total pressure, Energy flux ($E_{MD}$) of the AGMD is calculated by:

$$E_{MD} = \frac{1.5 \times 10^{-3} \times (T_{MD,H,in} - T_{MD,C,in}) \times \left(1 + 1.41 \times \ln \left(\frac{1 - X_c}{1 - X_h}\right)\right)}{b \left(\gamma \phi \sqrt{T_{MD,H,in}}\right) + \left(L \sqrt{T_{MD,C,in}}\right)}$$

$$\gamma = \frac{k_{membrane}}{\phi \times k_{air}}$$

$\phi$ is the porosity of the membrane material, $k_{air}$ is the thermal conductivity of air, $k_{membrane}$ is based on material type. The useful energy consumed by the membrane distillation is calculated as:

$$Q_{dis} = M_{dis} \times (\lambda_L)$$

Where $Q_{dis}$ is the useful energy required for evaporation of vapor, $\lambda_L$ is the latent heat of evaporation.

3.5. Energy efficiency

Energy efficiency is the ratio of useful energy produced by the system to the amount of energy supplied to the system. In this study, efficiency of gas turbine power plant (existing), combined cycle and tri-generation (power, cooling and desalination) are individually analyzed.

$$\eta_{GT} = \frac{W_{GT}}{\dot{m}_f \times LHV}$$

$$\eta_{CC} = \frac{W_{GT} + W_{ST}}{\dot{m}_f \times LHV}$$
\[ \eta_{Tri} = \frac{W_{GT} + W_{ST} + Q_{ch} + Q_{dis}}{m_f \cdot LHV} \]  

(29)

Where \( W_{GT} \) and \( W_{ST} \) are the work done by the gas turbine and steam turbine, \( m_f \) is the mass flow rate of the fuel, \( LHV \) is the lower heating value, \( Q_{ch} \) is the chilled energy produced in the absorption chiller and \( Q_{dis} \) is the useful energy utilized by membrane distillation unit.

4. Results and discussion

Thermodynamic modeling and optimization of complete tri-generation system for cooling, desalination and hot water is analyzed and discussed in this section.

4.1. Brayton cycle

The gas turbine cycle is numerically modeled with data and parameters provided by the Al-Hamra gas turbine power plant [19]. The reheat-brayton cycle operated in the Al-Hamra power plant is optimized to maintain a constant temperature of 1097K, at the inlet conditions of both the gas turbines irrespective of the air intake temperature. The pressure ratio and the rate of air intake at the compressor as well as the rate of fuel intake in the combustion and reheat chambers were optimized to achieve 1097K at the inlet of gas turbines. The mean operational parameters of gas turbine are shown in table 2. The performance of brayton cycle is analyzed for varying the air inlet temperatures as shown in figure 4.

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas turbine model (GE- manufacturer)</td>
<td>Model: LM2500 PE</td>
</tr>
<tr>
<td>Gas turbine inlet temperature</td>
<td>824°C</td>
</tr>
<tr>
<td>Mean intake air temperature</td>
<td>35°C</td>
</tr>
<tr>
<td>Mean exhaust gas flow</td>
<td>64.1 kg/s</td>
</tr>
<tr>
<td>Mean exhaust gas temperature</td>
<td>550°C</td>
</tr>
<tr>
<td>LHV</td>
<td>47,000 kJ/kg</td>
</tr>
</tbody>
</table>

Figure 4. Performance of gas power cycle for varying inlet temperatures
The air intake temperature is an influential factor in augmenting the performance of gas turbine cycle. Reduction in the air intake temperature can drastically increase the gas turbine power output. But on the contrary, reduction in air intake temperature increases the rate of fuel consumption in order to provide the desired operating temperature at the inlet of gas turbine. The mass of fuel consumption is derived based on the energy balances at the combustion and reheat chambers. Increase in fuel consumption has direct impact on the exhaust gas flow rate. The temperature of flue gases liberated from the gas turbine rises with the increase in air intake temperature. This results in reduction of pressures ratios with increase in air intake temperature to provide designed gas turbine inlet temperature.

