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SFC Optimization of Gas Turbine Cycle Using Air Pre-Cooling Unit: Performance Improvement

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ABSTRACT
A critical issue concerning the gas turbine cycle of combined cycle power station is the reduction in net power output considerably with increasing ambient air temperature. The present simulation study aims to improve the performance of gas turbine cycle of combined cycle power station through use the fresh air handling units of mechanical chiller. To obtains the influence of use the mechanical chiller as pre-cooling units on a performance of gas turbine cycle, the thermodynamic analysis are used to simulate the performance of gas turbine cycle for three different operating loads (66.67, 83.33, and 100% full load). Based on results of the simulation analysis, the mechanical chiller which is inserted as a pre-cooling unit can; (i) increase the average monthly net power output by values varying between 7.33-18.17 MW compared to the original case without pre-cooling units; (ii) the rate of improvement in the average monthly net power output varies between 4.6-11% compared to the case without pre-cooling units; (iii) reduce the specific fuel consumption (SFC) by a rate varying between 5.21-10.81 kg_{fuel}/MWh compared to the original case without pre-cooling units; and (iv) the saving in SFC reached to 4.4% compared to the original case without pre-cooling units.

Keywords: *Combined cycle, Gas turbine cycle, Mechanical chiller, Performance improvement, SFC optimization.*

1. Introduction

As a result of increasing the demand for electricity, whether for domestic or industrial use, research is being carried out continuously on higher-efficiency combined power plant that utilizes natural gas fuel [1-5]. And since the gas turbine represents a main component of the combined power stations, this requires continuous development to enhance their performance. Whereas, the recent developments on gas turbines have lead to an improvement of their efficiency in excess over 40%, as well as for combined cycle when using fuel with low heating value, which leads to an increase in efficiency over 60% [6-11].

The gas turbines have some defects during operation. The most important of these disadvantages is that they produce less net power output during hot ambient because their performance largely depends on a temperature of ambient air surrounding the gas turbine units. In order to increase the net power output during the hot weather, it's necessary to reduce a temperature of the air flowing to gas turbines. Use the air pre-cooling technologies are the most cost effective for improving the gas turbine performance [12-15]. The air pre-cooling technologies used in gas turbine may be absorption chillier, mechanical chillier, evaporative cooler, inlet fogging, etc. The evaporative cooler consists from air blower and water distribution system connected to water pump. The evaporative cooler is characterized by very low capital cost, but is defective in low cooling effect as compared to other cooling technologies [16]. The mechanical chillier is characterized by having a coefficient of performance higher than 6 [17]. Therefore, it has a higher cooling effect, but it has the drawback that it needs higher electrical energy to drive it. Likewise, for absorption chillier, it is characterized by the fact that it does not consume electricity, except for operation of pumps that need low electrical power, which can be neglected. In addition, the absorption chillier characterized by higher coefficient of performance such as absorption chillers with double-effect those have coefficient of performance reach to about 1.5 [18]. Absorption chillier integrated with combined cycle power plants to cool the inlet air was conducted by [18].

Ameri and Hejazi [19] examined the influences of absorption chiller use to cool the inlet air on a behavior of 16.60 MW Chabahar gas turbine plant. They found that the use of absorption chiller improves the net power by 11.3%. Also, the economical studied presented that a payback time is 4.2 years. Kakaras et al. [20] used a model to study the influences of absorption chiller on a reduction in the inlet air temperature of gas turbine. Mohammad and Mohamed [21] used the desiccant cooling technology to cool the air inlet to gas turbine system. Some studies were conducted to calculate the coolant flow rate [22-24]. Farzaneh-Gord and Deymi-Dashtebayaz [25] compared between three different cooling techniques (turbo-expanders, mechanical chiller, and evaporative media system) used to cool the air inlet to gas turbines. Moon et al. [26] examined the influence of the turbine inlet temperature on a performance of gas turbines. Kwon et al. [27] used the absorption chiller to pre-cool air inlet to gas turbine of combined cycle. They found that use the absorption chiller to pre-cool inlet air to gas turbine improve a net power by 8.2%. Mohapatra and Sanjay [28] used two cooling technologies (vapor compression cooling system and evaporative cooler) to pre-cool the air inlet to gas turbine. They conducted that the improvement in plant specific work reached to 18.4% and 10.48% for using vapor compression cooling system and evaporative cooling system, respectively.

The present simulation study aims to enhance a performance of gas turbine cycle of Talkha combined cycle power station. To inveterate this idea, we proposed a mechanical chiller installed on the air path inlet to the compressor to pre-cool the air before entering to gas turbine system. The pre-cooling unit utilized to cool the air consists of fresh air handling units connected to mechanical chiller. The thermodynamic analysis of gas turbine cycle of Talkha combined power station integrated with mechanical chiller to pre-cooling the air inlet to gas turbine cycles was investigated. Also, in this study the effect of mechanical chiller as a pre-cooling unit on a degree of coolant cooling, rate of air mass flow inlet to compressor, net power, specific fuel consumption, and thermal efficiency was investigated.

