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Performance analysis on various system layouts for the combination of an ambient pressure molten carbonate fuel cell and a gas turbine

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Abstract

This study analyses hybrid systems combining a molten carbonate fuel cell (MCFC) operating at ambient pressure and a gas turbine. Various possible system layouts, with the major difference among these layouts being the heating method of the turbine inlet gas, are proposed and their design performances are simulated and comparatively analyzed. Power of the MCFC in the hybrid system is explained in terms of the cathode inlet air temperature. Power of the gas turbine differs among various layouts because of large difference in the turbine inlet temperature. The direct firing in front of the turbine allows far higher turbine inlet temperature, and thus greater gas turbine power than the indirect heating of the inlet gas. The optimum pressure ratio of the directly fired system is higher than that of the indirectly fired system. The directly fired system allows not only much larger system power and higher optimum efficiency but also greater flexibility in the selection of the design pressure ratio of the gas turbine. In addition, the directly fired system can better accommodate the specifications of both current micro gas turbines and advanced gas turbines than the indirectly fired system.

Keywords: Molten carbonate fuel cell; Gas turbine; Hybrid system; Directly fired system; Indirectly fired system; Design performance

1. Introduction

Continuous reductions of fossil fuel energy resources and recent attention for environmental issues have promoted the development of advanced energy systems. The development of such systems is important especially in the electric power industry, which is the biggest primary energy consuming section. Even though renewable energy sources have been considered and some practical systems have begun to appear in the market, they have not been very cost effective and have problems mainly related to technical reliability. Consequently, fuel cells have received much attention as electric power sources. In particular, high temperature fuel cells are very suitable for stationary power generation systems because of their high efficiency and possibility of further performance upgrade through combination with other conventional heat engines. Because of their high performance potential, research has focused on the area of high temperature fuel cells such as molten carbonate fuel cells (MCFCs) and solid oxide fuel cells [1].

Since molten carbonate fuel cells operate at 600–700 °C, the high energy content of their exhaust gas is very useful. Thus, they can be applied to various advanced systems, such as cogeneration and hybrid systems. As a result, much research has been directed to the development of MCFC cell stack itself and entire systems for commercialization [2–5]. Even though MCFC provides sufficient efficiency, its performance can be further enhanced by hybridization with a gas turbine.

As in most combined systems between two independent sub-systems, there are many ways to design a hybrid system. Therefore, the system designer has various options available, and different configurations of the hybrid system have been proposed through fundamental research. Hybrid systems are classified as either an ambient pressure system or pressurized system depending on the operating pressure of the fuel cell. Both of these two basic configurations are feasible for MCFC, but the current development focus is given to the ambient pressure system. In this system, the exhaust gas from the gas turbine is fed to the MCFC. The MCFC is well suited for this bottoming

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Nomenclature

CIT	cathode inlet air temperature (°C)
F	Faraday constant
GT	gas turbine
HRU	heat recovery unit
$\bar{h}$	molar enthalpy (kJ kmol^{-1})
I	current (A)
LHV	lower heating value (kJ kg^{-1} K^{-1})
MCFC	molten carbonate fuel cell
$\dot{m}$	mass flow rate (kg s^{-1})
$\dot{n}$	molar flow rate (kmol s^{-1})
$\dot{Q}$	heat transfer rate (kW)
SCR	steam/carbon ratio
TIT	turbine inlet temperature (°C)
U_f	fuel utilization factor
V	cell voltage (V)
$\dot{W}$	power (kW)

Greek letter

η	efficiency

Subscripts

AC	alternating current
aux	auxiliary
C	compressor
cell	cell stack
conv	conversion
DC	direct current
FC	fuel cell
gen	generator
HS	hybrid system
i	each component
m	mechanical
ref	reformer
T	turbine

system because it needs to have carbon dioxide in the inlet air stream [6]. While most of the currently developed systems operate at ambient pressure, there also exist some efforts to realize pressurized hybrid systems [5]. Pressurized operation may take advantage of the theoretical high cell voltage. However, because of the practical technical problems inside the cell accompanied by pressurization [7,8], the ambient pressure system has been preferred up to now. The most important advantage of operating the fuel cell at ambient pressure is the fact that pressure ratio of the gas turbine can be designed independently to optimize the hybrid system performance.

The simplest way to develop a hybrid system is to construct a system based on existing MCFC [3] or gas turbine [9,10] and optimize the system under given conditions. In addition to such simple combinations, more complicated system designs such as adopting additional steam cycles [11] and steam injection [12] have also been investigated to further enhance system performance. Integration of coal gasification and MCFC has also been considered as an option since the carbon monoxide can be used as fuel in the MCFC [13].

In addition to the existing analyses for the MCFC/gas turbine hybrid systems, some of which are listed above, more comprehensive research is required to find the optimal system design in the immature field of hybrid systems technology. In particular, because comparative analyses for various possible combinations of the MCFC and the gas turbine are very useful for design engineers, this study presents design performance characteristics of various system layouts of the hybrid system based on the ambient pressure MCFC. Main focus is given to the performance differences according to the heating method of the turbine inlet gas (direct and indirect heating).

