AN INTEGRATED TRI-GENERATION SYSTEM FOR PRODUCING POWER, WATER AND COOLING/HEATING

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ABSTRACT

Multiple generation system can considerably improve the financial costs and environmental impacts of existing small power plants. The current work study and analyze an integrated tri-generation scenario for producing electric power, potable water and cooling/heating energy based on a small gas turbine and thermal desalination units. This aspect can be applied by retrofitting the existing small gas turbines with retrieval the waste heat energy to drive a developed multi-stage humidification-dehumidification (MSHDH) unit for seawater desalination in the combined system. Additionally, the plant can provide a simultaneous cooling and heating energy for various purposes.

The results of this study indicated that plant's combined efficiency could be enhanced to 72-76% for cogeneration technology owing to the best thermal utilization of fuel. But, this value could be increased to a range of 93% with the optimum utilization of waste to the possible minimum exhaust temperature in the case of using an additional low-scale MED plant for more water production.

Technically, the poly-HDH unit showed a stable operation than that the single-HDH unit under various circumstances. Also, the specific energy consumption for the two seawater desalination plants MSHDH and MED that ranged about 83 kW.hr/m^3, is reasonable and acceptable. Real climatic and technical conditions are applied in case study for petroleum site.

Keywords: Gas turbine, Humidification-dehumidification, Thermal vapor compression, Jet-ejector, Seawater desalination, Cooling and heating.

1. INTRODUCTION

The efficient cogeneration application for producing electric power and water prompted the engineering designers to extend this system to include new technologies of multiple simultaneous generation of thermal heating, cooling and chemical products. This new cogeneration scenario can be applied by using the modern gas-turbine for tri-generation technology. Tri-generation system can be defined as the simultaneous conversion of fuel energy into useful products: electric power, potable water and low temperature energy for cooling/heating purposes. Industrial small gas turbine engine operated on Brayton cycle typically exhausts large waste heat energy to atmosphere at relatively high temperatures in the range of 400-500°C with efficiency of approximately 35-55% [1]. Thus, two-third to half of the heat supplied by the high price fossil fuel is rejected to the atmosphere as waste. That thermal waste apparently offers large temperature difference above ambient which can be recovered throughout the integrated system and reused in the developed multi-stage humidification-dehumidification (MSHDH) and cooling/heating cycles to enhance the efficiency of fuel's utilization.

Many investigations have studied cogeneration systems for power, cooling and heating [2-4], but little studies involved seawater desalination in poly-generation systems.

Water in fact has always been the essence of life for all created things. The need for clean and suitable freshwater is stringent necessity and become an important political, social, economic and national issue in most countries. Therefore, shortage of water is one of the principal factors restraining the development of arid and hyper arid regions, particularly in Middle East. This reality is due to the large desert belt that compasses the Arab region from the Arabian Gulf easterly to the Atlantic shore westerly. In addition, rainfall in this vast desert is erratic and it cannot satisfy water requirements for growing population. Thus, there is a great dependence on desalination processes of seawater and brackish water to substitute the lack, where 52% of the world desalination plants are in the Arab region.

A novel thermal humidification-dehumidification system are primarily proposed by Saadawy et al [5] using the refrigeration vapor compression unit (HDHRVC) for the purpose of enhancing the plant capacity. That study discussed the concept, design and technical analyses of the new way features to solve the problem of low productivity and realize the large-scale capacity. Undoubtedly, the created super cooling action below the sink temperature increases the amount of daily water production over the conventional HDH systems by about 45% in cold climate and 6.5 fold in hot climate. Thus, this new proposed process has been shown to be viable for replacing the traditional humidification-dehumidification applications.

The idea of employing the staging technique in the humidification-dehumidification process is actually not new. But, few researchers have been studied evolution of the unit performance by using multi-stage technology based on the low-temperature solar energy. Chafik [6-8] developed a new air solar-plate collector for heating and humidifying air stream by passing it through hot seawater. Four stages were experimentally utilized to conduct heating/humidifying steps to maximize the final air temperature to 78°C and the specific humidity to 100 g H_2O/kg dry air. Also, Ben-Amro et al [9] tested experimentally the same collector design at different parametric conditions. The maximum air temperature reached 70°C for both indoor and outdoor tests.

Although the previously proposed single HDHRVC [5] and the multi-stage conventional units [6-9] had increased the amount water produced, but the commercial promising large-scale production by these techniques is still unattainable for seawater desalination units. One of its main drawbacks is the need for high temperature energy source to practically increase the potential freshwater productivity. Thus, the relatively low temperature of solar source limits the economic requirements for high production. Therefore, this study presents a new methodology concerning the idea of using the multi-stage technique by the waste thermal sources of relatively medium temperature such as that rejected from power generation plants.

Energy supply concerns some individual small societies far from civilization are often different and independent on the distributed energy via the unified electric network in most countries. These societies such as coastal tourist villages, separate and isolated residential communities (in desert, islands and elevated mountains) and the petroleum and mining sites, have special small centralized conventional power stations for energy needs. Such self-efficient generation systems had indicated a growing interest in the last few decades in Egypt as a result of the financial and economic development. Energy generation systems sized from tens kilo-Watts to few mega-Watts output power of single or dual-purpose have been used for providing the previously mentioned sectors with the requirements of electricity and/or water. These existing small power generation units are candidates to be retrofitted for the proposed new tri-generation system by exploiting the waste thermal energy for producing the required potable water for some sectors in additional to generating a cooling/heating energy for supplying the residents with comfortable space.

