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Experimental study on multi-layered type of gas-to-gas heat exchanger using porous media

T. Tomimura [a,*], K. Hamano [a], Y. Honda [b], R. Echigo [c]

[a] Institute for Materials Chemistry and Engineering, Kyushu University, 6-1 Kasuga-koen, Kasuga 816-8085, Japan
[b] Interdisciplinary Graduate School of Engineering Sciences, Kyushu University, 6-1 Kasuga-koen, Kasuga 816-8085, Japan
[c] Department of Mechanical Engineering, Shibaura Institute of Technology, Minato-ku, Tokyo 108-0023, Japan

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Abstract

Based on an effective energy conversion method between flowing gas enthalpy and thermal radiation, a multi-layered type of gas-to-gas heat exchanger using porous media has been proposed. A series of experiments have been conducted for the inlet temperature of high temperature gas 300–700 °C, the optical thickness of porous media 0–15.4, the number of layers 2–5 and two types of walls (bare or finned) placed in the system. As a result, a heat recovery section is shown to play an important role in lowering an outer wall temperature of the system and at the same time in increasing the total heat recovery rate $H_{\mathrm{tot},N}$. In addition, it is clarified that the optical thickness of about 8 is enough to obtain sufficient $H_{\mathrm{tot},N}$, and the finned walls are quite effective to promote $H_{\mathrm{tot},N}$ under the present experimental conditions.

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Keywords: Energy conversion; Heat exchanger; Porous medium; Thermal insulation; Thermal radiation

1. Introduction

Lots of methods have been proposed for heat transfer augmentation in high temperature facilities by utilizing thermal radiation. The effective energy conversion method from flowing gas enthalpy to thermal radiation emitted from porous metal plates proposed by Echigo [1] can be given as one of the promising methods. Applying the conversion method, Echigo [2] and Tanigawa et al. [3] have proposed a new type of gas-to-gas heat exchanger equipped with a pair of circular porous metal plates. From their analytical and experimental studies conducted on a high temperature combustion gas, it is shown that this type of heat exchanger has much higher overall heat transfer coefficients than the conventional heat exchangers [3], and further the heat exchanger also exhibits excellent reaction characteristics as a steam-reformer [4,5].

In succession from their studies, for the purpose of improving the heat exchanger efficiency and at the same time insulating the system effectively, the authors proposed a self-insulated type of gas-to-gas heat exchanger with a heat recovery section [6,7]. The heat exchanger has three-layered structure, and consists of the following sections; a low temperature section, a high temperature section, and a heat recovery section. Each section is equipped with a circular porous metal plate, and is separated by an opaque solid wall. The high temperature section is placed in between the heat recovery and low temperature sections. This arrangement, along with a directional effect of low temperature gas in the heat recovery section flowing toward the high temperature section, enables an outer wall of the heat exchanger to be self-insulated. This idea of the effective thermal

* Corresponding author. Tel.: +81-92-583-7824; fax: +81-92-583-7638.
E-mail address: tomi@cm.kyushu-u.ac.jp (T. Tomimura).

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Nomenclature

c_p	specific heat of working gas at constant pressure
$H_{\mathrm{tot},N}$	total heat recovery rate of multi-layered gas-to-gas heat exchanger defined by Eq. (1)
m	mass flow rate of working gas
N	number of layers
T	temperature or absolute temperature
u_{m}	mean velocity of working gas
V	volumetric flow rate of working gas
x	distance measured from surface of porous metal plate along gas flow

Greek symbols

ρ	density of working gas
τ_{o}	optical thickness of porous metal plate

Subscripts

hr	heat recovery section
h	high temperature working gas
ht	high temperature section
i	inlet
l	low temperature working gas
lt	low temperature section
m	mean
N	number of layers
o	outlet
w	outer wall of heat exchanger
1	first
2	second
∞	ambient

insulation came from an approaching flow type of the original heat exchanger [2], and is also the same as that of the active thermal insulation proposed by Maruyama et al. [8]. From the studies performed on a high temperature gas of 400–600 °C in experiments and 400–1200 °C in numerical analysis by a simple one-dimensional model [6,7], it has been clarified that the heat recovery section is essential to reduce the outer wall temperature of the heat exchanger, and the total amount of heat recovered by adding the heat recovery section is roughly doubled on average.