2. Molten carbonate fuel cell

2.1. MCFC system model

The performance of MCFC only system is first studied as a performance guideline. With this preliminary study, performance enhancement due to hybridization can be estimated. The models and parameters for the MCFC will also be applied to those of the hybrid systems in Section 3. The basic MCFC system considered in this work is shown in Fig. 1. Internal steam reforming, which allows higher system performance than the external reforming, is adopted. The steam required for the reforming process is supplied from the outside. Fuel and water are preheated at the heat recovery unit and fed to the reformer. The incoming air is partially burned with the redundant fuel (combustible substances) from the anode exit. The dotted lines denote a couple of modifications to the simple system, as will be explained in Section 2.2. The fuel is methane and the chemical reactions at

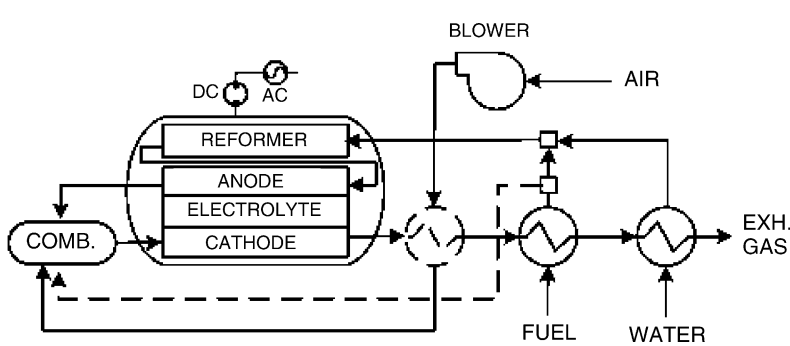

Fig. 1. MCFC only system (solid lines: simple system, dotted lines: modifications).

the reformer considered are as follows:

$$\text{Reforming}: \quad CH_4 + H_2O \rightarrow 3H_2 + CO \tag{1}$$

$$\text{Shift}: \quad CO + H_2O \rightarrow H_2 + CO_2 \tag{2}$$

$$\text{Total}: \quad CH_4 + 2H_2O \rightarrow CO_2 + 4H_2 \tag{3}$$

The amount of steam supplied is decided through the steam carbon ratio defined as follows:

$$\text{SCR} = \frac{\dot{n}_{H_2O}}{\dot{n}_{CH_4}} \tag{4}$$

Net electrochemical reaction at the fuel cell is described as follows:

$$\text{Anode}: \quad 2H_2 + 2CO_3^{2-} \rightarrow 2H_2O + 2CO_2 + 4e^- \tag{5}$$

$$\text{Cathode}: \quad O_2 + 2CO_2 + 4e^- \rightarrow 2CO_3^{2-} \tag{6}$$

$$\text{Overall}: \quad H_2 + \tfrac{1}{2}O_2 + CO_2\,(\text{cathode}) \rightarrow H_2O + CO_2\,(\text{anode}) + \text{electricity} \tag{7}$$

At the cell inlet, carbon monoxide (CO) is present due to the incomplete conversion of methane to hydrogen at the reformer. Carbon monoxide also produces the same amount of electric current per unit mole as hydrogen (Eqs. (5)–(7) with replacement of H_2 with CO). The fuel utilization factor at the cell is defined by the ratio between reacted to supplied fuels as follows:

$$U_f = \frac{(\dot{n}_{H_2} + \dot{n}_{CO})_{\text{reacted}}}{(\dot{n}_{H_2} + \dot{n}_{CO})_{\text{supplied}}} \tag{8}$$

In the fuel cell with an internal reformer, heat is transferred from the cell to the reformer to maintain the endothermic steam reforming reaction. Thus, the energy balances at the cell and the reformer are presented by the following equations:

$$\text{Cell}: \quad \sum_{\text{in}} \dot{n}_i \bar{h}_i + \dot{Q}_{\text{cell}} = \sum_{\text{out}} \dot{n}_i \bar{h}_i + \dot{W}_{\text{FC,DC}} \quad (\dot{Q}_{\text{cell}} < 0) \tag{9}$$

$$\text{Reformer}: \quad \sum_{\text{in}} \dot{n}_i \bar{h}_i + \dot{Q}_{\text{ref}} = \sum_{\text{out}} \dot{n}_i \bar{h}_i \quad (\dot{Q}_{\text{ref}} = -\dot{Q}_{\text{cell}} > 0) \tag{10}$$

The DC power generated from the cell stack is calculated as follows:

$$\dot{W}_{\text{FC,DC}} = VI = V \cdot (\dot{n}_{H_2} + \dot{n}_{CO})_{\text{reacted}} \cdot 2F \tag{11}$$

The final AC power from the cell stack is calculated as follows considering the DC to AC conversion loss:

$$\dot{W}_{\text{FC,AC}} = \dot{W}_{\text{FC,DC}} \cdot \eta_{\text{conv}} \tag{12}$$

One of the important design parameters that should be assigned for the analysis is the cell voltage. Cell voltages of MCFCs under development range between 0.7 and 0.85 V and as high as 0.9 V is expected in the future [14]. The cell operating temperature also varies between 600 and 700 °C. In this study, the cell temperature of 650 °C is used as a representative value. The corresponding voltage of 0.8 V is adopted as a reference value since it represents the average of reported values. This voltage value is also close to the one predicted by a correlation considering Nernst potential and voltage losses [15]. Table 1 lists reference design parameters for the fuel cell system. All of the heat exchanger sections at the heat recovery unit (HRU) are modeled as counter flow types with appropriate effectiveness values. Pressure losses are also given to all of the flow elements. The fuel cell system and the entire hybrid system of the next section are modeled and simulated with a process simulation software [16].

