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Comparison of Exergy Efficiency of Oxygen- and Air-combustion H_2O Turbine Power Generation Systems

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Abstract: This paper evaluates the thermodynamic characteristics of two proposed power generation systems (PGSs): one is a CO_2-capturing PGS with regenerative cycle based on an oxy-fuel combustion method and the other is a PGS with a similar structure but which uses air to combust fuel. In each of the proposed PGSs, steam (H_2O), which is produced by utilizing a heat energy resource outside of the system, is used as the main working fluid for a kind of gas turbine: this feature is different from a conventional gas turbine in which air is used. Exergy efficiency is used in evaluating the thermodynamic characteristics of the PGSs, since energy with different qualities, fuel and steam, are used as input energy in each system. It is estimated under assumed conditions that the oxygen-combustion PGS (OCS) has higher exergy efficiency than the air-combustion PGS (ACS). The reasons are as follows. The turbine outlet pressure of the air-combustion PGS is higher than that of the oxygen-combustion PGS. Additional energy is consumed for the air-combustion PGS; power to compress nitrogen gas included in the air for injecting it into a combustor, heat energy to raise the nitrogen gas to the turbine inlet temperature, and power to compress the condenser outlet gas to the atmospheric pressure for exhausting it to the atmosphere. For example, from the simulation study performed, the exergy efficiency of the OCS is estimated to be 54.4%, higher by 0.87%, compared to the highest exergy efficiency (53.8%) of the ACS. CO_2 reduction characteristics of the two PGSs are also discussed.

Key words: CO_2-capture; Oxy-fuel combustion; Regenerative cycle; High efficiency; Simulation

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1. INTRODUCTION

Reduction of carbon dioxide (CO_2) emission into the atmosphere has become increasingly important to mitigating global warming. Capturing CO_2 from exhaust gas at thermal power generation systems (PGSs) is considered to be effective for drastically reducing CO_2 emission into the atmosphere.

As for CO_2 capturing technologies, there exist pre-combustion, post-combustion and oxy-combustion methods[1-3]. Among them the oxy-combustion method has the following novel features: the system based on it becomes a semi-closed system and so can capture nearly 100% of produced CO_2 in principle [4,5], and generates no thermal nitrogen[6-8]. To produce oxygen (O_2), however, equipment for O_2 production and a considerable amount of additional energy are required. This is bad for efficiency and economics of an O_2 combustion PGS.

However, if a CO_2-capturing PGS with a significantly high fuel-to-electricity efficiency (denoted by η_f) can be realized compared with a conventional PGS with an ordinary net power generation efficiency (NPGE), it makes it possible to reduce necessary fuel consumption and the required quantity of O_2 for combusting fuel to generate the same amount of electric power energy. Here, η_f is defined as the ratio of the net generated electric power energy to the fuel energy consumed in the PGS. Based on this concept, the present author proposed oxy-combustion PGSs, where heat energy included in various kinds of energy resources outside of the PGS is utilized to increase the value of η_f by producing steam and using it as the main working fluid of a gas turbine [7-20]. A more specific explanation is as follows: Ref. [8] showed that a CO_2-capturing PGS with a relatively high-efficiency can be constructed by using saturated steam, and that possibilities to achieve economic feasibility are high in cases of using steam produced by utilizing solar thermal energy[9-12], waste heat from factories [13-17] or exhaust gas from a PGS [7,18-20], on account of high fuel-to-electricity efficiency. These results are based on the fact that each proposed oxygen-combustion PGS (referred to as OCS) is more efficient than an air-combustion PGS (ACS) in which air is used instead of O_2.

No estimation of the thermodynamic characteristics of ACSs was shown in the previous published works by the authors in [7-20]. Hence, it has not been shown that the efficiency of each OCS is higher than that of the corresponding ACS. The objective of the present paper is to clarify the reasons why the OCS becomes more efficient than the ACS, by performing a thermodynamic characteristics estimation as generally as possible.

Section 2 describes the proposed OCS and ACS. Section 3 explains the premises assumed to evaluate OCS and ACS. Estimated thermodynamic characteristics are described and reasons why the OCS achieves a higher efficiency than the ACS are explained using the index of exergy efficiency of the PGS. Use of exergy efficiency is indispensable in evaluating thermodynamic characteristics, since the investigated systems use steam and fuel as the input energy, and the qualities of steam and fuel are significantly different from each other. In section 4, the CO_2 reduction characteristics of the proposed systems are discussed. Section 5 presents our conclusions.

2. OUTLINE OF PROPOSED OXYGEN AND AIR COMBUSTION POWER GENERATION SYSTEMS

2.1 Outline of Proposed Oxygen-Combustion Power Generation System

Fig. 1 shows the schematic of the proposed CO_2-capturing PGS based on an oxy-fuel combustion method. The proposed system generates power by driving a kind of gas turbine (GT). As shown in Fig. 1, steam is used as the main working fluid of the H_2O turbine in the proposed system. The steam used as the main working fluid can be obtained by various ways: by making use of solar thermal energy [9-12], waste heat from factories [13-17], exhaust gas from a GT [7,18,19] or a fuel cell [20]. The temperature of the steam is raised by utilizing recuperator (regenerator), and then increased beyond 1000 ºC by combusting fuel in a combustor using pure O_2, not the air. The resulting high temperature gas, composed mainly of H_2O and CO_2 gas, is used for driving a generator connected to the GT. The energy included in the GT outlet gas is used to raise the temperature of the steam by making use of the regenerator, and then is cooled in a condenser. Most of the steam included in the condenser inlet gas is condensed at a condenser outlet. The condensate is compressed by a feed pump and returned to a steam generator (SG) to produce the steam; the excess water, not required to return, is compressed to the atmospheric pressure (1 atm was assumed to be 101 kPa) to discharge it from the system. The condenser outlet gas includes CO_2 gas, which is produced as the result of the combustion reaction of the fuel, surplus O_2 gas, which is injected into the combustor to secure complete combustion of the fuel, and H_2O gas, which is not condensed and remains as the state of saturated steam. This condenser outlet gas is usually treated in a liquefaction device to obtain liquefied CO_2. Details of the liquefaction process are explained in section 4. Here the condenser outlet gas was assumed to be compressed to 101 kPa to compare the thermodynamic characteristics of the air-combustion PGS described in the following section. The power consumed for compression of the condenser outlet gas and the surplus water up to 101 kPa is denoted as discharge fluid compression power in the following.