Recently, we designed a new multi-layered type of gas-to-gas heat exchanger equipped with porous metal plates, and conducted an investigation on basic heat transfer performance [9]. The significant differences between the previous and the present heat exchangers are the shape of the cross-section of the main body and the porous metal plate [circular (previous) or square (present)], and the shape and the total cross-sectional area of the piping for the high and low temperature gases [cylindrical tubes (previous, 12.6 cm^2) or rectangular ducts (present, 80 cm^2)]. The present heat exchanger is so designed that it has lower pressure loss in the piping and much simpler structure for layer extension. The major purpose of the present study is to present detailed heat transfer characteristics of the newly designed gas-to-gas heat exchanger. A series of experiments have been carried out for investigating the effects of the inlet gas temperature of the first high temperature section (300–700 °C), the optical thickness of porous metal plate (0–15.4), the heat exchanger structure (i.e. the number of layers, 2–5), and the type of outer, separating and insulated walls on the temperature distributions in the porous metal plates and in the heat exchanger system, the outer wall temperature, and the total heat recovery rate.

2. Basic concept of heat exchanger using porous media

As illustrated schematically in Fig. 1 [2], when a high temperature gas of T_i flows through a duct, wherein a porous metal plate with extremely high porosity is installed, a large temperature drop ΔT occurs in the porous plate with appropriate optical thickness τ_{o}, and larger amount of converted radiant energy $(-q_x^{\mathrm{R}}(0))$ propagates to upstream direction of the gas flow.

The phenomena are based on the heat transfer mechanism shown in Fig. 2 [1]. The enthalpy of the high temperature gas is effectively transferred to the porous metal plate via an extremely high heat transfer coefficient between the flowing gas and the porous plate with

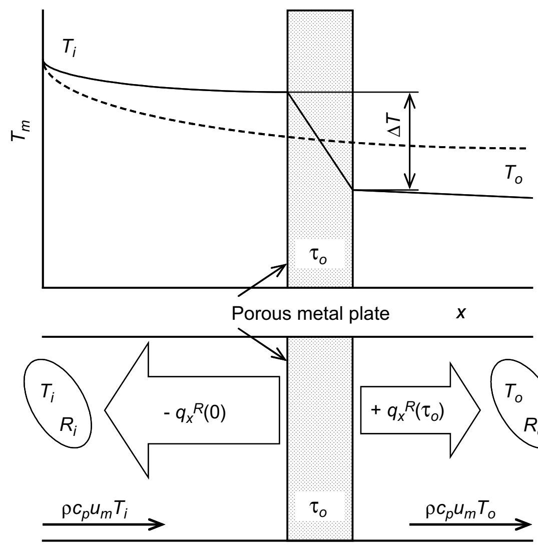


Fig. 1. Schematic sketch on effective energy conversion.

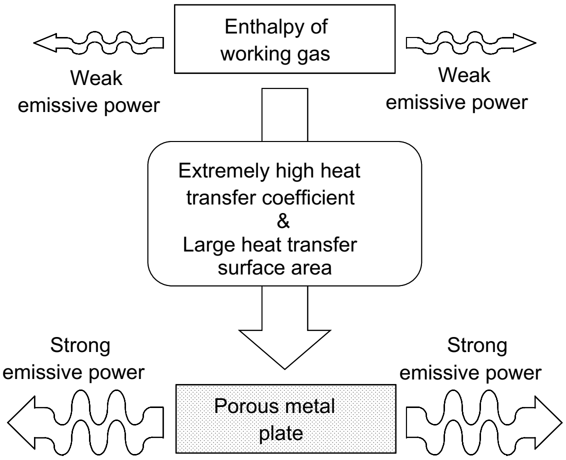


Fig. 2. Effective energy conversion mechanism.

fine mean pore size, and at the same time a substantially large surface area of the porous plate for heat transfer. Then the larger amount of energy transferred from the gas is emitted to upstream direction by the porous plate with strong emissive power.

From the above explanation and the fact that the reversed conversion from strong incident radiation to gas enthalpy is also feasible very effectively, a basic two-layered type of gas-to-gas heat exchanger can be comprised by putting the two ducts with porous metal plates to each other via a separating wall (see Fig. 3(a)). In the study, although experiments have been conducted using an opaque separating wall made of heat-resisting stainless steel as a first step, such a transparent separating wall like a quartz glass plate would be much more effective for the present type of heat exchanger.

3. Experiment

A series of experiments were carried out for four kinds of heat exchanger (HE) structures; i.e., two, three, four and five-layered heat exchangers. From them, three representative structures are shown schematically in Fig. 3. Here, the arrows drawn by solid and broken lines represent the flow directions of high and low temperature gases, respectively. In the actual HE system, however, the flow pattern between the high and low temperature gases is a cross-flow. The rectangle shown by pixel pattern represents a porous metal plate and the thick solid line between them shows a separating wall.