Table 1
Reference parameters for the fuel cell

Cell temperature (°C)	650
Cell voltage (V)	0.8
SCR	2.5
U_f	0.78
η_{conv}	0.96

2.2. MCFC system performance

The effect of different voltages on the performance is examined briefly, assuming that the cell voltage may vary among different developers. Then, the effect of modulating cathode inlet air temperature is examined. Fig. 2 shows the fuel cell system efficiency and power for different cell voltages. Other parameters except the voltage are kept constant. Air flow rate of $1.0\,\text{kg}\,\text{s}^{-1}$ is assumed. The fuel cell efficiency is defined as follows:

$$\eta_{\text{FC}} = \frac{\dot{W}_{\text{FC,AC}}}{(\dot{m} \cdot \text{LHV})_{CH_4}} \tag{13}$$

At the reference point of 0.8 V, the MCFC system efficiency is around 55%. For the variation of voltage from 0.65 to 0.9 V, MCFC efficiency varies by more than 20% point, and the power varies by more than two times. The cathode inlet temperature for the reference case (0.8 V) is estimated to be 554 °C, which is almost 100 °C lower than the cell operating temperature. If

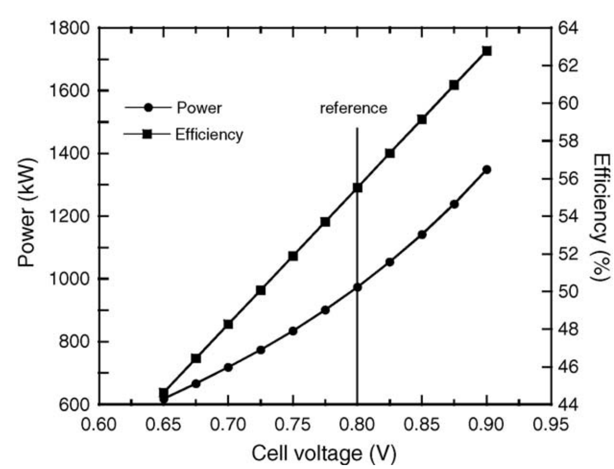

Fig. 2. Power and efficiency of the MCFC only system.

Table 2
Effect of increase of cathode inlet temperature on the MCFC performance

Method	Cathode inlet temperature (°C)	Efficiency (%)	Power (kW)
Simple (no modification)	554	55.52	955.5
Additional fuel at the combustor	600	39.26	476.7
Reduction of fuel utilization of the cell	600	49.27	730.4
Heat exchange between cathode exit and inlet	600	55.02	624.4

this inlet temperature seems too low (thus inlet and outlet temperature difference is too large) in view of chemical stability or mechanical rigidity of the cell, there are several possibilities to raise the inlet temperature. The simplest method is to burn additional fuel in front of the cell. Alternatively, the fuel utilization of the cell may be reduced to increase the combustible substances at the combustor. Another method is to adopt a recuperative heat exchanger between cathode exit and inlet gases. Modifications except the change in the fuel utilization are shown as dotted lines in Fig. 2. All of these three modifications were analyzed and their effects on the cell performance are summarized in Table 2. The cell voltages of all cases are 0.8 V and the cathode inlet temperature is raised to 600 °C by modulation of the design parameters of each case, such as additional fuel amount, fuel utilization, and degree (effectiveness) of heat exchange. Increase of the cathode inlet temperature means reduction of the cooling effect of the cell, which results in the reduction of fuel supply to the cell for a given cell operating temperature, and this, in turn, reduces the power output capability of the cell. A larger power capability requires a larger cell size if current density is comparable in all cases. Efficiency also varies much among different cases. In particular, the method of additional fuel supply gives the lowest efficiency. Reduction of fuel utilization also causes considerable loss of efficiency. However, the heat exchange method, which utilizes exhaust energy otherwise wasted, exhibits almost comparable efficiency to the base case.

3. MCFC/GT hybrid systems

3.1. Hybrid system layouts and analysis

The hybrid systems of this study are limited to combinations between an MCFC operating at ambient pressure and a gas turbine. Various system layouts are considered as shown in Fig. 3, and their design performances are comparatively analyzed. For all of the layouts, the ambient air is compressed first and then heated at the heat recovery unit (recuperative heat exchanger). Fuel is first provided to the MCFC and the remaining fuel (all combustible components) is supplied to a combustor, the location of which depends on the system layout. Thus, the processes between the HRU exit and the cathode inlet are diverse among the layouts.

In layout A, the heated air directly enters the turbine and then the turbine exit air is burned with the unreacted fuel components from the anode exit. Thus, the combustor locates after the turbine. Layout B is based on layout A. However, the gas from the combustor does not directly enter the cathode but it heats up the incoming air to the turbine before it flows into the cathode. This layout is basically similar to the configuration of hybrid systems for commercial development, targeting efficiency far higher than 60% [3]. A and B can be called indirectly fired systems since the high temperature at the turbine inlet is not achieved by combustion but by heating; that is, the turbine is operated by air instead of combustion gas as in usual gas turbines.