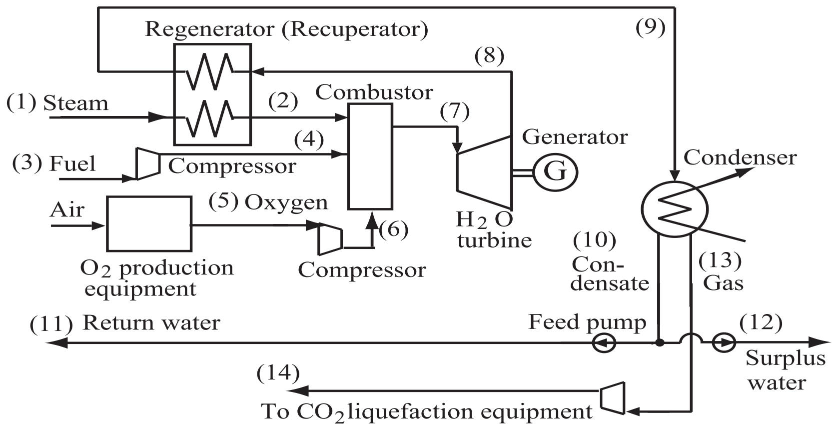


Fig. 1: Schematic of the proposed CO_2-capturing power generation system based on oxy-fuel combustion method (OCS)

In the proposed system as mentioned above, high-temperature steam whose thermodynamic property becomes gaseous is used as the working fluid of the GT. The GT in the proposed system is referred to as an H_2O gas turbine to distinguish it from a conventional GT in which air is used as the working fluid. An H_2O gas turbine is also referred to as an H_2O turbine or HT, for simplicity in the following.

2.2 Advantages of the Proposed System and Differences from Similar Pgss

The proposed system has the advantage that the GT turbine inlet temperature (TIT) can be raised higher than 1000ºC while it cannot in a conventional steam turbine power generation system (STPS). With use of a feed pump the pressure of the steam is made high when it is at the state of *liquid* (water), so that no compressing work of the H_2O *gas* by the H_2O turbine is required, although fuel and oxygen compression work is required. This feature is different from that of the conventional GT, in which a large amount of air is compressed with an air compressor; that is, from the intrinsic property of the Brayton cycle, approximately two thirds of turbine axial power output is consumed in this air compression process [21,22]. Both characteristics - the absence of an energy-consuming air compression process and the use of steam having a larger heat energy compared with air as the working fluid - make the efficiency of fuel use significantly high. Moreover, the H_2O turbine working fluid can be expanded down to a vacuum (for example, 9.81 kPa), producing a larger turbine axial power output. Hence, the value of η_f can be higher than 70%, as will be seen in the following section. Therefore, the quantity of required O_2 for fuel combustion becomes small, and thus the power required for producing O_2 becomes small to generate the same electric power energy. In the proposed system, the temperature of steam is raised by combusting the fuel, so that the generated power output becomes much greater compared with the power output obtained by using conventional STPSs. That is, obtaining much greater power is possible in the proposed system and the proposed system belongs among repowering systems [23-25]. This feature is fundamentally different from a system that captures CO_2 from flue gas by using chemical absorbent, in which the power output inevitably decreases and its power-selling income decreases [26]. The proposed system is based on GT technologies, so that it is easy to apply not only a small scale PGS [20] and a medium scale PGS [17], but also to a large scale power generation system [18,19].

The fuel is burned using O_2 and so the combustion reaction takes place in the combustor without nitrogen (N_2) gas. Hence, no thermal NO_x is produced in the proposed system when carbon hydride fuel is used [6-8].

Similar to the OCS, clean energy systems (CES) [27-31] and a PGS using the Graz cycle [31-35] have been proposed, in which H_2O (water or steam) is used as the main working fluid of turbines. It is an intrinsic property of water, however, that a large amount of heat energy is required to evaporate water [9-12], so that the energy efficiency of the CES degrades during which evaporation heat of water is supplied inside the system [31]. In a PGS using the Graz cycle, a sophisticated process is adopted to decrease fuel consumption by incorporating a heat recovery process of turbine exhaust gas and by using it for pre-heating water. Hence, the problem of needing a large heat requirement for water evaporation is alleviated for the PGS using the Graz cycle, and its efficiency can be improved over that of the CES [31]. However, this problem is not solved in the PGS using the Graz cycle as it is in the CES, whereas the OCS can avoid this problem by utilizing heat a resource outside the system.

2.3 Outline of the Air-Combustion Power Generation System

Fig. 2 shows the schematic of the proposed air-combustion PGS; this system is similar to the OCS, but is different from the OCS in the following point: fuel is burned using air, and an air compressor is needed to compress the air to supply it into a combustor. The system in Fig. 2 is referred to as ACS in the following. In the ACS, the condenser outlet gas including CO_2 gas is compressed to 1 atm and then discharged into the atmosphere as the exhaust gas; this is different from the OCS. The condenser outlet condensate is compressed and returned to a SG to produce the steam: the excess water, not returned to the SG, is compressed to 101 kPa to discharge it from the system.

