Energy and exergy comparison of three similar combined power cycles

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Abstract. Three similar combined power cycles were compared from energy and exergy aspect during one day. Enormous different data were collected every five minutes. In common exergy analysis, it is only possible to determine main sources of exergy destructions. These sources are mainly equipment in which severe irreversible process, i.e. combustion is going on. However, it is clear that nothing can be done to reduce these kinds of irreversibilities in an operating cycle. Here, it has been shown that in contrast to usual researches, the energy and exergy comparison of different power cycles lead to useful improvements in their performance. Moreover, since the ambient temperature was changed more than 10 °C, ambient temperature impacts on equipment were studied based on those collected data. The effect of evaporative coolers is the other objective, which was proceeded to. Since coolers only come into the service in the afternoon, those collected data were a valuable source to study the effect of adding moisture. Due to air high moisture content, it was not possible to use dry air assumption and more complicated equations were needed to simulate combustion process. Results show which equipment needs more attention during maintains which is useful in similar plants.

Keywords: Exergy destruction / combined cycle / thermodynamic optimization / entropy generation

1 Introduction

Electricity generation heavily relies on fossil fuels. Despite the undeniable improvements in renewable energy, the main contribution of energy consumption is devoted to fossil fuels. The oil demand growth is expected to be 47.5%, for natural gas is 91.6% and for coal is 94.7% [1]. On the other hand, the depletion of fossil fuels reserves and also environmental impacts of using fossil fuels such as climate change, necessitate the using fossil fuels more efficiently.

It is well-known that the 1st law analysis of a power plant suffers some inherent limitations. For example, it does not account for plant environment or quality degradation of energy. In this regard, exergy analysis is more superior.

The Exergy of a system at a certain thermodynamic state is the maximum amount of work that can be obtained when the system moves to a state of equilibrium with the surroundings [2] and Exergy analysis is based on the second law of thermodynamics.

Many researchers have studied gas or steam power plants [3–13]. Aljundi [7] asserted that the major exergy loss can be found in the boiler system and next to it, is the turbine. Regulagadda et al. [8] concluded that the maximum exergy destruction can be found to occur in the boiler. Hou et al. [9] found that the main units of exergy loss are the boiler and the steam turbine in the studied steam power plant. In the boiler, the exergy losses are mainly due to combustion process irreversibility and the heat transfer process. Ahmadi and Toghraie [10] realized the boiler as the main equipment destroying the exergy which is mainly due to the combustion process. They also showed that the steam turbine is the second equipment destroying the exergy.

More and less similar results may be found in researchers conclusions. Although exergy analysis for a combined power cycle is relatively new and less study may be found, the conclusions are approximately the same, i.e. that combustion chamber, duct burner and heat recovery steam generator are the main sources of irreversibility [11–15].

However, the fact is that nothing more can be done for heat transfer and combustion irreversibilities! It means that those valuable studies are useful only at the design step and it doesn’t seem reasonable to compare exergy destruction of the combustion chamber with the compressor.
This fact that exergy destruction of a combustion chamber or a steam generator is much more than a compressor or a pump, does not show that those pump or compressor operates at the expected performance.

So, unlike those studies, in the current research, it is intended to compare main parts of similar power plants with each other to recognize which parts of power plants operate out of the performance and which ones require maintenance.

For this reason, the Fars power station is selected for the study. This station has three similar combined power cycles. The performance of main components was compared with each other. Based on this comparison, the weakness of each component was recognized. It was shown that which one works better and which one needs maintenance. As results, some suggestions were advised to improve the efficiency. This study can easily show the possibility of this method in power cycle improvement. Moreover, since, this comparison was performed on a day length; the effect of ambient temperature was studied in reality.

This study is comprised of four main parts. In the second part, plants specifications are described in detail. In part three, the required and used equations are derived and introduced. Since there are evaporative coolers, the simple common equations based on dry air assumption may not be used. Results and discussions are presented in part four. In this part, similar components in different plants have been compared in detail and finally, a brief discussion can be found in part 5. Also, some supplementary information may be found in appendices.

