# Investigation of Water Balance in Water Recovery of Closed Steam Injection Gas Turbine Cycles

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**Abstract:** An industrial gas turbine is able to decrease output power on hot summer days while electricity demand peaks. Steam injection in gas turbines has been used for many years to prevent the loss of performance of gas turbines caused by high ambient temperature. Vodoley system is a Closed Steam Injection Gas Turbine Cycle (CSIGTC), which is known as a self-sufficient in steam production. In this study the influence of steam injection in Mashhad Power Plant GE-F5 gas turbine parameters was experimentally investigated. Moreover a computer modelling has been used to study water balance in a CSIGTC system. The results show that although the system has additional water treatment process, its application increases thermal efficiency up to 35.33 percents for 25 MW of power output at an ambient temperature of 40 ˚C. Also water balance considerations show for a direct contact condenser of 96% efficiency and the exhaust temperature of 420 ˚C at turbine outlet, about 7.8 kg/s of make-up water is needed under ISO rating of gas turbine (101.3 kPa, 15 ˚C) which must be provided for the CSIGTC system. This amount of water is about 51% of injected steam.

**Keywords:** CSIGTC System, Gas turbine, Steam injection

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## 1 INTRODUCTION

The first gas turbine in production for electrical power generation was introduced by Brown Boveri of Switzerland in 1937. It was a standby unit with a thermal efficiency of 17% [1]. Global installed gas-fired generation capacity is forecast to rise from 1,311 GW in 2010 to 2,008 GW in 2030 [2]. Many additional cycle configurations have been suggested to improve the performance of the simple gas turbine cycles. In Combined Heat and Power "CHP" system, patented by R. MacKay [3], a heat engine or a power station is used to simultaneously generate both electricity and useful heat. The CHP captures the product heat of the simple gas turbines high temperature exhaust gases for domestic or industrial heating purposes [4], [5].

In gas turbine with Air Bottoming Cycle "ABC" the exhaust flow of an existing, topping gas turbine in a gas-air heat exchanger is used to heat the air in the secondary gas turbine cycle [6]. In a Combined Cycle Power Plant "CCPP" a gas turbine generator generates electricity, and heat in the exhaust is used to make steam, which in turn drives an extra steam turbine to generate additional electricity. Thermal efficiency of the CCPP can reach a range of 50–58 % using more advanced gas turbines [7]. Electrical efficiencies and specific power outputs, and Humid Air Water Injected Turbine cycle "HAWIT" were also investigated by Traverso and Massardo, aiming to increase the thermal efficiency of the gas turbine cycle and decrease the irreversibilities [8], [9].

Despite all its unique advantages over other turbines, gas turbine output is a strong function of ambient air temperature and this is mostly due to the effects of ambient temperature on density or mass flow of the air intake to the compressor [7]. Widespread attempts have been made so far to prevent or reduce this drop of power and efficiency. First category of these methods incorporates cooling of air in the inlet or outlet of the compressor using evaporative coolers, chillers or any other cooling equipment. Ashraf M. Bassily [10], adopting a parametric approach, showed that inlet air cooling and cooling of the compressor discharge using water injection boost both efficiency and power of gas turbine cycles.

M. De Lucia et al. described the technical and economic advantages of providing a compressor inlet air cooling system to increase the gas turbine's power rating [11]. M. De Lucia et al. compared absorption and evaporative compressed air cooling systems performance and economic benefits with the dry low-NOx LM6000 gas turbine [12]. Different options have been represented (including evaporative cooling, mechanical chiller, absorption chiller and thermal energy storage) in gas turbine power augmentation using inlet air cooling [13].

Second category of the methods to prevent or reduce the drop of power and efficiency at higher inlet air temperature includes steam injection techniques. The performance characteristics of two types of regenerative steam-injection gas-turbine "RSTIG" systems have been analyzed and compared with the performances of the simple, regenerative, water injection and steam injected gas-turbine "STIG" cycles. The thermal efficiencies of the "RSTIG" systems are higher than those of the regenerative, water injection and "STIG" systems and the specific power is larger than that of the regenerative cycle [14]. Steam injection leads to temperature drop in combustion gases and allows using additional fuel to air ratio up to specific Turbine Inlet Temperature "TIT".

Mircea Cârdu et al. gave the results of the thermodynamic analysis on some installations with total water injection in the combustion chamber [15]. In the total water injection the heat tempering of the combustion gases in the combustion chamber is performed with water injection and the air quantity compressed by the compressor will be the one strictly needed for the fuel combustion. They showed that these kinds of gas turbines enjoy higher thermal efficiency, more reduced temperature for combustion chamber, and lower air excess for proper combustion than partial or without steam injection. F. J. Wang et al. implemented both steam injection gas turbine and inlet air cooling features by developing a computer code to simulate a Taipower_s Frame 7B simple cycle GENSET [16]. Under the condition of local summer weather, the benefits obtained from the system were more than 70% boost in power and 20.4% improvement in heat rate.

It should be mentioned that the expansion of steam inside the gas turbine to atmospheric pressure is less efficient than inside a steam turbine, where the steam leaves the turbine at much lower pressures and, thus, provides more power and higher efficiency. Furthermore, the design of modern heat recovery steam generators includes two or more pressure levels and reheaters, which allows advanced recovery of the flue gas thermal energy [7], [17]. A mixed air and steam turbine technology has a lower efficiency than a combined cycle. However, for small scale power generation (less than 50 MW), it is more cost effective to install a less complex power plant due to the adverse effect of the economics of the scale [7]. This makes steam injection a good modification to small scale simple cycle gas turbines. A major practical disadvantage of Cheng cycle, especially for regions with lack of water resources, is the large water consumption. The amount of water injected is released from the exhaust to the ambient and would be wasted. To overcome this shortage, Closed Steam Injection Cycle "CSIGTC", which is known as a self-sufficient cycle in steam production, can be used. In the CSIGTC, the required amount of water for steam injection is recovered by a condenser which is planted in the path of exhaust gases.

Water species normally creates in a combustion chamber of a gas turbine. Many attempts, accordingly, have been made to recover the initial steam injected from the exhaust gases and it is shown that an approximate 100% recovery of the injected steam is possible [18]. However, the efficiency of water recovery is practically lower than 100% for some difficulties. This failure to recover all of the required water for steam injection from the exhaust, results in necessary additional calculations to investigate the water balance in the whole system.

In this study, the experimental data of Mashhad Power Plant GE-F5 simple gas turbine cycle shows 30% efficiency drop with a 40 °C increase in temperature. In present work the influence of steam injection in Mashhad Power Plant GE-F5 gas turbine parameters (applying CSIGTC system due to poor water resources in Mashhad) is being observed by developing a computer code, which is based on experimental data of various components of the simple cycle gas turbine GE-F5 in different temperatures of ambient air. The procedure is given in Flowchart 1 in the appendix. Also the water balance of the system is discussed and the results show although a CSIGTC system has been used, but with a condenser of 96% efficiency, still some make-up water is necessary for the system.