The two-layered HE shown in Fig. 3(a) consists of a first high temperature section (HT1) and a low temperature section (LT) separated by an opaque solid wall. This system has the same structure proposed by Echigo [2] and Tanigawa et al. [3], and is a starting point in the present study. The term "Approaching flow" noted just above Fig. 3(a) is given to describe the flow direction of the low temperature gas in the LT against the HT1, and all the present experiments were performed under this flow direction. The plane view and the side view of each section (i.e., HT1 or LT in this case) are shown in Fig. 4 for reference. The three-layered HE shown in Fig. 3(b), on the other hand, is composed of the LT, HT1 and a first heat recovery section (HR1). This HR1 is added to increase heat recovery rate from the high temperature gas based on a phenomenon [1] that if a heat sink is

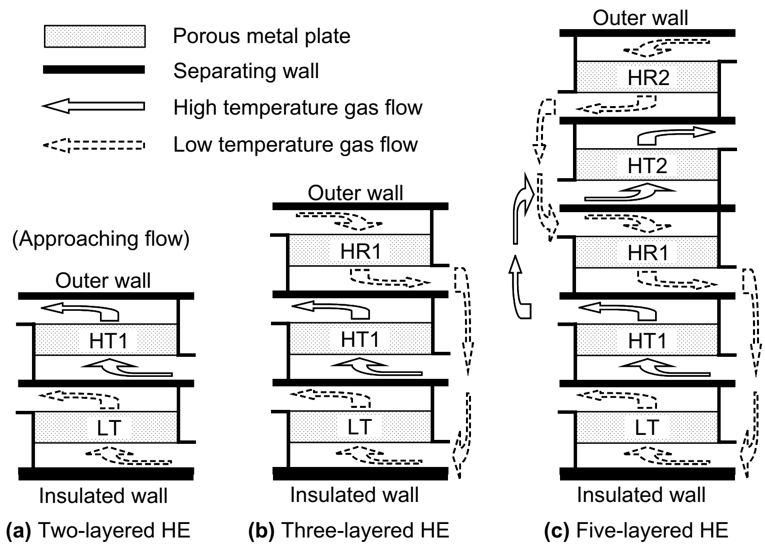


Fig. 3. Schematic of multi-layered type of gas-to-gas heat exchanger.

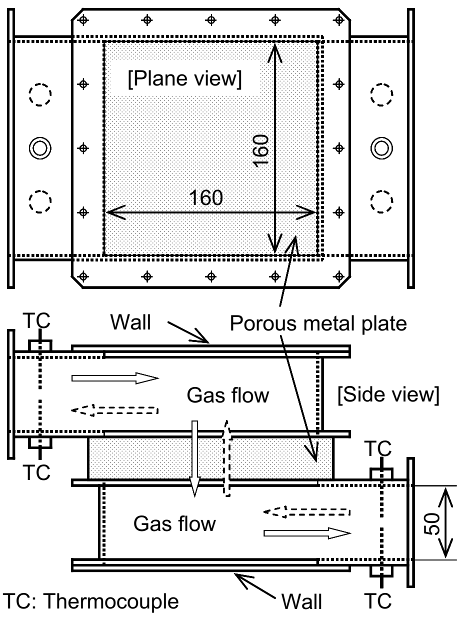


Fig. 4. Schematic view of each section composing gas-to-gas heat exchanger.

placed in the downstream section the temperature drop within the porous metal plate becomes more pronounced without inducing the temperature decrease in the upstream region. In addition to this advantage, the HR1 plays a role of effective thermal insulation of an outer wall of the HE system. For the purpose of increasing the heat recovery rate further, experiments were also conducted on the five-layered HE shown in Fig. 3(c), which is made up of the LT, HT1, HR1, and second high temperature and heat recovery sections (HT2 and HR2).

Fig. 5 shows a schematic example of the present experimental apparatus for the three-layered system. Each section has an interior cross-sectional area of 160 mm × 160 mm (see Fig. 4), and the heat-resisting black paint is applied on the inside surface of the body. The working gas for the HT1 drawn by a solid line is blown to an electric heater via a flow meter, and heated up to the prescribed temperature. After leaving the heater, the heated gas flows into and out of the HT1. Whereas, as shown by a broken line, the air of the ambient temperature is blown into the HR1 through a flow meter, and then sent into the LT.

