Strategies on energy loss reduction from high-temperature steam for stable hydrogen production using solid-recovered fuel

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Summary
A high-temperature steam generation system to supply steam to a water electrolysis system was designed and tested using solid-recovered fuel (SRF). The energy loss must be reduced to supply hydrogen production stably, which are conducted by three strategies: (a) using a double pipe, (b) installing a baffle inside the pipe to obstruct steam flow, and (c) bypassing the overflowing steam. Double pipe reduced energy loss by 25% compared to single pipe. Consequently, a heat source with a temperature of 973 K or higher was obtained. In addition, CFD simulation was performed over a temperature range of 373 to 973 K to investigate the change in energy loss with the temperature of the external fluid. When the three baffles were installed inside the double pipe, it reduced heat dissipation approximately 6%. Therefore, installing three or four inner baffles inside the double tube proved to be the most effective method, and it was confirmed that the temperature of the external fluid should be maintained above 573 K. It was concluded that the system using the double pipe with inner baffles can produce approximately 43.8 ton/year of hydrogen when generating high-temperature steam, and the CO₂ emission is reduced to 1.1 ton-CO₂/ton-hydrogen compared to liquefied natural gas.

KEYWORDS
energy loss reduction, high-temperature steam, hydrogen production, solid-recovered fuel

1 | INTRODUCTION

Climate change and global warming are closely related to greenhouse gas emissions, particularly resulting from the combustion of fossil fuels.¹ The COP 26 UN Climate Change Conference in 2021 continues to develop the UN Framework Convention on limiting global temperatures rising above the industrial level, with several countries committing to a 50% reduction of CO₂ emissions.² Andreoni and Galmarini³ studied indicators (environmental technologies, changes in the structure and efficiency of energy systems, and economic activities) that contribute to reduce CO₂ emissions of a country. Consequently, the efficiency improvement of energy production systems is the most important factor to reduce CO₂ emissions in line with policies. Singh and Kennedy⁴

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developed a tool to estimate the global CO$_2$ emissions based on a regression model. In particular, according to the CO$_2$ emissions by region by the IEA, in 2020, China had the highest CO$_2$ emissions of 11 759 million metric tons, followed by United States, India, and Russia.\textsuperscript{5} China and India have also shown increased emissions compared to a decade ago. Mardani et al\textsuperscript{6} reviewed the previous research on the CO$_2$ emissions of 55 countries in the past 20 years (1995-2017). It is worth noting that until 2017, China and the Middle East regions accounted for a large portion of the total volume, approximately 48%. It led to a deduction that CO$_2$ emissions were directly related to industrial modernization and economic development.\textsuperscript{7,8} Particularly, in China, the growth rate of CO$_2$ emissions is decreasing depending on the use of coal.\textsuperscript{9} In recent years, to understand CO$_2$ emission flows, comparative analysis has been conducted between regions, companies, and countries, and research is being actively performed with interest in policies\textsuperscript{10,11} and technologies that control energy consumption.\textsuperscript{12} However, these studies have suggested that the approach of reducing the consumption of fossil fuels has limitations in regard to addressing the depletion of resources and energy; thus, major focus should be on developing renewable energy sources that can replace the fossil fuels. Therefore, in order to reduce CO$_2$ emission, the use of renewable energy sources that can replace fossil fuels should be increased, but there are limitations in that it is difficult to supply continuously or the energy density is low.\textsuperscript{13}

Hydrogen is always a main concern as a green energy source that can overcome this limitation.\textsuperscript{14} The energy density per mass of hydrogen is approximately 142 MJ/kg, which is approximately 2.6 times higher than liquefied natural gas (LNG) and 6.2 times higher than Coal.\textsuperscript{15} Likewise, its relatively high energy density allows it to store more than 5 times energy per kg than lithium-ion batteries. Although hydrogen is being continuously produced nowadays, the large portion is generated by reforming fossil fuels. However, during this process, a large amount of CO$_2$ is discharged, significantly affecting the atmospheric environment.\textsuperscript{16,17} Thus, research is underway to develop technologies that can produce hydrogen without using fossil fuels.\textsuperscript{18} Solid oxide electrolysis cell (SOEC) is to produce hydrogen through electrolysis of high-temperature steam.\textsuperscript{19,20} It is an excellent method that utilizes the phenomenon that the electrical energy demand decreases due to an endothermic reaction as the steam temperature rises.\textsuperscript{21} Therefore, SOEC is presently gaining significant interest, which decreases energy demand by 15% compared to low-temperature water electrolysis.\textsuperscript{22} Yadav and Banerjee\textsuperscript{23} analyzed the electrolysis efficiency in the temperature range 873 to 1273 K to derive the optimal energy efficiency of the hydrogen production system. It increased to 9.1% at 873 K and 12.1% at 1273 K, resulting in enhanced system efficiency depending on the steam temperature. It was emphasized that in order for such a system to be used for industrial purposes, it was necessary to make a demonstration experiment on a pilot scale. However, the production of high-temperature steam requires a fuel having a higher heating value, and the SOEC using low-temperature steam should be developed.