The annual performance of the gas cycle in term of electricity generation and mass of fuel consumption is shown in figure 5. As discussed earlier, the air inlet temperature is the influential parameter in the performance of the gas cycle. The electrical performance of the gas cycle reduces by 10% in peak summer months. Maximum productivity of 34 MW is achieved in winter months. Annual variations in the amount of flue gases produced and its temperatures are shown in figure 6. In summer, the temperature of flue gases maximizes as the pressure ratio is reduced at higher air intake temperatures. The mass flow rate of exhaust gases liberated from the gas cycle is reduced in summer due to reduction in the amount of fuel consumption and rate of air intake.
Figure 5. Annual variations in power production, rate of fuel intake and exhaust gas parameters in gas cycle

4.2. Steam cycle

In HRSG, outlet conditions of the super-heated steam are optimized based on maximum permissible temperature and pressure ranges of the steam turbine. HRSG system is simulated for different inlet conditions of flue gases at feed pressure of 65 bar. Feed water flow rates are optimized for secure operation of steam cycle. Limiting the temperature of super-heated steam below the maximum acceptable temperature limit of the steam turbine is considered as the optimization criterion. Maintaining constant feed water flow rate in the steam cycle allows uniform heat recovery to drive cooling and desalination cycles throughout the year. Q-T profile of HRSG for the design conditions in the month of July is shown in figure 6. HRSG is simulated with a constant approach temperature and pinch point difference of 10K. Steam turbine is selected based on design conditions and requirements. The parameters of the steam turbine are shown in table 3.

The performance of the steam cycle in the tri-generation system is analyzed for the variations of air intake temperatures. The system is simulated for different inlet conditions of flue gases as shown in
figure 7(a). The electricity production and stack temperature of flue gases leaving the HRSG are chosen as the prime indicative parameters in this study. The electricity production in steam power cycle follows similar trend as gas power cycle. Reduction in the quantity of flue gases at higher air intake temperature is the major influencing factor for the decreasing trend of power production. Energy balance in the economizer indicates that the stack temperature decreases with reduction in the mass-flow rate of flue gases. Annual performance of steam cycle is shown in figure 7(b). The performance of the steam cycle improves during winter months due to higher mass-flow rate of flue gases.

Table 3. Technical specifications of Steam turbine [20]

<table>
<thead>
<tr>
<th>Description</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam turbine model (Siemens)</td>
<td>SST-100</td>
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<tr>
<td>Power output</td>
<td>8.5 MW</td>
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<tr>
<td>Inlet steam pressure</td>
<td>Up to 65 bar</td>
</tr>
<tr>
<td>Inlet steam temperature</td>
<td>Up to 480°C</td>
</tr>
<tr>
<td>Condensing pressure</td>
<td>1 bar</td>
</tr>
<tr>
<td>Exhaust area</td>
<td>0.22 m²</td>
</tr>
</tbody>
</table>

Figure 6. Q-T profile of HRSG in July

Figure 7. (a) Performance of steam power cycle for varying inlet temperatures (b) Annual performance of steam cycle
4.3. Absorption chiller

In the district cooling network, duplex houses in the Al-Hamra region are considered. Duplex houses are two identical houses sharing a common wall with a total floor area of 390m$^2$ (195m$^2$ for single house). The cooling load requirement of a duplex house binding the ASHRAE standards is estimated by modeling and simulating the duplex house in TRNBuilt software. The annual cooling load distribution (hourly distribution) for maintaining indoor set point temperature of 22°C is shown in figure 8. The building simulation is conducted with occupancy of 5 persons per building and infiltration rate of 0.25 ACH. The enlarged peak cooling requirement of a duplex building is shown in figure 8. Cooling energy requirements in the summer reach up to 36.5 kW for a duplex house, this peak cooling requirement is considered as the design condition for the distribution in the district cooling network. The COP of the chiller varies between 0.75 and 0.69, as the system is operated with a hot water inlet temperature of 90°C. The variation in COP is mainly due to the temperature of cold water supplied to the condenser.

Based on the energy balances of the heat recovery system, the total hot sea water production rate is determined to be 358 kg/s. This can be used to determine the chilling capacity of the absorption chiller. As discussed earlier, the peak load requirement is considered as the design parameter for sizing the distribution network. Performance of the absorption chiller plant during the peak load period with varying inlet hot water temperatures is shown in figure 9 (a). The chilled energy production of the absorption chiller maximizes at higher temperatures. At the supply temperature of 90°C, 4621.69 kW of chilled energy is produced. It is sufficient to meet the cooling demands of 124 duplex villas as the peak cooling demand per villa is 36.5 kW with a COP of 0.69. Reduction in the inlet hot water temperature by 10°C affects the performance of chiller by 39%; steep decrease in performance of absorption chiller is observed with reduction in supply temperature. Operation of the absorption chiller is optimized based on the cooling demand requirements of the building. Month-wise cooling energy production by the absorption chiller is shown in figure 9 (b). The production maximizes in the month of July as it corresponds to the cooling energy peak demand. **Figure 8.** Annual cooling load in a duplex house.
Figure 9. (a) Performance of absorption chiller with varying inlet hot temperature (b) Month-wise cooling energy production