2. System descriptions

The present study aims to enhance the performance of the gas turbine cycle. For a gas turbine operating cycle, a net output power is the difference between turbine power and compressor power. Therefore, the net output power of the gas turbine system depends to a large extent on the compressor input power. Therefore, the power consumed by the compressor is proportional directly with air inlet temperature. This means that, if a compressor inlet temperature is decreased and density of air inlet to compressor is increased, it will, in turn, affect the gas turbine cycle performance. The data in this study was obtained from the operational located at Talkha Combined Cycle power station. The parameters utilized for this work was generated from a log sheet during the period of January to December 2018. Fig. 1 shows the Talkha Combined cycle power station used in this study. In this study, the proposed cooling system (mechanical vapor compression chiller) was applied to cool the air inlet to the compressor of the turbine cycle as shown in Fig. 1.

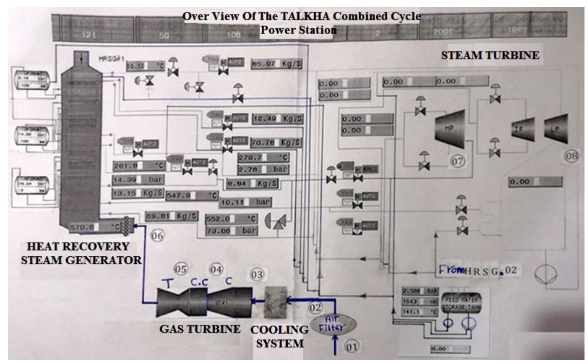


Figure 1- Layout diagram of Talkha Combined cycle power station

The simulation study was conducted with a pre-cooler installed on air path inlet to the compressor to pre-cool the air. The pre-cooler utilized to cool the air consists of air handling units connected to a mechanical chiller. The number of fresh air handling units used in this study was calculated based on the size of fresh air handling unit and compressor air flow rate at full load. Fig. 2 shows air handling unit that used in the present simulation study. Specifications of the gas turbine cycle and the fresh air handling unit used in this simulation study are presented in Table. 1. The fresh air handling unit is usually a large metal box containing the blower; filter racks, cooling coils; dampers, and sound attenuators.

Figure 2- Fresh air handling unit

Table 1- Specifications of gas turbine cycle of Talkha combined power station and the fresh air handling unit

Gas turbine cycle of Talkha Combined power station			
Components	**Parameters**	**Units**	**Values From Log sheet**
Compressor	Inlet Temperature, T_{o3}	K	Variable Values
	Outlet Temperature, T_{o4}	K	Variable Values
	Inlet Pressure, P_{o3}	bar	0.9998
	Outlet Pressure, P_{o4}	bar	17.3
	Air Mass flow rate, $\dot{m}_{air}$	kg/s	667.3
	Air volume flow rate	m^3/s	577.62
	Isentropic Efficiency	%	85
Combustion Chamber	Inlet Temperature, T_{o4}	K	Variable Values
	Max.Temperature,T_{o5}	K	1600
	Inlet Pressure, P_{o4}	bar	17.3
	Outlet Pressure, P_{o5}	bar	17.3
	Mass flow rate of fuel, $\dot{m}_{fuel}$	kg/s	13.9
Turbine	Inlet Temperature, T_{o5}	K	1503
	Outlet Temperature, T_{o6}	K	852.8
	Inlet Pressure, P_{o5}	bar	17.3
	Outlet Pressure, P_{o6}	bar	1.013
	Mass flow rate, $\dot{m}_{gas}$	kg/s	681.2
	Isentropic Efficiency	%	87
Exhaust	Gases temperature, T_{o6}	K	852.8
	Gases pressure,P_{o6}	bar	1.013
	Mass flow rate, $\dot{m}_{gas}$	kg/s	681.2
Fresh air handling unit of mechanical chiller			
Evaporator Blower	Nominal air flow rate	CFM (L/S)	(4721) 10000
	Minimum air flow rate	CFM (L/S)	(3776.8) 8000
	Maximum air flow rate	(CFM (L/S	(5665.2) 12000
	Motor power	kW	7.5
Evaporator Coil	Tube dia.	Inch	3/8
	Number of rows	-	6
	Face area,	Sq. ft. (Sq. m.)	28.7(2.66)
	Total cooling capacity <u>for each unit</u>	kW	98.67
	Sensible cooling capacity <u>for each unit</u>	kW	74.16
Outdoor unit	Compressor power <u>for each unit</u>	kW	25.64

$$\text{Number of fresh air handling unit} = \frac{\text{Compressor air flow rate at \%100 full load}}{\text{air flow rate for each fresh air handling unit}} = \frac{577.62\ m^3 s^{-1}}{5665.2\ \times 10^{-3}\ m^3 s^{-1}} = 101.959\ \equiv 102\ \text{units}$$

In the present simulation study, two options have been taken in designing the present configurations. In the first option, the gas turbine cycle of Talkha combined cycle power station working without pre-cooling units. In the second option, we installed the 102 fresh air handling units on the air path inlet to the compressor to pre-cool air before inlet to compressor of gas cycle of Talkha combined cycle power station. For two options the gas turbine cycle operated under three different operating capacities (in the first operating case the plant operated at 66.67% full load, in the second operating case the plant operated at 83.33% full load, and in the third operating case the plant operated at 100% full load). All fresh air handling units were connected in parallel. The outlet cold air from the fresh air handling unit represents the supply inlet air to the compressor.