The higher heating value of the solid-recovered fuel (SRF) used in this study is about 26 MJ/kg, which corresponds to first of the five grades of SRF presented in CEN EN 15359,\textsuperscript{24} and it can be manufactured by processing waste-based energy sources.\textsuperscript{25} This renewable energy source is extensively used in industrial facilities because of its large distribution and high thermal performance.\textsuperscript{26,27} However, Aracil et al\textsuperscript{28} reported the decrease in system efficiency by being treated through a general landfill or incineration process. In addition, it was emphasized that a review on the performance characteristics of SRF should be made considering thermochemical analysis of particle size, moisture content, and fuel composition parameters. In particular, Nasrullah et al\textsuperscript{29} reported that 62% of the total input energy content was recovered in the form of SRF. They optimized SRF recovery system by changing mass and materials, which recovered up to 75% of energy. Most of the research has focused on the SRF combustion system, and there is no research that uses it to produce high-temperature steam and applies steam to hydrogen production.

In this article, SRF, high-temperature steam generator (HTSG), and the integrated system linked to SOEC are investigated, and this is the first study attempted for industrial application. The process of generating high-temperature steam from HTSG and supplying it to SOEC under the influence of heat supply generated during SRF combustion is a new approach,\textsuperscript{30} shown in Figure 1. By supplying homogeneous steam, stable hydrogen production is possible, which not only reduces waste but also greatly contributes to CO$_2$ reduction.\textsuperscript{31,32} Energy loss reduction due to this can be expected for high-efficiency long-term operation in industrial fields. Therefore, in order for the SRF-SOEC integrated system to be utilized, high-temperature steam must be stably supplied to the hydrogen stack, and a stable vapor delivery strategy is an important factor to realize the energy loss reduction of the SRF-SOEC system and to increase the overall efficiency. In this study, waste treatment and energy acquisition were simultaneously attempted through the SRF-SOEC system, which was not previously linked. The possibility of producing high-temperature steam for SOEC was considered by using SRF, which is currently
underutilized and treated by landfill or incineration as fuel. In particular, since uniform steam must be stably supplied to SOEC, the stability of the system was experimentally verified by identifying the energy loss. Herein, the energy loss of combined SRF-hydrogen production system is evaluated during the high-temperature steam transport process. The heat energy produced by the system is 15 MJ/h, where the energy loss is estimated to be approximately 9.4%. The purpose of this study is to predict and reduce energy loss with maximizing hydrogen production.

2 | EXPERIMENTAL PROCEDURES

2.1 | Experimental apparatus

A system for producing high-temperature steam with 973 K or higher is constructed using SRF as shown in Figure 2. It is composed of a bed-type primary combustion furnace, a bed-type secondary combustion furnace, HTSG that generates high-temperature steam, a steam boiler, and posttreatment facilities. In particular, the exhaust gas contains ash and harmful elements, and posttreatment to prevent the formation of sediment is important. Therefore, in this study, a bag filter with excellent filtration efficiency was used, which is a device that can achieve a dust collection efficiency of more than 99% for fine dust (PM 2.5). The exhaust gas with posttreatment is discharged in the form of cleaned gas through a stack. High-temperature steam is generated through multistage approach. The low-temperature steam produced by the steam boiler flows the inside of the HTSG. The inlet conditions are as follows: steam temperature is 450 K, steam pressure is 0.3 MPa, and the mass flow rate of steam is 100 kg/hour. SRF, which is a heat source, is combusted in the primary combustion furnace, and the gas temperature generated during this process is over 1200 K. The gas flows to the secondary combustion furnace, and the gas heat is supplied to the
HTSG of the chamber. The low-temperature steam (under 450 K) in the HTSG is continuously affected by the gas heat to become a high-temperature steam (above 973 K). The high-temperature steam generated during this process is supplied to the SOEC through the steam outlet pipe of the HTSG.