2 Plant description

The Fars combine power station has a total installed power capacity of 1035 MW. It is located 1530 m above sea level near the city of Shiraz in Fars state. It has three similar combined power plants. Each power plant comprises of one steam cycle with a capacity of 98.2 MW and two gas units each one with a capacity of 123.4 MW.

The ambient air is filtered into the compressor entrance and its pressure decreases slightly during the filtration process. At the ambient temperature more than 40°C, the evaporative cooler located after the air filter comes into service. So, the air inlet temperature decreases by spraying water droplets and its humidity increases.

In the compressor, air pressure increases and the compressed air conducts to the combustion chamber and there, its temperature increases while its pressure remains constant. Combustion chamber in each gas turbine module is fired with natural gas. Exhausted gas generates work in turbine and then gets mixed with the exhausted gas from another gas unit in an unfired Heat Recovery Steam Generator (HRSG).

Required steam for steam turbines generates by heat transfer of flue gas in the HRSEG unit. Each boiler has three main parts, i.e. deaerator, IP drum and HP drum. In the deaerator, feed water gets heated slightly and oxygen and other dissolved gases get removed. The deaerator exit water is separated into two parts and feed to two IP and HP pressure levels by two distinct pumps. The exiting steam from HP drum enters the steam turbine and reaches to IP pressure level. At this stage, the exit steam from IP drum gets injected to the steam turbine. The turbine discharging steam which is the summation of IP and HP steam flows go to a flash type condenser. The cooled liquid water is sprayed onto the steam and condenses it. Then, a part of condensate water goes to Heller type dry cooling tower and the other returns to the boiler. A schematic diagram of the power plant can be found in Figure 1.

3 Analysis

Exergy analysis is based on the second law of thermodynamics. Unless otherwise specified, in considering the balances of mass, energy, entropy (including entropy generation) and exergy, the changes in kinetic and potential energies are neglected. Also, steady state flow is assumed. For a steady state process, the mass balance for each control volume can be written as:

$$ \sum_i \dot{m}_i = \sum_o \dot{m}_o, $$

where the subscript “i” indicates to the inlet and “o” indicates to the outlet.

The energy and entropy balances for a control volume are written as:

$$ \sum_i \dot{E}_i + \dot{Q} = \sum_o \dot{E}_o + \dot{W}, $$

$$ \sum_i \dot{S}_i + \frac{\dot{Q}}{T} + S_{gen} = \sum_o \dot{S}_o. $$

The exergy balances for a control volume are written as:

$$ \sum_i \dot{Ex}_i + \sum_j \left(1 - \frac{T}{T_j}\right) \dot{Q}_j = \sum_o \dot{Ex}_o + \dot{W} + \dot{Ex}_{des}, $$
where the stream exergy rate is:

\[ \dot{E}_x = \dot{m}(e_x) = \dot{m}(e^{tm}_x + e^{ch}_x). \]  

(5)

\( e^{ch} \) is the specific chemical exergy which will be discussed later and the specific thermal exergy, \( e^{tm}_x \), is given by:

\[ e^{tm}_x = (h - h_0) - T_0(s - s_0). \]  

(6)

The input data were collected by measuring the temperature and pressure of all components of six gas units and three steam units on 27th July. Measuring process started since 09:00 till 16:00 in every 5 min. A sample of measuring data for gas unit number 1 has been shown in Appendix A.

To estimate chemical exergy in the combustion chamber used in equation (5), air and flue gas are assumed as ideal gases. The following equations are used to simulate the power plant thermodynamically based on the simplified flow diagram of the power plant shown in Figure 2.