3. Calculation procedure

3.1. For pre-cooling units

Procedure of calculating the temperature of air inlet to compressor

The procedure used to calculate the inlet temperature of cooled air to the compressor includes two-step. The first step by given the ambient air data (ambient air temperature T_{o1} and relative humidity RH_{o1}) recorded in the operational log sheet within a period of January-December 2018 in Talkha combined Cycle power station. From a psychometrics chart shown, in Fig. 3 by given T_{o1} and RH_{o1} we calculate h_{o1}. In the second step by given TC, SC, and SHF given in equations (1:3) we obtain the temperature of cold air inlet to the compressor T_{o3}.

Total cooling capacity, TC of pre-cooling units could be calculated as follows:

$$TC = \dot{m}_{air}\left(h_{o1} - h_{o3}\right) \quad (1)$$

Sensible cooling capacity, SC of pre-cooling units calculated as follows:

$$SC = \dot{m}_{air}\, C_{p,air}\left(T_{o1} - T_{o3}\right) \quad (2)$$

The sensible heat factor, SHF calculated as follows:

$$SHF = \frac{Sensible\ Cooling\ Capacity, SC}{Total\ Cooling\ Capacity, TC} \quad (3)$$

For gas turbine cycle .3.2

After obtaining the temperature of cold air inlet to compressor in the above section and by given the specifications of the gas cycle of Talkha combined power station presented in Table 1 and the equations (4-12) we calculated the net power and thermal efficiency as presented as follows:

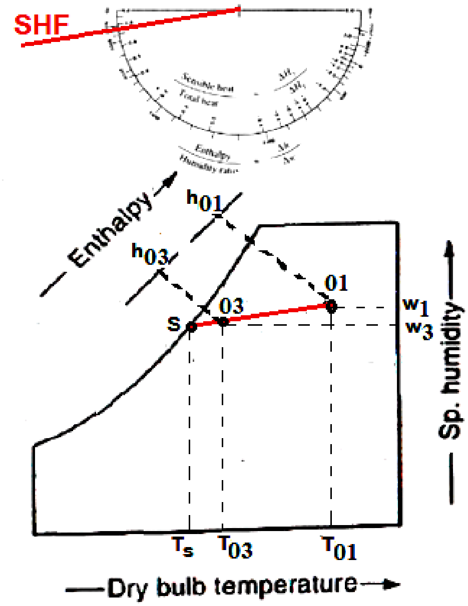


Figure 3 - Psychometrics chart

Air compressor

The actual discharge temperature ($T_{o4,a}$) calculated as:

$$T_{o4,a} = \frac{T_{o3}}{\eta_c}\left[\left(\frac{P_{o4}}{P_{o3}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_{o3} \quad (4)$$

The compressor power $\dot{W}_c$ calculated as follows:

$$\dot{W}_c = \dot{m}_{air}\, C_{p,air}\left(T_{4o,a} - T_{3o}\right) \quad (5)$$

Combustion chamber

Heat delivered by a combustor chamber is calculated from heat balance in it:

$$\dot{Q}_{in} = \eta_{comb} \times \dot{m}_{fuel} \times LHV = \left(\dot{m}_{air} + \dot{m}_{fuel}\right) C_{p,gas}\, T_{5o} - \dot{m}_{air}\, C_{p,air}\, T_{4o,a} \quad (6)$$

Mass flow rate of natural gas $\dot{m}_{fuel}$ is defined as:

$$\dot{m}_{fuel} = \frac{\dot{Q}_{in}}{\eta_{comb}\ LHV} \quad (7)$$

Turbine

The actual discharge temperature of gas leaving a turbine ($T_{06,a}$) can be written as:

$$T_{o6,a} = T_{o5}\left\{\eta_t \times T_{o5}\left[1-\left(\frac{1}{\left(P_{o5}/P_{o6}\right)}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\} \quad (8)$$

The power produced from the turbine $\dot{W}_t$ calculated as follows:

$$\dot{W}_t = \dot{m}_{gas}\, C_{p,gas}\,\left(T_{5o} - T_{6o,a}\right) \quad (9)$$

Where;

$$\dot{m}_{gas} = \dot{m}_{air} + \dot{m}_{fuel}$$

Net power produced from gas turbine cycle $\dot{W}_{net}$ calculated as follows:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c - \dot{W}_{illerhc} - \dot{W}_{AHUfans} \quad (10)$$