The pipes used in this system must be made of materials with high corrosion resistance to hydrogen and maintain their mechanical strength at high temperature. Stainless steel 316 was found to be the most suitable for high-temperature applications owing to its lowest thermal conductivity and a high corrosion resistance.33 Based on these results, stainless steel 316 was used in the present study to build the HTSG and the pipes. The outer and inner diameters of the pipes were 0.043 m and 0.037 m, respectively. Given that these pipes were intended for supplying high-temperature steam to the water electrolysis system, Cerakwool insulation was applied. The experimental results were collected by the data acquisition system using Labview 2018. The inlet and outlet gas temperatures of the combustion furnace, the inlet and outlet steam temperatures of the HTSG, the steam pressure, and the steam flow rate were measured. It is important to reach a steady state for stable hydrogen production, and after that, the system will be operated for a long time (at least 7 days). It takes approximately 30 minutes for the steam temperature to reach 973 K, but after that, it operates in a stabilized state that means steam can be uniformly supplied to the SOEC.

### 2.2 Data reduction

The amount of heat supplied was calculated based on the fuel composition of SRF. The amount of heat received by steam is calculated as Equation (1), where $\dot{Q}_s$ is heat transfer rate of steam, $\dot{m}_s$ is mass flow rate of steam, $h_{i,s}$ is enthalpy of inlet steam, and $h_{o,s}$ is enthalpy of outlet steam, respectively.

$$\dot{Q}_s = \dot{m}_s \cdot (h_{o,s} - h_{i,s})$$  \hspace{1cm} (1)

In order to calculate the amount of heat input of SRF, the chemical composition of the sample was analyzed at Korea laboratory accreditation scheme (KOLAS), which is shown in Table 1. Most of the SRF used in the experiment consisted of combustible waste and consisted of 3 kg per pack. In the process of sorting combustion waste, SRF is selected as a uniform component by a specialized waste company. Therefore, for chemical analysis, a portion of SRF was collected from three packs among all the SRFs used in the experiment and crushed to 1 to 2 mm for analysis. The calorific value, C, H, N, S, and O were measured by SRF quality test and analysis method of the Ministry of Environment in South Korea. In addition, ash, volatile, and fixed carbon were measured in dry basis by standard test methods specified in ASTM 7582-15.34 The SRF is mainly composed of C, H, N, S, and O, which account for 88.3%, and ash is 11.8%. Incombustibles are composed of inorganic components or trace metal components in the form of oxides and remain as ash. In addition, in proximate composition, moisture, ash, volatile content, and fixed carbon are 100% on a dry basis, which is the same standard. Here, the SD for the sample is an average of approximately 5.8%. The inlet flow rate of SRF is set to 48 kg/hour, and the heat transfer rate of SRF is 3.9 MJ/h in the experiment. The temperature of SRF and air was assumed to be 293 K, and the specific heat of static pressure between SRF and air was 0.3 and 0.31 kJ/kg K, respectively. The total combustion calorific value is 976 MJ/h based on the low calorific value.

### 2.3 Design of double pipe to reduce energy loss

A double pipe was manufactured and used in the experiments to reduce the energy loss of the high-temperature steam as presented in Figure 3. The diameter of the inner pipe was designed to be the same as that of the existing pipe at the outlet. Its detailed specifications are listed in Table 2. The space between the inner and outer pipes was filled with air at atmospheric pressure. The external surface of this double pipe was insulated with 0.1 m thick Cerakwool, similar to the single pipe surface. The energy loss refers to the difference in heat at the initial supply point and the end point of the HTSG pipe, which is based on Equation (2)
The overall heat transfer coefficient ($R$) is calculated using the Equation (3). In the equation, $h_i$ is the heat transfer coefficient of the steam inside the pipe, and $h_o$ is the air-side heat transfer coefficient of the double pipe. The pipe is subject to natural convection, which is considered as heat transfer coefficient of air-side outside the pipe ($h_o$). The outside the pipe is subject to the effects of convection and radiation.

$$Q = R A_p (T_s - T_a)$$

Steam flows inside the inner pipe, and the range of Reynolds number ($Re$) is 4200 to 4500. The steam-side heat transfer coefficient ($h_i$) and Nusselt number ($Nu_i$) are calculated through Equations (4) and (5). The Friction factor ($f$) and the Prandtl number ($Pr$) can be calculated through Equations (6) and (7), respectively. The Gnielinski correlation considering the forced convection was used.

$$h_i = \frac{Nu_G k_s}{D_1}.$$  

$$Nu_G = \frac{f}{8} Re Pr \left[ 1 + 5 \sqrt{\frac{f}{8} (Pr - 1)} \right]^{-1} .$$

where

$$f = (1.82 \log(Re) - 1.64)^{-2}$$ and

$$Pr = \frac{\nu}{\alpha} (0.5 < Pr < 2000) .$$

The air-side heat transfer coefficient ($h_o$) and the Nusselt number ($Nu_c$) of the natural convection are calculated through Equations (8) and (9), where the Rayleigh number ($Ra$) is calculated through Equation (10).