On the contrary, the compressed air of layout C is directly burned before it expands at the turbine. Thus, layout C can be called a directly fired system. The fuel for the combustor also comes from the anode exit. A compressor, called anode gas compressor, is required to pressurize the anode exit gas to the pressure of the combustor. In this basic layout (C1), there can be a practical difficulty in manufacturing the auxiliary compressor operating at high temperature because the inlet temperature of the compressor is sufficiently high (cell exit temperature, 650 °C), and thus the exit temperature may be very high depending on the pressure ratio of the gas turbine required. As a remedy to this problem, the cooling of anode exit gas before it enters the anode gas compressor is devised to decrease the inlet temperature (layout C2). An internal recuperative type heat exchanger is considered as denoted by the dotted lines in the figure to prevent a performance penalty due to external heat rejection. The anode exit gas is cooled by the fuel and steam mixture, which is still at a sufficiently lower temperature than that of the anode exit gas. Both of the two layouts (the basic one: C1, the one with cooling: C2) are considered in this work.

Table 3 lists major parameters used for the gas turbine and other component in the hybrid system. Reasonable values of pressure losses through all flow components (heat exchanger, combustor and so on) are also included. The fuel cell parameters are the same as those in Table 1.

The net power output from gas turbine is calculated as follows:

$$\dot{W}_{\mathrm{GT,AC}} = (\dot{W}_{\mathrm{T}} \cdot \eta_{\mathrm{m}} - \dot{W}_{\mathrm{C}}) \cdot \eta_{\mathrm{gen}} - \dot{W}_{\mathrm{aux}} \tag{14}$$

Powers of all the auxiliary components, most of which are turbo machines, are included in the gas turbine power. The total hybrid system power and efficiency are represented as follows:

$$\dot{W}_{\mathrm{HS}} = \dot{W}_{\mathrm{FC,AC}} + \dot{W}_{\mathrm{GT,AC}}, \qquad \eta_{\mathrm{HS}} = \frac{\dot{W}_{\mathrm{HS}}}{(\dot{m} \cdot \mathrm{LHV})_{\mathrm{CH_4}}} \tag{15}$$

This study intends to compare the design performances of indirectly and directly fired systems and examine their practical design requirements and limitations. The fuel cell operating

Table 3
Reference parameters for the gas turbine and other components of the hybrid system

Turbine isentropic efficiency	0.90
Compressor isentropic efficiency	0.85
Anode gas compressor efficiency	0.75
HRU effectiveness (water and fuel heater)	0.78
HRU effectiveness (air heater)	0.78
HRU2 effectiveness (high temperature air heater)	0.89

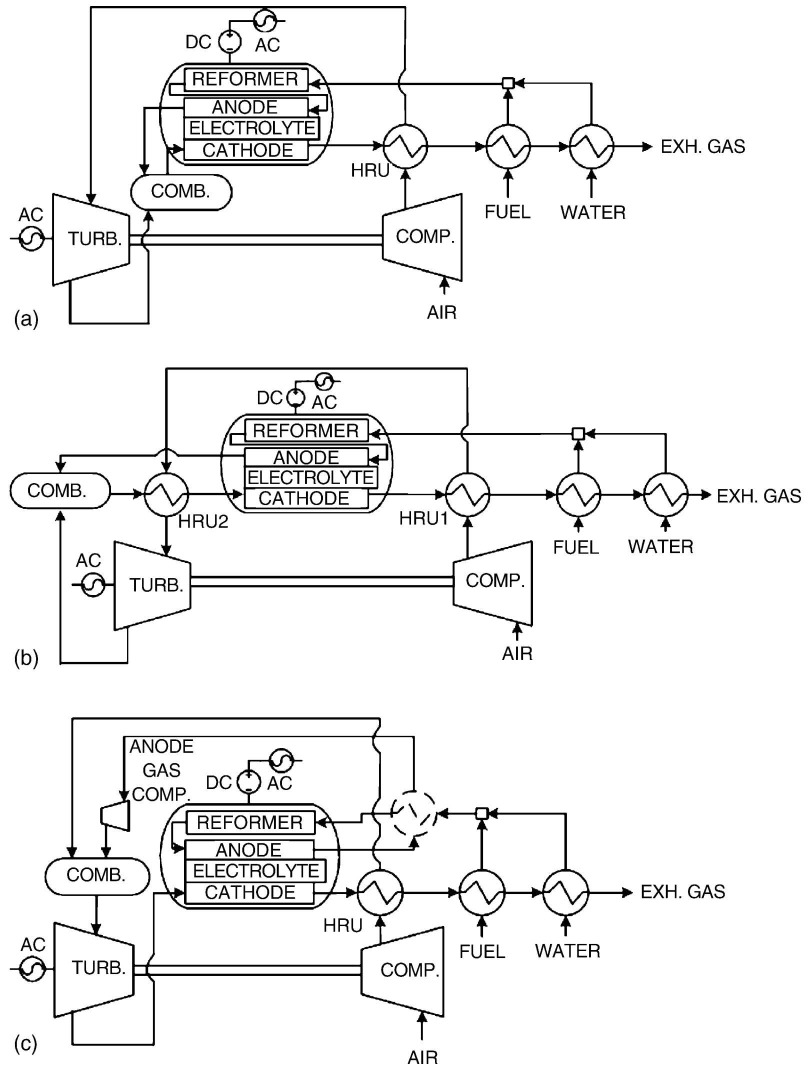

Fig. 3. MCFC/GT hybrid systems: (a) layout A, (b) layout B and (c) layout C1 and C2 (the dotted heat exchanger is for C2).

temperature is given as 650 °C, and the major design parameter is the pressure ratio of the gas turbine. Inlet air flow rate is set to the same as that of the MCFC only case, i.e. 1.0 kg s^{-1}. For each layout, variations in the design parameters and the hybrid system performance according to the pressure ratio are analyzed.