The insulated wall, the outer wall and the separating walls are made of heat-resisting stainless steel plates of 210 mm in length, 210 mm in width and 2 mm in thickness, and we call them bare walls here. In this study, for the purpose of increasing the total heat recovery rate especially in the lower temperature region, experiments using the walls with zigzag shaped fins like those shown in Fig. 6 were also conducted. The fins were

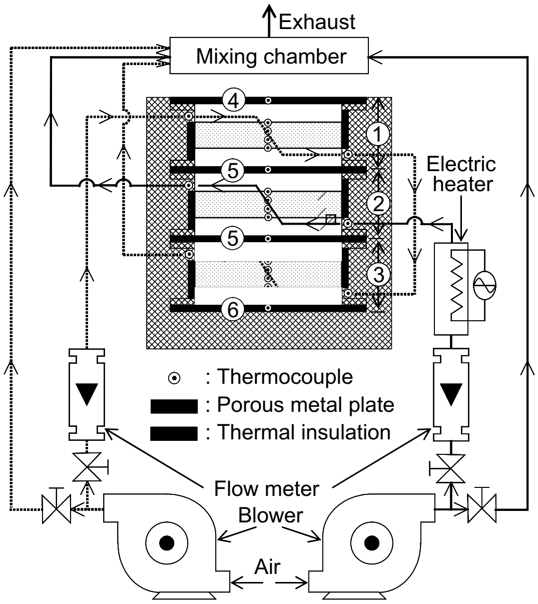


Fig. 5. Schematic example of experimental apparatus (three-layered heat exchanger).

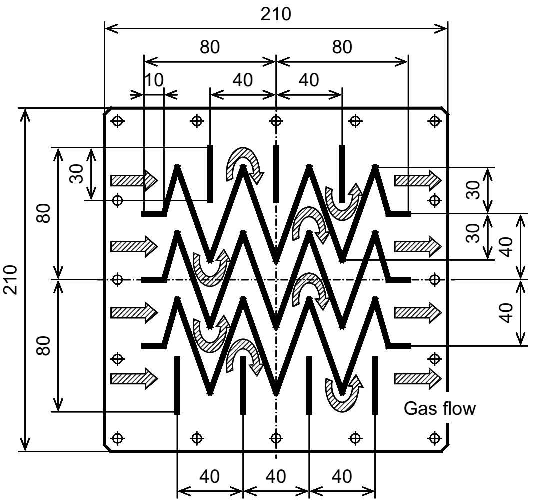


Fig. 6. Schematic of wall with zigzag shaped fins.

spot welded on the walls, and are called finned walls against bare walls. Here, the fin height were designed 3 mm higher than the internal height of each section (see Fig. 4) to minimize the gap between the fin tip and the porous metal surface, which arises from the connecting flange (2 mm in thickness) and the packing sheet (also, 2 mm in thickness) to stop the air leakage. While the insulated wall or the outer wall has the fins on one side of course, the separating walls have them on both sides intersected perpendicularly. On each surface of those walls heat-resisting black paint is applied to realize a black body surface approximately.

The porous metal plate placed in each section is made of nickel and chromium foamed alloy, and the overall dimensions for the length, width and thickness are 185, 185 and from 10 to 60 mm, respectively. The porosity, specific surface areas and the absorption coefficient of the porous plate are 93%, 1700 m^2/m^3 and 258 m^{-1} [1], respectively.

The axial temperature distributions of the porous plates were measured with 0.5mm OD chromel–alumel sheathed thermocouples placed along the plate central axis (see Fig. 5). Since the diameter of the thermocouple is approximately the same as an equivalent diameter (= 0.86 mm) of the porous plate, the measured temperature is regarded as the plate temperature. On the other hand, the gas temperatures at the inlet and the outlet in each section (see Figs. 4 and 5), the outer wall temperature, the separating wall temperatures, and the insulated wall temperature (see Fig. 5) were measured with 1.0 mm OD chromel–alumel sheathed thermocouples, respectively.

The key parameters in the present experiments are selected as follows; inlet gas temperature of the HT1 $T_{\mathrm{ht1},i}$: 300, 400, 500, 600 and 700 C (573, 673, 773, 873 and 973 K), optical thickness of porous metal plate τ_o = (absorption coefficient) × (thickness of porous plate): 0 (without porous plate) to 15.4, heat exchanger structure (i.e. number of layers) N: 2, 3, 4, 5, type of outer, separating and insulted walls: bare or finned, volumetric flow rate of high and low temperature gases V: 0.35 m^3/min, flow direction of low temperature gas in the LT: approaching flow.

4. Results and discussion

4.1. Temperature distributions in porous metal plates

Figs. 7 and 8 show examples of the temperature distributions in each porous metal plate for the two- and three-layered heat exchangers with bare walls for 700 °C (973 K) level of $T_{\mathrm{ht1},i}$ and $\tau_o = 7.7$. Here, the arrows drawn by solid and broken lines mean the flow directions of the high and low temperature gases in the porous plates, and the transverse axis x represents the distance along the gas flow measured from the upstream side of the porous plate.