## 2 COMPUTER MODEL

### 2.1. System description

A simple GE-F5 gas turbine is considered as a base unit in this study. The maximum net output power of this unit under ISO condition (101.3 kPa, 15 °C) is 25 MW. Several major performance specifications of the simple GE-F5 gas turbine are presented in Table 1. The exhaust temperature of the base gas turbine is in a range of 257-455 °C. This means there is a plenty of useful energy expelled to the atmosphere especially in higher net output powers. This amount of energy can be recovered to improve the overall performance of the simple gas turbine cycle.

In order to recover the exhaust wasted energy, a CSIGTC is employed to recapture most of the exhaust wasted energy to generate suitable steam for CSIGTC system.

A schematic diagram of the CSIGTC is shown in Fig. 1. The simple GE-F5 gas turbine unit is shown in the closed dashed line (a). Exhaust gases of the gas turbine pass through a heat recovery boiler in which the provided water converts into the required steam. The exhaust gases are then discharged into a direct contact condenser. In the direct contact condenser the condensation process occurs by injecting cold water (to reduce the amount injected water) into the gas flue. The condensed water is filtered in a cleaning block. The cleaned water is collected in a reservoir tank.

Extra purification for cleaned water can be done in this stage if needed (neither cleaning block nor the extra purification system has an effect on thermal efficiency in this paper). The water is then separated into two parts. One part is used as feed water for the steam injection. The other part is cooled down in a cooling system by being exposed to the air and then is used as the cooling water in the condenser. Two pumps are used to recycle the water from the reservoir tank into the condenser and the heat recovery boiler. The make-up water provided for the CSIGTC can be added to the cycle from reservoir tank. Table 2 represents process variables of different points of Fig. 1.

**Table 1** Experimental data of the simple GE-F5 gas turbine cycle

| Power, MW | 5 | 10 | 15 | 19 |
|---|---|---|---|---|
| $T_1$, °C | 4 | 4 | 4 | 4 |
| $P_1$, atm | 0.9007 | 0.9001 | 0.8995 | 0.8990 |
| $P_2$, atm | 7.49 | 7.91 | 8.41 | 8.71 |
| $T_2$, °C | 296.83 | 306.71 | 317.83 | 324.39 |
| $\dot{m}_f$, kg/s | 0.6564 | 0.9026 | 1.1761 | 1.3895 |
| $T_4$, °C | 258 | 326 | 394 | 455 |
| $P_{amb}$, atm | 0.91 | 0.91 | 0.91 | 0.91 |

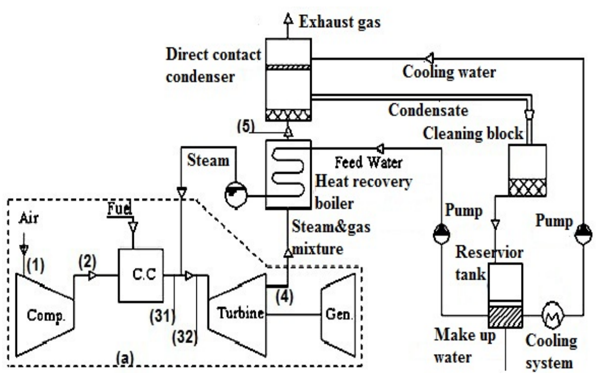



**Fig. 1** Schematic diagram of the closed steam injection gas turbine in this study

**Table 2** Process variables of different points of Fig. 1

| Variable | Statement |
|---|---|
| $T_1$, $P_1$ | T & P for state (1) |
| $T_2$, $P_2$ | T & P for state (2) |
| $m^{\circ}_{air}$, $m^{\circ}_{fuel}$ | $\dot m$ for air and fuel |
| $T_{steam}$, $m^{\circ}_{steam}$ | T & $\dot m$ for steam |
| $T_{31}$, $T_{32}$, $P_3$ | T & P for states (31),(32) |
| $T_4$, $P_4$ | T & P for state (4) |
| $W_{net}$, $W_{comp}$, $W_{turb}$, $W_{pump}$ | In nomenclature |
| $\eta_{comp}$, $\eta_{turb}$, $\eta_{comb}$, $\eta_{cond}$ | In nomenclature |
| $T_{cw}$, $m^{\circ}_{c\,w}$ | T & $\dot m$ for cooling water |
| $T_5$, $P_5$ | T & P for state (5) |
| $T_{dp}$ | Dew point Temperature |
| $m^{\circ}_{Condensate}$ | Recaptured water |
| $m^{\circ}_{make\text{-}up\ water}$ | $\dot m$ for make-up water |

### 2.2. Assumptions for thermodynamics consideration:

1. Combustion chamber pressure loss is negligible.
2. Pressure drop of the exhaust in CSIGTC is assumed 5% due to back pressure of heat recovery boiler and condenser.
3. Cooling water temperature is assumed 20 °C.
4. The minimum air to fuel ratio of simple gas turbine is about 67. So the air thermodynamic tables have been used for simple gas turbine cycle analysis.
5. Back pressure of compressor increases from 4% to 20% for different power outputs due to steam injection system, according to the Mashhad power plant reports.
6. Flue gases leave the direct contact condenser at the dew point of the condenser temperature.

### 2.3. Simple cycle thermodynamics consideration:

In order to have more accurate results in applying the CSIGTC, some preliminary thermodynamics considerations in the simple gas turbine cycle are inevitable. In the following calculations, subscripts 1, 2, 3, and 4 indicate the positions shown in the schematic figure of the simple cycle gas turbine, i.e. Fig. 2. It should be mentioned that the mass flow rate of the inlet air and fuel "$m'_f$, and $m'_{air}$", low heat value of fuel "$Q_{LHV}$", fuel enthalpy "$h_f$", exhaust enthalpy "$h_4$", net power output "$W'_{net}$" and inlet enthalpy of the compressor "$h_1$" are known from experimental results presented in Table 1. Low heat value of Fuel is considered 49884 kJ/kg according to Mashhad natural gas chemical analysis. Considering energy balance for the control volume indicated by a dashed closed line in Fig. 2, the air mass flow rate of the simple gas turbine can be calculated as follows:

$$\dot m_f \eta_{comb.} Q_{LHV} + \dot m_f h_f + \dot m_{air} h_1 = \left(\dot m_{air} + \dot m_f\right)\left(h_4\right) + W_{net} \quad (1)$$

Which gives

$$\dot m_{air} = (\dot m_f (\eta_{comb.} Q_{LHV} + h_f - h_4) - W_{net}) / (h_4 - h_1) \quad (2)$$