Specific fuel consumption SFC is calculated as:

$$SFC = \frac{3600 \times \dot{m}_{fuel}}{\dot{W}_{net}} \quad (11)$$

The plant efficiency of the gas turbine system is calculated as:

$$\eta_{plant} = \frac{\dot{W}_{net}}{\dot{m}_{fuel} \times LHV} \quad (12)$$

Percentage improvement in net power of gas cycle calculated as follows:

$$\text{Percentage improvement in net power output, \%} = \left[\frac{\left(\dot{W}_{net}\right)_{with\ cooling} - \left(\dot{W}_{net}\right)_{without\ cooling}}{\left(\dot{W}_{net}\right)_{withput\ cooling}}\right] \times 100$$

Percentage improvement in the plant efficiency of the gas turbine system calculated as follows:

$$\text{Percentage improvement in plant efficiency, \%} = \left[\frac{\left(\eta_{plant}\right)_{with\ cooling} - \left(\eta_{plant}\right)_{without\ cooling}}{\left(\eta_{plant}\right)_{withput\ cooling}}\right] \times 100$$

4. Results and discussions

Fig. 4 shows the average monthly compressor air inlet temperature of gas cycle with/without pre-cooling units for operating capacity equal to 66.67%, 83.33%, and 100% full load. For the operating capacity equal to 66.67% full load as shown in Fig. 4a, average monthly compressor inlet air temperature of gas turbine cycle varying between 292-305 K and 283-295 K for the case without and with cooling units respectively. The degree of coolant cooling of air inlet to compressor for using the pre-cooling units various between 6-12 K at operating capacity 66.67% full load. For increasing the operating capacity to 83.33% full load, the average monthly compressor inlet air temperature of gas cycle varying between 292-305 K and 283-294 K for the case without and with cooling units respectively as shown in Fig. 4b. Degree of coolant cooling of air inlet to compressor for using the pre-cooling units various between 6-11 K at operating capacity 83.33% full load. Also, with continues increases the operating capacity to 100 % full load as shown in Fig. 4c, the average monthly compressor inlet air temperature of gas turbine cycle varying between 292-305 K and 284-295 K for the case without and with cooling units respectively as shown in Fig. 4b. The degree of coolant cooling of air inlet to compressor for using the pre-cooling units various between 6-10 K at operating capacity 100% full load. As shown in Fig. 4, the results indicated that, the degree of coolant cooling reached to 12 K for using the pre-cooling units. This reduction in the compressor inlet temperature will increase the rate of air mass flow inlet to the compressor at the same volume flow rate. This will be increases a net power output and thermal efficiency, also, decrease specific full consumptions.

Figs. 5 shows the variations of average monthly compressor inlet air mass flow rate at operating capacity equal to 66.67%, 83.33%, and 100% full load. Fig. 5a shows the average monthly compressor inlet air mass flow rate varying between 426.27-455.61 kg/s and 451.84-473.85 kg/s for gas cycle without/with pre-cooling units respectively at operating capacity equal to 66.67% full loads. The improvement in air mass flow rate inlet to compressor varies between 2.75-6.83% for using gas cycle with pre-cooling units. With increasing the operating capacity to 83.33% full loads as shown in Fig. 5b, the average monthly compressor inlet air mass flow rate varying between 537.97-569.52 kg/s and 564.52-592.13 kg/s for gas cycle without/with pre-cooling units respectively. The improvement in the air mass flow rate inlet to compressor various between 2.75-6.25% for using gas cycle with pre-cooling units at the operating capacity to 83.33% full loads. But with increases the operating capacity to 100% full loads as shown in Fig. 5c, the average monthly compressor inlet air mass flow rate was varying between 645.56-683.42 kg/s and 671.07-709.29 kg/s for gas cycle without/with pre-cooling units respectively. The improvement in the air mass flow rate inlet to compressor various between 2.75-5.74% for using gas cycle with pre-cooling units at the operating capacity to 100% full loads. This result presented that the improvement in air mass flow rate inlet to compressor for using the gas turbine cycle with pre-cooling units reached to 6.25%.