In both systems, since the upstream side of the LT is thermally insulated (see Fig. 3(a) and (b)), the temperature profile in that region is almost uniform. On the other hand, in the HR1 of the three-layered HE system shown in Fig. 8, although the upstream side of the HR1

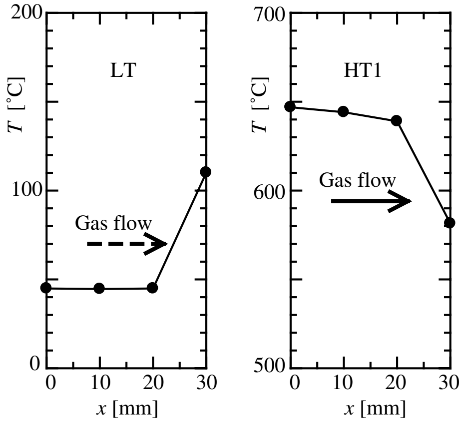


Fig. 7. Example of temperature distributions in each porous metal plate (two-layered heat exchanger with bare walls, $T_{\mathrm{ht1},i} = 704$ °C, $\tau_o = 7.7$, $V = 0.35$ m^3/min).

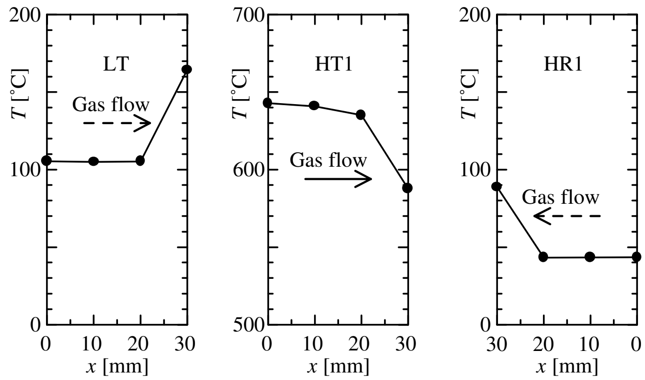


Fig. 8. Example of temperature distributions in each porous metal plate (three-layered heat exchanger with bare walls, $T_{\mathrm{ht1},i} = 698$ °C, $\tau_o = 7.7$, $V = 0.35$ m^3/min).

(i.e. the outer wall) is not insulated, almost uniform temperature distribution is attained in the upstream region. This virtual insulation of the HR1 outer wall is realized by reversed conversion from incident radiation emitted by the HT1 to sensible heat of the low temperature gas flowing toward the HT1 through the porous plate in the HR1. Further, from these figures, it can be seen that placing the HR1 in the downstream side of the HT1 is effective for increasing heat recovery rate further from the high temperature gas without disturbing the temperature field in the upstream region of the HT1 [1].

4.2. Temperature distributions in heat exchanger

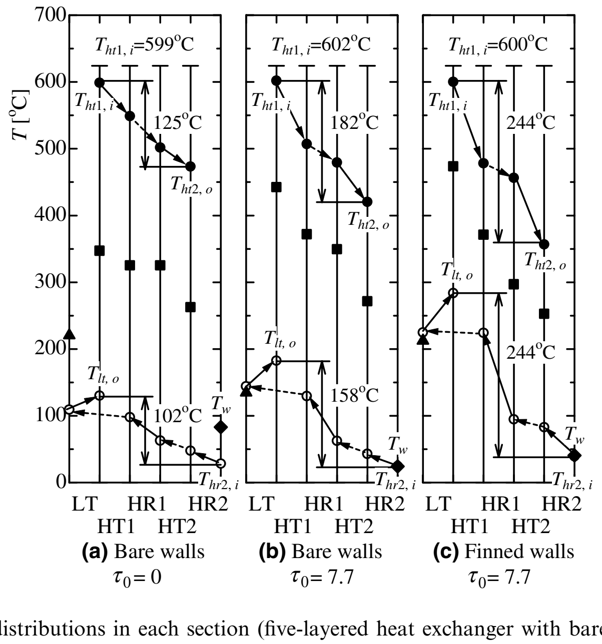
Fig. 9 shows an example of the inlet and outlet gas temperatures, and the separating wall, insulated wall and outer wall temperatures for the five-layered heat exchanger for 600 °C (873 K) level of $T_{ht1,i}$. Here, in the figure, each section corresponds to the space between the vertical lines which symbolically represent the separating wall, insulated wall and outer wall. The closed and open circles stand for the inlet or outlet gas temperatures of the high and low temperature gases. The closed square, triangle and diamond represent the temperatures of separating wall, insulated wall and outer wall, respectively. The arrow drawn by the solid line represents the flow directions of the high or low temperature gases in the HT1, HT2, HR2, HR1 and LT. Whereas, the arrow drawn by the broken line shows the directions of those gases flowing through the ducts which connect the HT1 to the HT2, the HR2 to the HR1, or the HR1 to the LT.