Thus, the main objective is studying the performance of the new combined tri-generation arrangement and the involved MSHDH operating at elevated air temperatures than that produced by solar energy.

2. DESIGN CRITERIA OF THE COMBINED SYSTEM

The constituents of the proposed tri-generation system as shown in Fig. 1 are: gas turbine unit, MSHDH unit and MED unit. The small gas turbine is assumed to operate on the modified open Brayton cycle with regeneration. The unit usually consists of: air compressor, combustor, recuperator (regenerator) and gas turbine. Retrieval of thermal waste is technically achieved by the heat recovery boiler HRB (vapor generator) which combines the gas turbine cycle with the desalination cycle. A modified multi-stage humidification-dehumidification MSHDH unit is suggested for production of potable water for the following reasons:

- a- It is more appropriate to be connected with the high temperature heat recovery boiler because it can operate at elevated temperatures up to 160°C for carrier air, which is considered higher than the working temperatures of other thermal processes such as MSF (105°C) or MED (110°C).
- b- The potential for scale formation in MSF and MED increases with high temperature of heat recovery boiler if they are directly connected with it.

c- Usually, the heat recovery boiler is used to supply superheated steam at high temperature for steam turbine in combined power cycle, and the high steam pressure and temperature insufficient for use in MED.

Therefore, it is preferable to use MSHDH to be connected with HRB instead of other thermal processes. Because the temperature of combustion gases leaving the HRB is still relatively high (170-180°C), it can be used again for producing additional potable water by connecting another thermal desalting unit such as MED.

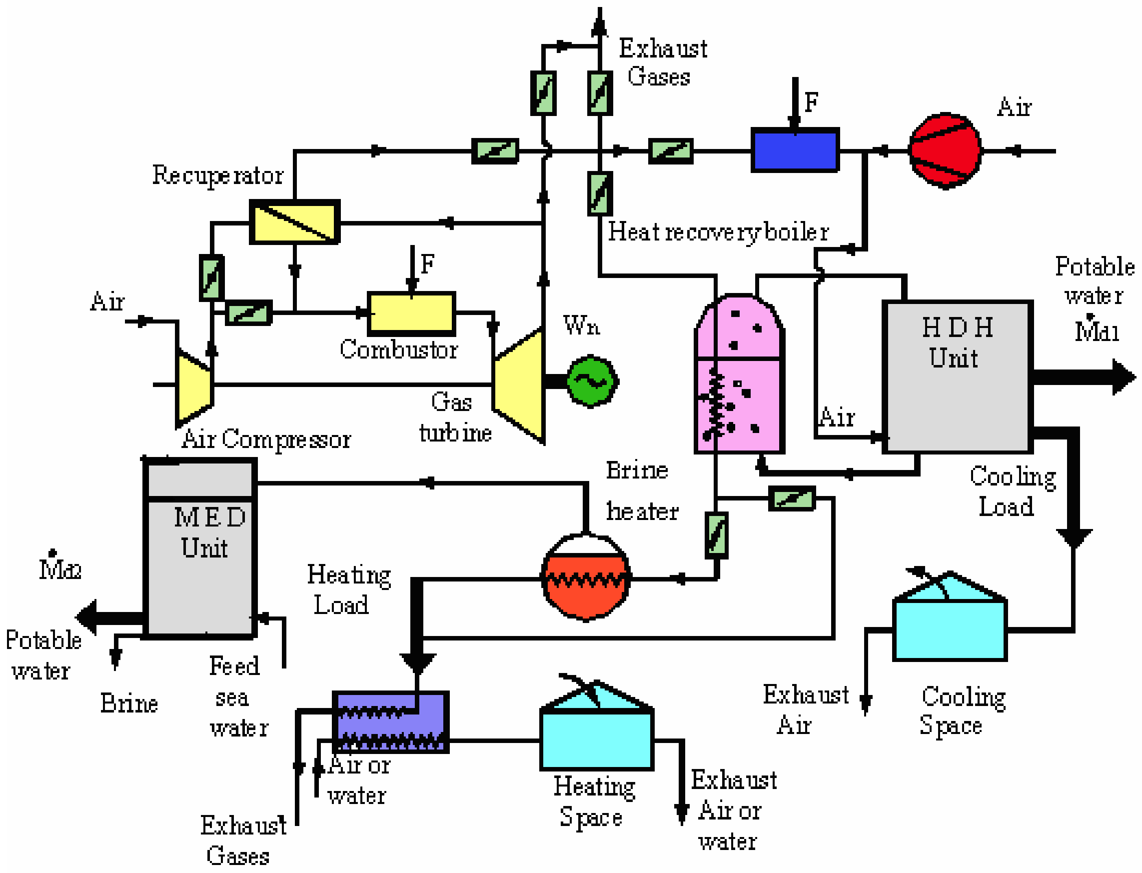

Fig. 1 Integrated Tri-generation System

Figure 2 indicates the idea of staging plotted on psychometric chart. The new MSHDH unit consists mainly of the thermal refrigeration vapor compression unit, in addition to several heater/humidifier units connected in series. The vapor-compression unit usually creates the necessary low supercooling action for the condensation of the moisture in the dehumidifiers. While, the consecutive heating/humidifying units are used to obtain a high humidity content which can finally be harvested as freshwater free of salt.

