A new Rankine cycle for hydrogen-fired power generation plants and its exergetic efficiency

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Abstract: A novel power generation cycle is proposed in this paper taking hydrogen as fuel and using steam generated by hydrogen firing as working fluid. The progress of the development work and side issues such as the application of hydrogen combustion turbines to environmentally clean fossil fuel power plants for early commercialisation of the system are reviewed. We propose the hydrogen-fired Rankine cycle as similar to (C) type developed earlier by Hisadome et al. and Sugishita et al. and then making a new design of it by increasing the performance characteristics and efficiencies with (reheating, regenerative and recuperation) of the working fluid of the bottoming cycle respectively, and in this case we present two types (C1 and C2). In the case of type C2 the cycle is called the 'New Rankine Cycle'. These cycles are also compared with the Rankine cycle of type (C) for hydrogen-fired to show the advantages of the performance characteristics of the new design at which the highest value of exergetic efficiency reaches 63.58% as HHV at 1700°C of the combustor discharge temperature. These cycles are analysed through thermodynamics, particularly by exergy analysis, and the performance characteristics of the cycles are also studied.

Keywords: exergy analysis; gas turbine cycle; hydrogen-fired; Rankine cycle; thermodynamics.

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1 Introduction

The world's population is expected to increase to ten billion in the 21st century, accompanied by a rapid increase in energy consumption. This will bring about a shortage of fuel resources and global warming caused by CO_2 emissions. The basic solution to these problems will be to use energy resources more efficiently and to switch from fossil fuels to nuclear energy and renewable energy (Eiichi et al., 2002; Hisadome et al., 1998; Hyprotech, 2002; Jerich et al., 1989, 1991; Jericha, 1985; NEDO-WE-NET, 1997; Soufi et al., 2003; Sugishita et al., 1996, 1997; Wang and Fujii, 1996). The WE-NET (World Energy Network), a part of 'New Sunshine Project', which is a Japanese government sponsored programme through New Energy and Industrial Technology Development Organization (NEDO) aimed at solving energy and environmental problems in the world. The 'Hydrogen Combustion Turbine', which is a subtask of WE-NET, has been investigated under contracts with Japan Power Engineering and Inspection Corporation (JAPEIC) and the Central Research Institute of Electric Power Industry (CRIEPI) since 1993 (Hisadome et al., 1998; NEDO-WE-NET, 1997; Sugishita et al., 1996, 1997). The hydrogen combustion turbine cycle used for an industrial power plant with the greater power capacity has not yet been realised in the world.

The hydrogen storage system and transportation system are also going to be prepared simultaneously (Hisadome et al., 1998). It will be effective in reducing greenhouse gases by not generating CO_2. Fossil fuel like natural gas and oil is not infinite, therefore, it is possible for industrial power plants to be adapted in the near future. In 1994 three different closed hydrogen combustion turbine cycles were evaluated. These were the Topping Extraction Cycle (A) (Jerich et al., 1989, 1991; Jericha, 1985) and the Bottoming Reheat Cycle (B) and Rankine Cycle (C) (NEDO-WE-NET, 1997). The results of this study showed the Topping Extraction Cycle (A) to be the best cycle (Sugishita et al., 1996). Furthermore, the Topping Recuperation Cycle (D) was designed to be a highly efficient and highly feasible cycle suitable for high combustion temperatures (1700°C) (Hyprotech, 2002). Subsequent to this research, the Closed Circuit Cooled Topping Recuperation Cycle (G) with the high temperature turbine cooled by steam was designed to be more efficient than the abovementioned cycles (Hisadome et al., 1998). This study in this paper presents the system configurations and performance characteristics of the proposed new design of the Rankine cycle (C) (Hisadome et al., 1998; Sugishita et al., 1997), with reheat and recuperation to show the higher efficiency and performance characteristics of this cycle according to the new design. We propose the hydrogen-fired Rankine cycle as similar to (C) type (Hisadome et al., 1998; Sugishita et al., 1997), and then we will modify it by increasing the performance characteristics and efficiencies (reheating, regenerative and recuperation) respectively of the working fluid of the bottoming cycle, and in this case we present two types (C1 and C2). In the case of type C2 the cycle will be called the 'New Rankine Cycle'. These cycles will also be compared with the Rankine cycle (C) for hydrogen-fired, to show the advantages of the performance characteristics of the new design which can reach the highest value of exergy efficiency at 63.58% as HHV at 1700°C of the combustor discharge temperature. These cycles will be analysed by the thermodynamics theory as well as for exergy analysis, and the performance characteristics of the cycles will be studied. The effects of the different parameters, such as the discharge temperatures and pressures of the combustors of high and low pressure and also the effect of pressure of the condenser on the performance of the cycles, will also be clarified.

This study will be one of the most advanced studies in this field as it depends on the calculation method of exergy analysis which cannot be found in the earlier adapted studies in this field, which considered only the energy analysis in order to calculate the balance of the hydrogen-fired plant.

2 Technology and arrangement of the hydrogen-fired cycle

2.1 The exergy analysis of the hydrogen-fired cycle

The Exergy Method is an alternative, relatively new technique based on the concept of exergy, loosely defined as a universal measure of the work potential or quality of different forms of energy in relation to a given environment. An exergy balance applied to a process or an entire plant tells us how much of the usable work potential, or exergy, supplied as the input to the system under consideration has been consumed (irretrievably lost) by the process. The loss of exergy, or irreversibility, provides a generally applicable quantitative measure of process inefficiency. Analysing a hydrogen-fired plant indicates the total plant irreversibility distribution among the plant components, pinpointing those contributing most to overall plant inefficiency.

Unlike the traditional criteria of performance, the concept of irreversibility is firmly based on the two main laws of thermodynamics. The exergy balance for a plant, from which the irreversibility rate of a steady flow process can be calculated, can be derived by combining the steady flow energy equation (First Law) with the expression for the entropy production rate (Second Law), Although the Second Law is not used explicitly in the Exergy Method, its application to process analysis demonstrates the practical implications of the Second Law. Thus studying different forms of irreversibilities and their effect on plant performance, gives a better and more useful understanding of the Second Law than studying its statements and corollaries (Kotas, 1995).