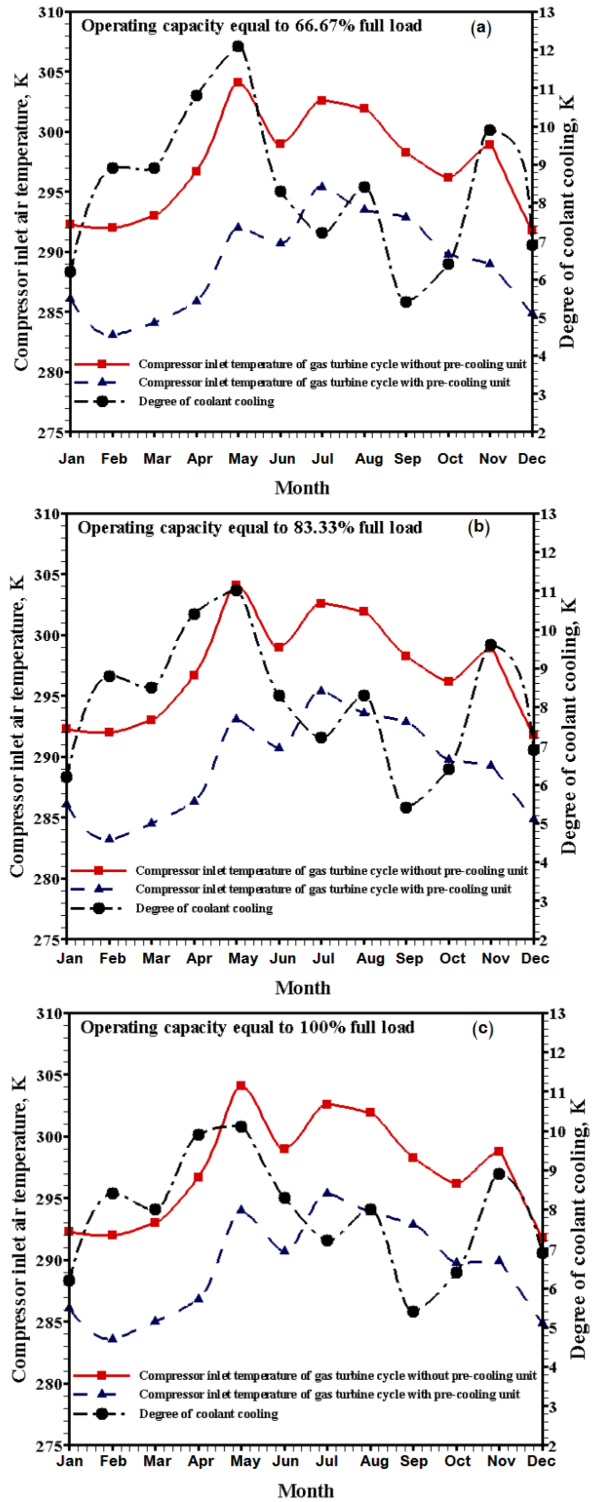


Figure 4 - Average monthly compressor inlet air temperature at operating load; (a) 66.67%, (b) 83.33%, and (c) 100% of full load

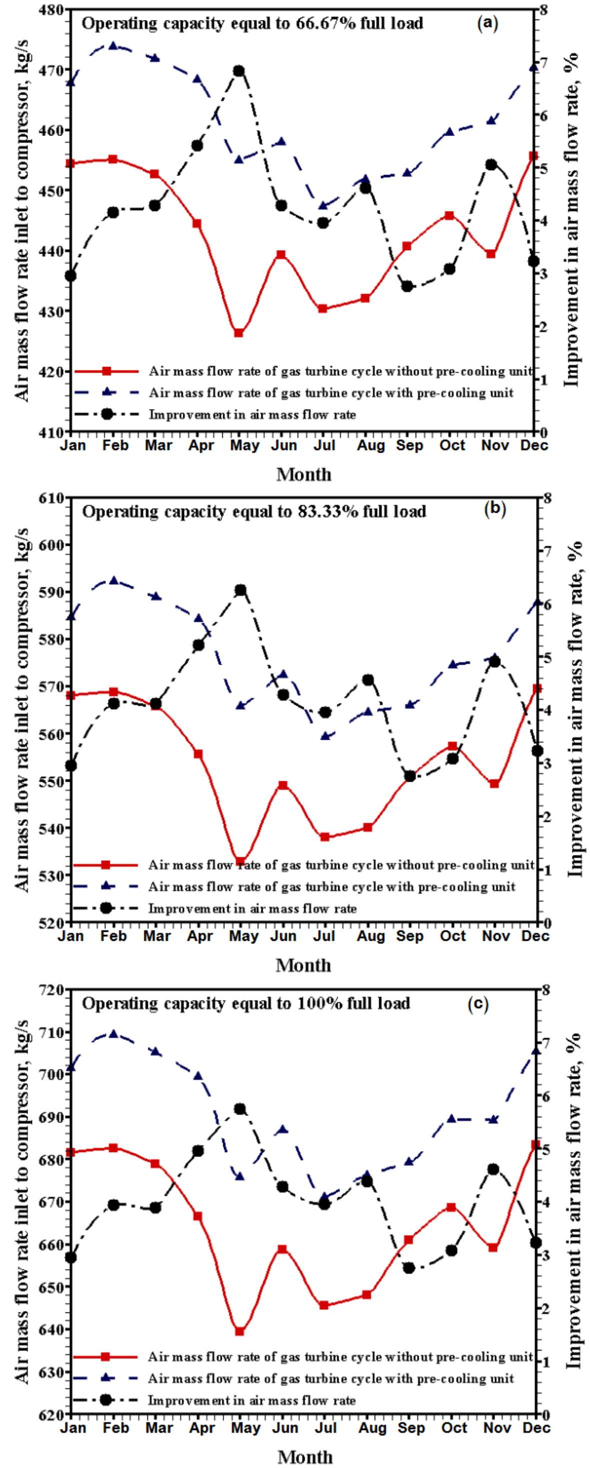


Figure 5 -Variation of average monthly compressor inlet air mass flow rate at operating load; (a) 66.67%, (b) 83.33%, and (c) 100% of full load.