In this stepwise heating/humidifying arrangement a small air to water ratio can be obtained 10:1 or less, and the necessary heat source temperature is less than 100°C. The maximum air temperature reached in this task not exceeds 85°C. The main idea in this new process aims to achieve a high freshwater production of low power consumption without a considerable increase of either air flow or the maximum operating temperature.

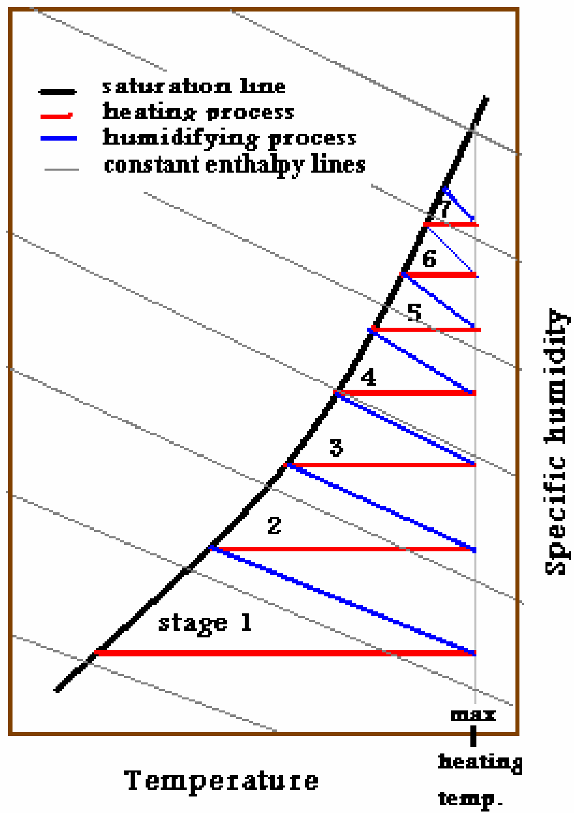

Fig 2. Stepwise heating/humidifying process

The intake air at initial condition of ambient temperature T1 and moisture content of less than 2% can be heated to a temperature T2 in the heat recovery units (high temperature dehumidifier and refrigerant condenser 1). In the first stage, air is further heated to a temperature T3 in first heater (HE1) by waste heat energy. Then, the air is brought to be humidified by injecting seawater in the first humidifier HU1 till the saturation condition (state 4). During the second stage, the air flows to the second heating step (HE2) and heated up to state 5 and then is loaded with further seawater to reach state 6 of 100% relative humidity. If it is desired to implement additional stages, the process proceeds in consecutive steps of heating and followed by humidification, where a part of seawater evaporates as water vapor leaving the salts behind. Finally, air can be moistened by water to a concentration of 12% or more, where the end hot air temperature approaches about 55-70°C (state 7). Thus, the moisture in the humid air can now be condensed as water free of salts in the dehumidification sections. The first is the high temperature dehumidifier (HTDH), and it is practically cooled using seawater as coolant, while the second is the low temperature dehumidifier (LTDH) which is cooled by the vapor compression unit. So, the air leaves the second dehumidifier at state 8 with very low temperature (0°C or less).

The feed seawater is heated at first via the heat recovery sections and finally is heated in high temperature water heater to a temperature T18 and then distributed in parallel to all humidifiers. Obviously, the new proposed system using the multiple staging techniques is designed and developed to be alternate to the traditional vapor-compression humidification-dehumidification system (HDDRVC) of single-stage.

3. COMPUTATIONAL PROGRAM

The tri-generation performance is calculated in this section. A detailed description for the methodology of modeling the tri-generation system including the power unit, MSHDH desalination unit with thermal vapor compression unit, MED unit and the cooling/heating energy is presented. The refrigeration vapor-compression humidification-dehumidification system (HDDVC) is intensively explained before in reference [5], with the controlling parameters and thermal relations. The previous computational equations are also applied here using the staging arrangement. The principles of the current tri-generation theoretical model are arranged in a computer program (TRIGEN-PROG). This computational program takes into account the number of stages of HDH, different types of working fluids, heat source characteristics, variation of thermo-physical properties of working fluids, jet-ejector configurations, meteorological conditions, humidifiers and dehumidifiers specifications, efficiency of different thermal equipments, heat pipe design features, operational limits of heat pipes, scale formation limits and seaweter properties.

The electric work generated by the gas turbine and that required for the compressor is evaluated:

$$W_t = \mathrm{m}^{\mathrm{o}}_{\mathrm{gas}}\ (h_{c,o} - \mathrm{h}_{\mathrm{t,o}}) \quad : \quad W_{com} = \mathrm{m}^{\mathrm{o}}_{\mathrm{gas}}\ (h_{com,o} - \mathrm{h}_{\mathrm{com,i}}) \tag{1}$$

The heat balance in the recuperator and its effectiveness are;

$$Q_{rec} = \mathrm{m}^{\mathrm{o}}_{\mathrm{gas}}\ (\mathrm{h}_{\mathrm{c,i}} - \mathrm{h}_{\mathrm{com,o}}) = \mathrm{m}^{\mathrm{o}}_{\mathrm{gas}}\ (\mathrm{h}_{\mathrm{t,o}} - \mathrm{h}_{\mathrm{exh}}) \tag{2}$$

$$\varepsilon_{rec} = \frac{\mathrm{T}_{\mathrm{c,i}} - \mathrm{T}_{\mathrm{com,o}}}{\mathrm{T}_{\mathrm{t,o}}\ \mathrm{T}_{\mathrm{com,o}}} \tag{3}$$