Knowing $T_1$, $P_1$ and $P_2$, outlet enthalpy of the compressor is obtained using Eqs. (3) and (4).

$$(S^0_{T_{2s}} - S^0_{T_1}) - R Ln \frac{P_2}{P_1} = 0 \quad (3)$$

$$h_2 = (h_{2S} - h_1)/\eta_{Comp.} + h_1 \quad (4)$$

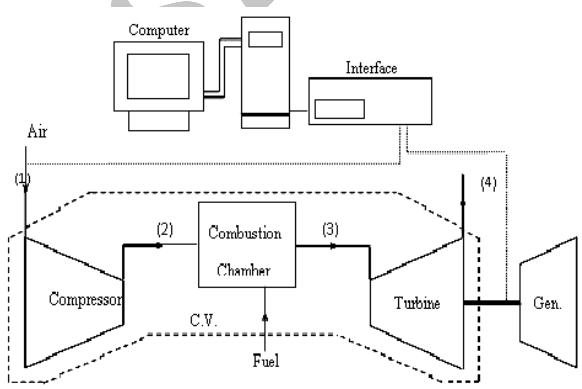



**Fig. 2** Schematic figure of the simple gas turbine cycle in this study

So outlet temperature of the compressor can be read from thermodynamic tables. The compressor and turbine works can be obtained as:

$$W_{comp.} = \dot m_{air} \left(h_2 - h_1\right) \quad (5)$$

$$W_{turb.} = W_{net} + W_{comp.} \quad (6)$$

The Turbine Inlet Temperature (TIT) can be obtained with considering the turbine as a control volume.

$$h_3 = \frac{\left(W_{turb.}\right)}{\left(\dot m_{air} + \dot m_f\right)} + h_4 \quad (7)$$

The turbine efficiency is calculated as:

$$P_3 \approx P_2 \quad (8)$$

$$(S_4^0 - S_3^0) - RLn\frac{P_4}{P_3} = 0 \tag{9}$$

$$\eta_{turb.} = \frac{h_3 - h_4}{h_3 - h_{4S}} \tag{10}$$

Thermal efficiency of the simple gas turbine cycle is obtained as follows:

$$\eta_{ther.} = W_{net} / \left(\dot{m}_f \, \eta_{comb.} Q_{LHV}\right) \tag{11}$$

And finally Back-Work "B.W." ratio is obtained using Eq. (12).

$$B.W. = \frac{W_{comp.}}{W_{turb.}} \tag{12}$$

### 2.4. The CSIGTC water balance consideration

In this study, the CSIGTC is applied for the GE-F5 gas turbine by developing a computer code in order to investigate the water balance in water recovery steam-injected gas turbines. The cycle considered here is shown in Fig. 3.

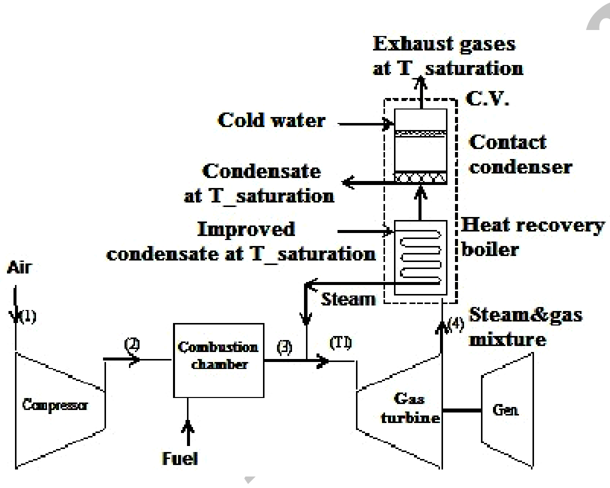



**Fig. 3** The applied CSIGTC for the GE-F5 gas turbine

Steam injection leads to temperature drop in combustion gases and allows using additional fuel to air ratio up to specific TIT. With increasing the mass flow rate of fuel in the CSIGTC compared to the simple cycle, the temperature of the combustion gases at point 3 in Fig. 3 is obtained as:

$$h_3 = \frac{\dot{m}_{air} h_2 + \dot{m}_f \left(h_f + \eta_{comb.} Q_{LHV}\right)}{\dot{m}_{air} + \dot{m}_f} \tag{13}$$

Respecting the temperature of the injected steam after the combustion chamber and considering a specific TIT, the required mass flow rate of steam injection is calculated as:

$$\dot{m}_{steam} = \frac{(\dot{m}_{air} + \dot{m}_f) \times (h_3 - h_{TITair})}{(h_{TITsteam} - h_{steam})} \tag{14}$$

$h_{TIT}$ can be written as:

$$h_{TIT} = \frac{((\dot{m}_{air} + \dot{m}_f) \times h_3 + \dot{m}_{steam} \times h_{steam})}{(\dot{m}_{air} + \dot{m}_f + \dot{m}_{steam})} \tag{15}$$

Knowing the isentropic efficiency of the turbine from the simple cycle, enthalpy of the exhaust gases can be obtained as follows:

$$S^0_{T_{4air}} = S^0_{T_{TITair}} + R \times Ln\left(\frac{P_4}{P_3}\right) \tag{16}$$

$$h_{4s} = \frac{((\dot{m}_{air} + \dot{m}_f) h_{4s\,air} + \dot{m}_{steam} h_{4s,steam})}{\dot{m}_{air} + \dot{m}_f + \dot{m}_{steam}} \tag{17}$$

$$h_4 = h_{TIT} - (h_{TIT} - h_{4s}) \times \eta_{turb.} \tag{18}$$

This calculated enthalpy must be compared with another enthalpy obtained from a relation mentioned bellow. So TIT can be corrected for a specific exhaust temperature.

$$h_4 = \frac{((\dot{m}_{air} + \dot{m}_f) h_{4air} + \dot{m}_{steam} h_{4steam})}{\dot{m}_{air} + \dot{m}_f + \dot{m}_{steam}} \tag{19}$$

Thermal efficiency of the gas turbine cycle with CSIGTC system is obtained as follows:

$$\eta_{ther.} = \frac{W_{net}}{\left(\dot{m}_f \, \eta_{comb.} Q_{LHV}\right) + W_{pump}} \tag{20}$$

A C.V. is considered including the contact condenser and the heat recovery boiler, closed dashed line in Fig. 3 to obtain the required amount of cold water injection in the contact condenser. In the condenser the temperature of the steam and gas mixture is reduced up to the saturation temperature by the cold water. Neglecting changes in temperature of the condensed water because of improving process, the feed water for steam in the heat recovery boiler can be considered at saturation temperature of the condenser. The cold water is considered 10 °C degrees below the ambient air temperature. Temperature of the steam and gas mixture at point 4 in Fig. 3 is known.