3.2. Results

First, a couple of important temperatures obtained from the analysis are examined. Fig. 4 shows the variations in the cathode inlet air temperature with the pressure ratio of the gas turbine. As the pressure ratio becomes high, the cathode inlet air temperature becomes lower because of the larger temperature drop at the turbine in front of the fuel cell. Layouts A, B and C1 have nearly same temperatures, while layout C2 exhibits relatively lower temperature. Compared with the result of Section 2.2, the cathode inlet air temperatures of the hybrid systems are higher than that of the simple MCFC only system. In layouts A, B and C1, the temperature is over 600 °C for a pressure ratio of eight. Even for layout C2, the temperature is higher than that of the simple MCFC only system (554 °C) for the entire pressure ratio

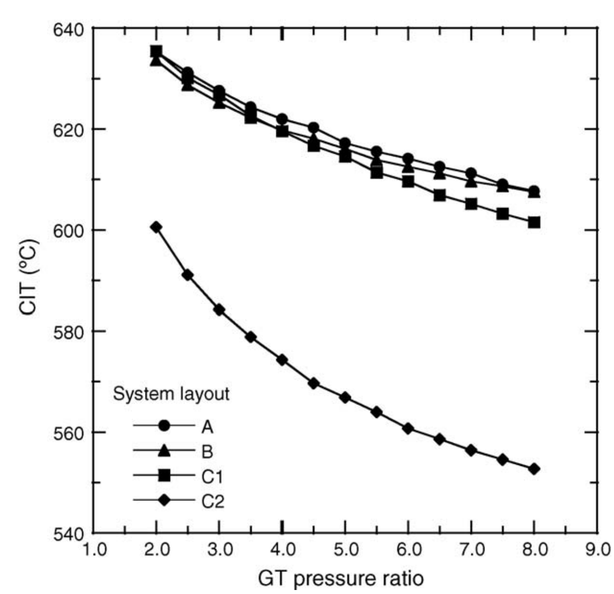

Fig. 4. Variation in cathode inlet air temperature with design pressure ratio.

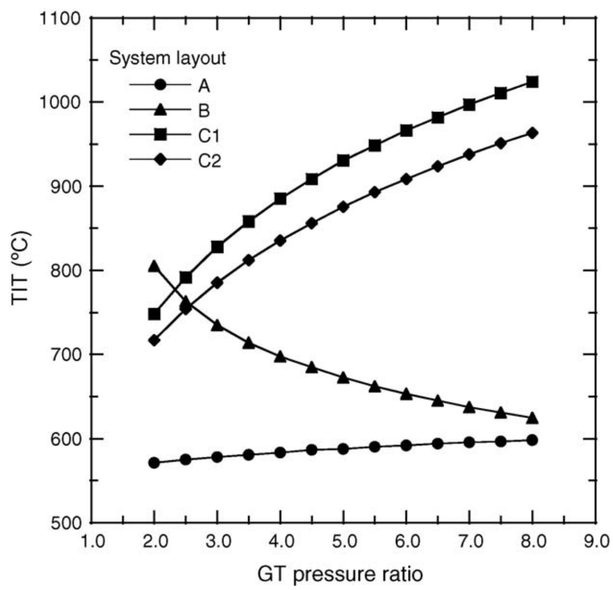

Fig. 5. Variation in turbine inlet temperature with design pressure ratio.

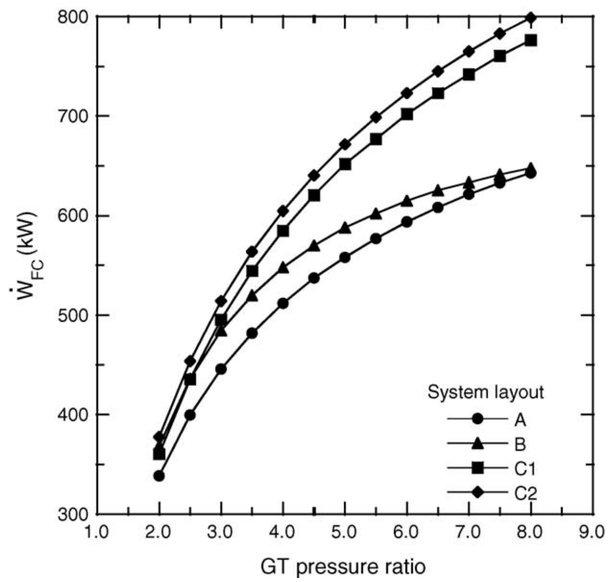

Fig. 6. Variation in MCFC power with design pressure ratio.

range of this study. The main reason for this high cathode inlet temperature in the hybrid systems is that all of them are equipped with heat exchangers which recover a part of the exhaust heat from the MCFC.