Where, $h_{c,o}$ is the enthalpy at exit and from the combustor at 1300 K and $\mathrm{h}_{\mathrm{c,i}}$ are the enthalpy at inlet to the combustor. While, $h_{com,o}$ and $\mathrm{h}_{\mathrm{com,i}}$ are the enthalpy at the exit and inlet from/to the compressor. Also, $\mathrm{h}_{\mathrm{t,o}}$ and $\mathrm{h}_{\mathrm{exh}}$ are the enthalpy at the exit from the turbine and the enthalpy of the exhaust gases exit from the recuperator. The modified heat input by the fossil fuel to the power unit and that recovered by the desalination and cooling/heating are calculated:

$$Q_f = \mathrm{m}^{\mathrm{o}}_{\mathrm{gas}}\,(\mathrm{h}_{\mathrm{c,o}} - \mathrm{h}_{\mathrm{c,i}}) \quad : \quad Q_{rec} = \mathrm{m}^{\mathrm{o}}_{\mathrm{gas}}\,(\mathrm{h}_{\mathrm{exh}} - \mathrm{h}_{\mathrm{rej}}) \tag{4}$$

Where, $\mathrm{h}_{\mathrm{rej}}$ is the enthalpy of rejected gas from the heat recovery boiler to the atmosphere which is higher than the ambient temperature.

The unit productivity per day for MSHDH is calculated as:

$$\overset{\bullet}{M}_{d,MSHDH} = \overset{\bullet}{M}_{d1} + \overset{\bullet}{M}_{d2} = \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,(\phi_6 - \phi_8) \tag{5}$$

Where, the production of the two dehumidifiers LTDH and HTDH is:

$$\overset{\bullet}{M}_{d1} = 86.4\ \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,(\phi_6 - \phi_7) \qquad : \qquad \overset{\bullet}{M}_{d2} = 86.4\ \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,(\phi_7 - \phi_8) \tag{6}$$

Condensation load in two dehumidifiers can be calculated as:

$$Q_{ev1} = \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,(h_6 - h_7) \qquad : \qquad Q_{ev2} = \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,(h_7 - h_8) \tag{7}$$

Total condensation load in dehumidifiers are:

$$Q_{ev} = Q_{ev1} + Q_{ev2} = \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,(h_6 - h_8) \tag{8}$$

Enthalpy of air exit from humidifier and entering the high temperature dehumidifier T_7 is:

$$h_7 = \mathrm{h}_8 + \frac{\overset{\bullet}{\mathrm{m}}_{\mathrm{s}}\,(\mathrm{h}_{15} - \mathrm{h}_{14})}{\overset{\bullet}{m}_a} \tag{9}$$

Where, $\mathrm{m}^{\cdot}_{\mathrm{s}}$ and $\mathrm{m}^{\cdot}_{\mathrm{a}}$ are the mass flow rate of both secondary refrigerant and air respectively. High temperature air heaters loads are:

$$Q_{htah} = \overset{\bullet}{\mathrm{m}}_{\mathrm{a}}\,[(h_3 - h_2) + (h_5 - h_4) + \ldots\ldots\ldots\ldots + (h_{2\mathrm{N}+1} - h_{2\mathrm{N}}) \tag{10}$$

Where, N is the number of stages. High temperature water heater load is:

$$Q_{htwh} = \overset{\bullet}{\mathrm{m}}_{\mathrm{tw}}\,[(h_{18} - h_{188}) \tag{11}$$

The total amount of seawater injected in the dehumidifiers is:

$$\overset{\bullet}{m}_{tw} = \overset{\bullet}{m}_{w1} + \overset{\bullet}{m}_{w2} + \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots + \overset{\bullet}{m}_{wn} \tag{12}$$

The unit coefficient of performance COP_{hd} is:

$$COP_{hd} = \frac{\mathrm{Q_{ev1}} + \mathrm{Q_{ev2}}}{\mathrm{Q_g} + \mathrm{Q_{htah}} + \mathrm{Q_{htwh}} + \mathrm{W_{pr}} + \mathrm{W_{pw}} + \mathrm{W_{pa}}} \tag{13}$$

The coefficient of performance of the refrigeration vapor compression unit is:

$$COP_{vc} = \frac{\mathrm{Q_{ev2}}}{\mathrm{Q_g} + \mathrm{W_{pr}}} \tag{14}$$

The amount of potable water produced by MED can be calculated by the following equation:

$$\overset{\bullet}{M}_{d,\mathrm{MED}} = \mathrm{PR}\ \frac{\varepsilon_{\mathrm{BH}}\ \mathrm{m^{o}_{gas}}\ (\mathrm{h_{bh,i}} - \mathrm{h_{exh}})}{4.2\,(\mathrm{T_{TBT}} - \mathrm{T_{amb}}) + \mathrm{h_{fg@TBT}}} \tag{15}$$

The efficiency of the power unit (turbine) and heat process (recover) are:

$$\eta_w = \frac{\mathrm{W_t}}{\mathrm{Q_f}} \quad : \quad \eta_h = \frac{\mathrm{Q_{recv}}}{\mathrm{Q_f}} \tag{16}$$

The overall efficiency of the combined tri-generation system is defined as the sum of the net turbine output and the recovered energy, it can be evaluated as:

$$\eta_{over} = \frac{\mathrm{W_t} + \mathrm{Q_{recv}}}{\mathrm{Q_f}} = \eta_{\mathrm{w}} + \eta_{\mathrm{h}} \tag{17}$$

4. RESULTS AND DISCUSSION

4.1 Tri-generation system characteristics

It is necessary to analyze the performance of the combined tri-generation plant as long as it can operate in variable climatic conditions allover the year with wide temperature range. The computational procedures are based mainly on the system data shown in Table (1). The effect of inlet (ambient) temperature to the air compressor on the combined system characteristics at ambient temperature range between -10°C to 45°C are shown in Figs. 3 and 4.