The average monthly net power from the gas cycle at operating capacity equal to 66.67%, 83.33%, and 100% full loads are shown in Figs. 6. As shown in Fig. 6a, average monthly net power output varying between 131.4 -148.72 MW and 142.2-158.9 MW for the gas turbine cycle without/with pre-cooling units respectively at operating capacity equal to 66.67% full loads. At this operating case the improvement in net power output various between 4.6-11% for using the gas turbine cycle with pre-cooling units. With increase the operating capacity to 83.33% full loads as shown in Fig. 6b, the average monthly net power output varying between 164.25-185.9 MW and 177.8-198.5 MW for the gas turbine cycle without/with pre-cooling units respectively. At this operating case the improvement in net power output various between 4.6-10% for using the gas turbine cycle with pre-cooling units. But for increase the operating capacity to 100% full loads as shown in Fig. 6c, the average monthly net power output varying between 197.1-223.1 MW and 213.3-237.4 MW for the gas turbine cycle without/with pre-cooling units respectively. At this operating case the improvement in net power output various between 4.6-9.2% for using the gas turbine cycle with pre-cooling units. The results of average monthly net power presented that the improvement in net power reached to 13.19% for using the gas turbine cycle with pre-cooling units as compared to case without pre-cooling units.

The average monthly plant efficiency of gas cycle at operating capacity 66.67%, 83.33%, and 100% full loads are presented in Figs. 7. Fig. 7a shows the average monthly plant efficiency varying between 35.9-36.55% and 36.1-36.74% for the gas turbine cycle operated without/with pre-cooling units respectively at operating capacity 66.67% full loads. At this operating case the improvement in plant efficiency various between 0.5-0.55 % for using the gas turbine cycle with pre-cooling units. With increase the operating capacity to 83.33% full loads as shown in Fig. 7b, the average monthly plant efficiency varying between 35.9-36.55% and 36.11-36.74% for the gas turbine cycle without/with pre-cooling units respectively. At this operating case the improvement in plant efficiency various between 0.49-0.57% for using the gas turbine cycle with pre-cooling units. But for increase the operating capacity to 100% full loads as shown in Fig. 7c, the average monthly plant efficiency varying between 35.9-36.55% and 36.1-36.74% for the gas turbine cycle without/with pre-cooling units respectively. At this operating case the improvement in plant efficiency various between 0.49-0.56% for using gas cycle with pre-cooling units.

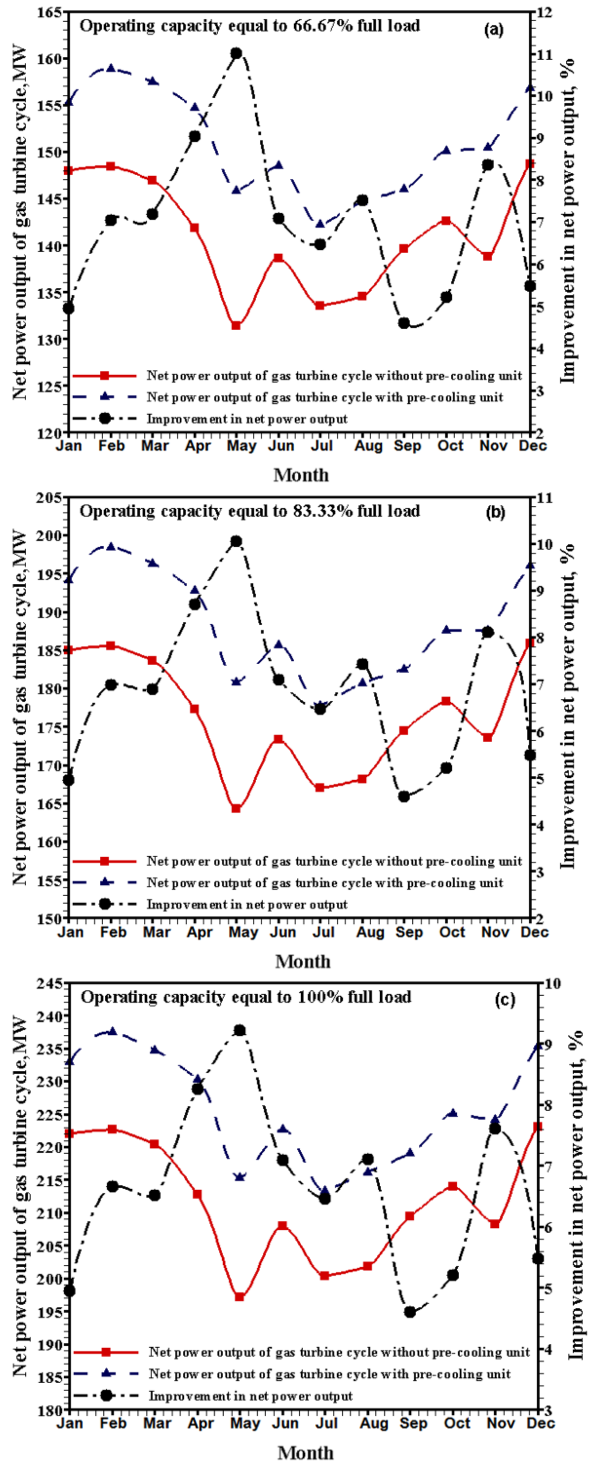


Figure 6- Average monthly net power output from the gas turbine cycle at operating load; (a) 66.67%, (b) 83.33%, and (c) 100% of full load

The average monthly specific fuel consumption SFC of gas cycle at operating capacity 66.67%, 83.33%, and 100% full loads are presented in Figs. 8.