Since moisture is added to air by evaporative coolers, neither fuel low heating value nor dry air assumption may be used and more complicated equations are needed to simulate combustion process. Therefore, for the wet air stream, one can express:

\[ h_1 = h_{\text{air}}1 + \frac{h_{\text{water}}1}{18\omega_1}, \]

(7)

in which \( \omega_1 \) is the specific humidity of air stream to compressor. The mass average specific heat capacity of the mixture is a function of temperature only is presented as:

\[ c_{p1\text{mix}} = \frac{c_{p\text{air}}(T = T_1) + c_{p\text{water}}(T = T_1)\omega_1}{1 + \omega_1}, \]

(8)

\[ c_{p2\text{mix}} = \frac{c_{p\text{air}}(T = \langle T_1, T_2 \rangle) + c_{p\text{water}}(T = \langle T_1, T_2 \rangle)\omega_1}{1 + \omega_1}, \]

(9)

in which \( \langle T_1, T_2 \rangle \) is the average temperature between \( T_1 \) and \( T_2 \).

\[ R_{\text{mix}} = R_{\text{air}} + R_{\text{water}}\omega_1. \]

(10)

It is clear that whenever the evaporative coolers are off, \( \omega = \omega_0 \) and otherwise \( \omega = \omega_1 \):

\[ \omega_0 = \frac{h_{\text{water}}(P = P_{\text{w}}, T = T_0) - h_{f\text{water}}(T = T_0)}{-c_{p\text{air}}(T = \langle T_0, T_1 \rangle)(T_1 - T_0)} \]

\[ \omega_1 = \frac{h_{\text{water}}(T = T_1) - h_{f\text{water}}(T = T_0)}{h_{\text{water}}(T = T_1) - h_{f\text{water}}(T = T_0)}. \]

(11)

\( h_{\text{water}} \) and \( h_{f\text{water}} \) are the water vapor and liquid specific enthalpy at saturated condition at temperature \( T_1 \) and \( T_0 \), respectively. The water partial pressure, \( P_{\text{w}} \), is calculated by [16]:

\[ P_{\text{w}} = \phi_0 P_{\text{sat,water}}(T_0). \]

(12)

\[ \phi_0 = \frac{\alpha_0 P_0}{(0.622 + \alpha_0)P_{\text{sat,water}}(T_0)}. \]

(13)

\( P_{\text{sat,water}}(T_0) \) is the water saturated pressure at temperature \( T_0 \). The inlet vapor flow rate to the gas unit is:

\[ \dot{m}_{\text{water},0} = \dot{m}_{\text{dry,air}}\gamma_0. \]

(14)

In the same way, the vapor flow rate into the compressor is:

\[ \dot{m}_{\text{water},1} = \dot{m}_{\text{dry,air}}\gamma_1. \]

(15)

and finally, total air flow rate is:

\[ \dot{m}_{\text{air}} = \dot{m}_{\text{dry,air}} + \dot{m}_{\text{water},1}. \]

(16)

The compressor isentropic outlet temperature can be calculated as:

\[ T_{S,2} = \left( \frac{P_2}{P_1} \right)^{\frac{c_{p1\text{mix}}}{c_{p2\text{mix}}}} \frac{T_1}{\langle T_1, T_2 \rangle}^{c_{p1\text{mix}}/c_{p2\text{mix}}} \langle T_1, T_2 \rangle, \]

(17)

in which \( \langle T_1, T_2 \rangle \) is the average temperature between \( T_1 \) and \( T_2 \).

Based on the fuel analysis, components of the used natural gas are presented in Table 1.

For the complete combustion, (this assumption was validated by analysis of exit gas from the stack), the flue gas contents are found in Table 2.

Knowing \( x_i \) as mass fraction, \( J_i \) as mole fraction and \( W_i \) as molar mass of \( i \)th component as well as \( W_{\text{fuel}} \) as molar mass, the thermodynamic properties of flue gas will be:

\[ \dot{c}_{p1\text{flu}} = \left\{ \left( x_{C_{2}H_{2}}c_{C_{2}H_{2}} + x_{H_{2}O}c_{H_{2}O} + (x_{N_{2}} + 3.76CW_{N_{2}})c_{N_{2}} \right) + 7.63C\omega_1W_{H_{2}O}c_{H_{2}O} + (C - J_{O_{2}})W_{O_{2}}c_{O_{2}} \right\} \]

\[ \left( 1 + \omega_1 \right)(137.33C) + W_{\text{fuel}}. \]

