

A novel efficiency assessment criterion for thermo-economic analysis of Gas-Steam Combined Cycle combined heat and power

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Abstract. Gas-steam combined cycle (GSCC) plants are among the best technologies for power production, in particular when operating in combined heat and power (CHP) generation. As far as we know, there is not any efficiency assessment criterion for thermo-economic analysis of GSCC-CHP plants until now. To solve this problem, this paper proposes a novel efficiency assessment criterion through establishing a mathematical model of heat-consume apportionment and then defining a set of thermo-economic performance indexes for the GSCC-CHP plants, based on the basic principle of heat quantity method by synchronically considering some predefined parameters such as energy utilization efficiency of gas turbine units and the unit comprehensive energy utilization efficiency. With the multi-condition analysis of the changed heat load, extraction pressure and water recovery rate, it verifies the feasibility of the efficiency assessment criterion.

Keywords: gas-steam combined cycle; combined heat and power; efficiency assessment criterion; thermo-economic performance index; thermal power plant.

1. Introduction

Gas-steam combined cycle (GSCC) plants are currently among the best available technology in terms of conversion efficiency [1], especially when operating in combined heat and power (CHP) mode [2]. The application of gas-steam combined cycle cogeneration increases the utilization efficiency of chemical energy in fuels and decreases the harmful impacts on the environment. It eliminates emissions of SO₂ and particulates and decreases the emissions of NO_x and CO₂ to the atmosphere [3]. In general, gas-steam combined cycle cogeneration has not only advantage of environmental protection, but also advantage of high energy conversion efficiency because of the cascade utilization of fuel compound energy [4].

In recent years, a number of GSCC-CHP plants have been built and thermo-economic performance indexes are necessary to evaluate the efficiency and thermal economy of the plants. About the study of performance indexes, many scholars put forward different views. Bolesław Zaporowski and Radosław Szczerbowski (2003) studied energy analysis of technological systems of GSCC-CHP plants and some efficiency indicators: energy efficiency of GSCC-CHP and efficiency of electric-energy generation were



proposed [3]. M.ATMACA (2011) defined new performance criterion named energy utilization factor (EUF) and artificial thermal efficiency (η_A). EUF has been defined as ratio of total heat quantity taken from a system and used for a process to heat the quantity given to the system [5]. Indicators used in evaluating efficiency of GSCC-CHP plants named gross global efficiency ($\eta_{el,gl}$) and fuel utilization efficiency (FUE) have been propounded by Matteo Jarre and Michel Noussan (2016) [6]. These proposed indicators were basically the overall energy utilization indicators without considering the difference between heat load and electric load. Moreover, these studies did not consider heat-supply thermal efficiency.

In order to objectively evaluate the energy conversion efficiency of GSCC-CHP plants, the total energy consumption should be divided into two kinds of products: heat load and electric load. Then, the thermal economy of plants can be evaluated respectively by the thermo-economic performance index of these two loads. Concerning the heat-consume apportionment of heat load and electric load, there are various kinds of methods, such as heat quantity method, actual enthalpy drop method, the exergy method, entropy change method [7] and net benefit method. Among them, the heat quantity method is based on the first law of thermodynamics and is the national statutory method of apportionment [8-9].

From above brief literature review, we can find that, until now there is not any efficiency assessment criterion for thermo-economic analysis of GSCC-CHP plants and the aforementioned proposed indicators did not consider the difference between heat load and electric load. The heat quantity method, in particular in China, is a popular tool for heat-consume apportionment on cogeneration plants so as to distinguish heat load and electric load. This method has been successfully applied to establish the performance index of thermal power cogeneration plants.

Therefore, as above motivated, this paper puts forward a novel efficiency assessment criterion. More specifically, we establish a mathematical model of heat-consume apportionment and then define a set of thermo-economic performance indexes on the basis of heat quantity method. Evidently, the main contribution of this paper is the novel efficiency assessment criterion for thermo-economic analysis of GSCC-CHP plants, which can be taken as a reference value for standardizing thermo-economic assessment system and can also be used to evaluate the thermal efficiency of GSCC-CHP plants.

The paper is organized as follows: Section II establishes the mathematical model of efficiency assessment criterion and then defines a set of thermo-economic performance indexes after introducing the GSCC-CHP plant. In Section III, some case studies are conducted to validate the proposed performance index by comparing with other kinds of performance indexes under different heat-supply conditions. The last section concludes the paper.

2. GSCC-CHP

The general configuration scheme of gas-steam combined cycle combined heat and power (GSCC-CHP) is shown in Fig.1.