In general, for the open-loop system, the available energy, that is called as the exergy, is expressed as:

$$e = h - h_0 - T_0(s - s_0) \tag{1}$$

where the subscript 0 denotes the external standard environmental state.

This value expresses the ability of the maximum work, which the working fluid possesses until the external standard state, based on the second law of the thermodynamics. The available energy, which is supplied to any system, is utilised as the useful work but a part is extinguished due to the irreversible process, which loss is called as 'Lost Work (LW)'.

$$LW = \Delta E_1 - \Delta E_2 \tag{2}$$

where $\Delta E = G\,(e_1 - e_2)$.

According to the abovementioned formulae the exergy balance and lost work for each respective component of the plant were calculated. Additionally, the entire calculations of the operating system were done by software of HYSYS (Hyprotech, 2002), for power generation plants.

2.2 The arrangement of the hydrogen-fired cycle

The stoichiometric flow of H_2 and O_2 reacting without cooling would lead to an excessively high combustion temperature of 3600°C. Although at this temperature level, the reaction of H_2 and O_2 would also be led to far to a high dissociation and the danger of poor combustion efficiency, it would be a high combustion efficiency and less dissociation if the combustion temperature is limited to 2000°C. It is clear that the gas turbine cycle may not be suitable for combustion of H_2/O_2. There will be lots of heat at a high temperature ejecting even though regenerative heating is undertaken. It is seen that the combined cycle may reach a higher temperature. If the product of reaction is directly used as the working fluid, it is evident that some water/steam, with a mass flow equal to H_2/O_2, has to be rejected. Thus the combined cycle must be an open circuit cycle. Certainly it may be an open circuit cycle/closed cycle using two working fluids, usually in an open 'Brayton'/closed 'Rankine' combination. If so, there may be heat ejected at a high temperature from the topping cycle. The G cycle mentioned above is an excellent cycle for H_2/O_2, as fuel. It is an integrated cycle of the 'Brayton' and 'Rankine' cycles. In this cycle, the merit of combustion of H_2/O_2 is utilised. The irreversibility in the combustion chamber has an important effect on the overall efficiency. The entry temperature of coolant (steam) for the combustion chamber should be as high as possible. In other words, this may reduce the irreversibility of heat transfer in the combustion chamber. As to cycle, rise of entry temperature of coolant (steam) for the combustion chamber means an increase in the mean temperature of the cycle; this may lead to greater efficiency (Wang and Fujii, 1996, 1997). To get a higher cycle efficiency, the following ways are usually utilised:

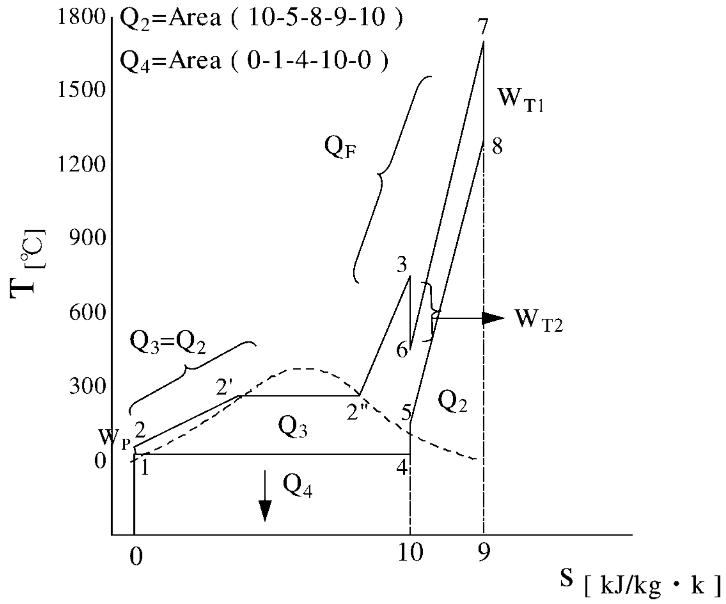
1. regenerative heating, maybe both increasing mean topping (high) temperature and decreasing bottoming (low) mean temperature of cycle
2. reheat, aim to increasing mean topping (high) temperature of cycle
3. arising the topping pressure
4. reducing the irreversibilities of components in cycle, e.g. rising the efficiencies of turbines and compressors
5. reducing the irreversibilities of the inner process, e.g. optimising heat transfer.

If all of the working fluid at point 5 expands to point 4 rather than using the compressor of topping cycle, it would be called the direct-expansion cycle as shown in Figure 1. In our study we will focus only on the type of direct-expansion as the basic condition of the proposed study. Now, let us compare these two cycles with the same states of working fluid in the cycles. Hence, only part of the supplied heat for the Rankine cycle will be ($Q_2 = Q_3$) which can be contributed by the exhausted gas from the gas turbine. All the heat supplied for the direct-expansion cycle is:

$$W_P + Q_F = W_{T1} + W_{T2} + Q_4 \tag{3}$$

where $Q_F = (W_{T1} + W_{T2} - W_P) + Q_4 = W_{GN} + Q_4$ and $W_{T1} + W_{T2} - W_P = W_{GN}$.

Figure 1 Direct-expansion cycle



So, the thermal efficiency of the cycle shown in Figure 1 can be expressed as:

$$\eta_{th} = \frac{W_{GN}}{Q_F} = \frac{Q_F - Q_4}{Q_F} = 1 - \frac{Q_4}{W_{GN} + Q_4}. \tag{4}$$

On the other hand, the relationship between the enthalpy and exergetic efficiencies of the direct expansion cycle can be expressed by the following formula:

$$\eta_{th} = \frac{W_{GN}}{Q_F} = \frac{E_F}{Q_F} \cdot \eta_{ex} = \frac{E_F}{Q_F}\left(1 - \frac{\sum(LW)_D}{E_F}\right) \tag{5}$$

$$\eta_{ex} = \frac{W_{GN}}{E_F} = \frac{E_F - \sum(LW)_D}{E_F} = 1 - \frac{\sum(LW)_D}{E_F}. \tag{6}$$

2.3 Calculations for combustion chamber

The chemical energy content in fuel can convert into heat energy by firing. This combustion process could not occur naturally and is therefore an irreversible process. The irreversible loss can be divided into two kinds:

1 adiabatic combustion irreversible loss

2 heat transfer irreversible loss by heat dissipation and/or cool.