(18)

and all used properties are evaluated at turbine outlet temperature. \( C \) is the participant mole of air (including excess air) to burn each mole of fuel. \( R \)-value of flue gas is also calculated as follows:

\[ R_{\text{flu}} = \left\{ R_{C_{2}H_{2}}x_{C_{2}H_{2}} + R_{H_{2}O}x_{H_{2}O} + (x_{N_{2}} + 3.76CW_{N_{2}})R_{N_{2}} + 7.63C\omega_1W_{H_{2}O}R_{H_{2}O} + (C - J_{O_{2}})W_{O_{2}}R_{O_{2}} \right\} \]

\[ (1 + \omega_1)(137.33C) + W_{\text{Fuel}}. \]

(19)

\( c_{p2\text{flu}} \) also may be evaluated at the combustion chamber outlet temperature in the same way.
The turbine outlet isentropic temperature can be calculated as:

\[ T_{S.A} = \left( \frac{P_1}{P_2} \right)^{\frac{\gamma_{11u}}{\gamma_{21u}}} \left[ \frac{T_3}{(T_3, T_4)} \right]^{\frac{q_{21u}}{q_{11u}}} (T_3, T_4) \]  

(20)

in which \( T_3 \) and \( T_4 \) is the average temperature between \( T_3 \) and \( T_4 \) and finally, the specific enthalpy of turbine exit gas is:

\[ h_4 = \frac{J_{C20}h_{C20} + J_{H2O}h_{H2O} + (C - J_{O2})h_{O2} + (J_{N2} + 3.76C)h_{N2} + C\omega_1 \frac{137.33}{10} h_{H2O}}{137.33c} \]  

(21)

in which all the thermodynamic properties are evaluated at temperature \( T_4 \). In the same way, the specific enthalpy of exit gas of the stack at temperature \( T_5 \) can be calculated.

Knowing the fuel component as well as fuel mass flow rate, \( m_{Fuel} \), dry air mass flow rate will be:

\[ m_{dry\_air} = \frac{m_{Fuel}}{W_{Fuel}C_{28.851} \times 4.76} \]  

(22)

and therefore, the electricity generated power is:

\[ W_E = m_{dry\_air}[(h_3 - h_4) - (h_2 - h_1)]\eta_{Gen} \]  

(23)

in which \( \eta_{Gen} \) is the generator efficiency.

Unfortunately, it was not possible to measure the combustion chamber exit temperature. So, a try and error procedure was used to calculate it indirectly. To calculate temperature \( T_3 \), \( C \) and after that, \( m_{dry\_air} \) are evaluated at an initial guess of \( T_3 \) and based on those values, \( W_E \) is calculated using equation (24). This procedure repeats till the calculated value of \( W_E \) gets equal to its measured value.

For an ideal gas, the entropy change between two points may be calculated using the following equation [16]:

\[ s_2 - s_1 = s_2^0 - s_1^0 - R\ln \left( \frac{P_2}{P_1} \right) \]  

(24)

in which \( s^0 \) is standard entropy and the compressor second law efficiency is:

\[ \eta_{II,C} = 1 - \frac{T_0(s_2 - s_1)}{h_2 - h_1} \]  

(25)

So,

\[ \eta_{II,C} = 1 - \frac{T_0}{h_2 - h_1} \times \left\{ \left[ \frac{s_{h2B}(T = T_2) - s_{h2B}(T = T_1)}{P_1} \right] - R_{h2B} \right\} \times \left\{ \ln \left[ \frac{(\omega_1 + 0.622)/(\omega_1 + 0.622)}{(\omega_0 + 0.622)/(\omega_0 + 0.622)} \right] \omega_1 + \left[ \frac{s_{air}(T = T_2) - s_{air}(T = T_1)}{P_0} \right] - R_{air} \right\} \]  

(26)
in the same way, the turbine second law efficiency is:

\[ \eta_{IT} = \frac{1}{(1 - \frac{(cp_{flu} \ln(T_2/T_3) + cp_{flu} \ln(T_2/T_3) - R_{flu} \ln(P_2/F_1))}{h_3 - h_4})} \]  

(27)

Finally, the gas unit efficiency can be calculated as:

\[ \eta_I = \left( \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} \right) \left( 1 - \frac{h_4 - h_1}{h_3 - h_2} \right) \]  

(28)

All used thermodynamic variables as functions of temperature have been presented in Appendix B.