As a process for verifying Churchill’s correlation, it is compared with McAdams’ correlation considering the low flow rate and isolation of the outer pipe (secondary flow), it is calculated by Equations (11) and (12).

$$h_o = \frac{Nu_o k_o}{D_{hyd}} .$$

$$Nu_c = \left\{ 0.6 + \frac{0.387Ra^{0.167}}{[1 + (0.559)^{0.563}]^{0.258}} \right\}^2 (Ra \leq 10^{12}) ,$$

where $Ra = \frac{g \beta (T_{sur} - T_a) D_{hyd}^3}{\nu \alpha}$.

$$Nu_f = 0.59 (GrPr)^{0.25} .$$

$$Gr = \frac{g \beta \rho^2 (T_{sur} - T_a) D_{hyd}^3}{\mu^2} .$$

Thermal resistances in pipe are inner fluid for forced convection ($R_1$), inner conduction ($R_2$), outer conduction ($R_3$), and outer fluid for free convection ($R_4$). Free convection has a double effect, one from the inner pipe wall to the insulated wall and the other from the insulated wall to the environment. Free convection must therefore have two components; however, since the thermal resistances of $R_2$ is 0.00085 W/m K and $R_3$ is 0.00081 W/m K, which are negligible and is considered a single value.

Since there is no flow inside the outer pipe, natural convection is considered. Thus, the air-side external heat transfer coefficient ($h_o$) can be obtained through

\[ \begin{align*}
R &= \frac{1}{\frac{D_{hyd}}{h_i} + \frac{D_{hyd} \ln(D_1/D_2)}{2k_i} + \frac{D_{hyd} \ln(D_1/D_2)}{2k_o} + \frac{L}{h_o}} \\

\end{align*} \]
Equation (13), where the radiant heat transfer coefficient ($h_r$) can be calculated using the Equation (14). $\sigma$ is the Stefan-Boltzmann constant of $5.67 \times 10^{-8}$ W/m$^2$ K$^4$, and $\varepsilon$ is the emissivity of 0.3. $T_{\text{sur}}$ is the surface temperature of the outer pipe, and $T_a$ is the external air temperature of 300 K.

\[
h_r = h_r + h_a. \tag{13}
\]

\[
h_r = \frac{\sigma \varepsilon (T_{\text{sur}}^4 - T_a^4)}{T_{\text{sur}} - T_a}. \tag{14}
\]

### 2.4 Uncertainty analysis

The uncertainty of the experimental system was calculated using the methods of Abernethy et al\textsuperscript{41} of Equations (15)-(18). The uncertainty method by Abernethy deals with the case where different dependent outcomes have different degrees of freedom when calculating the main outcome. It is suitable for use in the multisample experiment in this study because it reduces the error so that the measurement result becomes a more reasonable value. $U_c$ is the experimental uncertainty, $A$ is the bias uncertainty, $B$ is a random standard uncertainty, and $\sigma$ is the SD of the experimental values ($x$) of the temperature, pressure, and mass flow rate of the steam.

\[
U_c = 4 \sqrt{\sum_{n=1}^{N} \left(\frac{\partial X}{\partial x_n} B_{pn}\right)^2 + \sum_{n=1}^{N} \left(\frac{\partial X}{\partial x_n} A_{pn}\right)^2}, \tag{15}
\]

\[
B_p = \frac{\delta}{\sqrt{N}}, \tag{16}
\]

\[
X = X(x_1,x_2,x_3,\cdots,x_n), \quad \text{and} \tag{17}
\]

\[
\delta = \sqrt{\frac{1}{N-1} \sum_{n=1}^{N} (X_n - \bar{X})^2}. \tag{18}
\]

The measurement devices used in the HTSG system are summarized in Table 3, and the errors are listed in Table 4. The uncertainty was analyzed for about 30 minutes after the system was stabilized, and the experimental uncertainty in the energy loss is 5.19%.

### 3 Numerical Analysis

Double pipe was used to solve the problem of energy loss largely occurring in single pipe. Computational fluid dynamics (CFD) was used for optimal design through parametric analysis of double pipes. This section describes the CFD simulations conducted to determine the temperature change when air or steam flows through the outer pipe of the double pipe. A commercial CFD software package, ANSYS Fluent 19.0, is used for the simulation. The mesh is fine-tuned to examine the temperature change of the steam more accurately with respect to the outer fluid temperature. The inner steam is expected to undergo more significant changes when the outer fluid temperature is varied, and thus, more cells are added to this part, as shown in Figure 4A. The rate of convergence can be affected by time, grid size, and quality. To increase the precision, the mesh independence test was repeatedly performed. When the number of mesh elements was more than 2 077 500, it was less than 0.1% of the steam temperature for all cells. The average skewness is 0.22, and the average orthogonal quality is 0.87, all of which are sufficient to detect the flow of the steam. The Reynolds number at the pipe outlet is in the range of 11 747 to 12 455.