Figure 10. Distillate production for variation in hot and cold side temperatures

4.4. Membrane distillation

Performance of two-stage membrane distillation system is evaluated by varying the temperatures of hot and cold sides as shown in figure 10. The system is simulated for hot side temperatures from 60°C to 90°C and cold side temperatures between 10°C and 50°C with an increment of 5°C. The distillate productivity improves with increase in hot side temperature and decrease in cold side temperature. Maximum distillate productivity of a two-stage system is 41 kg/h with hot side temperature of 90°C and cold side temperature of 10°C.
Based on the energy balance in the heat recovery system, a total of 967 multi-effect membrane desalination units can be connected in parallel, each unit contain two membrane modules connected in series. The thermal energy supplied to the membrane distillation plant is optimized based on cooling energy requirements. Year round dynamic simulation of two-stage membrane distillation plant is conducted, productivity is affected by the cold side temperature and cooling energy demands as shown in figure 11 and 12. Sea water at ambient temperature is supplied to the cold side of the membrane distillation system. The productivity of desalination plant reduces gradually with increase in cooling energy demand and ambient temperature. Productivity of the plant drops by 12% in the summer due to higher cooling energy demands and cold side temperatures.

**Figure 11.** Effect of ambient temperature on distillate production
4.5. Energy efficiency

Month-wise energy distribution between different thermal cycles of the tri-generation system is shown in figure 13 (a). Useful energy produced by the tri-generation cycle is reduced in the summer months due to higher air intake temperature in the gas cycle leading to lesser fuel consumption. Month-wise energy efficiencies of gas cycle, combined cycle and tri-generation cycle is shown in figure 13 (b). The gas cycle efficiency and combined cycle efficiency reduces in summer months due to lower power production, while the efficiency of tri-generation cycle increases by 4% during this period due to full scale operation of absorption chiller plant as shown in figures 13 (a) and13 (b).
4.6. Emission analysis

Along with the energy benefits in terms of high thermal efficiencies, the tri-generation system also has a potential to reduce greenhouse gas emissions. Normalized emission of CO$_2$ and NO$_x$ per MWh is reduced by implementing poly-generation cycles. Variations of emissions for all three cycles – gas cycle, combined power cycle and trigeneration cycle are shown in figure 14. The CO$_2$ coefficient emission per kWh of electricity production is utilized for determining annual CO$_2$ emissions. CO$_2$ emission coefficient for United Arab Emirates is 600g of CO$_2$ per kWh of electricity production [21]. Normalized CO$_2$ emission of tri-generation system reduces by 51.5%, which proves the environmental sustainability of the system. This reduction in normalized CO$_2$ emission is achieved through total CO$_2$ avoided by driving the trigeneration system using waste heat recovery. In trigeneration scenario, production has been increased by 51.5% with same amount of fuel usage leading lower carbon emission. Along with CO$_2$, other GHG emissions like NO$_x$, SO$_x$ emissions were also reduced by 51.5%. In UAE, NO$_x$ contribute to 5% of total GHG emissions which leads to normalized production of 38g per kWh [27]. With trigeneration, the NO$_x$ emissions are reduced to 18.4g per kWh. Total CO$_2$ and NO$_x$ emissions of the existing gas turbine power plant is estimated by,

\[ \text{Total emissions} = \text{emission coefficient} \times \text{Annual electricity production} \]  

(30)

![Normalized CO$_2$ and NO$_x$ emissions](image)

4.7. Economic analysis

Economical benefit of implementing the tri-generation system for the waste heat recovery is discussed in this section. Lower operating costs are achieved in the tri-generation system as waste heat is utilized in the complete operation. The cost of individual components and other necessary parameters is shown in table 4. Total investment, operation and maintenance costs and annual benefits are estimated as shown in table 5. The cumulative net present value (NPV) and payback period (PB) of the tri-generation system are evaluated and shown in table 5. Annual benefits by implementing tri-
generation are estimated based on total annual production of electricity, cooling and freshwater as shown in Table 6. The net present value for the project is estimated by,

$$NPV = -C_0 + (B - C) \left[ \frac{(1 + i)^n - 1}{i(1 + i)^n} \right]$$

(31)