In addition to the injected steam after the combustion chamber, the amount of steam generated in the combustion chamber must be taken into account for the water balance study in the CSIGTC. The equation of combustion for methane can be written as:

$$aCH_4 + b(O_2 + 3.76N_2) \rightarrow n_{Co_2}Co_2 + n_{Co}Co + n_{H_2}H_2 + n_{O_2}O_2 + n_{H_2O}H_2O + n_{N_2}N_2 + n_{No}No \quad (21)$$

The balance of atoms for carbon, hydrogen, oxygen and nitrogen yields four equations with seven unknowns which include $n_{Co}$, $n_{Co2}$, $n_{H2}$, $n_{O2}$, $n_{H2o}$, $n_{N2}$ and $n_{No}$.

$$For \quad C: \quad a = n_{CO_2} + n_{CO} \quad (22)$$

$$For \quad H: \quad 4a = 2n_{H_2O} + 2n_{H_2} \quad (23)$$

$$For \quad N: \quad 7.52b = 2n_{N_2} + n_{NO} \quad (24)$$

$$For \quad O: \quad 2b = 2n_{CO_2} + n_{H_2O} + n_{CO} + 2n_{O_2} + n_{NO} \quad (25)$$

$a, b$ and the products temperature are known through CSIGTC system analysis. So three equilibrium equations are needed. Each equilibrium equation has an equilibrium constant, $K_f$, which is related with moles of its species. Values of these equilibrium constants have been tabulated for various temperatures. Here two equilibrium equations are used and the unknowns are to be obtained by a try-and-error procedure. The seventh equation which is used for correction of the guessed coefficient is to be the energy equation. These equations have been mentioned bellow:

$$\frac{1}{2}O_2 + \frac{1}{2}N_2 \rightleftharpoons NO, \quad K_f = \frac{n_{NO}}{(n_{N_2})^{0.5}(n_{O_2})^{0.5}} \quad (26)$$

$$CO_2 + H_2 \rightleftharpoons CO + H_2O, \quad K_f = \frac{n_{CO}.n_{H_2O}}{n_{CO_2}.n_{H_2}} \quad (27)$$

$$\sum_{P-R} n_i \{(H_T^o - H_{298K}^o) + \Delta H_{f,298K}^o\} = \dot{m}_f \eta_{comb}.Q_{LHV} \quad (28)$$

Where 'T' is the products temperature. First an estimation should be considered for mole of $H_2$. Using Eq. (23) coefficient of $H_2O$ in combustion equation is obtained. Solving Eq. (22) and Eq. (27) simultaneously, gives moles of $CO_2$ and CO. Finally, using remaining three Eqs. (24), (25), and (26) as a set, moles of O2, N2 and NO can be calculated. Equation (28) then is used to check the accuracy of the values and the first estimation would be corrected if needed.

Now that mole fraction of water produced in the combustion chamber is known, the additional mass flow rate of steam available in the combustion gases can be obtained. To obtain the mass flow rate of cold water injection in the condenser the following procedure can be used. First, the dew point temperature in the condenser is guessed. Then, considering the heat recovery boiler as a control volume, the outlet enthalpy of the condenser can be obtained using the first law of thermodynamic:

$$(\dot{m}_{air} + \dot{m}_f + \dot{m}_{steam})(h_4 - h_5) + \dot{m}_{steam}(h_{Tdp,f} - h_{steam}) = 0 \quad (29)$$

The mixture of exhaust gases and steam are considered to behave according to the ideal gas model, so Dalton model can be used [19]. For Dalton model of gas mixtures, the component properties of the mixture are considered as though each component exists separately and independently at the temperature and volume of the mixture [19]. Thus it may be written for the mixture of gases and steam which is leaving the condenser to obtain mole of $H_2O$ as below:

$$\frac{P_{H_2O}}{P_4} = \frac{n_{H_2O,ex}}{n_{emissions} + n_{H_2O,ex}} \quad (30)$$

Partial pressure of $H_2O$ is obtained from saturated water table using guessed temperature. Moles of all species except $H_2O$ have been represented as $n_{emissions}$ here. Now, output enthalpy of gas and steam mixture can be calculated

$$(\dot{m}_{emissions} + \dot{m}_{H_2O,ex})h_{output} = \dot{m}_{emissions} \times h_{Tdp,emissions} + \dot{m}_{H_2O,ex} \times h_{Tdp,g} \quad (31)$$

By definition of condenser efficiency, as the mass flow rate of recaptured water to mass flow rate of entering water and applying the first law of thermodynamic for the direct contact condenser, mass flow rates of cooling water and recaptured water can be obtained.

$$\eta_{cond.} = \frac{\dot{m}_{rec}}{\dot{m}_{cw} + \dot{m}_{H_2O}} \quad (32)$$

Which $\dot{m}_{H_2O}$ is the summation of mass flow rate of water produced in combustion and mass flow rate of steam injected after the combustion chamber.

$$\begin{aligned}(\dot{m}_{emissions} + \dot{m}_{H_2O})h_5 + \dot{m}_{cw} h_{cw} - \dot{m}_{rec} h_{Tdp,f} \\ + (\dot{m}_{H_2O} - \dot{m}_{H_2O,ex})h_{Tdp,fg} \\ - (\dot{m}_{emissions} + \dot{m}_{H_2O,ex})h_{output} = 0\end{aligned} \tag{33}$$

The fourth term of the above equation indicates the effect of water evaporation and condensation in the condenser. Since $\dot{m}_{H_2O,ex}$ should be less than $\dot{m}_{H_2O}$, in order to make the system feasible, the amount of water which is evaporated from the cooling water will be condensed. Mass conservation equation for water will be used to check the guessed temperature.

$$\dot{m}_{H_2O} + \dot{m}_{cw} - \dot{m}_{H_2O,ex} - \dot{m}_{rec} = 0 \tag{34}$$

### 2.5. Model validation

To validate the model, the results of the program are compared to those of the experiments performed by Ghazikhani et al. for the GE-F5 simple gas turbine [20]. Figures 4 and 5 show a comparison of model results with those of the experiments for the thermal efficiency and TIT against power output at an ambient temperature of 4 ˚C, where a good agreement is observed [20].