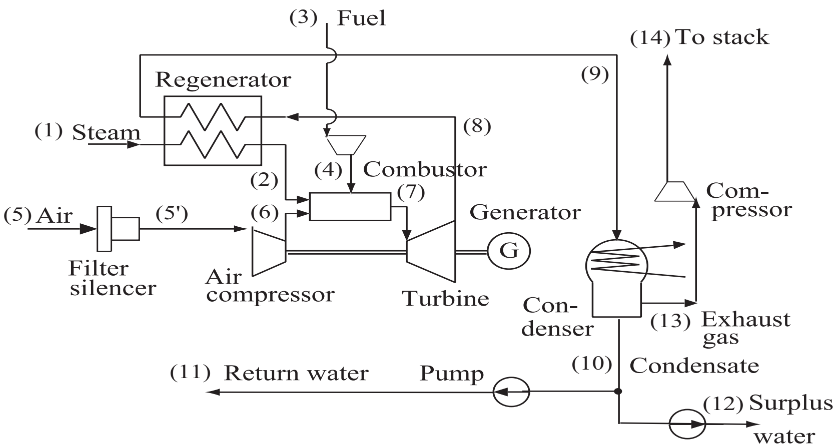


Fig. 2: Schematic of the proposed air-combustion power generation system (ACS)

3. ESTIMATION OF THERMODYNAMIC CHARACTERISTICS

3.1 Premises

Both the power generation systems investigated consist of a variety of units and fluid. The fundamental characteristics of a given system, such as the generated power and fuel consumption can be obtained by estimating the states of the fluid flowing between each unit based on the thermodynamic characteristics. Partly modified simulation models that were developed by the author of this study were employed to estimate the power generation characteristics of the two systems. For a more detailed explanation of the simulation methods used for characteristics estimation, refer to Ref. [36].

Tab. 1: Major exogenous variables and parameters used for the simulation models

Item		**OCS**	**ACS**
(a) Exogenous variables			
Saturated steam flow rate (t/h)		100	100
Saturated steam pressure (MPa)	changed from	0. 5 to 4.0 by 0.125	0. 5 to 4.0 by 0.125
Turbine inlet temperature (ºC)		1250	1250
Condenser outlet temperature (ºC)		32.55	32.55
Condenser outlet pressure (kPa)		9.81	49.0
Fuel		CH_4	CH_4
O_2, fuel and air temperature (ºC)		25	25
O_2, fuel and air pressure (bar)		1	1
(b) Exogenous parameters			
Turbine adiabatic efficiency (%)		90	90
Dryness of turbine outlet steam (%)		> 90	> 90
Compressor adiabatic efficiency (%)		75	75
Pump adiabatic efficiency (%)		75	75
Air compressor adiabatic efficiency (%)		-	85
Combustor combustion efficiency (%)		99	99
Combustor pressure loss rate (%)		5	5
Oxygen or air excess rate (%)		1	1
Generator efficiency (%)		98	98
Miscellaneous power consumption rate (%)		4	4
Unit oxygen production power (kWh/t-O2)		237.9-	
Fuel gas nozzle pressure loss rate (%)		20	20
Air injection nozzle pressure loss rate (%)		-	20
Regenerator temperature efficiency (%)		80	80
Regenerator heating side pressure loss rate (%)		5	5
Regenerator heated side pressure loss rate (%)		5	5
HRSG steam pressure loss rate (%)		10	10
CO_2 liquefaction equipment:			
Outlet temperature of chiller (ºC)		7	-
Coefficient of performance of chiller		3.5	-
Adiabatic efficiency of compressor (%)		75	-
Flow loss rate (%)		0	0

Tab. 1 shows the major exogenous variables and parameters used for estimating the characteristics of the two PGSs. The steam used as the main working fluid was assumed to be saturated steam, considering that higher pressure is desirable in general and storage of input steam using a steam accumulator becomes possible. The flow rate of the saturated steam was set at 27.8 kg/s (100 t/h), and its pressure was changed from 0.5 to 4 MPa by 0.125 MPa (the corresponding steam temperature changed from 152ºC to 250ºC) to search an optimal steam pressure that makes the maximum efficiency. The TIT was set at 1250ºC, although the higher the TIT the higher the resulting NPGE [37,38]. It should be noted that for simplification of simulation, no modeling of turbine blades cooling was carried out in performing the thermodynamic characteristics estimation, although cooling of turbine blades is carried out to keep the mechanical strength of the turbine blades under higher temperature for high temperature GTs. The reason no turbine blades cooling was assumed was that we can estimate efficiency of both systems, the OCS and ACS, with turbine blade cooling, that has a TIT of approximately 1350ºC, higher by 100 – 150 ºC compared to the set value of 1250 ºC, by performing simulation of no turbine blade cooling [18].

The condenser outlet temperature was set at 32.6ºC, but the condenser outlet pressure (P_{con}) was assumed to be 9.81 kPa, two times higher than with 4.90 kPa of the saturated water with a temperature of 32.6ºC, so that the condenser outlet gas at the OCS included noncondensable CO_2 gas approximately 50% in volume ratio. For the ACS, P_{con} was set at 49.0 kPa, five times higher, considering that N_2 gas, which is included in the air approximately four times lager in volume percentage, also remained in the condenser outlet gas.

The fuel was a natural gas consisting of CH_4, C_2H_6, C_3H_8 etc., but it was assumed to be composed solely of CH_4, although the simulate models used can deal with various kinds of fuels such as coal gas [4], liquefied petroleum gas [16], city gas [39], and pyrolysis gas [40,41]. Hence, the lower heating value of the fuel is 50.05 MJ/kg. The oxygen excess rate was set at 1%, two times higher than in the case of the oxy-combustion of hydrogen gas [42] to assure complete combustion of the fuel. The air excess rate of the ACS was set at 5%, five times higher than with the OCS, considering that O_2 only included approximately 20% in volume percentage compared with the pure O_2 gas. This assumption causes the effect that the volume composition of the residual O_2 at the condenser outlet gas of the ACS becomes approximately equal to that of the OCS, as will be shown in Sec. 3.4.