The solver for the analysis was set up for pressure-based, steady-state flow, and absolute velocity. The Boussinesq model, which assumes that the density is constant, is applied as the momentum equation. Simplec was used for in addition and pressure-velocity coupling, and

| Parameters      | A (%) | B (%) | $U_c$ (%) |
|-----------------|-------|-------|-----------|
| Steam temperature | 0.014 | 0.13  |           |
| Steam pressure   | 0.032 | 0.38  |           |
| Steam flow rate  | 1.75  | 0.86  | 5.19      |

**Table 3** The specifications of the measuring devices

|                        | Type               | Operating range | Accuracy   |
|------------------------|--------------------|-----------------|------------|
| Steam temperature [K]  | K-type             | –473-1523       | ±0.02      |
| Steam pressure [kPa]   | Pressure transducer| 0–1000          | ±4.71      |
| Mass flow rate of steam [kg/hour] | Volumetric flow meter | 10.0-47.6 kg/hour | ±0.35      |
the convergence criterion for the residual of the governing equation was set to $10^{-4}$.

The $k$-$\varepsilon$ standard model is used to analyze the steam flow rate based on Equations (19) and (20) for the turbulence kinetic energy ($k$) and the rate of dissipation ($\varepsilon$). The standard model is only valid for a fully turbulent case because it is assumed that the flow is completely turbulent and that the effect of molecular viscosity is negligible.

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho ku)}{\partial x} = \nabla \cdot \left( \left( \frac{\mu}{\sigma} \right) \nabla k \right) + G_k + G_b - \rho \lambda - Y_m + S_k.$$  \hspace{1cm} (19)

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u)}{\partial x} = \nabla \cdot \left( \left( \frac{\mu}{\sigma} \right) \nabla \varepsilon \right) + C_1 \frac{\varepsilon}{k} S_k - \rho C_2 \frac{\varepsilon^2}{k} + S_\varepsilon.$$  \hspace{1cm} (20)

Here, $G_k$ is the turbulence kinetic energy generated by the mean velocity gradients, $G_b$ is the turbulence

TABLE 5 The fluid conditions of the double pipe ($X_1$)

| Fluid conditions | Inner fluid (steam) | Outer fluid (steam) |
|------------------|---------------------|---------------------|
| Temperature (K)  | 973                 | 293-973             |
| Pressure (MPa)   | 0.3                 | 0.1                 |
| Mass flow rate (kg/hour) | 100             | 100                 |

FIGURE 4 Geometric model for numerical analysis; A, double pipe and B, double pipe with inner baffles, shaped as a type of rectangular plate.
kinetic energy generated by the buoyancy, and $C_1$ and $C_2$ are constants. The applied fluid conditions for the double pipe are listed in Table 5, and the outside of the outer pipe is assumed to be an adiabatic wall. Given that the temperature was measured at the center of the pipe cross-section during the experiments, for the simulation, temperature data from the same position are used. The physical properties of the steam were calculated by Engineering Equation Solver (EES).

In addition, the thermal resistance of air-side must be increased to minimize energy loss during the high-temperature steam transport. A method of obstructing the flow of the outer pipe was considered, which inhibits the growth of thermal boundary layer. Lal$^{42}$ proposed the method using inner baffles as a method for reducing the energy loss and improving the insulation performance. Similarly, simulations are conducted by installing radial-type inner baffles to the inside of the outer pipe. Concurrently, the inner baffles, which exhibited the lowest energy loss in the study by Lal$^{42}$ are selected and analyzed, and the effects of their thickness and number are examined. Lal$^{42}$ reported that the diameter and thickness of the baffles were assumed to be sufficiently thin to allow neglecting the respective temperature gradients. The assumption is verified, and thus, it is also applied in the simulation analysis. The geometrical conditions are shown in Figure 4B, and the diameter of a baffle is expressed as Equation (21).

$$D_o - \frac{D_o - D_i}{2}$$  \tag{21}

Rectangular-plate baffles are installed on the entire length of the outer pipe to obstruct the flow. The parametric analysis is conducted in terms of the baffles thickness (1-4 mm) to determine the threshold of temperature gradient. Moreover, the effect of number of baffles on energy loss reduction is evaluated in a double pipe based on the simulation analysis. The number of the inner baffles is varied as 3 to 6, as shown in Figure 5. The thickness is fixed at 1 mm, which is an optimal thickness determined in the parametric analysis.