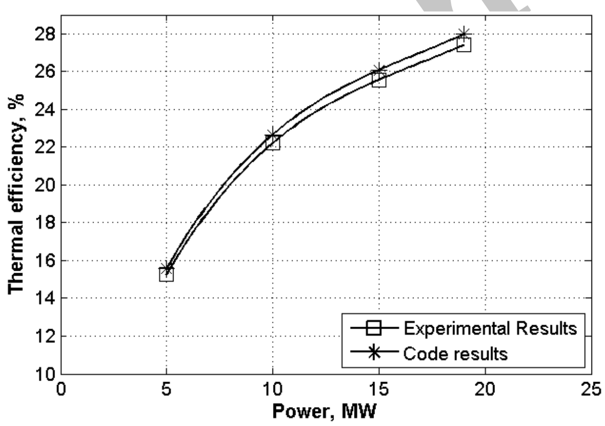



**Fig. 4** Comparison of the model and experimental results [20] of the GE-F5 simple gas turbine cycle

Figure 6 shows the results for the mole fraction of the exhaust gases in percent for 15 MW output power and 15 ˚C ambient temperature.

A SHKODA boiler in 350 °C temperature and 20 bar pressure had been used by power plant to produce the required steam for injection after the transition piece.

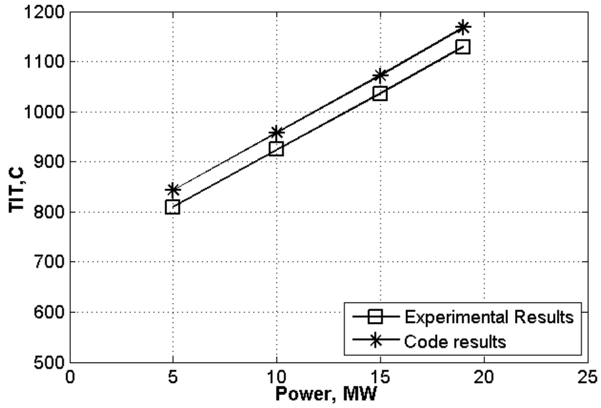
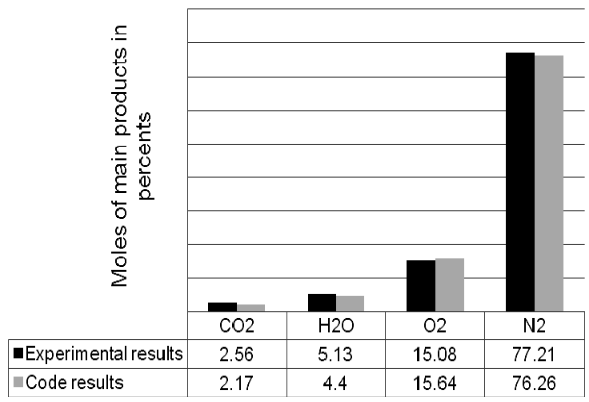



**Fig. 5** Comparison of the model and experimental results [20] of the GE-F5 simple gas turbine cycle




| | CO2 | H2O | O2 | N2 |
|---|---|---|---|---|
| Experimental results | 2.56 | 5.13 | 15.08 | 77.21 |
| Code results | 2.17 | 4.4 | 15.64 | 76.26 |

**Fig. 6** Comparison of the model and experimental results, reported by Mashhad power plant

## 3 RESULTS AND DISCUSSION

Figure 7 shows the magnitude of efficiencies of the compressor, turbine and combustion along with thermal efficiency of the simple gas turbine cycle. As it is shown in this figure, the combustion efficiency of the gas turbine is almost 98%. The efficiencies of the compressor and the turbine can be considered constant, 76% and 87%, respectively. The thermal efficiency of the simple gas turbine cycle is 15.09% in 5 MW and increases in higher net output powers. This may be due to higher TIT which is a result of higher mass flow rate of the fuel.

Table 3 shows the results for the mole fraction of the exhaust gases.

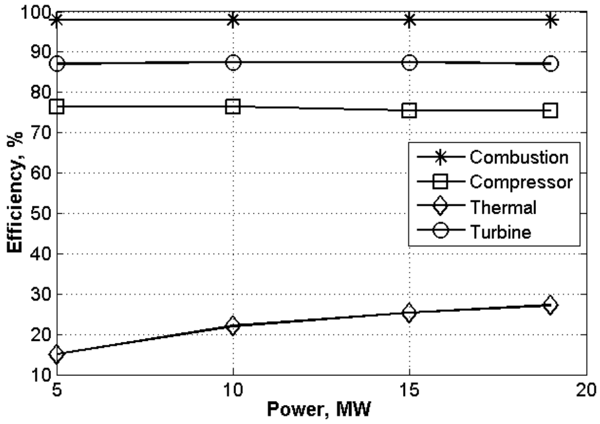



**Fig. 7** Combustion and thermal efficiency, isentropic efficiency of compressor and turbine of the simple gas turbine cycle

**Table 3** Results for the mole fraction of the exhaust gases (ISO condition and exhaust temperature of 420 ˚C)

| Power, MW | 10 | 15 | 19 | 25 |
|---|---|---|---|---|
| $n_{Co_2}$, % | 1.89 | 2.17 | 2.39 | 2.75 |
| $n_{Co}$, % | 0.56 | 0.53 | 0.53 | 0.48 |
| $n_{H_2}$, % | 1.19 | 0.98 | 0.87 | 0.69 |
| $n_{O_2}$, % | 16.3 | 15.64 | 15.10 | 14.30 |
| $n_{H_2O}$, % | 3.68 | 4.40 | 4.96 | 5.78 |
| $n_{N_2}$, % | 76.37 | 76.26 | 76.12 | 75.95 |
| $n_{No}$, % | 0.007 | 0.010 | 0.014 | 0.022 |

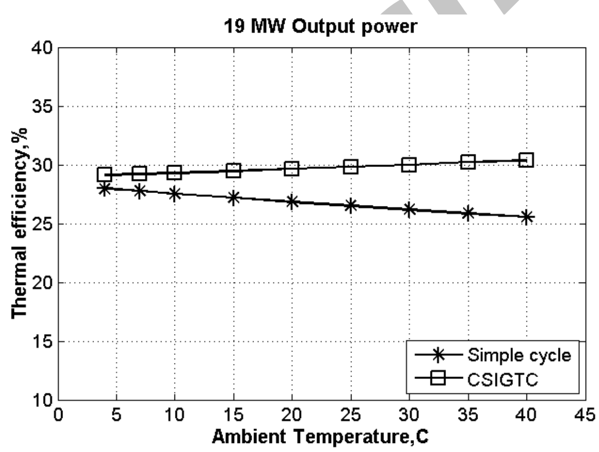



**Fig. 8** The variation of thermal efficiency against ambient temperature

Figure 8 represents the variation of thermal efficiency against ambient temperature at a constant power output. Figure 9 compares TIT for both simple gas turbine and closed steam injection cycles versus ambient temperature for a specific power output. In this model variation of ambient temperature affects on mass flow rates of fuel and injected steam as it is observed in Fig. 10. Reduction in power output because of decreasing of TIT can be compensated with injecting more steam.