The unit oxygen production power was assumed to be 237.9 kWh/(t-O_2) for the OCS. The miscellaneous power consumption rate was assumed to be 4% [7-20,25]. Here, the miscellaneous power designates various kinds of small power, such as the lubricant oil pump power and lighting, which are consumed in the PGS but are not dealt with in the simulation models. The flow loss rates of the units, which are all negligibly small in a large scale PGS, were all assumed to be zero, for simplicity of discussion.

A small amount of CO_2 gas dissolved in the condensate was assumed to be zero, for simplicity of discussion on the CO_2 reduction effect. It should be noted that this assumption and the assumption of the no flow loss of each unit in the PGSs make it easy to calculate captured CO_2 in the OCS.

The other values in Tab. 1 are based on previous analyses performed by the present author on a variety of PGSs, and are considered to be realizable by applying present technologies [6-20].

In evaluating the efficiency of both the PGSs, exergy efficiency was adopted, since the quality of energy used as the input energy of the PGSs is different between fuel and steam. The value of the exergy of the fluid (gas, steam or water) at various points in the PGS can be calculated by the following equation.

$$E = h - h_o - T_o (s - s_o) \tag{1}$$

where:

E – exergy of the fluid (kJ/kg),

h – enthalpy of the fluid (kJ/kg),

h_o – enthalpy of the fluid under the standard ambient conditions (kJ/kg),

s – entropy of the fluid (kJ/(kgK)),

s_o – entropy of the fluid under the standard ambient conditions (kJ/(kgK)),

T_o – temperature of the standard ambient (K).

In calculating the value of exergy, 1 atm (101 kPa) and 25 ºC were assumed to be the standard ambient conditions, and electric energy was assumed to be all convertible to work or exergy.

3.2 Estimated thermodynamic characteristics when steam pressure was changed

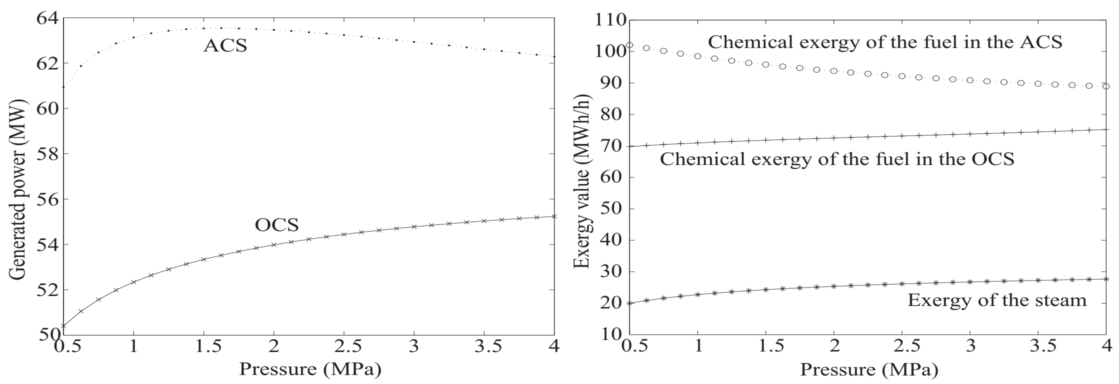
Fig. 3 shows estimated thermodynamic characteristics of the proposed PGSs, when saturated steam pressure (P_{stm}) was changed. As shown in Fig. 3(a), the net generated power (NGP) of the OCS is estimated to increase with P_{stm}. However, the NGP of the ACS is estimated to increase until P_{stm} = 1.625 MPa but then to decrease with P_{stm}. Here the NGP of the OCS and the ACS is calculated in the following equations, respectively:

$$W_{OCS} = W_{HT} - W_{FC} - W_{DP} - W_{FP} - W_{M} - W_{O2P} - W_{O2C} \tag{2}$$

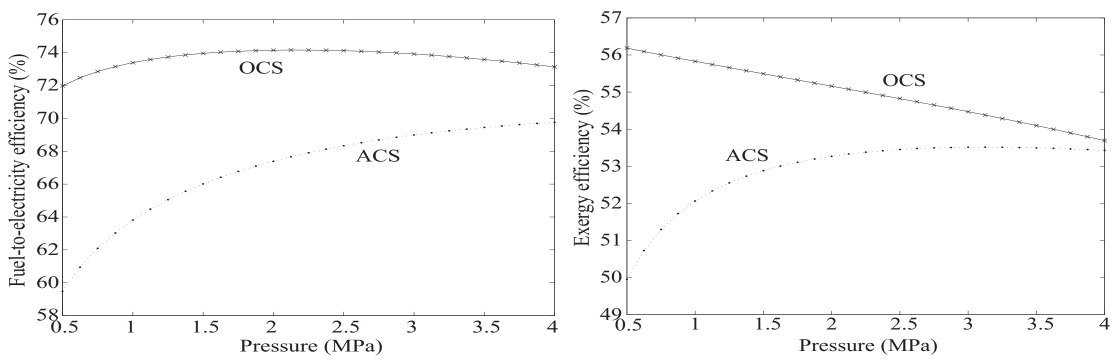
$$W_{ACS} = W_{HT} - W_{FC} - W_{DP} - W_{FP} - W_{M} \tag{3}$$

where:

W_{OCS} – NGP of the OCS (kWh/h),
W_{HT} – generated power at the HT generator (kWh/h),
W_{FC} – fuel compression power (kWh/h),
W_{DP} – discharge fluid compression power (kWh/h),
W_{FP} – feed pump power (kWh/h),
W_{M} – miscellaneous power (kWh/h),
W_{O2P} – O_2 production power (kWh/h),
W_{O2C} – O_2 compression power (kWh/h),
W_{ACS} – NGP of the ACS (kWh/h).