Table (1) Tri-generation System Data

1. Power unit (gas turbine)	
Air compression ratio in compressor	8
Max gas temperature inlet to gas turbine	1027°C (1300 K)
Recuperator effectiveness	0.80
Air compressor/gas turbine efficiencies	80 /85 %
Air mass flow rate to compressor	17 kg/s
Combustion gas temperature at exit from recuperator	330-406°C
Exhaust gas temperature to atmosphere	35-90 °C
2. First desalination unit (MSHDH)	
Number of stages	4
Number of dehumidifiers	2
Max. heating temperature of air	75 °C
Humidity ratio	100 %
Ambient temperature range	-10 - 45 °C
Evaporator temperature of thermal vapor compression unit	7 °C
Generator (primary) temperature of thermal vapor compression ratio	110- 140 °C
Reject air temperature exit from dehumidifier	10 °C
3. Second desalination unit (MED)	
Number of effects	14
Top brine temperature range	90- 105 °C
Performance Ratio (PR)	12
Total temperature difference through MED unit	45-85 °C
Temperature drop/stage	3.2-6 °C
4. Heating and Cooling energy	
Max. heating temperature in heating space	25 °C
Cooling air temperatures (supply/exit) in cooling space	10/26 °C

The simulation results showed generally the great dependence of the system operation on the seasonal condition, where power, water and cooling/heating wholly vary with ambient temperature as indicated in Fig. 3. It is interesting to note that over 50% of the power produced by the gas turbine is consumed to drive the compressor alone. Additionally, in case of high climatic temperature in hot summer seasons, that consumed power is practically increases, thus resulting in a relatively low net produced electric power and the pertinent plant efficiency. The generated net power decreased by about 22% in the temperature range -10-45°C. This in turn increases the released heat energy and reduces the power to heat ratio PHR of the plant from 0.82 to 0.588 as shown in Fig. 4. As the availability of the recovered heat energy reduces with the ambient temperature, production of water continuously degrades. In contrary, the heating and cooling loads increase as a result of raising the allowable temperature drop of exhaust gases in the air heater and cold air mass flow rate respectively.

It is obvious from the results that the electrical efficiency of such small gas turbines is in the range 34-42% for the single power generation purposes. But, when the proposed tri-generation aspect is introduced to get the most economic exploitation for the waste heat energy, the overall plant performance can be readily increased. Therefore, the overall efficiency of the integrated system increased to the range of 70-76% in the case of using the thermal HDH unit as water production unit in addition to a cooling energy. If a supplementary MED unit is attached to the system for producing additional water, because the exhaust gases have excessive temperatures higher than ambient, the overall efficiency is improved up to 93%. Although the great changes in net heat generated and the recovered heat energy, but generally the system overall efficiency nearly remains constant and independent of the ambient temperature, because the changes not exceed 2% allover 55°C change in climatic temperature.

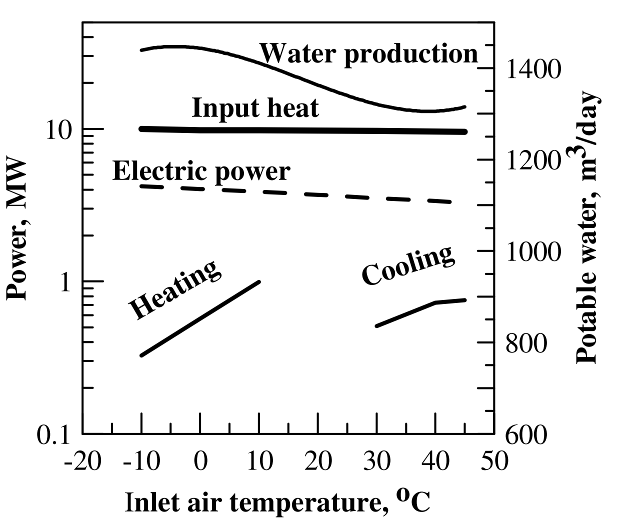

Fig. 3 Outputs of tri-generation plant

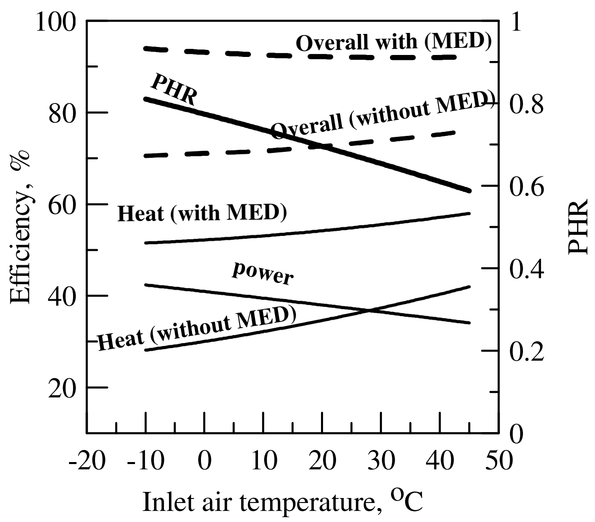

Fig. 4 Characteristics of tri-generation

Subsequently, as it is indicated from this analysis, the importance of integrated tri-generation system is undoubted the fuel savings due to heat recovery of 58% of the input fuel energy which equals 1.7 of the generated power (PHR = 0.558 at 45°C). But, the feasibility cost of this project is mainly based to a great extend on the fossil fuel price. On the other hand, the fuel prices has been shown a dramatic variation which increased from less than 40$ to more than 140$ and finally dropped to about 45$.