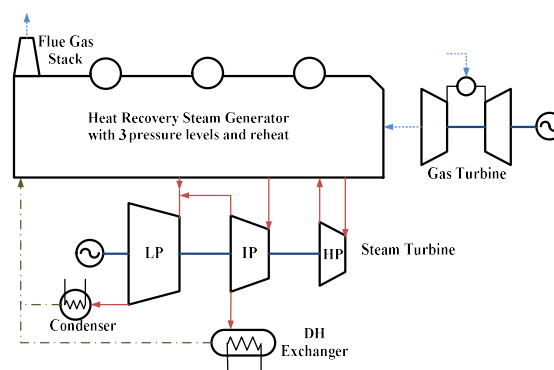


Fig.1 General Configuration scheme of GSCC-CHP.

GSCC-CHP is a kind of typical distributed energy system. It consists of the Brayton cycle of the gas turbine, the Heat Recovery Steam Generator (HRSG) and the Rankine cycle of the cogeneration steam turbine. In the Brayton cycle, natural gas and compressed air are mixed in the gas turbine for combustion and work, in addition to power consumption for driving the compressor. In the Rankine cycle, the exhaust gas from gas turbine transfers heat and produces steam in the HRSG and does work in the steam turbine, meanwhile the extraction or exhaust steam for heat supply. Compared with the traditional steam turbine cogeneration, gas turbine and HRSG instead of the traditional coal-fired or oil-fired boiler in this cogeneration way, the rear turbine generator sets are basically the same [10].

The objective of this paper is to put forward a novel efficiency assessment criterion for thermo-economic analysis of GSCC-CHP plants. To this end, we need to establish the mathematical model of heat-consume apportionment and then to define a set of thermo-economic performance indexes. More details will be discussed in the consequent Section 3. Furthermore, some important notations are summarized in the following table in order to benefit the consequent presentation.

Nomenclature		η_{hs}	heat-supply network transmission efficiency
<i>General abbreviations</i>		η_{GT}	energy utilization efficiency in gas turbine units
GSCC	gas-steam combined cycle	Q_0	steam turbine heat consumption(GJ/h)
CHP	combined heat and power	η_{bp}	HRSG and pipeline efficiency
HRSG	heat recovery steam generator	η	unit comprehensive energy utilization efficiency
<i>Technical abbreviations</i>		Q_{tp}	overall heat consumption(GJ/h)
m_{NG}	mass flow rate of natural gas(t/h)	$Q_{tp(h)}$	heat-supply heat consumption(GJ/h)
LHV_{NG}	lower heating value of natural gas(kJ/kg)	$Q_{tp(e)}$	power generation heat consumption(GJ/h)
Q_{air}	input heat by air to gas turbine(GJ/h)	Q	heat load(GJ/h)
Q_{gas}	exhaust gas heat quantity to HRSG(GJ/h)	Q_h	heat-supply quantity(GJ/h)
$Q_{loss,GT}$	heat loss in gas turbine units(GJ/h)	η_{hs}	transmission efficiency of heat-supply network
$P_{el,GT}$	electric power output of gas turbine(MW)	$\eta_{tp(e)}$	power generation efficiency
$P_{el,ST}$	electric power output of steam turbine(MW)	$q_{tp(e)}$	power generation heat consumption rate(kJ/kWh)
η_m	mechanical efficiency in gas turbine units	$b^s_{tp(e)}$	power generation standard coal consumption rate(kg/kWh)
η_g	generator efficiency in gas turbine units	$\eta_{tp(h)}$	heat-supply thermal efficiency
ζ_{GT}	energy loss ratio in gas turbine units	$b^s_{tp(h)}$	heat-supply standard coal consumption rate(kg/GJ)

3. The proposed criterion: mathematical model and performance index

The key procedure of heat-consume apportionment is calculating the heat-supply heat consumption by the needed heat load. For traditional coal-fired cogeneration units, the heat-consume apportionment method of our country's legal principle is: the actual heat supply of cogeneration steam turbine is converted into the unit heat consumption (considering some energy loss in the coal-fired boilers and in the process of transmission between boilers and pipelines), which is as the heat-supply heat consumption of the cogeneration unit. The power generation heat consumption equals to the overall heat consumption minus the heat-supply heat consumption. The heat-consume apportionment is on the basis of heat quantity method and the heat-supply heat consumption is determined by the heat load [11]:

$$Q_{tp(h)} = \frac{Q_h}{\eta_{bp}} = \frac{Q}{\eta_{bp}\eta_{hs}}, \quad (1)$$

Where η_{bp} is boiler and pipeline efficiency of coal-fired cogeneration units.

For GSCC-CHP plants, the energy conversion is more complicated than that of coal-fired cogeneration units, so the heat-supply heat consumption cannot computed by formula (1). To calculate

the heat-supply heat consumption, we need to analysis the balance relationship between the input and output energy of gas turbine firstly. The energy-balance diagram of GSCC-CHP is shown in Fig.2.