(a) Net generated power (b) Fuel chemical exergy and steam exergy



(c) Fuel-to-electricity efficiency (d) Exergy efficiency

Fig. 3: Major estimated thermodynamic characteristics of the oxygen and air combustion power generation system when P_{stm} is changed

The reason for the increase in the NGP for the OCS is that the expansion work of the HT increases with P_{stm}. The main reason for the decrease in the NGP for the ACS when P_{stm} is higher than 1.625 MPa is that the fuel consumption of the ACS decreases with P_{stm} as shown in Fig. 3(b), although the turbine axial power of the HT increases with P_{stm}. Another reason is that air and fuel compression power increases with P_{stm}.

As shown in Fig. 3(b), the exergy of steam is estimated to increase with P_{stm}. The chemical exergy of the OCS is estimated to increase with P_{stm}. This is because the heat energy required for raising the temperature of the combustor inlet steam up to 1250ºC increases with P_{stm}, since the HT outlet (the regenerator inlet) gas temperature decreases with P_{stm}. On the other hand, chemical exergy of the ACS is estimated to decrease with P_{stm}, although the regenerator inlet gas temperature decreases with P_{stm}. This is because the heat energy required for raising the temperature of the combustor inflow fluid, steam, fuel and air, up to 1250ºC decreases with P_{stm}, since the air flow rate is remarkably greater than the O_2 flow rate of the OCS, and the air compressor outlet air temperature increases with P_{stm}, although the combustor inlet steam temperature decreases with P_{stm}.

As shown in Fig. 3(c), the fuel-to-electricity efficiency of the OCS, which can be calculated from Eq. (4), is estimated to increase until P_{stm} = 2.125 MPa but then to decrease with P_{stm}. On the other hand, the fuel-to-electricity efficiency of the ACS, which can be calculated from Eq. (5), is estimated to increase with P_{stm}.

$$\eta_{f\_OCS} = W_{OCS}/(G_f \times L_f) \times 100 \tag{4}$$

$$\eta_{f\_ACS} = W_{ACS}/(G_f \times L_f) \times 100 \tag{5}$$

where:

η_{f_OCS} – fuel-to-electricity efficiency of the OCS (%),

G_f – fuel flow rate (t/h) of the system,

L_f – lower heat value of the fuel (kWh/t) of the system,

η_{f_ACS} – fuel-to-electricity efficiency of the ACS (%).

The reason for the increase in η_{f_OCS} until P_{stm} = 2.125 MPa is that W_{OCS} increases with P_{stm} but the required fuel quantity also increases with P_{stm} as seen from Fig. 3(b). The reason that η_{f_ACS} increases with P_{stm} is as follows: the fuel quantity required is decreased with P_{stm} as seen from Fig. 3(b), although W_{ACS} decreases with P_{stm} when P_{stm} surpasses 1.625 MPa.

As shown in Fig. 3(d), the exergy efficiency of the OCS, which can be calculated from Eq. (6), is estimated to decrease with P_{stm}. This is because the required fuel amount is increased owing to the reason mentioned above, although the NGP is increased with P_{stm}. For a more detailed discussion on the estimated of the exergy efficiency of the OCS, see the following section.

$$\eta_{E\_OCS} = W_{OCS}/(G_f \times C_f + G_f \times E_f + G_{stm} \times (E_{stm} - E_w) + G_{O2} \times E_{O2}) \times 100 \tag{6}$$

where:

η_{E_OCS} – exergy efficiency of the OCS (%),

C_f – chemical exergy of the compressor inlet fuel (kWh/t),

E_f – exergy of the compressor inlet fuel (kWh/t),

G_{stm} – flow rate of the regenerator inlet steam (t/h),

E_{stm} – exergy of the regenerator inlet steam (kWh/t),

E_w – exergy of the return water (kWh/t),

G_{O2} – flow rate of the compressor inlet O_2 (t/h),

E_{O2} – exergy of the compressor inlet O_2 (kWh/t).

The exergy efficiency of the ACS, which can be calculated from Eq. (7), is estimated to increase until P_{stm} = 3.125 MPa but then to decrease slightly with P_{stm}. This is because the NGP has a maximum value when P_{stm} is changed and the required fuel amount decreases with P_{stm}. The exergy efficiency of the ACS is estimated to have a maximum value of 53.5% at P_{stm} = 3.125 MPa.

$$\eta_{E\_ACS} = W_{ACS} /(G_f \times C_f + G_f \times E_f + G_{stms} \times (E_{stm} - E_w) + G_a \times E_a) \times 100 \qquad (7)$$

where:

η_{E_ACS} – exergy efficiency of the ACS (%),

G_a – flow rate of the filter silencer inlet air (t/h),

E_a – exergy of the filter silencer inlet air (kWh/t).

3.3 Discussion on Exergy Efficiency of the OCS

As can be seen from Fig. 3(d), it is estimated that the higher the P_{stm}, the lower the exergy efficiency of the OCS when P_{stm} is changed from 0.5 to 4 Mps. This feature of the OCS might be difficult to understand. The followings are more detailed explanation on the estimated exergy efficiency.

When the value of P_{stm} is small, the HT outlet temperature is high and thus combustor inlet steam temperature becomes high: this makes required fuel quantity to raise the steam temperature up to 1250ºC small compared to a case with higher P_{stm}, as seen from Fig. 3(b). That is, the lower the P_{stm}, the fewer the required fuel consumption. This is the main reason why the value of η_{E_OCS} is high at lower P_{stm}. It is needless to say that there exists an optimal P_{stm} that makes η_{E_OCS} the highest. It was estimated that the OCS has a maximum exergy efficiency of 56.5% at remarkably low P_{stm}, that is P_{stm} = 0.06 MPa, when optimal P_{stm} was searched by changing the value of P_{stm} by 0.01 MPa. It should be noted that an OCS with higher exergy efficiency does not always mean a favorable PGS, since its NPG becomes smaller as seen from Fig. 3(a). The NGP of the OCS with the highest exergy efficiency was estimated to be 43.2 MW, smaller by 21.2% compared with that when P_{stm} was set, for example, at 3.125 MPa.