### 4 Results and discussion

#### 4.1 Evaporative coolers

Ambient temperature and compressors entrance temperatures have been shown in Figure 3. Based on this figure, the ambient temperature reaches from about 30°C at 9:00 to about 40°C at noon. It is noticeable that the inlet temperature of compressor unit 1 is around 3°C to 4°C more than other units. This shows compressor unit 1 is located so that warmer air enters the compressor. So, it is an object of improvement. At 12:00, evaporative coolers come into service. These evaporative coolers cause the reduction in compressors consumption power which will be discussed later. Interestingly, the performances of all evaporative coolers are not the same and a large deviation may be observed among their outlet temperatures. While the evaporative cooler of the 1st unit decreases the temperature from 40°C to 22°C, cooler number 3 or number 6, reduce the air temperature from 38°C to 30°C. So, this variation in operation is subject to question.

Since the air inlet temperature to all coolers is approximately the same, the difference in their performance must be investigated in their structure. To show this, the relative humidity of compressor inlet air has been shown in Figure 4. According to this figure, the relative humidity of cooler number 1 reaches to about 93% while this value for coolers number 2, 4 and 5 is between 60% to 70%. For cooler 3 and 6, this value is lesser and is about 45%. This means that only 15% is added to ambient air humidity.

#### 4.2 Compressors

To investigate the effect of air inlet temperature to compressor discharge pressure, variation of discharge pressure versus inlet air temperature has been shown in Figure 5 for all six compressors. Aggregated points are related to working period of evaporative coolers where the inlet air temperature to the compressors is independent of ambient temperature and relatively constant. In this way,
the scatters points belong to the morning working condition. First of all, it is clear that by decreasing the inlet temperature, discharge pressure increases. However, this increase is not the same for all compressors. As an example, in compressor number 2, for a temperature reduction of 7°C from 37°C to 30°C, the discharge pressure increases from 8.4 bars to 9.0 bars, however, although in compressor number 1, the temperature reduction is more than this value, the increase in pressure discharge is less. This shows that compressor number 1 needs more consideration during overhaul.

Another important result can also be deduced from this figure. Due to the less efficient operation of cooler number 6, there can be found some point in which exit temperature (i.e. inlet temperature into the compressor) is equal to air temperature at the mooring. However, its relative humidity is more than morning air relative humidity. These points are indicated in the Figure 5 by a circle. According to this figure, although the temperature is the same, the compressor discharge pressure in the morning is less than the compressor discharge pressure in the afternoon. This shows that increasing the relative humidity increases the compressor discharge pressure, even though the temperature remains constant. The other fact is that compressors number 4 and 5 are less acceptable in comparison to others because they have lower discharge pressure in both the absence and presence of coolers.

All in all, it can be concluded that if all compressor acted like compressor number 2 and all coolers same as cooler number 1, discharge pressure would reach to a value more than 9.1 bars. This means that a considerable gain in efficiency. This deduction may be supported by Figure 6 in which plants power outputs have been shown against compressor discharge pressure. According to this figure, 7% increase in discharge pressure number 2, results in 15% increase in output power or 5% increase in discharge pressure number 6, results in 7% increase in output power. These examples show the output power sensitivity to compressor discharge pressure.

Finally, exergetic efficiencies of compressors have been shown in Figure 7. Based on Figure 7, in the absence of coolers, compressors numbers 2, 4 and 6 seem more efficient in comparison to others.

### 4.3 Turbines

In the same way, turbines exergetic efficiencies are shown in Figure 8. According to this figure, turbine 4 has lowest efficiency while evaporative coolers are not in service. When they are in service, the lowest efficiency belongs to turbine 5. Turbine 2 shows relative better performance in both working condition. So, it can be introduced as a standard to compare with the others during overhaul.