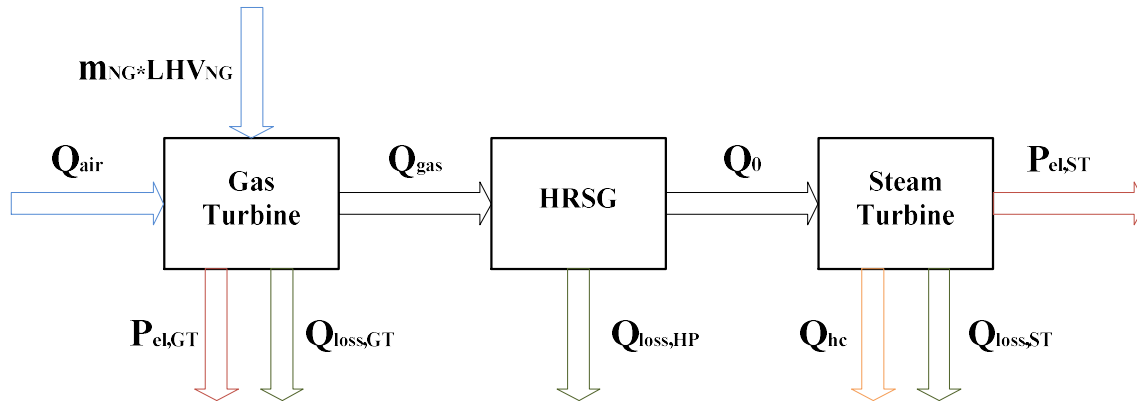


Fig.2 The energy-balance diagram of GSCC-CHP.

For gas turbine units, the energy balance formula is:

$$m_{NG} * LHV_{NG} + Q_{air} = Q_{gas} + 3600P_{el,GT}/(\eta_m\eta_g) + Q_{loss,GT}. \quad (2)$$

By considering the heat loss in gas turbine units ($Q_{loss,GT}$), which influence the heat-supply heat consumption calculation of GSCC-CHP, we define the two parameters in gas turbine units: energy loss ratio and energy utilization efficiency, which are presented respectively as follows

- Energy loss ratio in gas turbine units:

$$\zeta_{GT} = \frac{Q_{loss,GT}}{m_{NG} * LHV_{NG} + Q_{air}}, \quad (3)$$

- Energy utilization efficiency in gas turbine units:

$$\eta_{GT} = 1 - \zeta_{GT}. \quad (4)$$

The relationship of the gas heat quantity from gas turbine and the steam turbine heat consumption is:

$$\eta_{hp} = \frac{Q_0}{Q_{gas}}. \quad (5)$$

With the help of (4) and (5), we can now define the parameter unit comprehensive energy utilization efficiency:

$$\eta = \eta_{hp}\eta_{GT}, \quad (6)$$

Which lays the foundation for heat-consume apportionment.

On the basis of the above parameters, we can establish a mathematical model of heat-consume apportionment and then define a set of thermo-economic performance indexes. The specific model-building process are shown as below.

(1). Heat-consume apportionment model

Analogizing the heat-supply heat consumption calculation formula of coal-fired cogeneration units, the heat-supply heat consumption calculation formula of GSCC-CHP is:

$$Q_{tp(h)} = \frac{Q_h}{\eta_{bp}\eta_{GT}} = \frac{Q}{\eta\eta_{hs}}. \quad (7)$$

The overall heat consumption (Q_{tp}) equals to the input energy of gas turbine and the power generation heat consumption equals to the overall heat consumption minus the heat-supply heat consumption.

$$Q_{tp} = m_{NG} * LHV_{NG} + Q_{air}, \quad (8)$$

$$Q_{tp(e)} = Q_{tp} - Q_{tp(h)}. \quad (9)$$

(2). Sub-item performance index

With the heat-supply heat consumption and power generation heat consumption, we are ready to calculate the sub-item performance index, which are defined respectively as follows.

Power generation efficiency ($\eta_{tp(e)}$) is defined as the ratio between the total gross electric power output from both gas and steam turbines ($P_{el,GT} + P_{el,ST}$) and the power generation heat consumption ($Q_{tp(e)}$), described by

$$\eta_{tp(e)} = \frac{3600 * (P_{el,GT} + P_{el,ST})}{Q_{tp(e)}}. \quad (10)$$

Power generation heat consumption rate ($q_{tp(e)}$) is defined as the ratio between the power generation heat consumption ($Q_{tp(e)}$) and the total gross electric power output from both gas and steam turbines ($P_{el,GT} + P_{el,ST}$) in form of

$$q_{tp(e)} = \frac{Q_{tp(e)}}{P_{el,GT} + P_{el,ST}} = \frac{3600}{\eta_{tp(e)}} (\text{kJ/kWh}). \quad (11)$$