3.4 Comparison of Estimated Power Generation Characteristics at the Same P_{stm}

Tab. 2: Estimated state values at major points in the OCS

Points Kind of fluid	Temperature (ºC)	Pressure (kPa)	Enthalpy (kJ/kg)	Flow rate (t/h)
(1) Regenerator inlet steam	236	3125	2804	100
(2) Combustor inlet steam	293	2969	2977	100
(3) Compressor inlet fuel	25.0	101	0.00	5.34
(4) Compressor outlet fuel	410	3711	1100	5.34
(5) Compressor inlet oxygen	25.0	101	0.00	21.5
(6) Compressor outlet oxygen	655	3711	632	21.5
(7) Turbine inlet gas	1250	2820	2579	127
(8) Turbine outlet gas	308	10.3	513	127
(9) Condenser inlet gas	235	9.81	376	127
(10) Condenser outlet water	32.6	9.81	136	106
(11) Return water	39.6	3472	169	100
(12) Surplus water	39.2	101	164	5.87
(13) Condenser outlet gas	32.6	9.81	8.65	21.0
(14) Condenser outlet compressed gas	303	101	346	21.0

Tab. 2 and 3 show the estimated results of the state values of the fluid at major points of the OCS and the ACS, respectively, when P_{stm} is set at 3.125 MPa, the pressure makes the exergy efficiency of the ACS the highest. It should be noted in Tab. 2 and 3 that the enthalpy of water and steam are expressed by taking the enthalpy of the saturated water at temperature 0ºC as zero, and the enthalpy of the gas by taking the enthalpy of the gas at temperature 25ºC as zero, according to the custom in industries. We can see from Tab. 3 and 4 that the temperature of the compressor outlet fluid becomes higher than that of the compressor inlet fluid and that a large amount of H_2O gas included in the condenser inlet gas is condensed at the condenser.

Tab. 3: Estimated state values at major points in the ACS

Points Kind of fluid	**Temperature** (ºC)	**Pressure** (kPa)	**Enthalpy** (kJ/kg)	**Flow rate** (t/h)
(1) Regenerator inlet steam	236	3125	2804	100
(2) Combustor inlet steam	380	2969	3186	100
(3) Compressor inlet fuel	25.0	101	0.00	6.54
(4) Compressor outlet fuel	410	3711	1100	6.54
(5) Air	25.0	101	0.00	119
(5') Compressor inlet air	25.0	98.3	0.00	119
(6) Combustor inlet gas	621	3711	641	119
(7) Turbine inlet gas	1250	2820	2088	226
(8) Turbine outlet gas	416	51.6	593	226
(9) Condenser inlet gas	309	49.0	424	226
(10) Condenser outlet water	32.6	49.0	136	109
(11) Return water	33.0	3472	141	100
(12) Surplus water	32.7	101	137	8.97
(13) Condenser outlet gas	32.6	49.0	8.02	117
(14) Exhaust gas	121	101	103	117

Tab. 4: Estimated results of partial pressure in the condenser outlet gas

Item	**OCS**	**ACS**
Combustor inlet steam pressure (MPa)	3.125	3.125
Condenser inlet gas		
Pressure (kPa)	9.81	49.0
Temperature (ºC)	235	309
Volume composition (%)		
H_2O	94.8	63.7
CO_2	5.08	4.03
N_2	0	31.8
O_2	0.102	0.403
Condenser outlet gas		
Pressure (kPa)	9.81	49.0
Temperature (ºC)	32.55	32.55
Volume composition (%)		
H_2O	50.1	10.0
CO_2	49.0	9.99
N_2	0	79.0
O_2	0.979	0.999
Partial pressure (kPa)		
H_2O	4.91	4.91
CO_2	4.80	4.90
N_2	0	38.7
O_2	0.096	0.490

Tab. 4 shows the estimated volume compositions of the condenser inlet and outlet gas, and the partial pressure of the condenser outlet gas at both the systems, when the value of P_{stm} was set at 3.125 MPa. In Tab. 4, rare gases such as argon were included in the N_2 gas. We can see from Tab. 4 that the O_2 volume composition is nearly equal in both the PGSs: 0.979% and 0.999%, respectively, since a large amount of H_2O gas included at the condenser inlet gas is condensed at the condenser outlet. It is also seen that the partial pressure of the CO_2 gas in the condenser outlet gas is nearly equal in both the PGSs, 4.80 and 4.90 kPa, and that the partial pressure of the H_2O gas is estimated to be equal to 4.91 kPa.

Tab. 5 shows the major estimated thermodynamic characteristics of the proposed systems OCS and ACS, when P_{stm} is set at 3.125 MPa. A detailed explanation of Tab. 5 is presented in the following section.

Tab. 5: Major estimated characteristics of the proposed systems OCS and ACS

Item	OCS	ACS	Ratio*
Optimal saturated steam pressure (MPa)	3.125	3.125	1.00
Steam temperature (ºC)	236	236	1.00
Steam exergy (MWh/h)	26.9	26.9	1.00
Fuel consumption (t/h)	5.34	6.54	1.22
Chemical exergy of Fuel (MWh/h)	74.0	90.6	1.22
O_2 or air flow rate (t/h)	21.5	119	5.54
Turbine inlet gas flow rate (t/h)	127	226	1.78
Turbine inlet pressure (MPa)	2.82	2.82	1.00
Turbine expansion ratio	273	54.6	0.200
Turbine axial power output (MW)	72.6	93.6	1.23
Air compression power (MW)	-	21.2	-
Generator power output (MW)	71.1	70.9	0.997
Specific power (MW/(t/h))	0.560	0.314	0.560
Inhouse power (MW)	16.2	8.06	0.496
Inhouse power rate (%)	22.9	11.4	0.497
(Fuel compression power rate (%))	2.30	2.82	1.28
(O_2 prod. and comp. power rate (%))	12.5	-	-
(Discharge fluid comp. power rate (%))	2.77	4.35	1.57
(Feed pump power rate (%))	1.27	0.198	0.156
(Miscellaneous power rate (%))	4.00	4.00	1.00
Net generated power (MW)	54.9	62.9	1.15
Fuel-to-electricity efficiency (%)	73.8	69.1	0.936
Exergy efficiency (%)	54.4	53.5	0.984

*Ratio denotes the value of each estimated value of the OCS divided by the corresponding value of the ACS.