4.2 Performance of the new MSHDH

The modified features of the proposed unit MSHDH is presented through a comparative analysis for a two-stage unit of the MSHDH and a conventional single-stage unit HDRVC [5] operating at similar operational and design conditions.

The performance characteristics of the two units presented by the amount of daily water production and coefficient of performance COP are plotted in Fig. 5 at variation of generator temperature from 60-145°C. For the same conditions, the two units are operated at: ejector configuration $A_3/A_1 = 125$; $A_2/A_1 = 27$ and $d_i = 0.028$ m, generator temperature $T_g = 120°C$, $T_{amb} = 20°C$, and maximal air and water temperatures of 80 and 60°C respectively. It is evident from the figure that the amount of water produced by the two-stage unit is kept constant at 206 m^3/d and independent on generator temperature T_g.. But, the single-stage unit has an opposite trend to the two-stage unit, where the unit performance increases with the generating temperature. Then, the trend of the two units approaches quickly, till the two units can produce the same amount of water at generator temperature of about 113°C for the same given operating and design conditions. But, from the point of economic view, it is preferable to operate the single-stage unit at low generating temperature (90-110°C).

Figure 6 illustrates the relationship between the air mass rate $\dot{m}_a$ (carrier gas) and the characteristics of the two-stage unit at variation the air in the range of 1-6 kg/s. The designate operational and configuration parameters are: $A_3/A_1 = 125$; $A_2/A_1 = 27$ and $d_i = .03$ m, T_g= 90°C, T_{amb}= 10°C, and maximum allowable air and water temperatures are 50°C and 40°C respectively. As it is depicted from the figure, both the unit productivity and thermal efficiency (COP) of the MSHD-RVC are increased linearly with air flow rate. Where, the potential amount of freshwater produced may nearly grow by six fold in the studied range of intake air rate as a result of increasing the rate of the accompanied water that can possibly be injected into the humidifier and thereby the specific air humidity (Eqns. 1 and 2). Also, the economic performance of the multistage unit is intensively improved, as the COP increased by 4.67 fold with air rate variation due to increasing the thermal load of the dehumidifier. In the meantime, the single-stage unit shows a slight declination in its performance with air rate change.

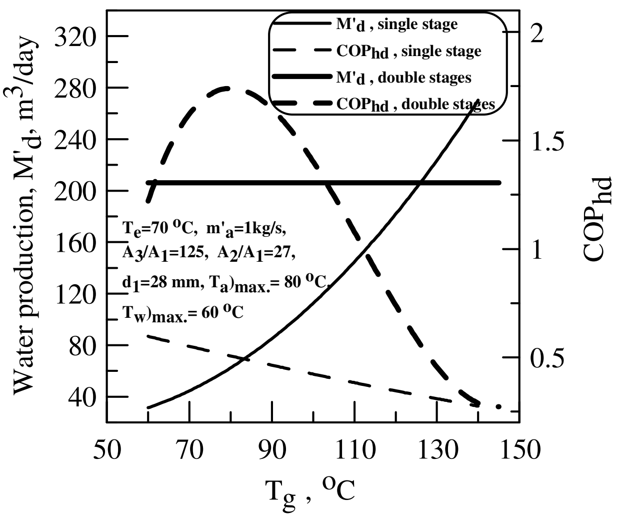

Fig. 5 Characteristics of the two units at variation of generator temp.

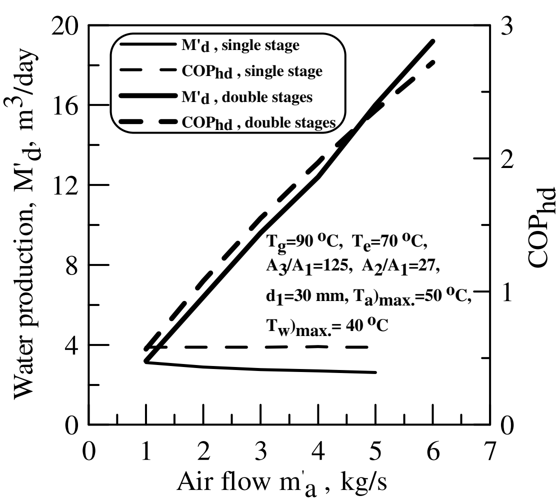

Fig. 6 Characteristics of the two units at variation of air flow rate

Thus, the resulting high performance clearly illustrates the main goal of using the multistage unit instead of the single-stage unit. But, the rise in air rate actually increases the pertinent condensation load of the high temperature dehumidifier Q_{ev1} to a great extend at the expense of the load of the other low temperature dehumidifier

Q_{ev2}. This means that at higher flow rates of air, the size of HTDH becomes very big, while the refrigeration vapor compression unit will be compact owing to the smallest load of its evaporator (LTDH).