Fig. 9(a) shows the temperature distributions of the heat exchanger with bare walls for $\tau_o = 0$, i.e., without porous metal plates, and Fig. 9(b) shows the corresponding temperature distributions for $\tau_o = 7.7$. Further, Fig. 9(c) gives those with finned walls for $\tau_o = 7.7$.

As seen from Fig. 9(a) and (b), while the temperature rise of the low temperature gas ($= T_{lt,o} - T_{hr2,i}$, here $T_{lt,o}$ is the outlet gas temperature of the LT and $T_{hr2,i}$ is the inlet gas temperature of the HR2) for $\tau_o = 0$ is 102 °C, that for $\tau_o = 7.7$ becomes 158 °C. This difference reflects the effective energy conversion between flowing gas enthalpy and thermal radiation by the porous metal plates. In addition, when all the bare walls are replaced with finned ones, the temperature rise $T_{lt,o} - T_{hr2,i}$ increases up to 244 °C as shown in Fig. 9(c). This explains that the finned walls are quite effective to promote heat recovery from the high temperature gas in the present range of $T_{ht1,i}$.

4.3. Outer wall temperature

Placing the porous metal plate or applying the finned wall in the heat recovery section is expected to help lower the outer wall temperature T_w as shown partially in the above Fig. 9. First, the effects of $T_{ht1,i}$, τ_o and N on T_w of the heat exchanger with bare walls are shown in Fig. 10. The open and closed circles represent $\tau_o = 0$ and 7.7, respectively. The dash-dotted, broken, two-dot chain and solid lines stand for the two, three, four and five-layered systems.



(a) Bare walls $\tau_0 = 0$ **(b)** Bare walls $\tau_0 = 7.7$ **(c)** Finned walls $\tau_0 = 7.7$

Fig. 9. Example of temperature distributions in each section (five-layered heat exchanger with bare or finned walls, $T_{ht1,i} \cong 600$ °C, $\tau_o = 0$ or 7.7, $V = 0.35$ ms^3/min).

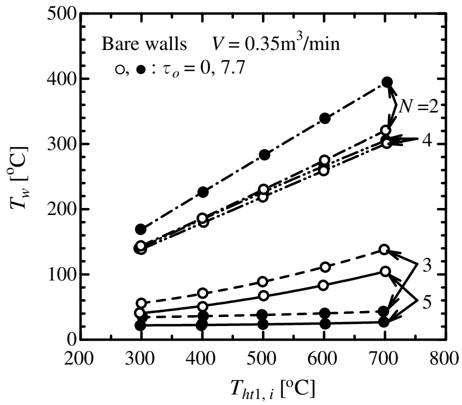


Fig. 10. Effects of $T_{\mathrm{ht1},i}$, τ_o and N on outer wall temperature T_w of heat exchanger with bare walls.

In the case of the two or four-layered heat exchanger, i.e. without the heat recovery section on the HT1 or the HT2 (see Fig. 3 for instance), T_w becomes extremely high. This is simply because the outer wall of the HT1 or the HT2 is exposed directly to the surrounding air. Further, in the case of $\tau_\mathrm{o} = 7.7$, since the outer wall is heated additionally by radiation from the porous plate placed in the HT1 or the HT2, T_w becomes higher than that for $\tau_\mathrm{o} = 0$.

On the other hand, in the case of three or five-layered heat exchanger, T_w is effectively lowered just by adding the HR1 or the HR2 to the system. In addition to this effect, through combined effects of the absorption of incident radiation from the HT1 or the HT2 and the flow of the low temperature gas toward the HT1 or the HT2, T_w can be further reduced by simply placing the porous metal plate in the HR1 or the HR2.

The effects of $T_{\mathrm{ht1},i}$, τ_o and the type of wall on T_w of the five-layered heat exchanger are also shown in Fig. 11. Here, the meaning of the open and closed circles is the same in Fig. 10, and the broken and solid lines represent the bare and finned walls. As seen from the figure, although T_w of the system with finned walls for $\tau_\mathrm{o} = 0$ or 7.7 is slightly higher than that with bare walls for $\tau_\mathrm{o} = 7.7$, almost the same effect on T_w reduction is obtained.