### 4.4 Gas units efficiencies

The efficiency of each gas unit based on equation (27) was calculated. The efficiency may be used to compare units
with each other totally. Results have been shown in Figure 9. Since evaporative coolers are not working all the day, three types of efficiency were calculated; the first one for times with evaporative coolers (in the afternoon), the second one for times without evaporative coolers (in the morning) and the last one for one day.

According to this figure, although the performances of all units are approximately the same, unit 3 has lower efficiency. This result was expected due to fewer performances of turbines and a compressor of these unites. Moreover, due to the higher performance of unit 1 cooler, the efficiency of unit 1 in the presence of coolers is higher than the others.

4.5 HRSG

Inlet and outlet temperatures of HRSGs have been shown in Figures 10 and 11. According to these figures, water injection in evaporative coolers causes a considerable reduction in inlet flue gas temperature. Moreover, a variation about 10°C can be observed in HRSG outlet temperature because the water coming from cooling towers (i.e. inlet flow to LP boiler) getting warmer. Furthermore, a meaningful difference may be observed between outlet temperatures of HRSGs numbers 4 and 1 with the others. It must be tried to reduce all HRSGs outlet temperatures to values of HRSGs numbers 4 and 1.

Second law efficiencies of HRSGs have been shown also in Figure 12. According to this figure, in the absence of coolers (before noon), efficiencies of HRSGs numbers 1 and 4 are slightly less than the others. However, when coolers start, the efficiency of HRSG number 5 decreases significantly. All in all, HRSG number 6 shows better performance than the others.

Since the inlet and outlet temperatures of HRSGs are affected by ambient temperature, second law efficiencies of HRSGs against air inlet temperature to compressors have been shown in Figure 13. Bunches of points are related to times when coolers are at the service and air inlet temperature is relatively independent of ambient temperature and is constant. As it is clear, water injection decreases HRSG efficiency.

4.6 Cooling towers

There are three cooling towers and each one is comprised of six sectors. Since they are similar in structure and inlet temperature, it is expected that the outlet temperature is
the same. However, based on Figure 13, the outlet temperature from sector 6 of cooling tower 2 is higher and this, decreases cooling tower performance. With the exception of this one, others are operating well (Figs. 14–16).

5 Steam turbines

Second law efficiencies of steam turbines were calculated and have been shown in Figure 17. Based on this figure, no meaningful difference can be observed among steam turbines.

5.1 Power plants efficiencies

Finally, total efficiencies of power plants have been shown in Figure 18. This figure is useful to compare overall performance of power plants. According to this figure, the efficiencies of all power plants in the absence of coolers are approximately the same. However, when evaporative coolers are in the service, the efficiency of unit “A” increases considerably.

6 Conclusion

In this study, three similar combined power cycles in one day were compared from energy and exergy point of view. In spite of common method, the exergy destruction of different types of equipments is not compared with each other, but each equipment is compared with its counterpart in similar plants. Some concluded results are:

– the performances of all evaporative coolers are not the same. Cooler number 1 is the best while coolers number 3 and 6 do not work as well as number 1;
– decreasing the compressor inlet temperature increases its discharge pressure. Moreover, the compressor number 1 needs more consideration during overhaul;
– adding humidity increases the compressor discharge pressure, even though the temperature remains constant;
– compressors numbers 2, 3 and 6 seem more efficiently in comparison to others;
– evaporative cooler increases the gas unit efficiency;
– water injection in evaporative cooler causes a considerable reduction in inlet flue gas temperature to HRSG and decreases HRSG efficiency;
– increasing the ambient temperature decreases HRSG efficiency;
– outlet temperature from sector 6 of cooling tower 2 is higher and must be corrected.

These are some results of comparing similar cycles. These results simply show the efficiency of this method in cycle improvement. By casting this method periodically, valuable guidelines may be obtained for overhaul.