The effect of changing the throat diameter d_I of the nozzle on the thermal performance is examined in the range of 1 – 5.5 cm, and the results for the single and multiple stages units is shown in Fig. 7. The other parameters are kept constant at $A_3/A_1 = 125$; $A_2/A_1 = 27$, T_g= 160°C, T_{amb}= 20°C, T_e= 60°C and maximum allowable air and water temperatures are 90°C and 60°C respectively. As is depicted from the figure the freshwater production remains constant at 198 m^3/d regardless of the throat diameter value. This stable behavior may be attributed to the unchanged thermal hydraulic properties of both air and water states (flow rate and temperature), in addition the two dehumidifier's temperatures (inlet to HTDH) and (exit from LTDH) remain constant at 75 and 60^o C respectively. But, the inter-stage temperature changes with variation of the throat diameter which changes the ratio of thermal load divided between the two dehumidifiers. For small throat diameters, the low temperature dehumidifier has relatively small thermal loads, which substantially induce a very compact refrigeration vapor compression unit. In the meantime, this in turn will enhance the unit COP to reach 5.9 at d_I = 1 cm. Therefore, it is convenient to construct a jet-ejector of small size in order to improve the unit economics. From the other side, for single-stage unit, water productivity increases with throat diameter, and it can produce the same amount of that two-stage unit at diameter of about 6.6 cm. But for such specifications, the jet-ejector will be so large to become the pertinent vapor compression unit very bulky which makes the economic issues are very costly.

Generally, the proposed multi-stage humidification-dehumidification MSHDH process showed stable operation than the single stage process because its performance is independent on variation of the generator temperature and throat nozzle diameter of the jet-ejector.

4.3 Selection of the appropriate working fluid (refrigerant)

Most thermo-physical characteristics of working fluids of vapor compression unit are temperature-dependent; therefore it is difficult to operate effectively at wide range of temperature change. Thus, it seems that there is a temperature limits in which the fluid operates well and can realize the best performance of unit. Figure 9 shows the suitable operational temperature range for each working fluid for a certain jet-ejector design. Based on this figure, R-134a seems to have a low generator temperature range between 25-40°C. Thus, the system is suitable to be coupled with external heat sources as: simple flat-plate solar collector, geothermal energy and waste cooling water of thermal equipments. But, R-123 as a working fluid can be applied in the medium temperature range, where its operating temperature can be varied from 50-90°C. Water lies in the high temperature range and it is more desirable to be used practically between 100-170°C. Therefore, for such high range, the external heat source may be supplied in the form of industrial flue gases and waste steam. The system is possible to be driven by:

enhanced solar heaters, diesel engine's exhaust gas and low temperature industrial waste heat. Ethanol and methanol are properly employed as organic working fluids in the medium range between 70-130°C. On the other hand, for solar applications the countries that lie in the northern part of the globe and having very cold climatic condition can use R-134a and R-123 as working fluids. On the contrary, ethanol and water seem to be the viable and appropriate fluids for the countries that lie in the southern part of the globe and having hot weather. It is clear that utilization of different working fluids represents wider application for the MSHDH technology at various climate conditions. Also, if there are several working fluids can operate at a specified temperature range, this gives a great scope for selecting the appropriate one. As depicted from Fig. 9, utilization of methanol as a working fluid gives the proper water productivity at operational temperature range of 90-100°C, which represents12%, 138% and 297% higher than production of R-123, ethanol and water respectively.