**RESULTS AND DISCUSSION**

**4.1 | Energy loss of pipes**

Figure 6 shows the experimental results for temperature change of steam with time for a single pipe. $X_1$ is the temperature at the outlet of HTSG, and $X_2$ is the inlet temperature of SOEC. The temperature of the steam in a single 2-m-long pipe decreased by approximately 70 K at the steam outlet. In the combined system, SRF is supplied periodically to the HTSG system, and it produces fluctuation, in other words, the prediction and control of fluctuation are very difficult in that the calorific value of SRF is not constant. Although there are technical limitations controlling industrial-scale system, the maximum uncertainty in this study is about 5.2%, which is reasonable. Thus, as the temperature at $X_1$ rises, the heat loss during transport process of high-temperature steam increases based on that the thermal resistance is proportional to the temperature difference.

The experimental conditions and measuring instruments used for the single pipe tests were also utilized for the double pipes. Thus, it is found that the temperature drops by approximately 39 K over the $\Delta X$ distance. This suggests that when the double pipe is applied, the temperature drop is approximately 35% less than that when using the single pipe. In addition, the energy losses from these insulated single and double pipes are calculated and compared with respect to the $\Delta X$ distance based on position-specific temperature data. Here, the air temperature is assumed to be 293 K. The physical properties of steam as a function of temperature have a substantially important influence on the calculation of energy loss. In this study, the properties of the steam state were used in the steam table.$^{43}$

![Figure 5](image1.png)  
**FIGURE 5** Cross-section view of the double pipe with respect to the number of inner baffles  

![Figure 6](image2.png)  
**FIGURE 6** The temperature variation during transport process of high-temperature steam
The energy loss of single and double pipes in experiments is compared in Figure 7. The temperature is measured over the total distance of 2 m in intervals of 0.4 m. As the experimental apparatus is industrial scale, small variation is sensitive enough to affect the entire system. The high-temperature steam transport process using single pipe cannot be reduced to a stable state because it is greatly affected by SRF feeding, and the temperature fluctuation range is irregular. Although the heat loss seems to increase in Figure 7, it is a fluctuation that is smaller than 5.2%. On the other hand, the comparison of heat loss between single and double pipe should be performed in terms of average heat loss as presented in Figure 7. The overall heat loss of double pipe is 25% smaller than that of single pipe. In addition, to verify Churchill’s correlation in the double pipe, it was compared with McAdams’ correlation considering the low flow rate and the isolation of the outer pipe. The error was 2.5% as shown in Figure 7, and it was confirmed that Churchill’s correlation was reasonable in the double pipe. However, the temperature change is not significant when moving from X₁ to X₂. The results confirm that the double pipe is more effective than the single pipe because its air layer provides sufficient thermal insulation, thereby significantly reducing the energy loss from the pipe. Moreover, it should be considered that the average value of heat loss is significantly reduced due to the reinforcement of the insulation in the double pipe.

### 4.1.1 Temperature variation

Simulations are conducted under the same conditions as the experiment assuming that the outer fluid is air, and the results are compared with the experimental data. The temperature distribution of double pipe, and the temperature decreases by 35 K over the ΔX. Even when the double pipe is used for improved insulation, energy loss still occurs, particularly along the inner pipe wall. The double pipe filled with air is unable to provide sufficient insulation for the entire steam system. The double pipe structure significantly reduces the energy loss of the steam present inside the pipe. However, it is still difficult to consider it as an efficient system.

Therefore, it is necessary to examine the extent at which any change in the outer fluid temperature can improve the insulation performance of the system. To this end, case studies are conducted. The physical properties of the outer fluid (steam) are listed in Table 6. It is found that as the outer fluid temperature increases, both the inner steam temperature and the inner pipe wall temperature increase, whereas the variation in the temperature of the outer fluid decreases. Figure 8 presents a comparison of the energy loss measurements at X₁ and X₂ with respect to the outer fluid temperature. It can be

![FIGURE 7 Overall energy loss in terms of pipe types](image)

![TABLE 6 Properties of the outer fluid in the double pipe](image)
seen from the simulation results that heat loss is reduced by approximately 15% when the outer fluid temperature is 293 to 573 K but is reduced by approximately 65% when the outer fluid temperature is 573 to 973 K. The energy loss tends to decrease with increasing the temperature. Notably, as the temperature exceeds 573 K, the slope starts to change, and the energy loss over the distance is found to decrease noticeably. This suggests that the temperature of the fluid inside the double pipe must be maintained at 573 K or higher if the steam inside the inner pipe is to be maintained at a high temperature.