3.5 Comparison of exergetic efficiencies of OCS and ACS

We can see from Fig. 3(d) that the exergy efficiency of the OCA is estimated to be higher than that of the ACS, when the two systems are compared at the same P_{stm}. The estimated results might be difficult to understand, considering that extra energy is required for the OCS to produce the O_2 compared with the ACS. However, it should be noted that the ACS has the following shortcomings compared with the OCS:

(a) More fuel is required for the ACS than for the OCS to raise the combustor outlet gas temperature up to the designated TIT, because an approximately four times larger N_2 gas volume is included in the air to combust the fuel compared with the O_2 gas.

(b) The turbine expansion ratio of the ACS becomes smaller than that of the OCS, because the HT outlet pressure of the ACS is higher than that of the OCS on account of the N_2 gas inclusion in the condenser outlet gas.

(c) The generator power output per unit working fluid flow (specific power) of the ACS becomes significantly smaller than that of the OCS. This is because a considerable amount of turbine axial power is consumed to compress the N_2 gas included in the air for injecting it into the combustor for the ACS, and from the shortcoming of (b) mentioned above.

(d) In the ACS, power is required for compressing the condenser outlet *gas*, which includes a considerably large amount of N_2 gas, to discharge it into the atmosphere; this gas compression power becomes small for the OCS, since no N_2 gas is included in the condenser outlet gas of the OCS.

Owing to these shortcomings of the ACS, the exergy efficiency of the ACS becomes lower than that of the OCS.

The followings are specific explanations of the results shown in Tab. 5 showing that the OCS is more efficient than the ACS.

The regenerator inlet steam conditions and the TIT are the same for both the systems, but the fuel consumption of the ACS is estimated to be larger than that of the OCS, owing to the shortcoming of (a); that is, 1.48 kg/s (6.54 t/h) of the ACS is 1.22 times larger than that of the OCS. Hence, the air flow rate of the ACS is also estimated to be larger (5.54 times larger) than the O_2 flow rate of the OCS. These make the turbine inlet flow of the ACS 1.78 times larger compared to the OCS. Thus, the turbine axial power output of the ACS is estimated to become 1.23 times larger than that of the OCS, despite the shortcoming of (b) of the ACS; that is, the turbine expansion ratio of the ACS is one fifth that of the OCS. The estimated generator power output 70.9 MW of the ACS is estimated to be smaller than 71.1 MW of the OCS, although the turbine axial power output of the ACS is 1.23 times greater than that of the OCS. This is because an air compression power of 21.2 MW is consumed for the ACS (shortcoming of (c)). The specific power (MW/(t/h)) of the ACS is estimated to be 0.314 compared with that (0.560) of the OCS .

For the ACS, the discharge fluid compression power rate is estimated to be 1.57 times larger than that of the OCS, owing to the shortcoming of (d), as shown in Tab. 5. This makes the difference between the inhouse power rates of the ACS and the OCS small; that is, the inhouse power rate of the ACS is estimated to be 11.4% compared with 22.9% of the OCS, although the O_2 production and compression power rate is estimated to be 12.5% for the OCS. It is thus true that the inhouse power rate of the ACS is estimated to be smaller than that of the OCS, and that the NGP 62.9 MW of the ACS is calculated to be larger than that (54.9MW) of the OCS. However, the difference between the NGP of the ACS and that of the OCS is not enough to compensate for the larger consumption of the fuel in the ACS. As shown in Tab. 5, the fuel-to-electricity efficiency is estimated to be 69.1% for the ACS and 73.8% for the OCS, and the exergy efficiency 53.5% of the ACS is estimated to be lower than that (54.4%) of the OCS.

4. DISCUSSION ON CO_2 REDUCTION CHARACTERISTICS

It is easy to capture CO_2 from the condenser outlet gas for the OCS. The amount of captured CO_2 gas is large, so that the captured CO_2 is liquefied for volume reduction in most cases. In this section, the method used for CO_2 liquefaction is briefly explained. The estimated CO_2 reduction characteristics are also explained.

4.1 Premises

An outline of the CO_2 liquefaction process adopted in this research is as follows. The separated gas from the condensate at the condenser outlet is first cooled, for example to 7ºC, with use of a refrigerating machine (chiller) to remove the saturated steam included in the gas for reducing compression power, and then is compressed to a high pressure, for example to 140 kg/cm^2 (13.7 MPa) using a multi-stage compressor. The compressed gas is adiabatically expanded at the compressor outlet to obtain liquefied CO_2. A small amount of O_2 gas, which has remained in the combustor outlet gas to secure complete combustion of the fuel, can also be obtained at the multi-stage compressor outlet. The obtained O_2 gas can be reused to decrease O_2 production power.

The liquefied CO_2 will be used for oil recovery enhancement or reused as carbon resource for synthesizing chemicals such as methanol from hydrogen if it can be inexpensively obtained by using renewable energy, for example solar energy in the future [43,44], or will be sequestrated under the ground or on the deep sea bottom [45].

Although various methods have been proposed to estimate the CO_2 reduction effect, the method described in Ref. [7] was adopted in the present study. That is, the amount of CO_2 reduction is calculated as follows.