Power generation standard coal consumption rate ($b_{tp(e)}^S$) can be defined as the ratio between the Power generation heat consumption rate ($q_{tp(e)}$) and the lower heating value of standard coal (29271 kJ/kg):

$$b_{tp(e)}^S = \frac{q_{tp(e)}}{29271} = \frac{0.123}{\eta_{tp(e)}} (\text{kg/kWh}), \quad (12)$$

Heat-supply thermal efficiency ($\eta_{tp(h)}$) can be defined as the ratio between the heat load (Q) and the heat-supply heat consumption ($Q_{tp(h)}$), i.e.,

$$\eta_{tp(h)} = \frac{Q}{Q_{tp(h)}} = \eta\eta_{hs}. \quad (13)$$

Heat-supply standard coal consumption rate ($b_{tp(h)}^S$) is defined as the ratio between the heat-supply heat consumption ($Q_{tp(h)}$) and the product of heat load (Q) and lower heating value of standard coal (29271 kJ/kg). More precisely, we have.

$$b_{tp(h)}^S = \frac{Q_{tp(h)} * 10^6}{29271 * Q} = \frac{34.1}{\eta_{tp(h)}} (\text{kg/GJ}). \quad (14)$$

These performance indexes can be used as the efficiency assessment criterion for thermo-economic analysis of GSCC-CHP plants. That is to say, we can evaluate the thermal economy by the numerical

value of indexes. In the next section, we mainly discuss the feasibility of these performance indexes by the experiments design.

4. Experiments: validation and discussion

This section consists of experiments design and performance index calculation. In section 4.1, experiments are designed for different kinds of heat-supply conditions. In section 4.2, performance index calculations under various heating modes are conducted. In section 4.3, performance indexes under different heating modes are compared and discussed to verify the feasibility of index model.

4.1. Experiments design

The Huaneng Jinling Gas Turbine Power Plant S109FA unit is used to compute in the present work. In connection with the operating parameters in ISO condition in 100% load, three heating modes are hypothesized: extraction steam from IP exhaust steam, extraction steam from HP exhaust steam(before reheating) and extraction from IP steam admission(after reheating) are conducted respectively for heat supply.

Table 1 summarizes the main technical data and operating parameters of S109FA unit in ISO condition in 100% load. In Table 2, the thermal parameters calculated by formula (2)-(6) in gas turbine of S109FA unit in ISO condition in 100% load are reported.

The experiments design considers the influencing factors, such as heat load, extraction pressure and water recovery rate. To verify the feasibility of index model, we designs the following comparison conditions: (1) the same heat load with different extraction pressures; (2) the same heat load with different water recovery rates; (3) different heat loads with the same extraction pressure and water recovery rate. The calculation assumption of these different heat-supply conditions is: the main steam flow is kept unchanged and steam turbine heat consumption is changed. When considering the heat load kept the same, the extraction steam flow with different extraction pressures is different.

Table 1. Main technical data and operating parameters of S109FA unit in ISO condition in 100% load.

Plant component	Value	S109FA
GSCC	Gross electric power(MW)	397.25
Gas turbine	Fuel type	Natural gas
	LHV of natural gas(kJ/kg)	48686.3
	Nominal gross electric power(MW)	255.52
Steam turbine	Nominal gross electric power(MW)	141.73
Ambient Conditions	Barometric pressure(kPa)	101.35
	Air temperature(°C)	15
	Relative humidity(%)	60

Table 2. Thermal parameters in gas turbine of S109FA unit in ISO condition in 100% load.

Parameter	Value	unit
m_{NG}	50.7	t/h
$m_{NG} * LHV_{NG}$	2468.39541	GJ/h
Q_{air}	72.41013	GJ/h
$P_{el,GT}$	255.52	MW
η_m	0.98	
η_g	0.98	
Q_{gas}	1555.36425	GJ/h
$Q_{loss,GT}$	27.24129	GJ/h
Q_0	1360.24733	GJ/h
ζ_{GT}	0.01072	
η_{GT}	0.98928	
η_{hp}	0.87455	
η	0.86517	

The principal thermodynamic system diagram of GSCC-CHP under three heating modes is shown in Fig.3.