Where $C_0$ is the total investment cost, $B$ is the annual benefits, $C$ is annual operation and maintenance cost, $i$ is the interest rate and $n$ is the life time of the project. The payback period of the project is determined by,

$$PB = \frac{\ln(B - C) - \ln(\frac{(B - C) - iC_0}{i})}{\ln(1 + i)}$$

(32)

**Table 4.** Parameter and costs required for economic analysis

<table>
<thead>
<tr>
<th>Component</th>
<th>Abbreviation</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat recovery steam generator [22]</td>
<td>$C_{HRSG}$</td>
<td>Equation 33</td>
</tr>
<tr>
<td>Steam turbine [22]</td>
<td>$C_{ST}$</td>
<td>Equation 37</td>
</tr>
<tr>
<td>Heat recovery unit [23]</td>
<td>$C_{HRS}$</td>
<td>$2000/m^2$</td>
</tr>
<tr>
<td>Feed water pump (Steam cycle) [24]</td>
<td>$C_{ST,PUMP}$</td>
<td>$881 W_p^{0.4}$</td>
</tr>
<tr>
<td>Membrane distillation unit [25]</td>
<td>$C_{AGMD}$</td>
<td>$1375/unit$</td>
</tr>
<tr>
<td>Membrane replacement cost [23]</td>
<td>$C_{MD,R}$</td>
<td>15% of $C_{AGMD}$</td>
</tr>
<tr>
<td>Feed pump (Heat recovery) [24]</td>
<td>$C_{HRS,PUMP}$</td>
<td>$881 W_p^{0.4}$</td>
</tr>
<tr>
<td>Absorption chiller [26]</td>
<td>$C_{AC}$</td>
<td>$400/kW$</td>
</tr>
<tr>
<td>Hydraulics [25]</td>
<td>$C_{HYD}$</td>
<td>5% of component costs</td>
</tr>
<tr>
<td>Installation cost [25]</td>
<td>$C_{I}$</td>
<td>5% of component costs</td>
</tr>
<tr>
<td>Interest rate</td>
<td>$i$</td>
<td>10%</td>
</tr>
<tr>
<td>Life time of the project</td>
<td>$n$</td>
<td>20 years</td>
</tr>
<tr>
<td>Cost of electricity</td>
<td>$C_F$</td>
<td>$0.12/kWh$</td>
</tr>
</tbody>
</table>

The investment of HRSG and steam turbine are determined by [22],

$$C_{HRSG} = 4131.8 \left( f_p \cdot f_{T,steam} \cdot f_{T,gas} \left( \frac{Q}{\Delta T_{in}} \right)^{0.8} \right) + 13380( f_p \cdot \dot{m}_{steam} ) + 1489.7 \dot{m}_{gas}^{1.2}$$

(33)

$$f_p = 0.0971 \frac{Pr}{30 \text{ bar}} + 0.9029$$

(34)

$$f_{T,steam} = 1 + \exp \left( \frac{T_{out,steam} - 830}{500} \right)$$

(35)

$$f_{T,gas} = 1 + \exp \left( \frac{T_{out,gas} - 990}{500} \right)$$

(36)

$$C_{ST} = 3880.5P_{ST}^{0.7} \left[ 1 + \left( \frac{0.05}{1 - \eta_{ST}} \right)^3 \right] \left( 1 + 5 \exp \left( \frac{T_{in} - 866}{10.52} \right) \right)$$

(37)

Where $Q$ is the heat transfer in HRSG, $\Delta T_{in}$ is temperature difference across HRSG and $Pr$ is the pressure of feed water to the HRSG. The total investment of cost ($C_o$) of the project is calculated by,
\[ C_D = C_{HRSG}(P_{HRSG}) + C_{ST}(P_{ST}) + C_{HRS}(A_{HRS}) + C_{ST,PUMP}(P_{ST,PUMP}) + C_{AGMD}(N_{AGMD}) \\
+ C_{HRS,PUMP}(P_{HRS,PUMP}) + C_{AC}(P_{AC}) + C_{HYD} \] (38)

\( P_{HRSG}, P_{ST}, P_{ST,PUMP} \) and \( P_{HR,PUMP} \) are the total capacity of HRSG, steam turbine and feed pumps of the steam cycle and the heat recovery system. \( A_{HRS} \) is the area of the heat recovery system and \( N_{AGMD} \) is the number of membrane distillation modules. The annual operation and maintenance cost for tri-generation project includes electricity requirements for the feed pumps and membrane replacement charges.