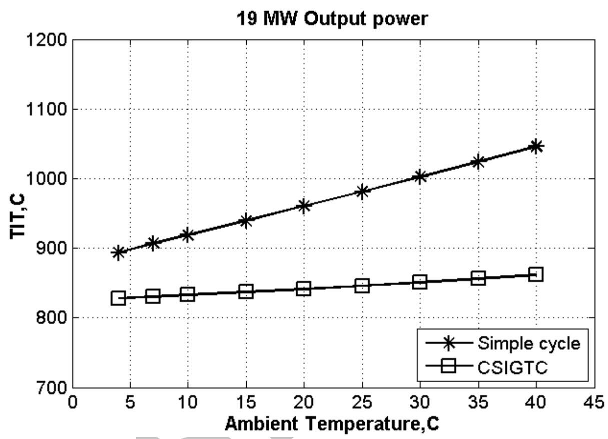



**Fig. 9** The variation of TIT against ambient temperature

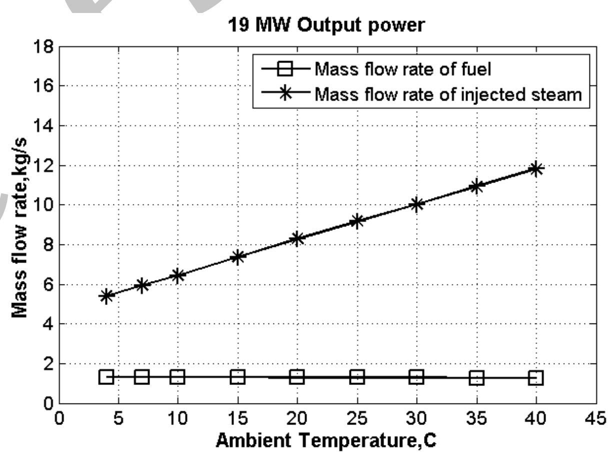



**Fig. 10** The variation of fuel and steam mass flow against ambient temperature

Figures 11 and 12 represent the variation of the compressor outlet pressure and temperature respectively against ambient temperature for both simple gas turbine and closed steam injection cycles. Exhaust temperature in CSIGTC system is 420 ˚C. The most important advantage of CSIGTC is the fact that the simple gas turbine cycle equipped with CSIGTC system can yield maximum net power output (25 MW) of course with a high thermal efficiency (with respect to the simple cycle) at even high ambient temperatures up to 40 ˚C. According to the reports of Mashhad power plant, this power output can be obtained at ambient temperatures less than 10 ˚C without using the CSIGTC system. Figure 13 displays the changes in thermal efficiency against ambient temperature in various exhaust temperatures in CSIGTC.

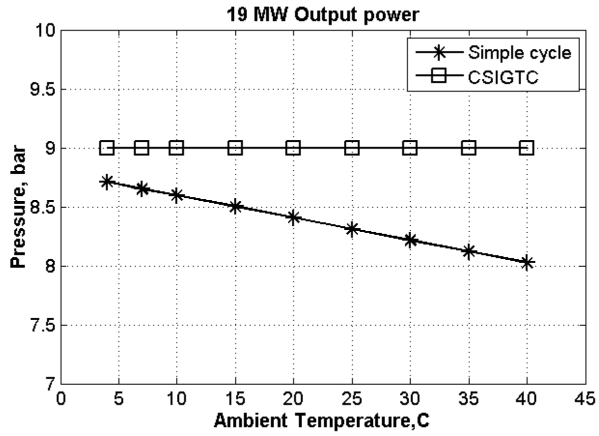



**Fig. 11** The variation of compressor discharge pressure against ambient temperature for a constant power output

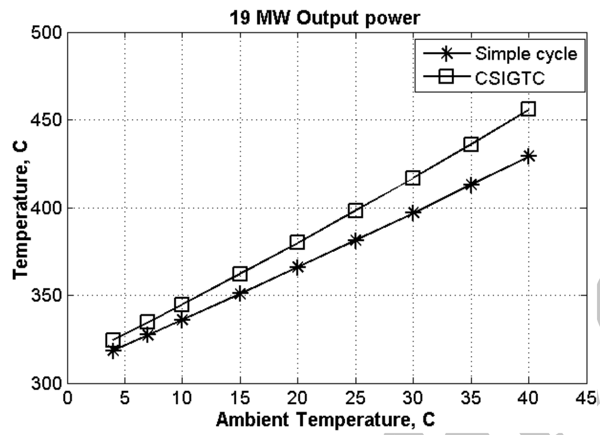



**Fig. 12** The variation of compressor outlet temperature against ambient temperature for a constant power output

In Fig. 14 the mass flow rate of water which is lost from the CSIGTC has been shown against ambient temperature for different networks output. This lack of water is due to the deviation of the direct contact condenser efficiency from 100% and must be provided for the CSIGTC. The Mass flow rate of water produced in combustion process also has been shown in Fig. 15 versus ambient temperature. These values of mass flow rates do not change extremely because fuel consumption is almost constant and the mass flow rate of steam injected will be increased with rise in ambient temperature as it is shown in Fig. 10.
It is understood from Figs. 14 and 15 that the amount of water needed to provide for the CSIGTC can be obtained by subtracting the Mass flow rates of water produced in combustion from Mass flow rates of water which are lost. Figure 16 shows these values of make-up water.