For the first term, assuming it is

$$I_{CC,AD} = E_{FUEL} - E_{AD} \tag{7}$$

where E_{AD} is the exergy of products of combustion at adiabatic combustion temperature.

For the second term, the irreversible loss can be written as

$$I_{CC,H} = E_{AD} + E_{IN} + E_{OUT}. \tag{8}$$

E_{OUT} is output exergy of combustion chamber. E_{IN} is exergy of coolant introduced into combustion chamber. Combining the above two equations yields

$$I_{CC} = I_{CC,AD} + I_{CC,H} = E_{FUEL} + E_{IN} + E_{OUT}. \tag{9}$$

The exergetic effectiveness of combustion chamber is defined as

$$\eta_{EXCC} = \frac{I_{CC}}{E_{FUEL}} = 1 - \frac{E_{OUT} - E_{IN}}{E_{FUEL}}. \tag{10}$$

The simplified formula for gross heat rate of combined cycle plant is as follows:

$$HR = \frac{Q_F \times HHV}{W_{GN}} \text{ (kcal/kWhr)}. \tag{11}$$

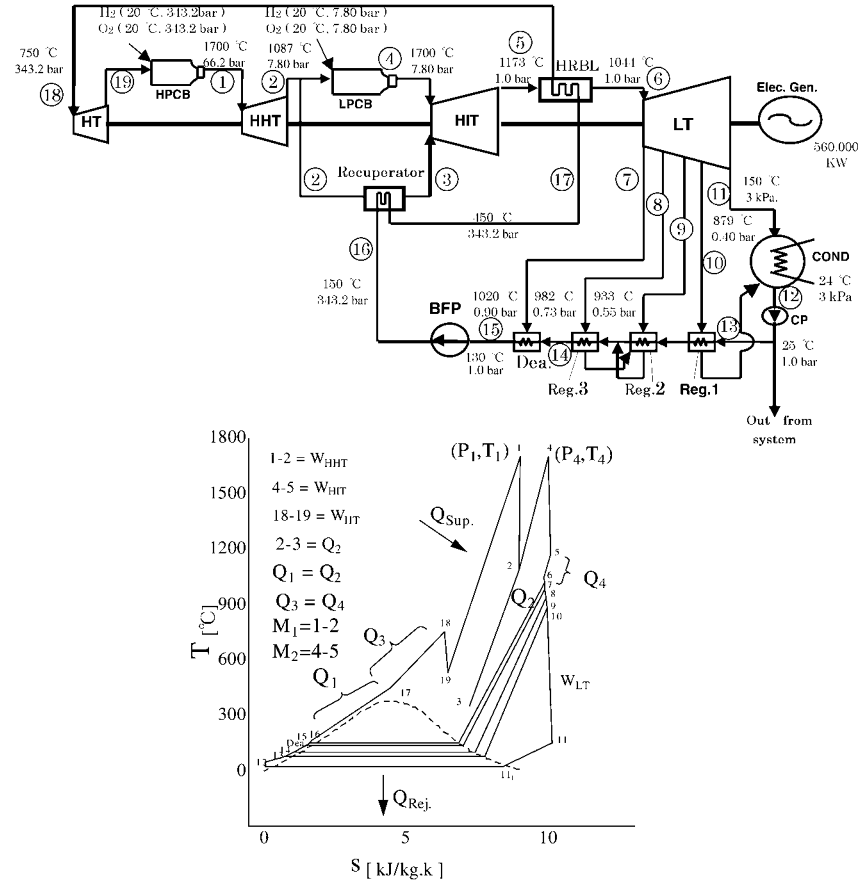
3 Configuration of hydrogen-fired power plant

3.1 Hydrogen-fired power plant (new Rankine cycle)

Based on the consideration in Sections 1 and 2 above, an integrated cycle is proposed, in which the product of fuel firing-temperature are shown in Figure 2. This cycle (C2) will be called the 'New Rankine cycle'. In this cycle the discharge steam M_1 (mass flow rate) out of high pressure combustion chamber with temperature T_1 and P_1 expands to low pressure P_2 through a high temperature steam turbine (HHT). After expanding, the steam is bifurcated to two lines. The first line (15%) of M_1 will be used as cooling steam for the turbine airfoil of (HIT) but before that it enters the high pressure recuperator heat exchanger where the steam loses some of its heat to reheat the feed water (which will be entering the low pressure heat recovery boiler). After recuperation the cooling steam enters the intermediate high temperature steam turbine (HIT) with the same pressure P_2 and temperature T_3. The second line (85%) of M_1 enters the low pressure combustion chamber with temperature T_2. The discharge steam of low pressure combustion chamber M_2 (mass flow rate) is expanded to more low pressure P_5 and low temperature T_5 into intermediate high temperature steam turbine (HIT). With these parameters the exhaust steam enters the low pressure heat recovery boiler (HRBL), Where the steam loses its heat to reduce the steam which will enter the high temperature steam turbine (HT) in the point 18. After that the steam after (HRBL) with pressure P_6 and T_6 is expanded in the low pressure steam turbine (LT) to be condensed after that in the condenser with pressure P_{11} and temperature T_{11}. After the condenser some of the mass flow rate is out of the system according to the operating conditions, and the main mass flow rate of the water is pumping to enter the regenerative heat exchangers (Reg. 1, 2 and 3 respectively) to be reheated by the bleeding low pressure steam coming from the low temperature turbine (LT). The reheated water then enters the de-aerator to be ready to be pumped by the (BFP). The reheated water at high pressure after (BFP), enters the high pressure recuperator, where the product enters the (HRBL) and exits with superheated steam to enter the high temperature turbine (HT) at point 18. The exhausted steam from (HT) is bifurcated to two lines. The first line (15%) of M_1 will be used as cooling steam of the turbine airfoil of (HHT). The second line (85%) enters the (HPCB) at point 19, and the cycle is repeated, where the processes are as follows:

1–2	high temperature steam turbine (HHT)
2–3	high pressure recuperator heat exchanger
2–4	low pressure combustion chamber (LPCB)
3,4–5	intermediate high temperature steam turbine (HIT)
5–6	low pressure heat recovery boiler (HRBL)
6–7	low pressure steam turbine (LT)
7–11	bleeding of low pressure steam to regenerative heat exchangers (RHE)
11–12	condensate process in the condenser
13–14	regenerative heat exchangers (RHE)
14–15	de-aeration process
15–16	boiler feed pump, (BFP)
16–17	recuperation process
17–18	low pressure heat recovery boiler (HRBL)
18–19	high temperature steam turbine (HT)
19–1	high pressure comb. chamber (HPCB).

Figure 2 Schematic chart and T-s diagram of (500 MW) new Rankine cycle with reheat and recuperation power plant. Type (C2)



4 Basic condition and assumption of plants

The fundamentals of calculating condition and assumption for the new design two cycles mentioned above, in Figures 2 and 3, respectively, are similar to the Rankine cycle (C) data which are shown below and expressed in Figure 4. As the temperature T_1 is an important parameter for the cycle performance so the design temperature will be 1700°C and with the restriction of components and combustion, the discharge temperature in the combustor will be restricted within a range of 1500–2000°C for the Rankine hydrogen-fired power plant with a 500 MW steam turbine:

- S/T output (net): 504 MW
- gross heat rate of H_2 on (HHV): 12,789 kJ/m^{3}N, (Hyprotech, 2002)
- efficiency of the S/T: 90%
- efficiency of CEP: 75%
- high pressure comb. pressure ratio = 47.50 (4750 kPa)
- low pressure comb. pressure ratio = 13.00 (1300 kPa)

- efficiency of the HRB: 90%
- efficiency of the generator: 98%
- efficiency of the BFP: 75%
- output pressure of BFP: 4750 kPa
- TIT = 1700°C. Maximum value of effectiveness of the heat exchanger was assumed as 85%
- efficiency of recuperator 85%
- fuel exergy mass flow rate of H_2 on the HPCB: 11771 kg/h (Hyprotech, 2002)
- fuel exergy mass flow rate of H_2 on the LPCB: 14400 kg/h (Hyprotech, 2002)
- fuel exergy mass flow rate of O_2 on the HPCB: 2097.50 kg/h (Hyprotech, 2002)
- fuel exergy mass flow rate of O_2 on the LPCB: 5039.9 kg/h (Hyprotech, 2002)
- the condensing process is holding by the cooling water
- the cooling steam of turbine airfoil of high temperature turbines in the all three cycles is assumed to be 15% of the respective mass flow rate
- radiation heat is neglected. The temperature of the H_2 and O_2 on the inlet of the comb. chamber: 20°C.

Figure 3 Schematic chart and T-s diagram of (500 MW) Rankine cycle with reheat and recuperation power plant. Type (C1)

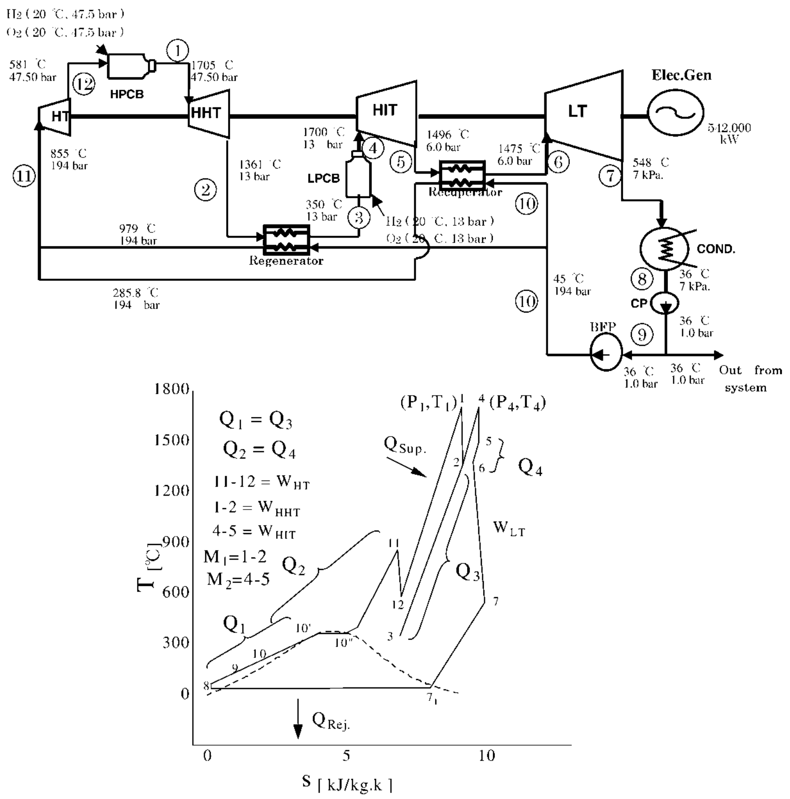
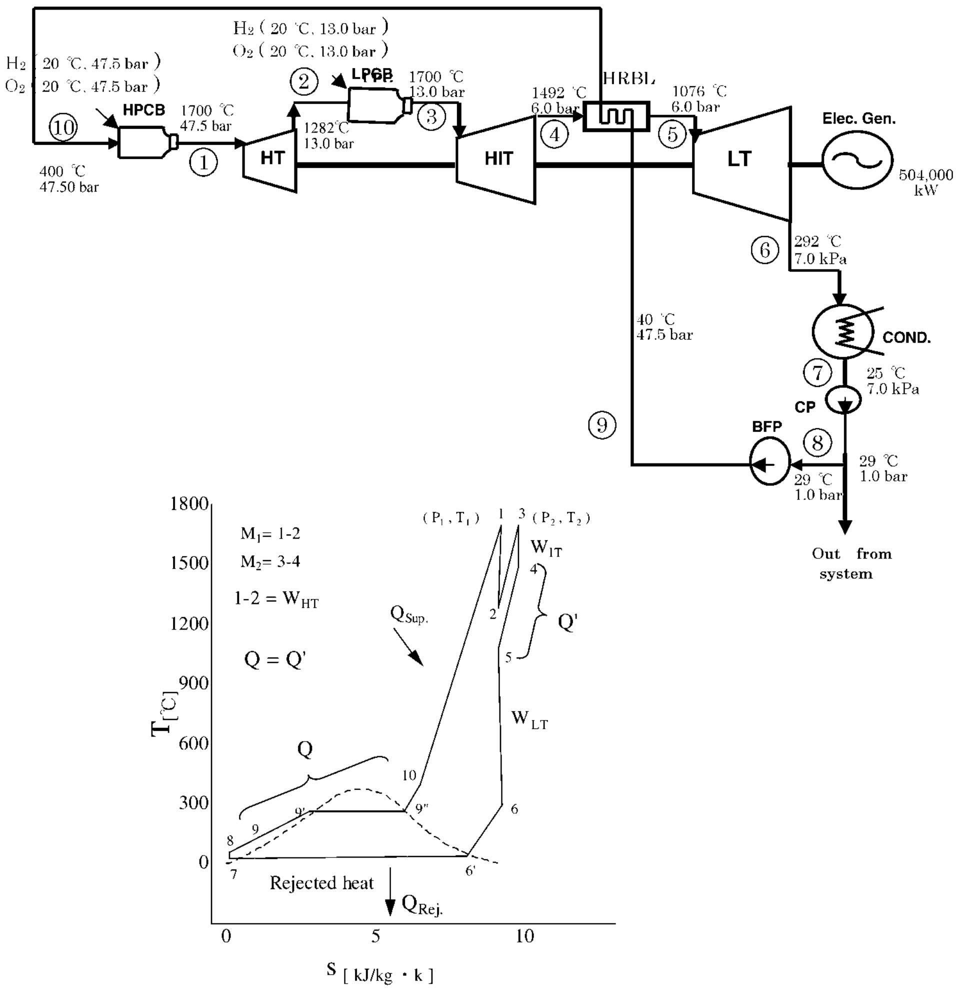