In terms of current fuel cost in the region, the system has a short payback of 1.38 years and greater cumulative profit of $66.10 Million. Economical analysis with possibility of changes in electricity prices are analyzed as shown in figure 15. The annual cash flows of the project is simulated with an interest rate of 10% as shown in figure 16.

**Table 5. Economic analysis of tri-generation system**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Investment cost</td>
<td>$11,079,548</td>
</tr>
<tr>
<td>Operating and maintenance cost</td>
<td>$443,494</td>
</tr>
<tr>
<td>Annual benefits</td>
<td>$9,413,460</td>
</tr>
<tr>
<td>Payback period</td>
<td>1.38 years</td>
</tr>
<tr>
<td>Net present value</td>
<td>$66,102,281</td>
</tr>
</tbody>
</table>

**Table 6. Annual benefits**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Total benefits</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity production</td>
<td>$0.12/kWh</td>
<td>$7,289,423.73</td>
</tr>
<tr>
<td>Cooling benefits</td>
<td>$0.10/kWh</td>
<td>$1,582,854.46</td>
</tr>
<tr>
<td>Cooling demand charges</td>
<td>$42.3/kW.year</td>
<td>$211,500.00</td>
</tr>
<tr>
<td>Water cost</td>
<td>$1/m^3</td>
<td>$329,681.90</td>
</tr>
</tbody>
</table>
A tri-generation system based on the integration of power, cooling and desalination thermal cycles was presented. A detailed numerical case study was developed for the implementation in Al-Hamra area, Ras Al Khaimah, UAE. The tri-generation system is modeled for maximizing the utilization of waste heat by driving four thermal cycles based on quality and temperature of thermal energy. The absorption chiller plant is optimized based on cooling energy requirement of duplex villas in the region. The simulation results show that energetic efficiency of tri-generation system maximizes up to 5.

**Conclusion**
85% in summer due to lower fuel consumption and full scale operation of absorption chiller. The thermal cycle performances are largely affected by air intake temperature, but due to lower cooling requirement in winter, the overall thermal efficiency reduces at lower air intake temperatures. Other important conclusions drawn from the project of recovering waste heat from existing power plant are as follows:

- The absorption chiller plant can provide district cooling for 124 duplex houses with a production capacity of 4600kW during peak requirements.
- The production capacity of membrane distillation plant varies between 33 and 37m$^3$/hour based on daily cooling requirements and ambient temperature.
- The normalized CO$_2$ emission per MWh is reduced to 291 kg/MWh from actual scenario of 600kg/MWh by implementation of waste heat recovery system.
- In terms of economy, the system has a rapid payback period of 1.38 years with a cumulative saving of $66 million.

Nomenclature

- \( A \) – Area [m$^2$]
- \( ACH \) – Air changes per hour
- \( AGMD \) – Air gap membrane distillation
- \( b \) – Membrane thickness [mm]
- \( B \) – Benefits [$]
- \( C \) – Cost [$]
- \( C_p \) – Specific heat capacity
- \( CC \) – Combustion chamber
- \( CCHP \) – Combined cooling, heat and power
- \( CCCWP \) – Combined cooling, clean water and power
- \( CHP \) – Combined heat and power cycle
- \( COP \) – Coefficient of performance
- \( E \) – Energy flux [kJ]
- \( HRSG \) – Heat recovery steam generator
- \( NPV \) – Net present value [$]
- \( i \) – Interest rate [%]
- \( GHG \) – Green house gases
- \( h \) – Enthalpy [kJ/kg]
- \( K \) – Thermal conductivity [W/m.K]
- \( L \) – Air gap width [mm]
- \( LHV \) – Lower heating value
- \( M \) – Mass flow [kg/h]
- \( MENA \) – Middle East and North Africa
- \( \dot{m} \) – mass flow rate [kg/s]
- \( P \) – Power Capacity [KW]
- \( p \) – Partial pressure of vapor
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Conflicts of Interest

“The authors declare no conflict of interest”.

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