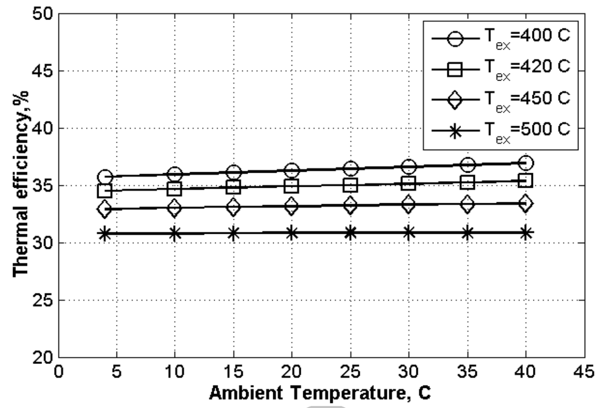



**Fig. 13** The variation of thermal efficiency against ambient temperature for CSIGTC ($W_{net}$=25 MW)

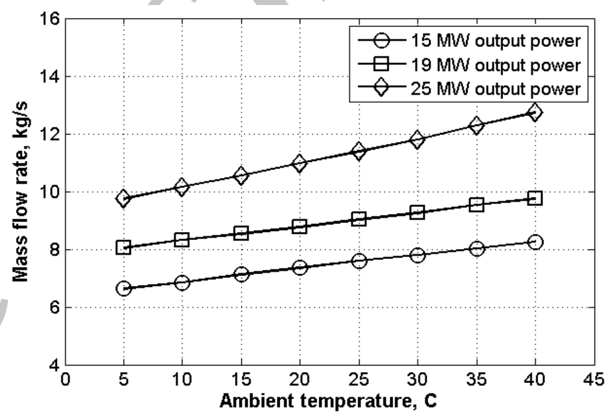



**Fig. 14** The variation of mass flow rate of water loses from CSIGTC against ambient temperature

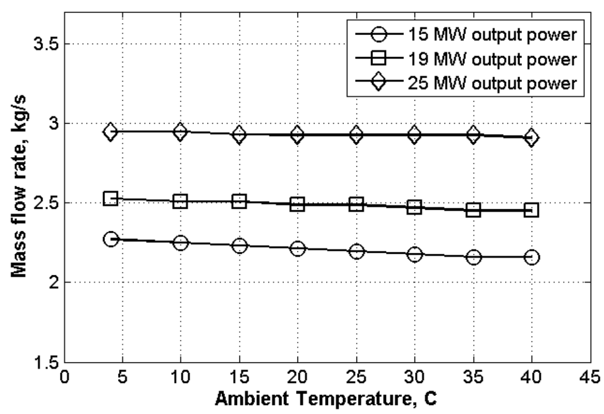



**Fig. 15** The variation of mass flow rate of water produced in combustion process against ambient temperature

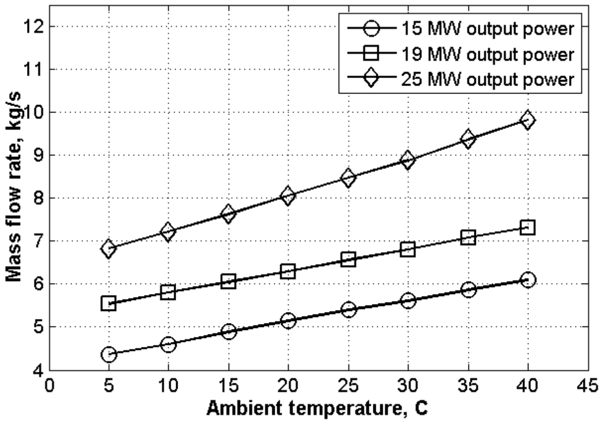



**Fig. 16** The variation of mass flow rate of make-up water against ambient temperature

In order to check the feasibility of using CSIGTC system, the mass flow rate of steam leaves the direct contact condenser should be less than the entering flow rate of steam. Figure 17 represents the mass flow rates of injected steam. Comparing Fig. 14 with Figs. 15 and 17 determines some operating condition that it is not economic to use CSIGTC.

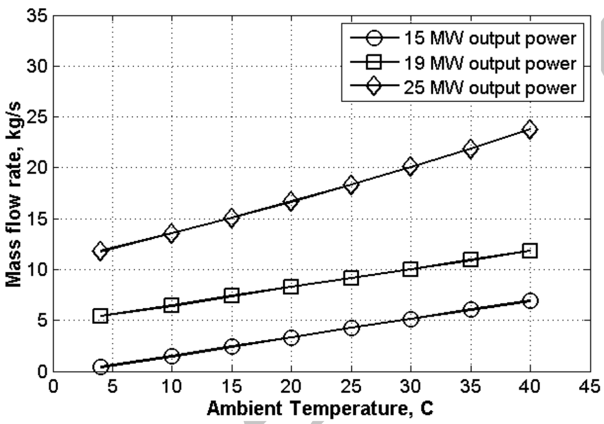



**Fig. 17** The variation of mass flow rate of injected steam against ambient temperature

Figures 18 and 19 represent the mass flow rates of cooling water and recaptured water in condenser respectively. The most important disadvantage of CSIGTC system is caused by injecting steam after the combustion chamber. This injection leads to an increment in pressure after the compressor. An axial thrust force is the consequence of this increment which can damage bearings. In present model, increasing the back work ratio is correlated with this disadvantage. Figure 20 represents B.W. ratio against ambient temperature for various output works in both simple cycle and CSIGTC.