Figure 4 Schematic chart and T-s diagram of (500 MW) Rankine power plant. Type (C2) cycle



5 Results and discussion

Figure 5(a), (b) and (c) respectively show the results of sankey diagrams of the three cycles of the Rankine cycle and the Rankine cycle with reheat and recuperation (type C1) and new Rankine cycle (type C2). First of all, according to the results in the sankey diagram, the hydrogen-fired power generation cycle has a great advantage over the efficiency and performance characteristics of the conventional Rankine power plant. The exergy efficiency in Figure 5(a) is the greatest at 57.5% of 100% of input fuel exergy: the conventional steam Rankine cycle recently reached a maximum of 42% due to a high temperature of 1700°C as TIT. However, we have to consider here that the hydrogen-fired cycle could be considered the combined cycle of Brayton and Rankine due to the working fluid of high pressure and high temperature and that is why the efficiency is much higher than the conventional steam cycle. In these sankey diagrams it is clear that the cycle with the new Rankine (C2) has the advantage compared with the other two cycles, (C) and (C1). The maximum exergy efficiency is assumed to be 63.6% of the input fuel exergy. It means about 6% up to the Rankine cycle (C) and 3% up to the Rankine cycle (C1) with reheat and recuperation, with the advantage that the input exergy fuel of the three cycles was found to be the same value. So, (type C2) as the new Rankine cycle, can increase the output power of the plant from the net output power of 504 MW to 560 MW on the same rate of input fuel exergy. It means an increasing of power output about 11.11% of output power of the hydrogen power plant (type C) of 504 MW. And up to 3.4% from the output power of type (C1) as shown in the exergy flow sheet of the new Rankine cycle in Figure 6. In sankey diagrams we can also observe that the value of the exergy loss LW is the greatest in the combustor, which is about 29.6% per the input exergy of the fuel as for the Rankine cycle (C). However, this value is in a good range compared with the conventional combustion chamber of the gas turbine. On the other hand, the second advantage of the new Rankine cycle (C2) after increasing the efficiency is less value of lost work of the combustion chamber. It is assumed to be 26.03%. However, the best value was found on the Rankine cycle with reheat and recuperation, C1. It assumed to be 22%, and this is because of the more reheating and recuperation of the working fluid. So, it is up to 7.6% compared with the Rankine cycle (C). Next, Figure 6 shows the exergy results of the calculation method of the respective components of the 500 MW hydrogen-fired power generation plant and flow sheets of the new design as type C2. The effect of the different parameters of the plant can be expressed in the following figures. Figure 7 is expressing the relationship between the thermal and exergy efficiencies with the TIT. We can see the great effect of increasing the temperature inlet turbine on the cycle efficiency. However, the advantage of the new Rankine cycle can be clearly achieved with the greatest value of thermal and exergy efficiencies, which can be obtained only on the new Rankine cycle (C2), assuming the value of exergy efficiency to be 63.58%. The same value in the Rankine cycle (C) is about 57.50% and with the Rankine cycle with reheat and recuperation (C1) about 60.50% and all these values are achieved on the parameter of design temperature of 1700°C.

Next is the relationship between the thermal and exergy efficiencies according to the pressure of the condenser. From Figure 8 it is clear that by decreasing the pressure of the condenser the energy and exergy efficiencies are increasing and the relationship is proportional.