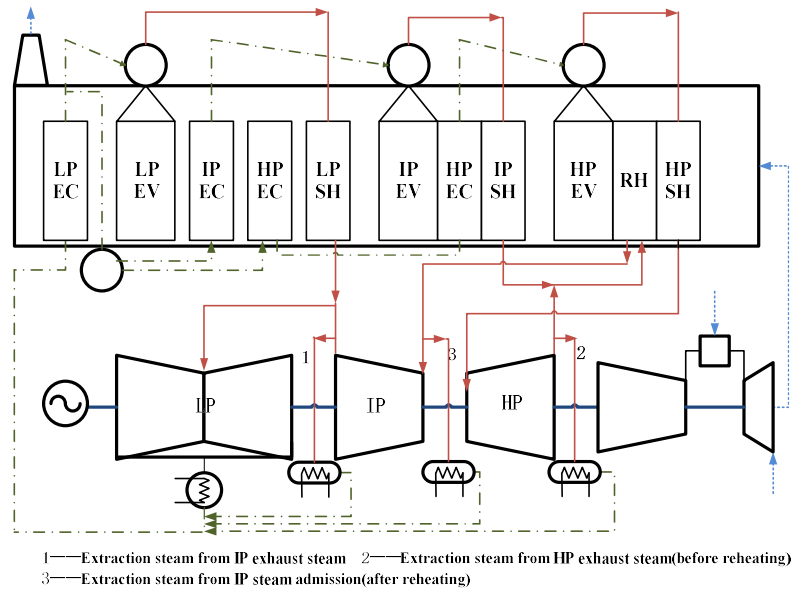


Fig.3 Principal thermodynamic system diagram of GSCC-CHP under three heating modes.

Specifically, when water recovery rate is 100%, three heating modes: extraction steam from IP exhaust steam, extraction steam from HP exhaust steam(before reheating) and extraction from IP steam admission(after reheating) are numbered as Heating mode 1, Heating mode 2 and Heating mode 3. When recovery rate is 0%, three heating modes: extraction steam from IP exhaust steam, extraction steam from HP exhaust steam(before reheating) and extraction from IP steam admission(after reheating) are numbered as Heating mode 4, Heating mode 5 and Heating mode 6.

When discussing the effect of changed heat load, the thesis chooses the heating mode of extraction steam from IP exhaust steam and the extraction steam flow respectively is 0(heating mode 7), 0.05D0(heating mode 8) and 0.1D0(heating mode 9), keeping the water recovery rate 100%.

Make sure unit's steam and water balance under different heat-supply conditions, we calculate steam turbine heat consumption, electric power output of steam turbine. When water recovery rate is 100%, we assume thermal network water temperature is 100°C, then backwater enthalpy $h_{wh} = 418\text{kJ/kg}$. Some thermal parameters for performance index calculation are computed as follows.

● Heating mode 1:

The extraction steam flow is $0.1D_0 = 36.6\text{t/h}$.

Condensation water enthalpy $h_{wc} = 125.3\text{kJ/kg}$.

Feed-water enthalpy: $h_{fw} = 0.9h_{wc} + 0.1h_{wh} = 154.57\text{kJ/kg}$

Steam turbine heat consumption:

$$Q_0 = (285.6 * (3540.4 - 154.57) + 40.2 * (3016.6 - 154.57) + 40.9 * (3061.1 - 154.57) + 313.4 * 3611.6 - 312.7 * 3144.5) / 10^3 = 1349.514021\text{GJ/h}$$

Overall heat consumption:

$$Q_{tp} = \left(\frac{Q_0}{\eta_{hp}} + \frac{3600P_{el,GT}}{\eta_m\eta_g} \right) / \eta_{GT} = 2528.488959\text{GJ/h}$$

Electric power output of steam turbine:

$$P_{el,ST} = (285.6 * 3538.3 - 272.5 * 3165.9 + 313.4 * 3607.5 + 40.9 * 3057.2 - 366 * 2382.4 - 36.6 * (3094.5 - 2382.4)\eta_m\eta_g) / 10^3 = 485.221872\text{GJ/h} = 134.78\text{MW}$$

● Heating mode 2:

To ensure the heat load is equal to that of 0.1D0 extraction steam flow from IP exhaust steam for heat-supply, then

$$D_h = \frac{0.1 * (3094.5 - 418)}{3165.9 - 418} * 366 = 35.649 \text{ t/h}$$

$$\text{Feed-water enthalpy: } h_{fw} = [(366 - 35.649)h_{wc} + 35.649h_{wh}]/366 = 153.81 \text{ kJ/kg}$$

Steam turbine heat consumption:

$$Q_0 = (285.6 * (3540.4 - 153.81) + 40.2 * (3016.6 - 153.81) + 40.9 * (3061.1 - 153.81) + 313.4 * 3611.6 - 312.7 * 3144.5 - 35.649 * (3611.6 - 3144.5))/10^3 = 1333.141065 \text{ GJ/h}$$

Overall heat consumption:

$$Q_{tp} = (\frac{Q_0}{\eta_{hp}} + \frac{3600P_{el,GT}}{\eta_m\eta_g})/\eta_{GT} = 2509.563429 \text{ GJ/h}$$

Electric power output of steam turbine:

$$P_{el,ST} = (285.6 * 3538.3 - 272.5 * 3165.9 + 313.4 * 3607.5 + 40.9 * 3057.2 - 366 * 2382.4 - 35.649 * (3607.5 - 2382.4)\eta_m\eta_g)/10^3 = 468.3155713 \text{ GJ/h} = 130.09 \text{ MW}$$

● Heating mode 3:

To ensure the heat load is equal to that of 0.1D0 extraction steam flow from IP exhaust steam for heat-supply, then

$$D_h = \frac{0.1 * (3094.5 - 418)}{3607.5 - 418} * 366 = 30.713 \text{ t/h}$$

Feed-water enthalpy:

$$h_{fw} = [(366 - 30.713)h_{wc} + 30.713h_{wh}]/366 = 149.86 \text{ kJ/kg}$$

Steam turbine heat consumption:

$$Q_0 = (285.6 * (3540.4 - 149.86) + 40.2 * (3016.6 - 149.86) + 40.9 * (3061.1 - 149.86) + 313.4 * 3611.6 - 312.7 * 3144.5)/10^3 = 1351.241178 \text{ GJ/h}$$

Overall heat consumption:

$$Q_{tp} = (\frac{Q_0}{\eta_{hp}} + \frac{3600P_{el,GT}}{\eta_m\eta_g})/\eta_{GT} = 2530.485383 \text{ GJ/h}$$

Electric power output of steam turbine:

$$P_{el,ST} = (285.6 * 3538.3 - 272.5 * 3165.9 + 313.4 * 3607.5 + 40.9 * 3057.2 - 366 * 2382.4 - 30.713 * (3607.5 - 2382.4)\eta_m\eta_g)/10^3 = 474.1243094 \text{ GJ/h} = 131.70 \text{ MW}$$

When water recovery rate is 0%, we consider the temperature of make-up water is 20°C, then backwater enthalpy $h_{wb} = 83.6 \text{ kJ/kg}$ and the feed-water enthalpy is various of three heating modes. The specific calculation process is omitted and concrete data of thermal parameters under three heating modes are shown in Table 3. In the same way, the thermal parameters under Heating mode 7,8,9 are presented in Table 3.

Table 3. Thermal parameters for performance index calculation under different heating modes.

Heating mode	$D_h(\text{t/h})$	$h_{fw}(\text{kJ/kg})$	$Q_0(\text{GJ/h})$	$Q_{tp}(\text{GJ/h})$	$P_{el,ST}(\text{MW})$
1	36.6	154.57	1349.514021	2528.488959	134.78
2	35.649	153.81	1333.141065	2509.563429	130.09
3	30.713	149.86	1351.241178	2530.485383	131.70
4	36.6	121.13	1361.727917	2542.607024	134.78
5	35.649	121.24	1345.124821	2523.415475	130.09
6	30.713	121.80	1361.53078	2542.379153	131.70
7	0	125.3	1360.24733	2540.80554	141.73
8	18.3	139.935	1354.880676	2534.692285	138.26
9	36.6	154.57	1349.514021	2528.488959	134.78

4.2. Performance index calculation

In this section, we use the thermal parameters shown in Table 3 to calculate the performance index under different heating modes.

Case 1: Performance index calculation under different heating modes when water recovery rate is 100%

In this case, the indexes will be calculated under three heating modes when water recovery rate is 100%, so as to compare and analyze the influence of extraction pressure on performance indexes.

Heat-supply quantity: $Q_h = D_h(h_h - h_{wh}) = 36.6 * (3094.5 - 418)/10^3 = 97.9599 \text{ GJ/h}$.

Assume transmission efficiency of heat-supply network $\eta_{hs} = 1$, heat load $Q = Q_h = 97.9599 \text{ GJ/h}$.

The performance indexes are calculated by formula (10)-(14) and calculation results are presented in Table 4.

Table 4. Performance index under different heating modes when water recovery rate is 100%.

Heating mode	1	2	3
$Q_h(\text{GJ/h})$	97.9599	97.9599	97.9599
$Q_{tp(h)}(\text{GJ/h})$	113.22619	113.22619	113.22619
$Q_{tp}(\text{GJ/h})$	2528.48896	2509.56343	2530.48538
$Q_{tp(e)}(\text{GJ/h})$	2415.26277	2396.33724	2417.25919
$P_{el,ST}(\text{MW})$	134.78	130.09	131.70
$\eta_{tp(e)}$	0.58175	0.57930	0.57668
$b_{tp(e)}^s(\text{kg/kWh})$	0.21143	0.21233	0.21329
$\eta_{tp(h)}$	0.86517	0.86517	0.86517
$b_{tp(h)}^s(\text{kg/GJ})$	39.4142	39.4142	39.4142

Among the three kinds of heating modes, the heating mode 1 is low pressure extraction steam and heating mode 2 and 3 are high pressure extraction steam. In the latter two heating modes, which are extraction steam before and after reheating, extraction pressures are similar, but the extraction temperature after reheating is much higher.