4.4. Total heat recovery rate

Fig. 12 shows the effects of $T_{\mathrm{ht1},i}$ and τ_o the total heat recovery rate $H_{\mathrm{tot},3}$ of the three-layered heat exchanger with bare walls, which is defined generally by the following equation.

$$H_{\mathrm{tot},N} = \frac{m_\mathrm{l} c_\mathrm{pl}(T_\mathrm{lt,o} - T_\infty)}{m_\mathrm{h} c_\mathrm{ph}(T_{\mathrm{ht1},i} - T_\infty)} \quad (N = 2\text{–}5) \tag{1}$$

Here, m_h and c_ph, m_l and c_pl are the mass flow rate and the specific heat at constant pressure of the high and low temperature gases, respectively. $T_{\mathrm{ht1},i}$ is the inlet gas temperature of the HT1, and $T_\mathrm{lt,o}$ is the outlet gas temperatures of the LT. Further, T_∞ represents the inlet temperature of the low temperature working gas to the system, i.e. $T_\infty = T_\mathrm{lt,i}$, $T_{\mathrm{hr1},i}$, $T_{\mathrm{hr1},i}$ or $T_{\mathrm{hr2},i}$ for two, three, four or five-layered system, respectively. Here, $T_\mathrm{lt,i}$, $T_{\mathrm{hr1},i}$ and $T_{\mathrm{hr2},i}$ are the inlet temperatures of LT, HR1 and HR2.

In the low temperature case of $T_{\mathrm{ht1},i} = 300$ °C, the rate $H_{\mathrm{tot},3}$ increases almost linearly with τ_o. As the temperature level of $T_{\mathrm{ht1},i}$ becomes higher, however, the rate $H_{\mathrm{tot},3}$ rises more rapidly with increasing τ_o, and the optical thickness of about 8 seems enough to obtain sufficient heat recovery rate $H_{\mathrm{tot},N}$ under the present experimental conditions.

The effect of $T_{\mathrm{ht1},i}$ and N on the total heat recovery rate $H_{\mathrm{tot},N}$ of the heat exchanger with bare walls is shown in Fig. 13 for the optical thickness $\tau_\mathrm{o} = 0$ and 7.7. As seen from the figure, in the case of two-layered heat

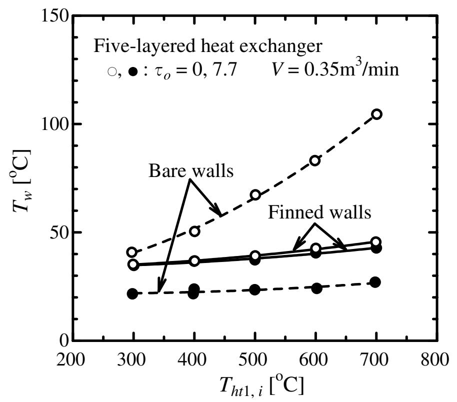


Fig. 11. Effects of $T_{\mathrm{ht1},i}$, τ_o and type of wall on outer wall temperature T_w of five-layered heat exchanger.

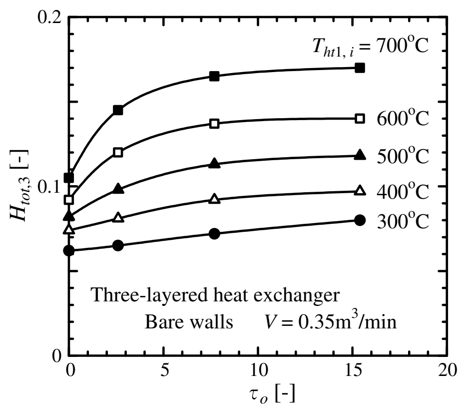


Fig. 12. Effects of $T_{\mathrm{ht1},i}$ and τ_o on total heat recovery rate $H_{\mathrm{tot},N}$ of three-layered heat exchanger with bare walls.

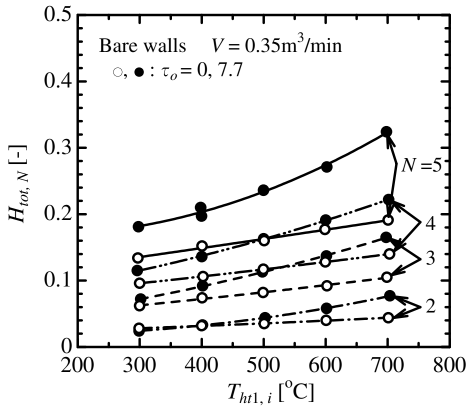


Fig. 13. Effects of $T_{ht1,i}$ and N on total heat recovery rate $H_{tot,N}$ of heat exchanger with bare walls for $\tau_o = 0$ and 7.7.