Another important parameter in the hybrid system is the turbine inlet temperature (TIT), whose variations are shown in Fig. 5. The TIT variations show two tendencies. In layouts C1 and C2 as well as A, raising the design pressure ratio leads to a higher turbine inlet temperature. In these layouts, the compressor outlet temperature directly affects the turbine inlet temperature. Thus, a higher pressure ratio results in a higher compressor exit temperature, and thus a higher turbine inlet temperature. In layout B, however, a larger expansion ratio at the turbine causes a lower inlet temperature at the high temperature heat exchanger (HRU2), and thus causes a lower temperature at the turbine inlet, which is heated by the heat exchanger. Layout A has the lowest TIT since it is heated only by a single heat exchanger in front of the turbine. Layouts C1 and C2 have relatively higher TIT due to the direct heating (combustion) of the turbine inlet gas. Layout C2 exhibits a slightly lower TIT than layout C1 because of the lower exit temperature of the auxiliary compressor that delivers the combustible components from the MCFC exit. This results in the reduction of cathode inlet (i.e. turbine exit) air temperature, as already shown in Fig. 4.

The MCFC powers are shown in Fig. 6. In all layouts, raising the design pressure ratio makes the temperature difference of the cell greater (lower cell inlet temperature). This in turn requires a greater amount of fuel flow, which enables a larger MCFC power. The directly fired systems (C1 and C2) allow greater cell power. In comparison with the result of Fig. 2, the fuel cell powers of all of the hybrid systems are smaller than the power of the simple MCFC only system with the same cell voltage (the reference case of Fig. 2) and air flow rate. This is due to the smaller temperature difference at the cell (higher cell inlet air temperature) depicted in Fig. 4.

The gas turbine power defined by Eq. (14) is shown in Fig. 7. The directly fired systems (C1 and C2) enable larger power to be produced from the gas turbine except at a very low pressure ratio range. The higher turbine inlet temperature of these layouts shown in Fig. 5 explains this result. The lowest TIT of layout A leads to the smallest gas turbine power. B as well as A exhibits a maximum gas turbine power at a relatively lower pressure ratio than C1 and C2 do, since TIT remains almost constant (A) or decreases with pressure ratio (B). On the other hand, in the directly fired systems, the gas turbine power tends to increase up to a considerably high pressure ratio due to the steady increase of TIT. Layout C2 results in a slightly higher gas turbine power than C1 does, even though its TIT is lower than that of C1. This is because the cooling of the inlet of the anode gas compressor reduces its power consumption.

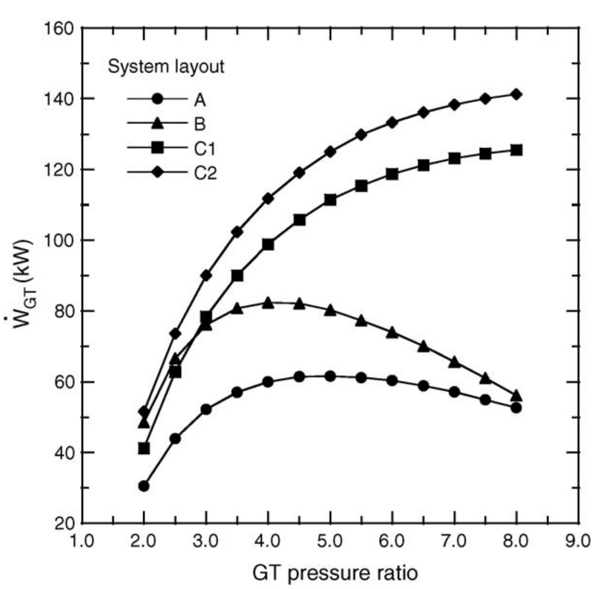

Fig. 7. Variation in gas turbine power with design pressure ratio.

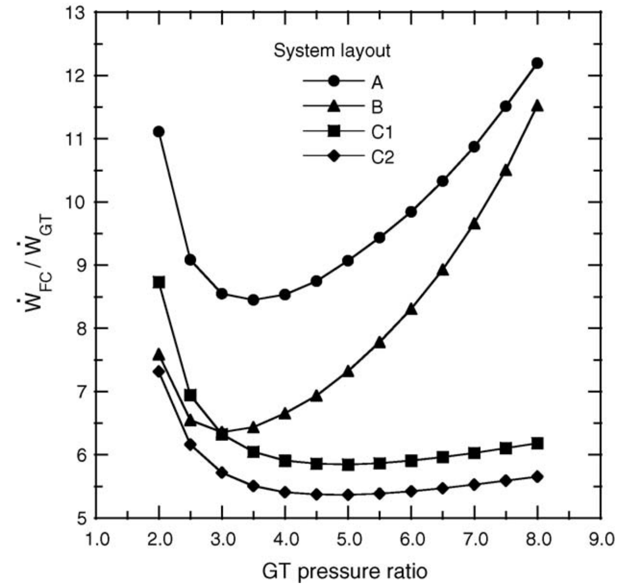

Fig. 8. Variation in power ratio with design pressure ratio.

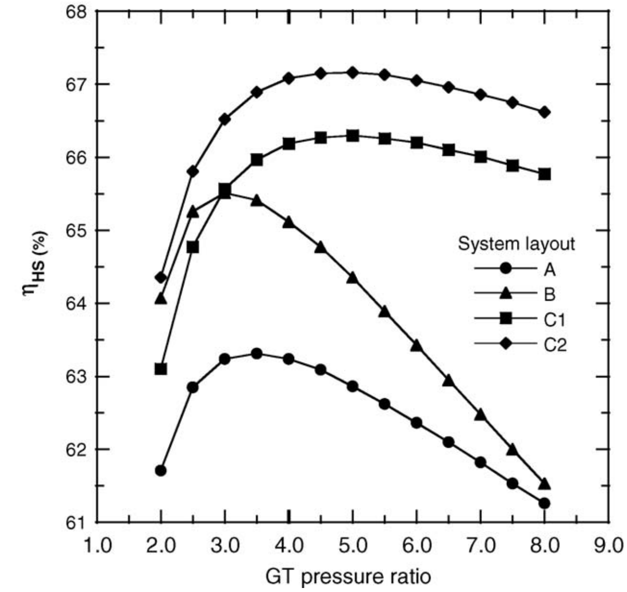

Fig. 10. Variation in hybrid system efficiency with design pressure ratio.