According to indexes presented in Table 3, we can find that when the heat load is constant, the heat-supply thermal efficiency in different heat-supply conditions is unchanged, because this index system is on the basis of heat quantity method, in which the heat-supply heat consumption is determined by the required heat load. The heat-supply thermal efficiency is the product of unit comprehensive energy utilization efficiency η and heat-supply network transmission efficiency η_{hs} . Consequently, the heat-supply thermal efficiency is the same, owing to the working condition kept in 100% load in this calculation example.

Visibly, the power generation efficiency of low pressure extraction steam is higher than that of high pressure extraction steam. Low pressure extraction steam for heat-supply is beneficial to thermal economy of GSCC-CHP plants.

In the calculation example, the main steam flow is kept unchanged and the heat consumption is changed in different heat-supply conditions. Table 3 presents that the overall heat consumption Q_{tp} of heating mode 2 is minimum, because of the extraction steam before reheating reducing the caloric receptivity of steam in reheater. In this situation, since the extraction steam flow is far few than the main steam flow, flue gas damper is adjusting to regulate the reheat temperature and another adjustment of attemperator spray is ignored.

When the heat load is same, the power generation heat consumption $Q_{tp(e)}$ of heating mode 2 is far fewer than the others. That is the reason why that although the electric power output of heating mode 2 is the fewest, the power generation efficiency is not the lowest. The power generation efficiency of extraction steam before reheating is higher than that after reheating.

Case 2: Performance index calculation under different heating modes when water recovery rate is 0%

In this case, the indexes will be calculated under three heating modes when water recovery rate is 0%. The performance index can be compared with that in case 1, so as to compare and analyze the influence of water recovery rate on performance indexes.

The performance indexes are calculated by formula (10)-(14) and calculation results are presented in Table 5.

Table 5. Performance index under different heating modes when water recovery rate is 0%.

Heating mode	4	5	6
$Q_h(\text{GJ/h})$	97.9599	97.9599	97.9599
$Q_{tp(h)}(\text{GJ/h})$	113.22619	113.22619	113.22619
$Q_{tp}(\text{GJ/h})$	2542.60702	2523.41547	2542.37915
$Q_{tp(e)}(\text{GJ/h})$	2429.38083	2410.18928	2429.15296
$P_{el,ST}(\text{MW})$	134.78	130.09	131.70
$\eta_{tp(e)}$	0.57837	0.57597	0.57386
$b_{tp(e)}^s(\text{kg/kWh})$	0.21267	0.21355	0.21434
$\eta_{tp(h)}$	0.86517	0.86517	0.86517
$b_{tp(h)}^s(\text{kg/GJ})$	39.4142	39.4142	39.4142

As well as the trend of performance indexes shown in Table 3, Table 4 indicates that the heat-supply thermal efficiency in different heat-supply conditions is unchanged and the power generation efficiency of low pressure extraction steam is higher than that of high pressure extraction steam.

The specific comparison and discussion of performance indexes with different water recovery rate are conducted in next section.

Case 3: Performance index calculation under different heat loads

In this case, the indexes will be calculated under three heat loads, so as to compare and analyze the influence of heat load on performance indexes. The performance indexes are calculated by formula (10)-(14) and Table 6 presents performance indexes under various heat loads.

Table 6. Performance index under different heat loads

Heating mode	7	8	9
$Q_h(\text{GJ/h})$	0	48.97995	97.9599
$Q_{tp(h)}(\text{GJ/h})$	0	56.61313	113.22619
$Q_{tp}(\text{GJ/h})$	2540.8055	2534.69229	2528.48896
$Q_{tp(e)}(\text{GJ/h})$	2540.8055	2478.07916	2415.26277
$P_{el,ST}(\text{MW})$	141.73	138.26	134.78
$\eta_{tp(e)}$	0.56285	0.57206	0.58175
$b_{tp(e)}^s(\text{kg/kWh})$	0.21853	0.21502	0.21143
$\eta_{tp(h)}$	0	0.86517	0.86517
$b_{tp(h)}^s(\text{kg/GJ})$	0	39.4142	39.4142

As can be seen from Table 6, the heat-supply thermal efficiency in different heat loads is unchanged and the power generation efficiency increases with the heat load rising. The comparison and discussion of performance indexes concerning power generation in detail are conducted in next section.

4.3. Comparisons and discussions

Discussion 1: Performance index comparison concerning power generation with changed water recovery rate

To compare the performance index about power generation of different water recovery rates, the related index data selected from Table 4 and Table 5 are drawn in Fig.4.

From Fig.4, it is obvious that when the heat load remains constant, the lower water recovery rate is, the lower the power generation efficiency becomes, because the temperature of backwater is higher than that of make-up water. For this reason, the steam turbine heat consumption of low water recovery rate is more than that of high water recovery rate. Besides, changed feed-water enthalpy has almost little impact on electric power output of steam turbine. Therefore, high water recovery rate is of benefit to thermal economy of GSCC-CHP plants.