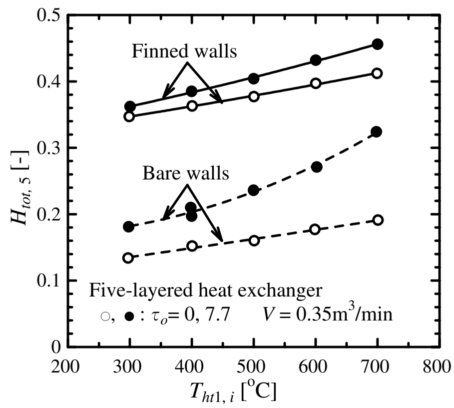


Fig. 15. Effect of type of wall on total heat recovery rate $H_{tot,N}$ of five-layered heat exchanger for $\tau_o = 0$ and 7.7.

exchanger, the difference between $\tau_o = 7.7$ and $\tau_o = 0$ (i.e., with and without porous metal plates) is quite small. However, when it comes to the three, four and five-layered systems, the effect of the HR1, HT2 and HR2 addition on $H_{tot,N}$ becomes remarkable, and the difference in$H_{tot,N}$ with and without porous metal plates gets larger with increasing $T_{ht1,i}$.

The effect of the type of wall on the total heat recovery rate $H_{tot,N}$ of the three and five-layered heat exchangers for $\tau_o = 0$ and 7.7is shown in Figs. 14 and 15, respectively. As clearly seen from these figures, the rate $H_{tot,N}$ is effectively increased by adopting finned walls in the present range of $T_{ht1,i}$. As a result, in the case of three-layered heat exchanger, $H_{tot,3}$ is increased by 3.1–2.4 times for $\tau_o = 0$ and 2.8–1.7 times for $\tau_o = 7.7$ in the temperature range of $T_{ht1,i} = 300$–700 °C. Similarly, $H_{tot,5}$ is increased by 2.6–2.2 times for $\tau_o = 0$ and 2.0–1.4 times for $\tau_o = 7.7$.

In Fig. 14, the results obtained in the previous study [6,7] by using the three-layered heat exchanger equipped

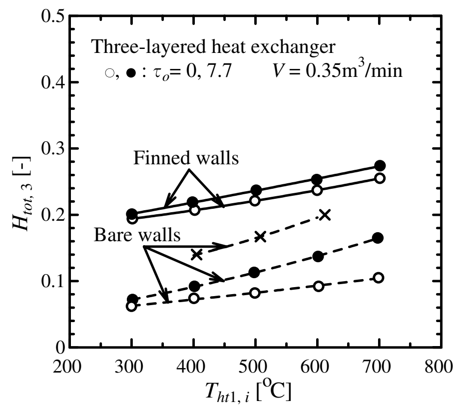


Fig. 14. Effect of type of wall on total heat recovery rate $H_{tot,N}$ of three-layered heat exchanger for $\tau_o = 0$ and 7.7.

with bare walls and circular porous metal plates of the same cross-sectional area as the present heat exchanger are also shown by the cross symbol for reference. [In the previous study [6,7], the mean gas velocities are mistakenly shown as 0.11 and 0.33 m/s. Their correct figures are 0.08 and 0.23 m/s, respectively.] Compared with the present results for $\tau_o = 7.7$, the rate $H_{tot,3}$ previously obtained is about 1.5 times higher on average in spite of thinner optical thickness of $\tau_o = 6.5$. The significant factors affecting this difference could be the shape of the cross-section of the main body [circular (previous) or square (present)], and the total cross-sectional area of the pipes [(previous, 12.6 cm^2) or ducts (present, 80 cm^2) which connect the HR1 and the LT and comprise the inlet and/or outlet portion of the HT1, HR1 and LT]. However, precise explanation to the cause has not been given yet, and that is the subject for a future study.

5. Conclusions

A series of experiments have been performed to investigate heat transfer characteristics of the newly proposed multi-layered type of gas-to-gas heat exchanger using porous media. The major findings of the present study are summarized as follows:

1. Placing the heat recovery section in the downstream side of the high temperature section is effective to lower the outer wall temperature T_w of the heat exchanger system and at the same time to increase the total heat recovery rate $H_{tot,N}$ further from the high temperature gas.
2. The optical thickness τ_o of about 8 of the porous metal plate is enough to obtain sufficient total heat recovery rate $H_{tot,N}$ under the present experimental conditions.

3. The effect of the heat exchanger structure, i.e. the number of layers N on $H_{tot,N}$ is remarkable, and the difference in $H_{tot,N}$ with and without porous metal plates gets larger with increase in the inlet gas temperature of the first high temperature section $T_{ht1,i}$.
4. The finned walls are quite effective to promote heat recovery from the high temperature gas. In the case of three-layered heat exchanger, compared to the heat exchanger with bare walls, the total heat recovery rate $H_{tot,3}$ is increased by 3.1–2.4 times for $\tau_o = 0$ and 2.8–1.7 times for $\tau_o = 7.7$ in the temperature range of $T_{ht1,i} = 300$–700 °C. Similarly, in the case of five-layered heat exchanger, $H_{tot,5}$ is increased by 2.6–2.2 times for $\tau_o = 0$ and 2.0–1.4 times for $\tau_o = 7.7$.

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