Nomenclature

\( c_p(T) \) Specific heat capacity at temperature \( T \) (kJ/kg K)
\( E \) Energy rate (kW)
\( Ex \) Exergy rate (kW)

Greek symbols

\( \varphi \) Relative humidity
\( \omega \) Specific humidity
\( \eta_I \) First law efficiency
\( \eta_{II} \) Second law efficiency

Subscripts

\( C \) Compressor
\( Gen \) Generator
\( gen \) Generation
\( i \) Inlet
\( mix \) Mixture
\( moist \) Moisture
\( o \) Outlet
\( S \) Isentropic
\( sat \) Saturated
\( T \) Turbine
\( 0 \) Dead state condition
## Appendix A: Sample of measuring data for gas unit number 1.

| Time       | Ambient Comp. | Ambient Temp.(°C) | Ambient Turb. (%) | Ambient Outlet Air Temp.(°C) | Ambient Power (barg) | Inlet Power (barg) | Outlet Power (barg) | Press. Eff. (%) | Temp. IP (°C) | Temp. HP (°C) | Press. IP (barg) | Press. HP (barg) | Press. Outlet Temp.(°C) | HP (4) |
|------------|---------------|-------------------|-------------------|----------------------------|----------------------|-------------------|-------------------|---------------|---------------|---------------|----------------|----------------|-------------------------|-------|
| 8:00:00    | 0.30          | 30.1               | 30.4              | 30.4                        | 24.6                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |
| 8:05:00    | 0.30          | 30.4               | 30.7              | 30.2                        | 24.2                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |
| 8:10:00    | 0.30          | 30.2               | 30.7              | 30.2                        | 24.2                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |
| 8:15:00    | 0.30          | 30.2               | 30.7              | 30.2                        | 24.2                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |
| 8:20:00    | 0.30          | 30.2               | 30.7              | 30.2                        | 24.2                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |
| 8:25:00    | 0.30          | 30.2               | 30.7              | 30.2                        | 24.2                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |
| 8:30:00    | 0.30          | 30.2               | 30.7              | 30.2                        | 24.2                  | 0.86              | 0.86              | 0.86          | 552.4         | 552.4         | 0.85          | 0.85          | 0.85                      | 504   |

### Table 1: Measuring data for gas unit number 1.

- **Time:** The time at which the measurement was taken.
- **Ambient Comp.:** The ambient composition.
- **Ambient Temp.(°C):** The ambient temperature.
- **Ambient Turb. (%):** The ambient turbidity percentage.
- **Ambient Outlet Air Temp.(°C):** The ambient outlet air temperature.
- **Ambient Power (barg):** The ambient power level.
- **Inlet Power (barg):** The inlet power level.
- **Outlet Power (barg):** The outlet power level.
- **Press. Eff. (%):** The pressure efficiency percentage.
- **Temp. IP (°C):** The temperature at the inlet pipe.
- **Temp. HP (°C):** The temperature at the high pressure.
- **Press. IP (barg):** The pressure at the inlet pipe.
- **Press. HP (barg):** The pressure at the high pressure.
- **Press. Outlet Temp.(°C):** The pressure outlet temperature.
- **HP (4):** The high pressure level.

This table provides a sample of the measuring data obtained for gas unit number 1, showing various parameters including ambient conditions, power levels, and temperatures at different locations within the unit.
Appendix B: Thermodynamic variables as functions of temperature

Introducing $\theta = T/100$ (T in Kelvin)

\[ c_{p_{air}} = (1.04841 - 3.8371 \frac{\theta}{10^3} + 9.4537 \frac{\theta^2}{10^6} - 5.4931 \frac{\theta^3}{10^9} + 7.9218 \frac{\theta^4}{10^{12}}) \text{ (kJ/kgK)} \]  
(B1)

\[ c_{p_{N2}} = \left[39.06 - 512.79\theta^{-1.5} + 1072.7\theta^{-2} - 820.4\theta^{-3}\right]/28.013 \text{ (kJ/kgK)} \]  
(B2)

\[ c_{p_{O2}} = \left[37.432 + 0.020102\theta^{-1.5} - 178.57\theta^{-1.5} + 236.88\theta^{-2}\right]/31.999 \text{ (kJ/kgK)} \]  
(B3)

\[ c_{p_{CO2}} = \left[-3.7357 + 30.529\theta^{0.5} - 4.1034\theta + 0.024198\theta^2\right]/44.01 \text{ (kJ/kgK)} \]  
(B4)