The power portion of the MCFC is far larger than that of the gas turbine for all layouts. However, the detailed power distribution between the MCFC and the gas turbine is diverse among various layouts as shown in Fig. 8, where the power ratio is defined as power of the MCFC divided by power of the gas turbine. The directly fired systems exhibit relatively smaller MCFC power share than layout B does because of the larger gas turbine power. Every case has a lowest power ratio value and the corresponding pressure ratio is higher in the directly fired systems than the indirectly fired systems. Variation in the total hybrid system power is shown in Fig. 9. Since the MCFC produces more power than the gas turbine, the tendency of total power variation follows that of the MCFC power. However, the difference in the total power between the directly fired and indirectly fired layouts becomes even greater as the pressure ratio increases because the difference in the gas turbine power between the two layouts becomes larger.

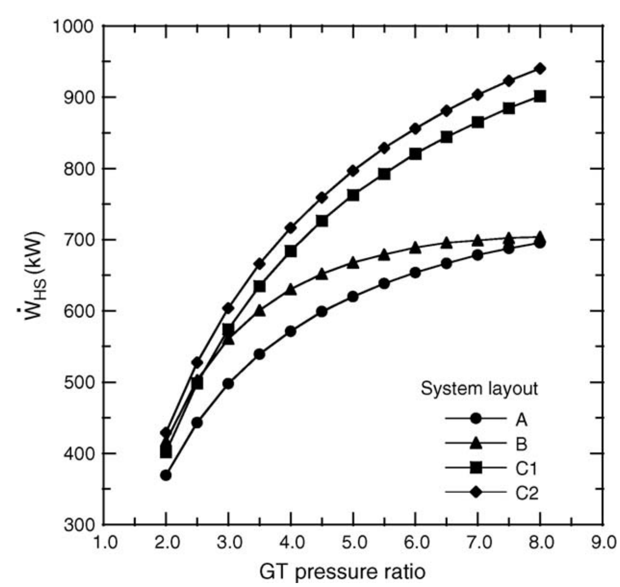

Fig. 9. Variation in hybrid system power with design pressure ratio.

Fig. 10 shows the variation in the hybrid system efficiency. For all layouts, supplied fuel flow rate increases with increasing pressure ratio, and thus, the fuel cell power and the total power also increase. However, the incremental increase of the total power continues to reduce with pressure ratio due to the gas turbine power behavior of Fig. 7. Accordingly, the hybrid system efficiency decreases beyond a certain pressure ratio. The optimum efficiency ranges from 63 to 67% depending on the system layout. The less effective power contribution of the gas turbine in the indirectly fired systems (A and B), as shown in Fig. 7, causes a relatively lower optimal pressure ratio (around 3 and 3.5 for A and B, respectively). The steady increase of the gas turbine power of the directly fired systems pushes the optimal pressure ratio to a rather high value (around 5), and thus, the maximum system efficiency is higher than those of the indirectly fired systems. Layout C2 exhibits the highest system efficiency. The optimal pressure ratio almost corresponds to the condition of the lowest power ratio (compare Figs. 8 and 10).

In addition to the slightly higher maximum efficiency, the directly fired systems have more advantages. Firstly, their efficiencies are very insensitive to the design pressure ratio. The efficiencies of C1 and C2 are almost flat for a wide pressure ratio range as shown in the figure. Thus, the design pressure ratio can be selected to be rather higher than the optimal (best efficiency) pressure ratio, which allows much greater system power. For example, 40% more power can be achieved for layouts C1 and C2 by setting the pressure ratio to 7.0 instead of 3.5, while the efficiency varies within less than 0.5% point. Secondly, the directly fired systems can accommodate specifications of practical gas turbines more easily. To now, developers have targeted the development of small hybrid systems (200–300 kW), adopting micro gas turbine of tens of kW. These micro gas turbines have design TIT values range from 850 to 900 °C, which do not necessitate turbine cooling. The design pressure ratios are also sufficiently low (3.0–4.0). The indirectly fired layout B cannot fully accept even this current commercial micro gas turbine specification since its optimal design requires far lower TIT (less than 750 °C at pressure ratio of 3). On the other hand, the directly fired layouts (C1 and C2) can accept the design TIT of current micro turbines. Furthermore, layout B is not suitable for larger hybrid systems. Several tens of MW or larger size hybrid systems are the ultimate target. If so, the gas turbine size must also be at least several MW. In such large gas turbines, the turbine inlet temperature and the pressure ratio are higher than those of micro gas turbines. Accordingly, layout B cannot accommodate the design parameters of the high performance gas turbines. As a result, the two major components (MCFC and gas turbine) may not be smoothly coupled and the gas turbine must be degraded to match the possible design. On the other hand, the directly fired system can be more favorably coupled with those MW class gas turbines because it can better match or accommodate the gas turbine parameter values.