Figure 5 Exergy sankey diagram of new Rankine cycle (type C2), new design (type C1), and Rankine cycle (type C)

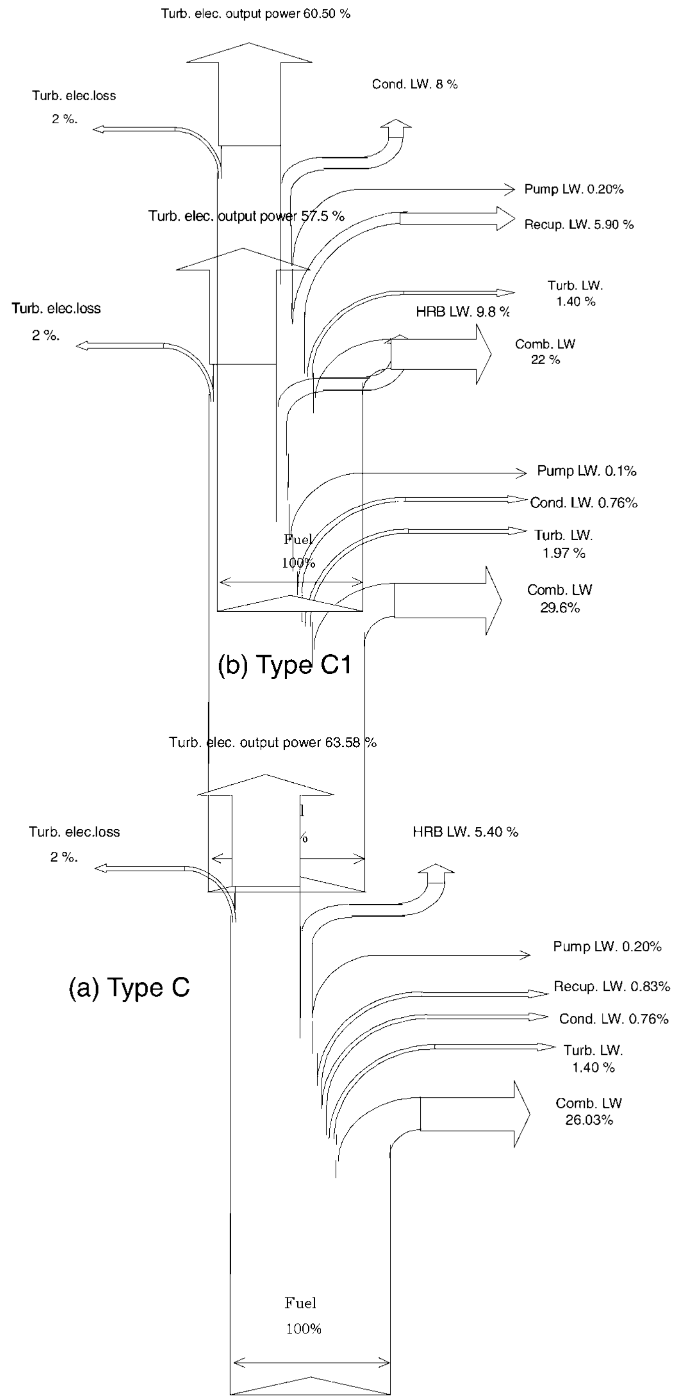


Figure 6 New Rankine cycle with reheat and recuperation of exergy flowsheet. Type (C2)