\[ c_{p_{H2O}} = \left[143.05 - 183.54\theta^{2.5} + 82.751\theta^{3.5} - 3.6989\theta\right]/18.015 \text{ (kJ/kgK)} \]  
(B5)

\[ c_{p_{CH4}} = \left[-672.87 + 439.74\theta^{2.5} - 24.875\theta^{3.75} + 323.88\theta^{0.5}\right]/16.04 \text{ (kJ/kgK)} \]  
(B6)

\[ c_{p_{C2H6}} = \left[6.895 + 17.26\theta - 0.6402\theta^2 + 0.00728\theta^3\right]/30.07 \text{ (kJ/kgK)} \]  
(B7)

\[ c_{p_{C3H8}} = \left[-4.042 + 30.46\theta - 1.571\theta^2 + 0.03171\theta^3\right]/44.09 \text{ (kJ/kgK)} \]  
(B8)

\[ c_{p_{C4H10}} = \left[3.954 + 37.12\theta - 1.833\theta^2 + 0.03498\theta^3\right]/58.124 \text{ (kJ/kgK)} \]  
(B9)

\[ h_{air} = 4.666416 + 0.96831\theta + 0.00003961\theta^2 + 0.00000003264\theta^3 \text{ (kJ/kg)} \]  
(B10)

\[ h_{N2} = \begin{cases} 
-4084835.458 + 13666978\theta^{1.0024978}/(4711.4676 + \theta^{1.0024978}) & \text{for } \theta \leq 4 \\
8670 + 2923.5942\theta - 12.044984\theta^2 + 2.136434\theta^3 & \text{for } \theta < 10 \text{ (kJ/kmol)} \\
\exp\left(8.0789435 - \frac{4.8921249}{\theta} + 1.0352864\ln(\theta)\right) & \text{for } \theta \geq 10 
\end{cases} \]  
(B11)

\[ h_{O2} = \begin{cases} 
-8683.2986 + 2915.5857\theta - 11.791232\theta^2 + 3.6922661\theta^3 & \text{for } \theta < 4 \\
-8349 + 2672\theta + 43\theta^2 & \text{for } \theta \leq 4 \text{ (kJ/kmol)} \\
-7939.7229 + 2451.347\theta + 80.132035\theta^2 - 1.8838384\theta^3 & \text{for } \theta < 10 \\
-8879.333 + 2882.7507\theta + 31.329828\theta^2 - 0.28252785\theta^3 & \text{for } \theta \geq 10 
\end{cases} \]  
(B12)

\[ h_{CO2} = \begin{cases} 
-9363.977 + 2939.3076\theta - 83.011986\theta^2 + 50.577468\theta^3 & \text{for } \theta < 4 \\
-9525 + 2640\theta + 184\theta^2 & \text{for } \theta \leq 4 \text{ (kJ/kmol)} \\
-9584.9286 + 2538.8571\theta + 240.08333\theta^2 - 6.4167\theta^3 & \text{for } \theta < 10 \\
\exp\left(8.6546661 - \frac{5.9395707}{\theta} + 1.0233514\ln(\theta)\right) & \text{for } \theta \geq 10 
\end{cases} \]  
(B13)

\[ h_{O2} = \begin{cases} 
-9903.9842 + 3250.3251\theta + 42.806253\theta^2 - 6.2183832\theta^3 & \text{for } \theta < 4 \\
-9598 + 3049\theta + 42\theta^2 & \text{for } \theta < 4 \text{ (kJ/kmol)} \\
-9494.7937 + 3083.6892\theta + 31.208332\theta^2 + 1.9537038\theta^3 & \text{for } \theta < 10 \\
-0.04166667\theta^4 & \text{for } \theta < 10 \\
-10205.43 + 2943.4614\theta + 73.011089\theta^2 - 0.63909098\theta^3 & \text{for } \theta \geq 10 
\end{cases} \]  
(B14)

\[ z_{air}^0 = 3.15161(T - 14.9972)^{0.13767} \]  
(B15)

\[ s_{air}^0 = 3.6088(T)^{0.18536} \]  
(B16)
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