The total heat resistance of the pipe can be divided into internal heat resistance (forced convection of inner fluid), heat conduction of the pipe, and external heat resistance (natural convection of outer fluid, air). The heat conduction of the tube is not affected by the fluid temperature in that the physical property almost constant with respect to the temperature, while the internal and external heat transfer coefficients were significantly affected by the temperature change. However, as the fluid temperature increases to more than 573 K, the thermal resistance of convection becomes not dominant factor in that thermal conduction, and radiation is gradually improved. In summary, when the fluid temperature is smaller than 573 K, the convection is dominant factor that significantly affects the overall heat loss. However, after that point, other factors including radiation/conduction are also important, which decreases the effect of fluid temperature. Therefore, it is important to increase the outer thermal resistance, a method for installing a baffle inside a double pipe was proposed in this study. It is significantly efficient to install the baffle inside the outer pipe, which obstructs flow and inhibits the growth of thermal boundary layer.

4.2 | Optimal design of double pipe

4.2.1 | Inner baffles of double pipe

Figure 9 shows the heat loss and the temperature variations in the fluid at $X_1$ and $X_2$ with respect to the baffle thickness. The inner temperature of the double pipe tends to decrease as the baffle thickness increases from 1 to 4 mm. A thick baffle obstructs air flow significantly; however, it increases the effective heat transfer area with the outside. Thermal resistance is calculated by considering the thermal conductivity according to the thickness of the baffles. When the thickness of the baffle is 1 mm, the overall heat loss is approximately 24.9 W/m K, and when the thickness is 4 mm, it is approximately 26.4 W/m K. When the thickness of the baffle is 1 mm, the heat loss reduced approximately 1.96%, and it is relatively effective compared to 4 mm. When installing the optimum thickness of 1 mm, heat loss can be reduced by 4.6% compared to the double pipe without baffles.

4.2.2 | Variation in number of inner baffles in double pipe

Figure 10 shows the temperature distribution inside the double pipe at $X_2$ with respect to the number of the inner baffles. It was expected that installing many baffles inside the double pipe would reduce the energy loss by blocking the steam flow. However, when the number of the inner baffles is 6, the outer fluid temperature and the inner steam temperature are found to decrease noticeably. In addition, in terms of the inner steam, which is fed to the water electrolysis system, the energy loss reduction decreases in the order of (a) > (b) > (c), as shown in Figure 11. The energy loss reduction ratio is approximately 1.1% higher in (a) than in (b), and the difference is approximately 2.1% when compared with (c).

The high-temperature steam transport system is found to be reasonable because the fluid temperature is maintained above 573 K, which can be used for hydrogen production. Even when the steam generation system is
located away from a hydrogen production facility where high temperature steam is needed, it is possible to prevent the inner steam temperature from dropping sharply. In addition, when using an internal baffle as a method to properly block the vapor flow inside the pipe, the strategy for the least heat loss is to construct no more than four baffles.

4.3 Effect of energy loss reduction on hydrogen production

A method for minimizing the energy loss was derived in Section 4.2. This affects the amount of hydrogen production and electricity consumption based on the steam temperature. The heat capacity in terms of three pipe types: (a) single pipe, (b) double pipe (outer fluid: air), and (c) double pipe with inner baffles is shown in Figure 12. Based on the heat capacity of double pipe with baffle installed, it is approximately 79% reduced compared to single pipe, and approximately 62% reduced compared to basic double pipe. The cases without baffles cannot be applied to water electrolysis system because the steam temperature at \( X_2 \) is less than 573 K. The water electrolysis system is generally operated only when steam with a temperature above 973 K is continuously supplied. At a temperature lower than 573 K, the electric energy demand exceeds 35%, and the significant decrease in the hydrogen production efficiency cannot be excluded.

According to the hydrogen technology development roadmap 2019 of the Ministry of Science and ICT, assuming a hydrogen conversion efficiency of 40%, if high-temperature steam is continuously supplied, 5 kg of hydrogen can be generated with 100 kg of steam. In this study, 100 kg of steam per hour is generated, and it is possible to generate approximately 5 kg of hydrogen per hour (= 43.8 ton/year).