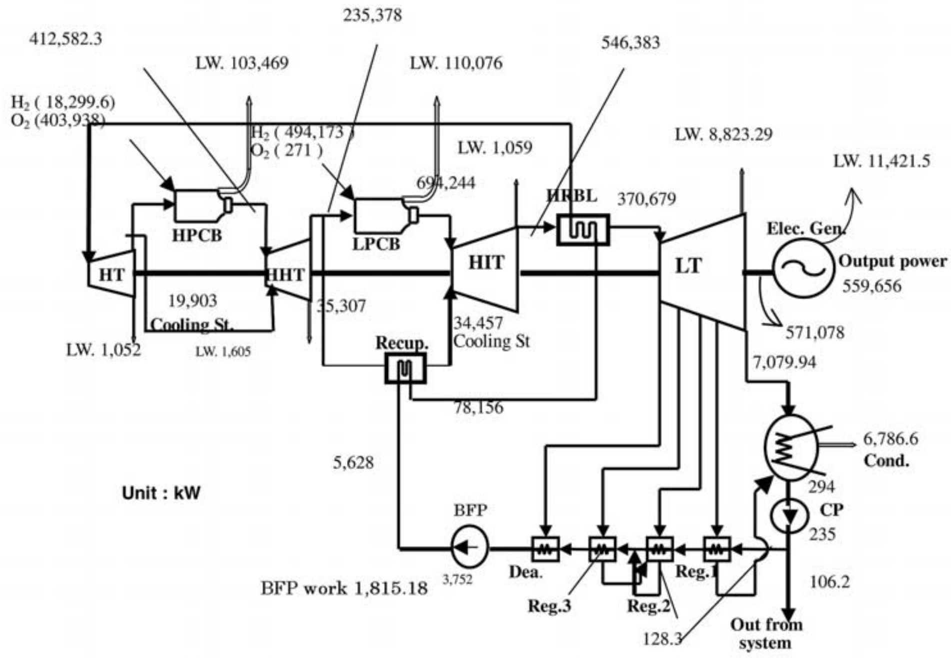


Figure 7 η_{th}, η_{ex} of 500 MW hydrogen-fired plant according to TIT

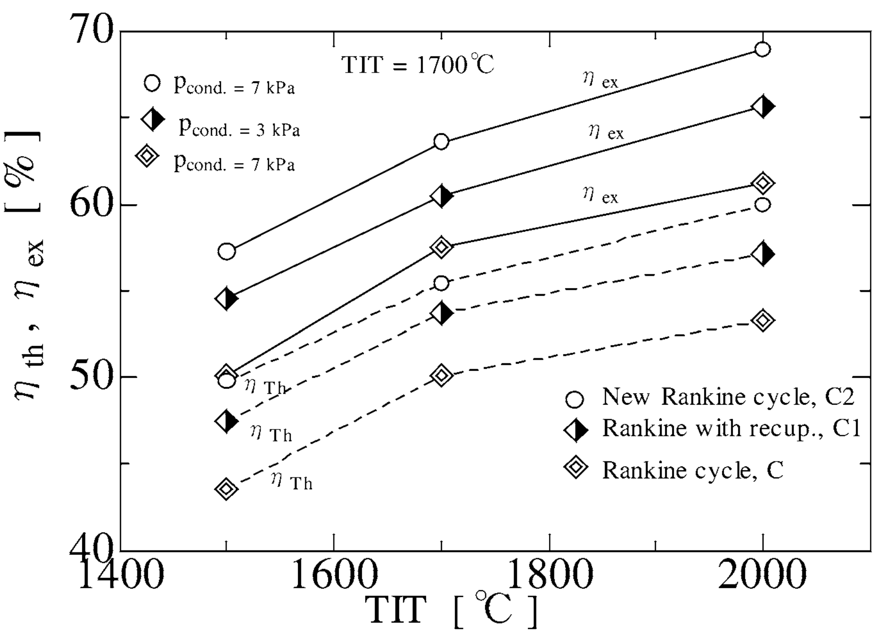
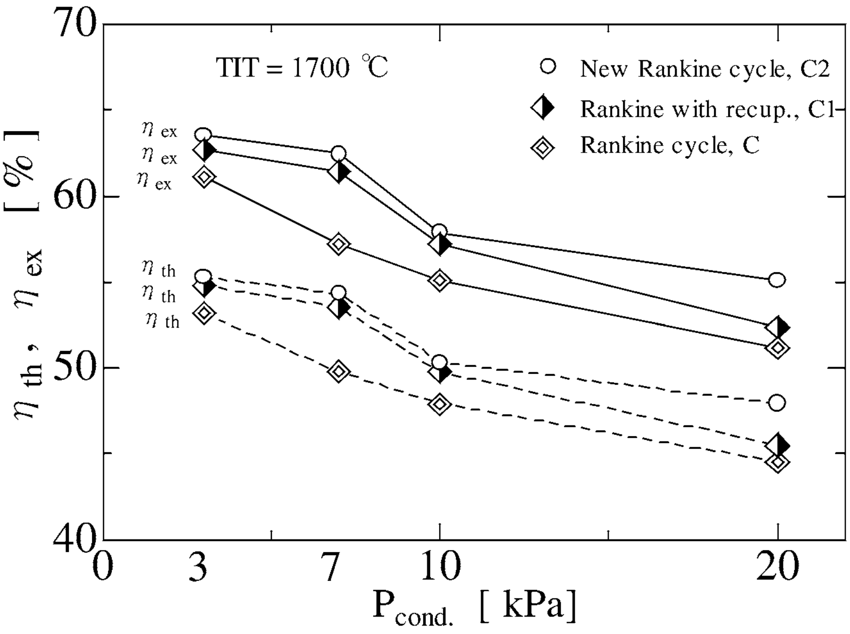


Figure 8 η_{th}, η_{ex} of 500 MW hydrogen-fired plant according to condenser pressure



The maximum value of exergy efficiency can be obtained only on the new Rankine cycle (C2). It is assumed to be 63.58% with a condenser pressure of 3 kPa and parameter of TIT as 1700°C (Hyprotech, 2002). From Figure 9 we can express that the effect of the TIT on the gross output power ratio is proportional. With an increasing TIT the ratio of output power is increased. The maximum value on the design point is obtained on the new Rankine cycle (C2) and assumed to be 11.11%. On the other hand, the exergy fuel ratio of the combustion chamber is proportional with an increase in TIT, and this is because of an increase in the mass flow rate of fuel exergy. One of the advantages of these two new design cycles is that the mass flow rate and exergy fuel values are the same for both as well as for the basic condition of the Rankine cycle (C). The relationship between the gross output power ratio and the pressure of the condenser is shown in Figure 10, which is proportional. The high ratio can only be obtained on the low condenser pressure. The maximum value is assumed to be 11.11% on the 3 kPa of pressure of condenser when the cycle is running with the new Rankine (C2). Finally the effect of the new design cycles type C2 and C1 on the lost work of the combustion chamber is expressed in Figure 11. The lost work is decreasing with an increase in TIT. The relationship is proportional. However, the minimum value of the lost work can be obtained with the Rankine cycle with reheat and recuperation (C1). It is assumed to be 22%. Mention that the same value for the new Rankine cycle (C2) is 26.03%, and 29.6% for the Rankine cycle (C).

Figure 9 The output power ratio $\Delta W/W_0$ and fuel ratio $\Delta Q/Q_0$ of the 500 MW hydrogen-fired plant according to the TIT

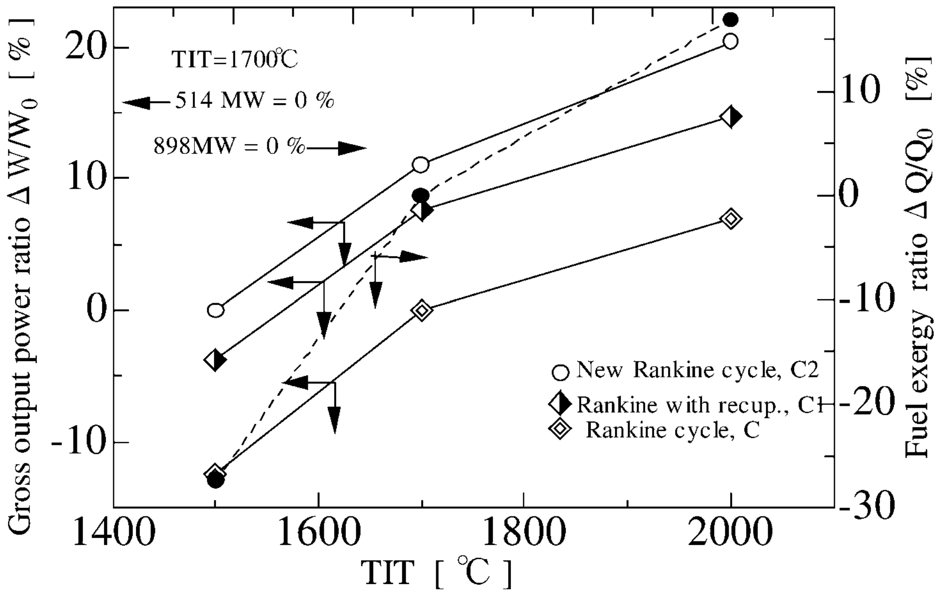


Figure 10 The output power ratio $\Delta W/W_0$ of the 500 MW hydrogen-fired plant according to the pressure of condenser