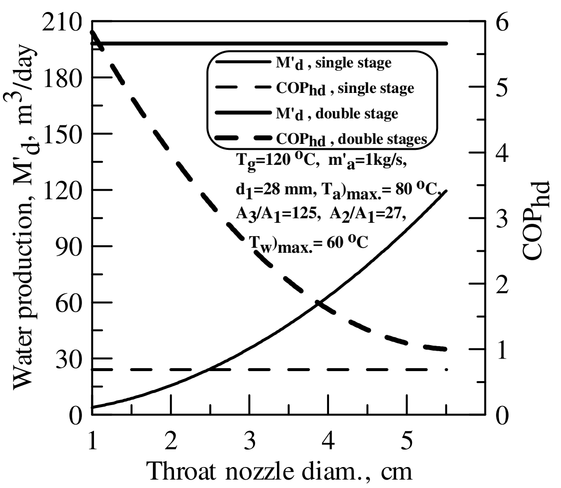

Fig. 7 Characteristics of the two units at variation of throat nozzle diameter

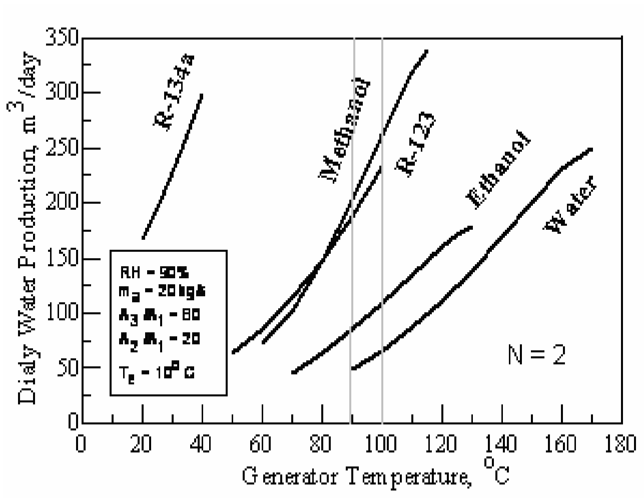

Fig. 8 Effect of various working fluids on MSHDH performance for two-stage unit

4.4 Case study

This study is practically applied at a real climatic condition for petroleum site (29°E – L 31°N) lies 120 km west Alexandria City in Egypt. The small gas turbine engine generates a nominal power 3.74 kW (manufacturer specifications), and if we want to enhance the economics and technical performance of the unit by retrofitting this single purpose unit with the proposed tri-generation unit by recovering the waste heat. The connected water production MSHDH and MED units is assumed to have 6-stages and 14-effect. The recorded climatic conditions for this region, data of desalination units and exhaust gases condition are shown in Table 2.

Table 2 Input data

	Climatic conditions		Desalination units		Exhaust gases
	Ambient temp.	Relative humidity	Generator temp. MSHDH	TBT for MED	Reject temp.
Summer	32° C	60%	160° C	110°C	77°C
Winter	12 ° C	80%	130° C	95°C	57°C

The performance of the modified plant (poly-generation) for this site is summarized in Fig. 9 for both summer and winter seasons. As seen the values of generated power, water produced heat recovered are close and the changes are small. But, based on the economic considerations, summer season results is better than that winter season due to the low heat input by fuel and the specific energy consumption in seawater desalination plants.

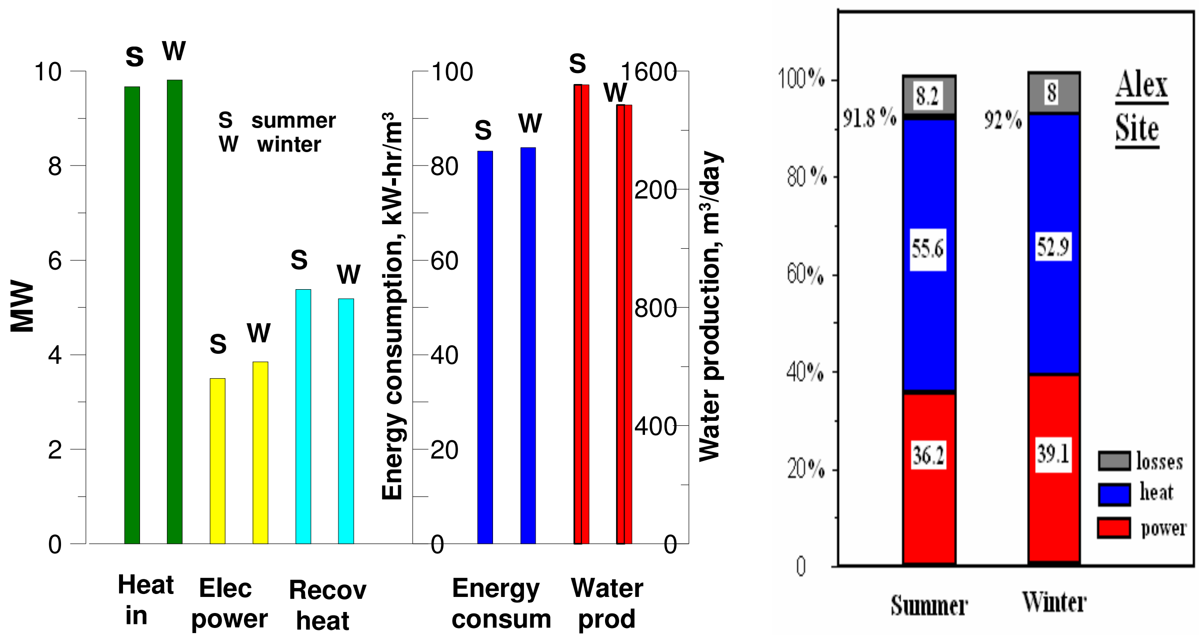

Fig. 9 Results of tri-generation on Alexandria site (petroleum)

5. CONCLUSION

In this paper the tri-generation system analysis for gas-turbine powering multi-stage humidification-dehumidification MSHDH and MED units was presented. The new poly-generation plant is proposed for retrofitting the existing small gas turbine to regenerate various products. The following conclusions and significance are:

Tri-generation (power, water and cooling/heating) is the most way to reduce fuel consumption or increase fuel savings. Where, the recovered heat (which was rejected as waste to atmosphere) is reached about 58% of input fuel energy that equal 1.7 of the generated power (PHR = 0.558).

The discussion of the simulation results showed the great dependence of the system operation on the seasonal condition, where power, water and cooling/heating wholly vary with ambient temperature.

The economical importance of the tri-generation concept resulted an enhancement of the plant overall efficiency to 72-76%.

When MED unit is added to the system to assure sufficient water production throughout the day and best fuel exploitation, the amount of water produced increased by 38-215%, and increased simultaneously the overall efficiency up to 93%.

The proposed multi-stage humidification-dehumidification MSHDH process showed stable operation than the single stage process because its performance is independent on variation of the generator temperature and throat nozzle diameter of the jet-ejector.

The results indicated hints for the choice of the working fluid (refrigerant) for the thermal vapor compression unit and the appropriate operational temperature ranges.

There is still an important lack of field data on the real economic cost of small capacity tri-generation systems.

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