In addition to TIT, other critical temperature requirements of each system need to be reviewed to check if any practical difficulty exists. In layout B, the highest temperature occurs at the combustor exit. Since the gas from the combustor flows into the high temperature heat exchanger (HRU2), the combustor exit temperature affects the design of the heat exchanger, especially, the selection of the heat exchanger material. In the directly fired layouts C1 and C2, the design of the anode gas compressor is important since its exit temperature may be very high. Fig. 11 illustrates the dependence of these critical parameters (combustor exit temperature of layout B, and anode gas inlet and exit temperatures of layouts C1 and C2) on the pressure ratio. In layout B, the HRU2 inlet (combustor exit) temperature varies from 600 to 800 °C. At the optimal design point (pressure ratio of 3.0), the temperature is about 750 °C. This temperature exceeds the allowable design temperature of a normal surface type recuperator made of steel. Only the high temperature alloy can accommodate this temperature. In layout C1, the basic directly fired system, the exit temperature of the anode gas compressor may be very high. At the pressure ratio of 5.0, the required temperature is about 1000 °C. Even though the allowable operating temperature of the turbomachinery depends on the life cycle expected, 1000 °C seems too high considering that normal uncooled turbines, made of alloy, are designed for operating temperature of less than 950 °C as in most micro gas turbines. Therefore, it is more practical to design a system with lower compressor exit temperature. This can be done by the modified system C2, where the compressor inlet gas is cooled as explained before. The compressor inlet temperature depends on the degree of cooling. In this analysis, the effectiveness of the heat exchanger is set to 0.78, similar to those of other heat exchangers considered in this work. The cooling allows the inlet temperature to stay at about 400 °C and also makes the exit temperature far less than that of layout C1. At the optimum pressure ratio of 5.0, the exit temperature remains at about 800 °C. The temperature does not exceed 900 °C even at a high enough pressure ratio of 7.0. Conclusively, layouts B and C2 require a high temperature heat exchanger and an auxiliary compressor operating at high temperature, respectively, both of which must be based on high temperature alloy. Cons and pros regarding these different requirements can be investigated in a more detailed research. However, the expansibility of the directly fired system (C2) explained in the previous paragraph is a definite advantage.

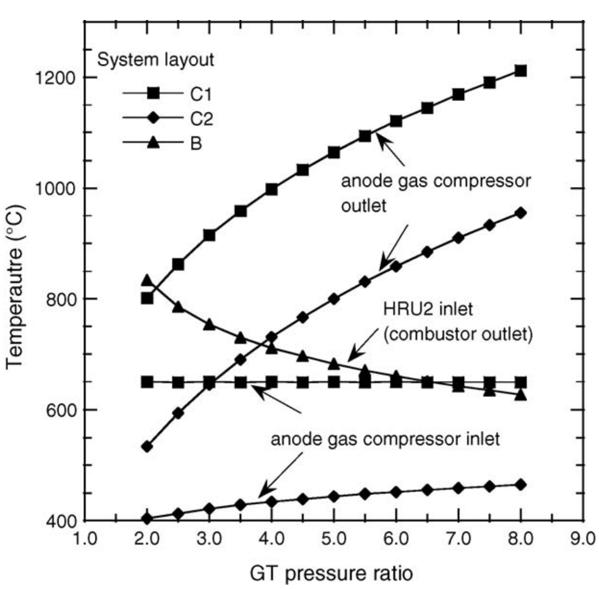

Fig. 11. Variations in critical temperatures for layouts B and C.

4. Conclusions

In this work, design analyses have been carried out for various layouts (indirectly and directly firing of the turbine inlet gas) of the ambient pressure MCFC/GT hybrid system. Research was focused on the comparison of performance among the layouts and different design requirements and limitations. The results are summarized as follows:

(1) The MCFC/GT hybrid systems exhibit higher cathode inlet temperature, and thus smaller MCFC power than the simple MCFC only system. With increasing design pressure ratio, the cathode inlet temperatures of the hybrid systems become lower, resulting in increased MCFC power.
(2) The turbine inlet temperature of the directly fired system becomes higher as the pressure ratio increases, while that of the indirectly fired systems is almost constant or decreasing. The relatively higher turbine inlet temperature of the directly fired system results in larger gas turbine power, which contributes to the far larger overall system power.
(3) The optimal pressure ratios for directly fired systems are higher (over 5) than those of indirectly fired systems (around 3). The hybrid systems may have higher efficiency than the MCFC only system by 8–12% point. The directly fired system exhibits similar or higher optimal efficiency to that of the indirectly fired system. Moreover, its efficiency was relatively insensitive to the pressure ratio, so the designer would have greater flexibility in selecting the design pressure ratio of the gas turbine.

(4) The most critical parameters in the indirectly and directly fired systems are the recuperator inlet temperature and the exit temperature of the anode gas compressor, respectively. Since both of the temperatures are sufficiently high, they can only be satisfied by high temperature alloy. In the directly fired system, the cooling of the anode gas in front of the auxiliary compressor using a recuperator may lessen the problem greatly.

(5) The indirectly fired system cannot fully accommodate the specifications of not only current commercial micro gas turbines but also advanced gas turbines. The directly fired system can accommodate the parameter values of the advanced gas turbines more favorably since it is optimized at a far higher turbine inlet temperature and pressure ratio than the indirectly fired system.

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