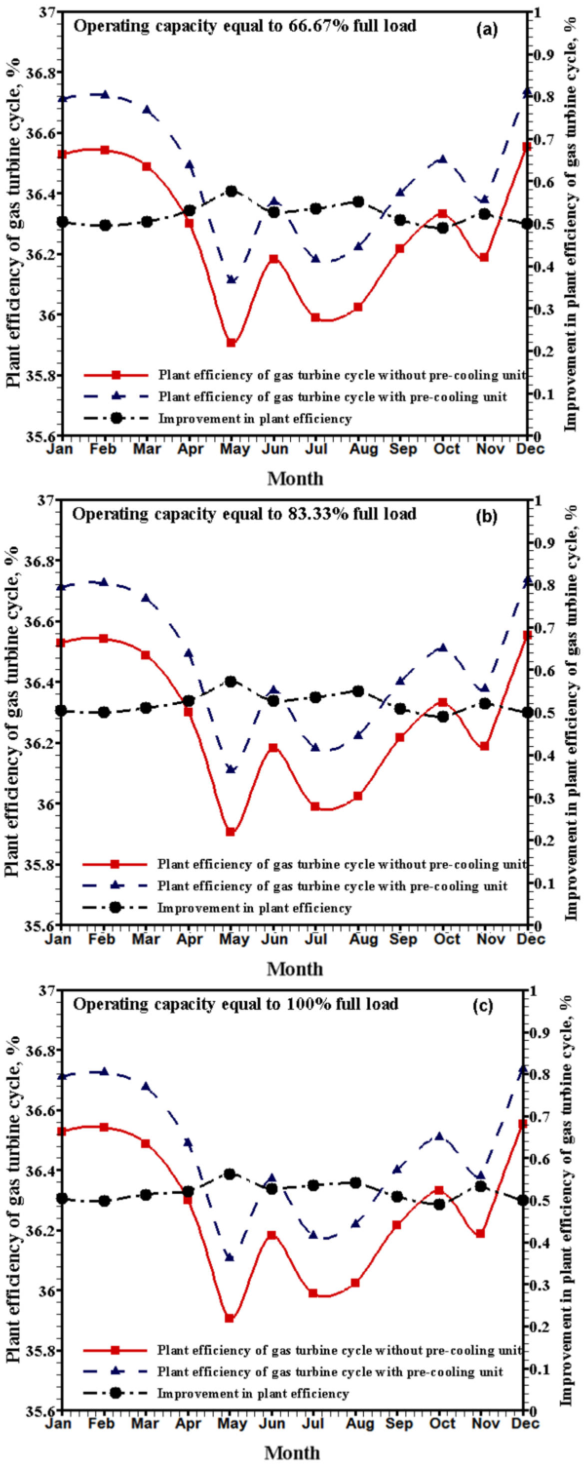


Figure 7- Average monthly plant efficiency of gas cycle at operating load; (a) 66.67%, (b) 83.33%, and (c) 100% of full load

Fig. 8a shows the average monthly SFC varying between 230-243.59 kg_{fuel}/MWh and 222.59-234.88 kg_{fuel}/MWh for the gas turbine cycle operated without/with pre-cooling units respectively at operating capacity 66.67% full loads. At this operating case, the gas turbine cycle with pre-cooling units reduces the SFC by 2.2-4.4% as compared with the case without pre-cooling units. With increase the operating capacity to 83.33% full loads as shown in Fig. 8b, the average monthly SFC varying between 229.75-243.59 kg_{fuel}/MWh and 222.63-234.88 kg_{fuel}/MWh for the gas turbine cycle without/with pre-cooling units respectively. At this operating case, use the gas turbine cycle with pre-cooling units reduces the SFC by 2.2-4.1% as compared to the case without pre-cooling units. But for increase the operating capacity to 100% full loads as shown in Fig. 8c, the average monthly SFC varying between 229.75-243.59 kg_{fuel}/MWh and 222.91-234.33 kg_{fuel}/MWh for the gas turbine cycle without/with pre-cooling units respectively. At this operating case, use the gas turbine cycle with pre-cooling units reduces the SFC by 2.2-3.8% as compared to the case without pre-cooling units. The results presented that, using the gas turbine cycle with pre-cooling units are more effective which the percentage reduction in the SFC reached to 5.51% for using gas turbine cycle with pre-cooling units as compared to the gas turbine cycle without pre-cooling units.