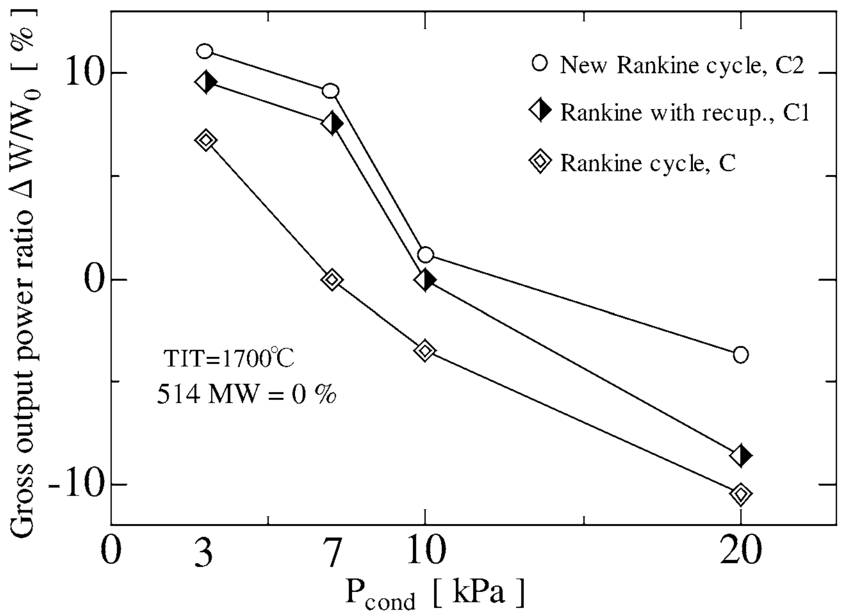
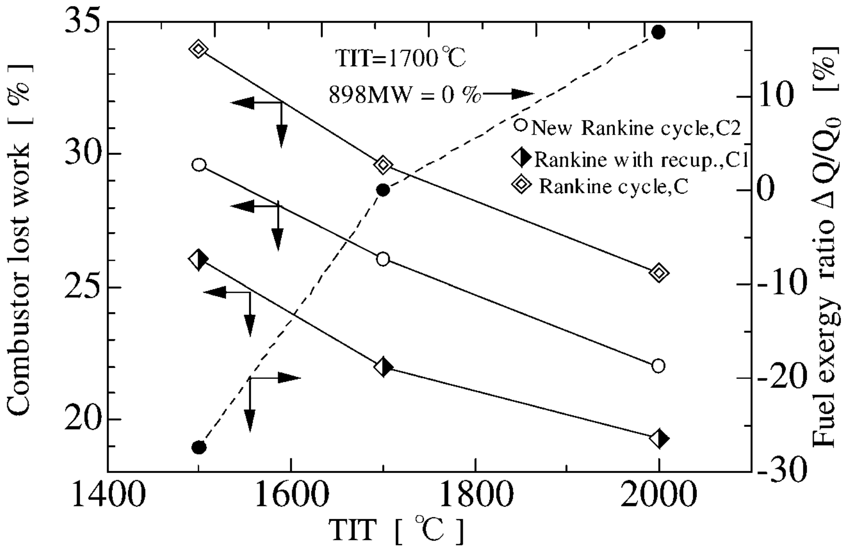


Figure 11 The combustor lost work ratio and fuel ratio $\Delta Q/Q_0$ of the 500 MW hydrogen-fired plant according to the TIT



6 Conclusions

We can draw the following concluding remarks:

- The proposed two new design cycles are suitable cycles for hydrogen as fuel, and they can run at a high temperature from 1500 to 2000°C. The exergy efficiency of the new Rankine cycle can achieve above 63.5% at 1700°C temperature level on the basis of a higher heating value. It is demonstrated that the exergetic and energy efficiencies of the new Rankine cycle are superior to those of the other cycles adapted (Hisadome, et al., 1998; Hyprotech, 2002; Jerich et al., 1989; Jericha, 1985; NEDO-WE-NET, 1997; Sugishita et al., 1996, 1997).
- Most of the irreversible losses in the cycle may occur in the combustion chamber, and the entry temperature of the coolant (i.e. steam) and the discharge temperature of the combustion chamber may be the most important factor in improving the performance of the cycle. They should be as high as possible within the acceptable range of components.
- The low pressure of the condenser has a great effect on the efficiencies and output power of the plants. The low pressure is the key of the cycle premium (Hyprotech, 2002).
- The proposed cycles are suitable for the highly efficient and highly feasible hydrogen combustion turbine used for an industrial power plant from the point of view of the thermal and exergetic efficiencies and the feasibility of components.
- Provide the solution to both energy and environmental problems in order to get an alternative clean energy source and develop a new energy system that has a higher efficiency of conversion from heat energy to power and lower pollutant emission due to CO_2, which is almost exhausted by fossil fuel burning.

Nomenclature

C_p	Specific heat capacity
E	Exergy
e	Specific exergy
G	Mass flowrate
h	Specific enthalpy
I	Irreversible losses
LW	Lost work of exergy
P	Pressure
Q	Heat quantity
q	Specific heat quantity
T	Temperature
W	Work (power output)
η	Efficiency
Δ	Difference or deviation

Subscripts

D	Direct-expansion cycle
ex	Exergy
Exh	Exhaust
F	Fuel
GT	Gas turbine
g	Gas
h	Heat
ST	Steam turbine
Th	Thermal

Abbreviations

BFP	Boiler feed pump
CC	Combustion chamber
Comp	Compressor
CP	Condensate pump
GN	Total generation output power of turbines
TIT	Turbine inlet temperature

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