In hydrogen production using SRF, 767 MWh of energy is harvested per year. In addition, it is possible to estimate the amount of CO2 emission reduction when using SRF. The CO2 emission per ton of hydrogen produced using SRF is estimated as 9.92 ton-CO2/ton-hydrogen, and CO2 conversion factor is 15.272 (tC/TJ). The calculation method is presented in Equations (22)-(24).

\[
\phi_F = P_{FC} \cdot G_{HG} \quad (22)
\]
\[
E_C = \phi_F \cdot \left( \frac{PCV}{106} \right) \quad (23)
\]
\[
E_{CE} = E_C \cdot 3.67 \quad (24)
\]

Hydrogen can be produced using a variety of energy sources (high-temperature steam produced from SRF, LNG, fossil fuels, and coal gas). The CO2 emissions when producing hydrogen by energy source are summarized in Table 7. When the high-temperature steam suggested in this study is used, the CO2 emission is reduced by 1.1 ton-CO2/ton-hydrogen compared to LNG, which can save approximately 10%. In particular, it is reduced by 19.28 ton-CO2/ton-hydrogen compared to low-

| Hydrogen of 1 ton/CO2 emissions (tCO2) |
|----------------------------------------|
| High-temperature steam                  | 9.92 |
| Low temperature steam                   | 28.2 |
| LNG reformer                            | 11.02|
| Coal gasification                       | 18.53|

Figure 11: Energy loss difference between \( X_1 \) and \( X_2 \) according to the number of inner baffles from the double pipe

Figure 12: Comparison of steam heat capacity according to pipe types (\( X_1 \) to \( X_2 \))
temperature steam, which is a significant difference enough to save about approximately 65%. Therefore, it is evaluated as an eco-friendly solution that can harvest energy using SRF and at the same time produce hydrogen, a clean energy source.

5 | CONCLUSIONS

High-temperature steam of 973 K or higher is produced using SRF in a combustion furnace system, which can be fed to a water electrolysis system. To reduce the energy loss of high-temperature steam, a double pipe structure is designed and used for the outlet pipe in the experiments. Experimental data and simulation results are compared and verified, and case studies are performed to minimize the energy loss. Inner baffles are installed in the pipes to increase the heat transfer area, and the changes in the energy loss are examined. The major findings of the present study are as follows:

1. When generating the high-temperature steam of 973 K or higher, a double pipe can reduce the energy loss by approximately 25% relative to that when a single pipe is used.
2. Inner baffles are installed to obstruct flow, which inhibits the growth of thermal boundary layer. The application of the inner baffles reduces the energy loss by approximately 4.6% compared to double pipe at the optimal conditions.
3. The energy loss reduction with four inner baffles or fewer is approximately 1.1 to 2.1% larger than that with six inner baffles. This configuration is determined to be the optimal approach to ensure a continuous supply of high-temperature steam to the water electrolysis system.
4. The system using the double pipe with inner baffles (outer fluid: 973 K) can produce approximately 43.8 ton/year of hydrogen when generating high-temperature steam, and the CO2 emission (per ton of hydrogen) is reduced approximately 10% compared to LNG and is reduced approximately 65% compared to the low-temperature steam.

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NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| A      | bias uncertainty |
| Ap     | area (m²) |
| B      | random standard uncertainty |
| D      | diameter (m) |
| D1     | inside diameter of the inner pipe (m) |
| D2     | outside diameter of the inner pipe (m) |
| D3     | inside diameter of the outer pipe (m) |
| D4     | outside diameter of the outer pipe (m) |
| Ec     | carbon emission (tC) |
| Ece    | CO2 emissions (tCO2) |
| f      | friction factor |
| FCF    | CO2 conversion factor (tC/TJ) |
| Gr     | Grashof number |
| g      | gravity (m/s) |
| GHG    | heat generation (MJ /Nm³) |
| ha     | air-side heat transfer coefficient (W/m K) |
| hil    | inside heat transfer coefficient (W/m K) |
| hio    | outside heat transfer coefficient (W/m K) |
| hr     | radiant heat transfer coefficient (W/m K) |
| k_s    | solid thermal conductivity (W/m K) |
| Nuc    | Nusselt number in natural convection |
| NuG    | Nusselt number in forced convection |
| m      | mass flow rate (kg/hour) |
| P      | pressure (MPa) |
| Pr     | Prandtl number |
| PFC    | fuel consumption (Nm³) |
| Q      | heat loss from the pipe (W) |
| Qs     | heat transfer rate of steam (kJ/hour) |
| R      | overall heat transfer coefficient (W/m K) |
| Re     | Reynolds Number |
| Ra     | Rayleigh Number |
| T      | temperature (K) |
| U      | experimental uncertainty |
| X      | experimental value |
| X1     | first point of the steam outlet pipe (m) |
| X2     | end point of the steam outlet pipe (m) |

GREEK SYMBOL

| Symbol | Description |
|--------|-------------|
| α      | thermal diffusion coefficient (m²/s) |
| β      | thermal expansion coefficient |
| δ      | standard deviation |
| ε      | emissivity |
| λ      | rate of dissipation |
| μ      | viscosity (kg/m s) |
| ρ      | density (kg/m³) |
| σ      | Stefan-Boltzmann constant |
| ν      | kinematic viscosity (m²/s) |
| ϕ_f    | fuel heating value (MJ) |
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