It was assumed that a PGS with the NPGE of 50% was adopted as a reference system to estimate the CO_2 reduction effect. Let denote the amount of CO_2 generated when 1 kWh of the fuel is burned as U_{CO2} [kg-CO_2/kWh], the NGP of the OCS as W_{OCS} [MW], the NGP of the ACS as W_{ACS} [MW], the NPGE of the reference system as η_{REF}, the NPGE of the OCS as η_{OCS}, and the NPGE of the ACS as η_{ACS}. It should be noted that η_{OCS} and η_{ACS} are identical to η_{f_OCS} and η_{f_ACS} defined in Eqs.(4) and (5), respectively, and that η_{REF} is equal to fuel-to-electricity efficiency of the reference system, denoted by η_{f_REF}, which can be calculated from the following equation.

$$\eta_{f\_REF} = W_{REF} /(G_f \times L_f) \times 100 \tag{8}$$

The annual amount of CO_2 reduction in the OCS, denoted as G_{RED_OCS} [t/y], and the amount of CO_2 reduction of the ACS, denoted as G_{RED_ACS} [t/y], are respectively calculated as follows.

$$G_{RED\_OCS} = W_{OCS} / \eta_{f\_REF} \times U_{CO2} \times 24 \times 365 \times R_{OP} /1000 \tag{9}$$

$$G_{RED\_ACS} = W_{ACS} (1/ \eta_{f\_REF} - 1/ \eta_{f\_ACS}) \times U_{CO2} \times 24 \times 365 \times R_{OP} /1000, \tag{10}$$

where R_{OP} denotes the annual operation rate of the PGS.

In Tab. 1, the values of the exogenous parameters used for the CO_2 reduction characteristics estimation are also shown. In calculating the annual CO_2 reduction quantities, R_{OP} was assumed to be 0.667, considering the actual values of Japan in recent years.

4.2 Estimated CO_2 reduction characteristics

Fig. 4 shows the estimated amount of CO_2 reduction in the OCS and the ACS. We can see from Fig. 4 that the OCS has an excellent CO_2 reduction effect compared with the ACS.

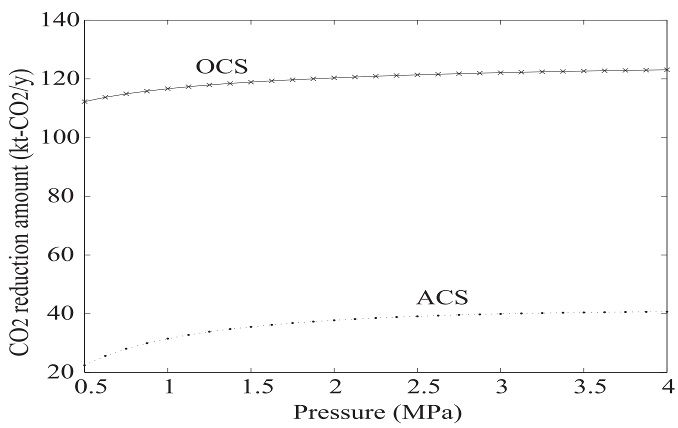


Fig. 4: Estimated CO_2 reduction amount of the oxygen and air combustion power generation system when P_{stm} is changed

The major estimated CO_2 reduction characteristics of the OCS and the ACS are shown in Tab. 6 when P_{stm} is set at 3.125 MPa. As shown in Tab. 6, the amount of CO_2 reduction in the OCS is estimated to be 122 kt-CO_2/y, 3.05

times greater than 40.1 kt-CO_2/y of the ACS, but the exergy efficiency of the OCS is estimated to be 52.6%, compared with 53.5% of the ACS, degradation by 0.933%, owing to liquefaction power of the captured CO_2.

Tab. 6: Major estimated characteristics of the CO_2 liquefaction equipment and CO_2 reduction effect

Item	OCS	ACS
Obtained condensate (t/h)		-
Recovered O_2 (t/h)		-
Captured liquefied CO_2 (t/h)		-
CO_2 liquefaction power (MW)	1.86	-
(CO_2 liquefaction power rate (%))	2.62	-
Net generated power (MW)	53.0	62.9
Amount of captured CO_2 (kt-CO_2/y)	85.6	-
CO_2 reduction amount (kt-CO_2/y)	122	40.1
Exergy efficiency (%)	52.6	53.5

5. CONCLUSIONS

In the present paper, the thermodynamic characteristics of the proposed oxygen- and air-combustion power generation systems, in which saturated steam is used as the main working fluid of an H_2O turbine, were evaluated. The evaluated systems were the O_2-combustion and the air-combustion H_2O turbine PGS with a TIT of 1250ºC.

From the estimated thermodynamic characteristics, the following characteristics were indicated:

(1) Between the OCS and ACS using steam with the same pressure, denoted by P_{stm}, the exergy efficiency of the ACS became lower that that of the OCS. The reasons were as follows: (a) heating energy of the N_2 gas, which is included in the air to combust the fuel, from the air compressor outlet temperature to the TIT is required for the ACS; (b) the condenser outlet pressure of the ACS becomes higher that of the OCS, owing to N_2 gas included in the air; (c) additional compression power of the N_2 gas is required for the ACS to inject it into the combustor; (d) compression power of the N_2 gas, which is included in the condenser outlet gas, is required for the ACS to discharge it into the atmosphere. It was shown from the estimated results obtained by changing the value of P_{stm} that these characteristics were the reasons the exergy efficiency of the OCS was estimated to be higher than that of the ACS. Exergy efficiency of the OCS was estimated to be 54.4%, higher by 0.87%, compared to that (53.5%) of the ACS, when P_{stm} was set at 3.125 MPa that makes exergy efficiency of the ACS the highest.

(2) The amount of CO_2 reduction in the OCS was estimated to be significantly large compared with that of the ACS. From the simulation study, the amount of CO_2 reduction in the OCS was estimated to be 122 kt-CO_2/y, 3.05 times greater than that of the ACS, when P_{stm} was set at 3.125 MPa; exergy efficiency was estimated to be 52.6% compared to that (53.5%) of the ACS, degradation by 0.93%, owing to liquefaction power of the captured CO_2. Hence, the OCS can be expected to be one of excellent PGSs when severe CO_2 emission constraints are imposed on PGSs.

The author is expecting that the results will contribute to a wide installation of oxy-fuel combustion PGSs with CO_2-capture in the field of power generation.

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