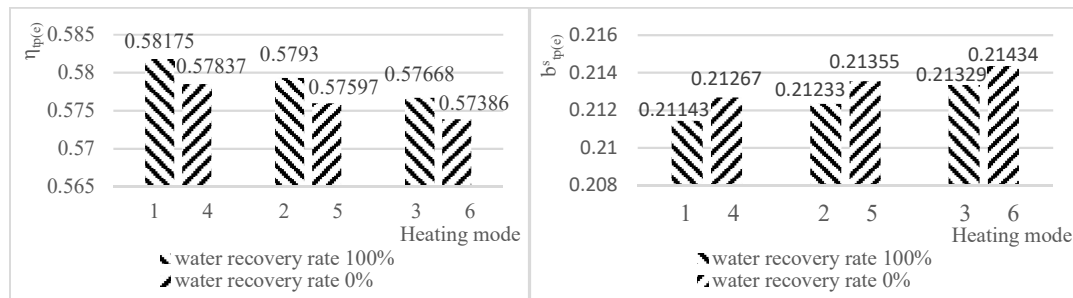


Fig.4 Variation of index $\eta_{tp}(e)$ and $b_{stp}(e)$ with different heating modes and water recovery rate.

Discussion 2: Performance index comparison concerning power generation with changed heat load

To compare the performance index about power generation of different heat loads, the related index data selected from Table 6 are drawn in Fig.5.

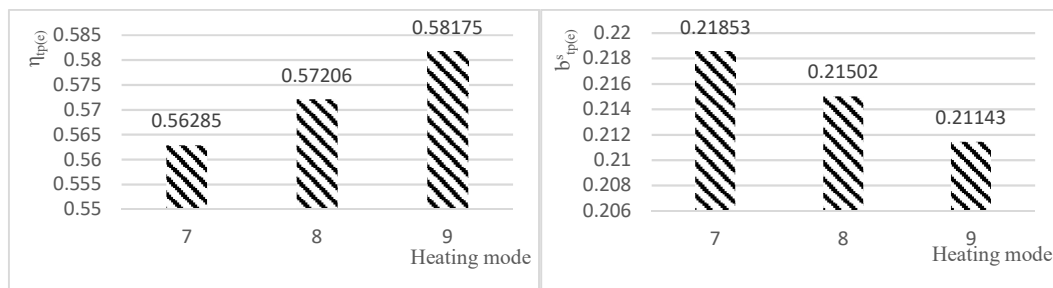


Fig.5 Variation of index $\eta_{tp}(e)$ and $b_{stp}(e)$ with different heat load.

When other conditions remain unchanged, the more heat load is, the higher power generation efficiency becomes. Though the increase of heat-supply quantity results in steam turbine power generation decreasing, the increase of heat-supply heat consumption results in power generation heat consumption reducing, of which the degree of decrease is greater than that of steam turbine power generation. It reflects the benefits of cogeneration as well.

Remark 1 It is well known that more heat load, low extraction pressure and high water recovery rate are conducive to the power generation efficiency of thermal power cogeneration plants. Similarly, we can find that the heat-supply condition of GSCC-CHP plants who has more heat load, low extraction pressure and high water recovery rate has higher power generation efficiency in the case study. Therefore, Comparative results of each index with different heat loads, extraction pressures and water recovery rates show that the variation trend of performance indexes is correct and the mathematical model of performance index has been verified to be rational.

As can be seen from the above analysis, the heat load, the extraction pressure, as well as the water recovery rate, affect the thermal economy of GSCC-CHP plants. If we want to improve the thermal economy of GSCC-CHP plants while heat load is constant, the major measure is to improve the power generation efficiency. We can take the following measures: making sure the reasonable choice of extraction steam parameter, giving priority in use of low pressure extraction steam; when extraction pressure is similar, extracting steam before reheating; trying to ensure all recovery of thermal network water, otherwise improving make-up water temperature.

5. Conclusion

This paper proposes a novel efficiency assessment criterion of GSCC-CHP plants. The key to this evaluation system is defining the two parameters in gas turbine units: energy loss ratio and energy utilization efficiency by studying the balance relationship between the input and output energy of gas turbine, which is the foundation for heat-consume apportionment. This work analogizes the heat-

consume apportionment of thermal power cogeneration plants based on the heat quantity method and it has a certain theoretical basis.

The proposed efficiency assessment criterion can be used to compute the power generation efficiency of GSCC-CHP plants. Comparisons of each index with different heat loads, extraction pressures and water recovery rates show that the variation trend of performance indexes is correct and the mathematical model of performance indexes has been verified to be rational. This study demonstrates the correctness of the assessment criterion.

Acknowledgments

This work was supported in part by National Key Technology Research and Development Program of the Ministry of Science and Technology of China under Pei-hong Wang's Grant (No.2015BAA03B02). This work was supported by Evidence-based early warming key technology and system development for power generation equipment (No.YNYJ2016043).

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