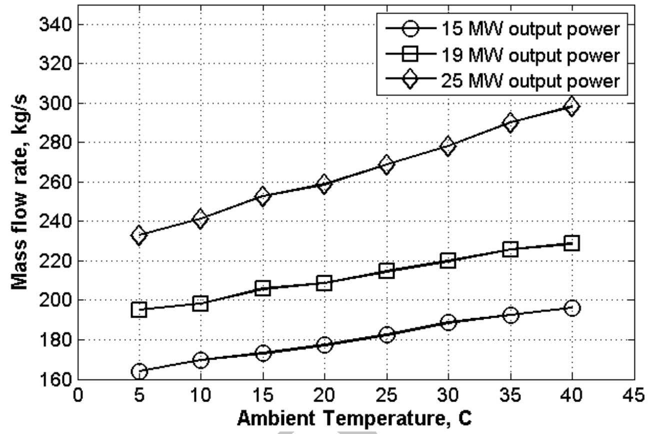



**Fig. 18** The variation of mass flow rate cooling water against ambient temperature

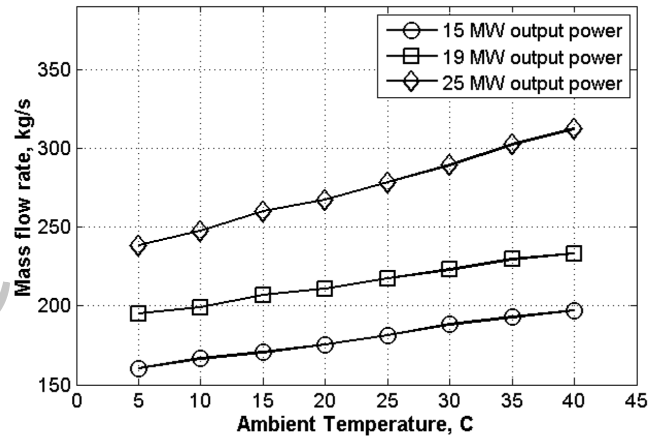



**Fig. 19** The variation of mass flow rate of recaptured water against ambient temperature

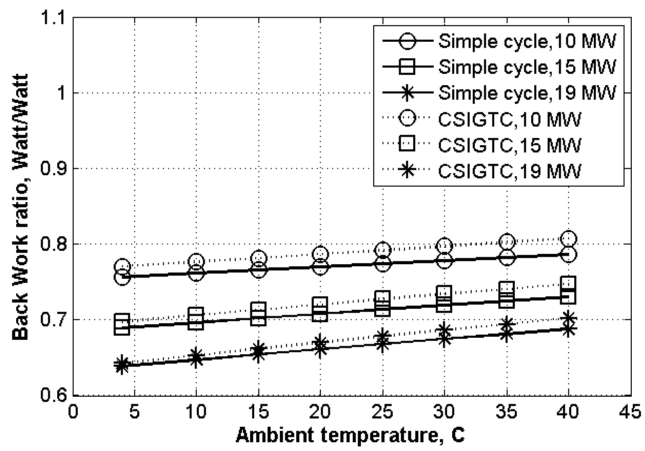



**Fig. 20** B.W. ratio against ambient temperature for various outputs works in both simple cycle and CSIGTC

## 4 CONCLUSION

1. Water balance considerations show for a direct contact condenser of 96% efficiency and the exhaust temperature of 420 °C at turbine outlet, 42% to 58% of injected steam is needed as make-up water which must be provided for the CSIGTC system for different ambient temperatures and 25 MW power output. Comparing the amount of water existing in the exhaust gas flow at inlet and outlet of the direct contact condenser specifies the feasibility of using CSIGTC. This system entails water treatment cost, but in comparison to simple GE-F5 power station applying it can increase thermal efficiency up to 35.33 percents for 25 MW of output power at an ambient temperature of 40 ˚C.
2. The water loss amounts as the percentage of circulated water for 15 MW, 19 MW, 25 MW of output powers are 3.99, 3.98, and 4.00 respectively.
3. Compared with GE-F5 simple cycle, GE-F5 cycle equipped with CSIGTC has less turbine inlet temperature, TIT, in constant power output and high power output even in high ambient temperature of course with an acceptable thermal efficiency.
4. In CSIGTC system compressor discharge pressure and temperature increase due to injection of steam after the combustion chamber. So the Back Work ratio in CSIGTC case is higher than simple gas turbine cycle. An axial thrust force is considered as the consequence of this increment.

## 5 NOMENCLATURE

| | |
|---|---|
| $\dot{m}$ | Mass flow rate |
| $h$ | Enthalpy |
| $W_{net}$ | Net work |
| $Q_{LHV}$ | Lower heating value of the fuel |
| $T$ | Temperature |
| $P$ | Pressure |
| $\eta_{Comp.}$ | Isentropic efficiency of compressor |
| $s^0$ | Standard entropy |
| $R$ | Gas constant |
| $W_{comp.}$ | Compressor work |
| $W_{turb.}$ | Turbine work |
| $TIT$ | Turbine inlet temperature |
| $\eta_{turb.}$ | Turbine efficiency |
| $\eta_{ther.}$ | Thermal efficiency |
| $K_f$ | Equilibrium constant |
| $\eta_{cond.}$ | Condenser efficiency |
| $B.W.$ | Back work |
| $\eta_{comb.}$ | Combustion efficiency |

## ACKNOWLEDGMENTS

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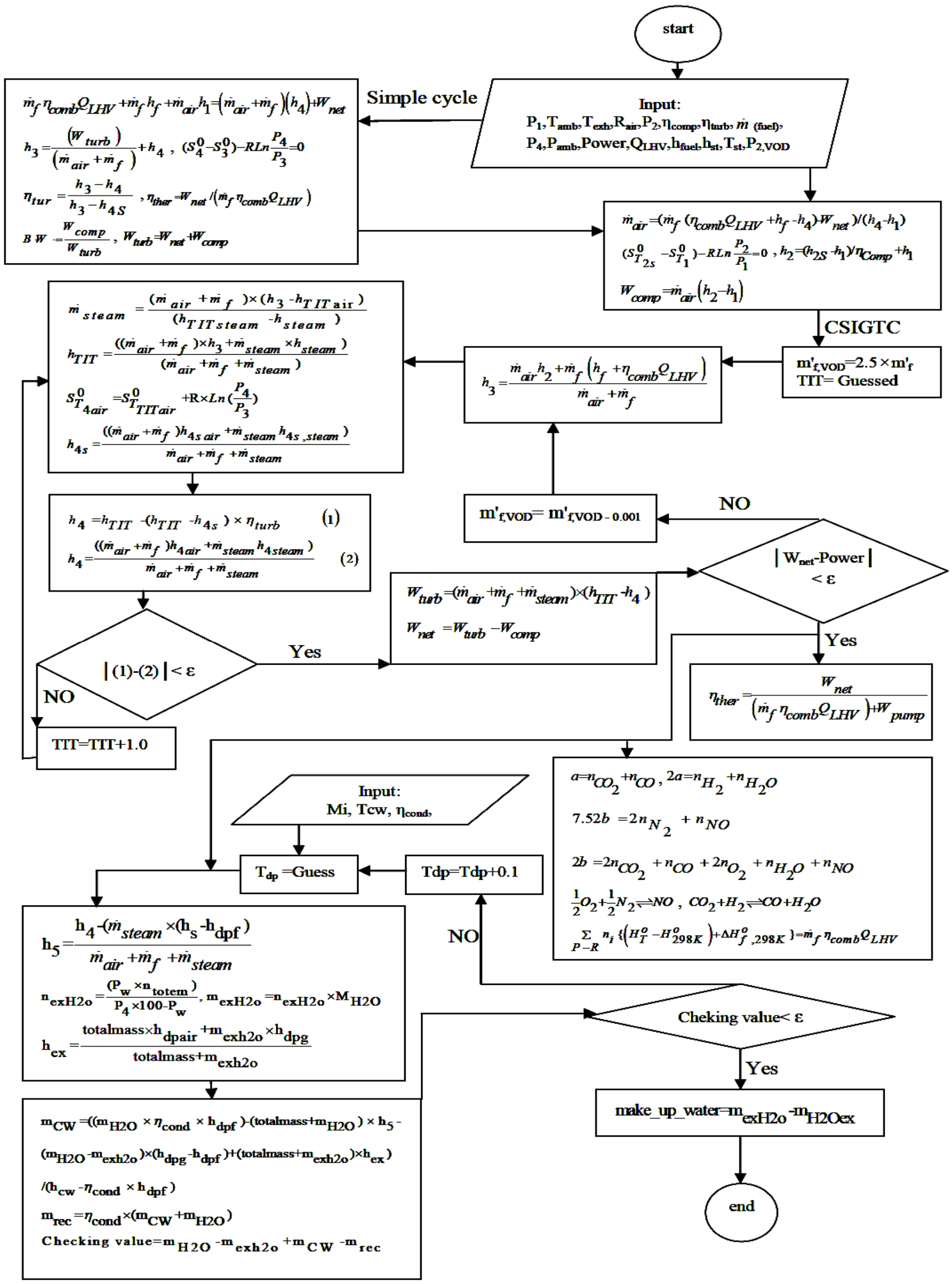
## APPENDIX




**Flow chart:** The procedure of finding unknown parameters in presented model for simple gas turbine cycle and CSIGTC