5. Conclusions

Based on the thermodynamic analysis of gas cycle of Talkha combined power station integrated with mechanical chiller to pre-cooling the air inlet to gas cycles, the main conclusions are summarized as follows:

- The improvement in the rate of air mass flow inlet to compressor for using the mechanical chiller as a pre-cooling unit reached to 6.83, 6.25, and 5.74% at operating capacity equal to 66.67%, 83.33%, and 100% full load, respectively.
- The increases in average monthly net power output for using the mechanical chiller as a pre-cooling units varying between 7.33-14.47, 8-16.5, and 9.63-18.17 MW at operating capacity equal to 66.67%, 83.33%, and 100% full load, respectively.
- The improvement in average monthly net power output for using the mechanical chiller as a pre-cooling unit reached to 11, 10, and 9.2% at operating capacity equal to 66.67%, 83.33%, and 100% full load, respectively.
- Improvement in plant efficiency of gas turbine cycles various between 0.49-0.55 % for using the gas turbine cycle with pre-cooling units.
- The saving in specific fuel consumption in gas turbine cycle for using the mechanical chiller as a pre-cooling units varying between 5.21-10.81,

5.21-9.99, and 5.21-9.26 kg_{fuel} /MWh at operating capacity equal to 66.67%, 83.33%, and 100% full load, respectively.

- The improvement in specific fuel consumption for using the mechanical chiller as a pre-cooling unit reached to 4.4, 4.1, and 3.8% at operating capacity equal to 66.67%, 83.33%, and 100% full load, respectively.

Nomenclature

Symbols

$_{p}C$	Heat capacity at constant pressure, kJ/kg K
h	Specific enthalpy, kJ/kg
LHV	Lower heating value, kJ/kg
$\dot{m}$	Mass flow rate, kg/s
P	Pressure, bar
SC	Sensible cooling capacity, kW
SFC	Specific fuel consumption, kg_{fuel} /MWh
T	Temperature, K
TC	Total cooling capacity, kW
$\dot{W}$	Power, kW

Greek Symbol
s

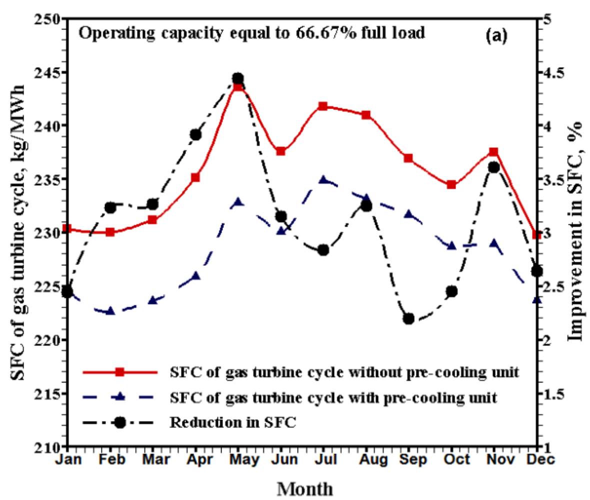
η	% ،Efficiency
γ	Specific heat ratio

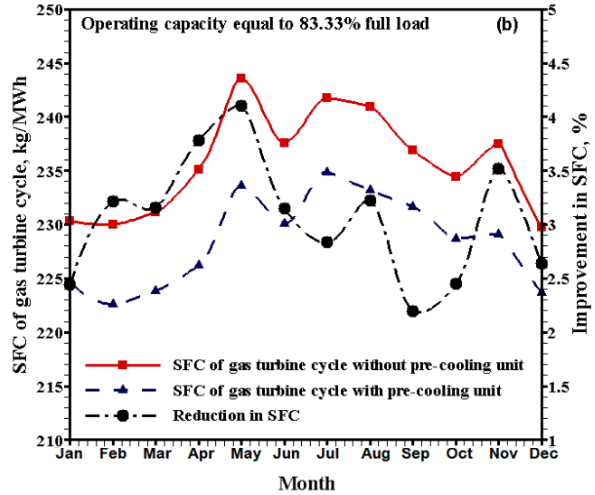
Abbreviation

SHF	Sensible heat factor
AHU	Air handling unit

Subscript

a	actual
c	compressor
in	inlet
t	urbineT





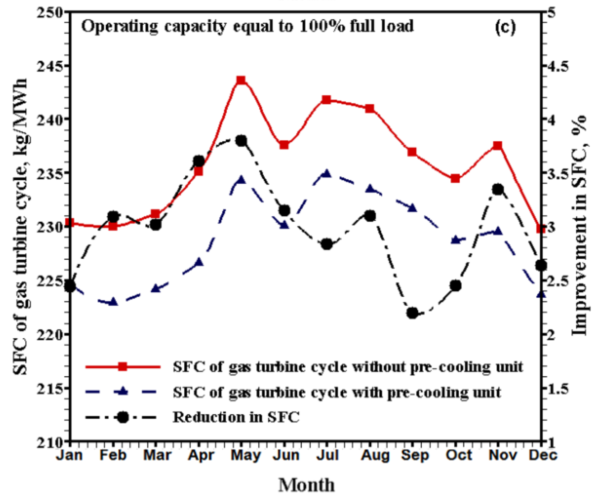


Figure 8- Average monthly SFC of the gas cycle at operating load; (a) 66.67%, (b) 83.33%, and (c) 100% of full load

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