Heat Transfer Enhancement to Cooling Water Pipe by a Surface Combustor Heater Equipped with a Convection-Radiation Converter

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ABSTRACT

This experimental study discusses the operation of a surface combustor heater (SCH) which has the potential to enhance heat transfer in fossil-fired industrial furnaces and equipment. The SCH is a combustion heater device involving relatively cold heat exchange surfaces (or tubes) embedded in a stationary bed of fibrous material in which a gaseous fuel is burned. The primary purpose of this experimental study is to apply porous medium technology or a convection-radiation converter (CRC) to a surface combustor heater (SCH) so as to enhance the rate of heat transfer to the cooling water pipe. The effects of various parameters, which are expected to control the performance of the SCH such as operating conditions and radiation properties of the porous bed, are clarified. The results show that the coupled CRC and SCH system yields a Nusselt number of about 15 times higher than that of a force convection only system.

1. INTRODUCTION

During the last two decades, business competition and economic problems have forced the manufacturing and material processing industries to improve productivity and product quality. This means that production costs have to be lowered while the quality of the product is maintained. In various manufacturing industries, several techniques have been used to reduce the production costs. The cost of energy in the manufacturing process should be considered to be the main part of the whole cost of the product. Effective heat transfer enhancement techniques are therefore of continual interest to industry. Increasing the heat transfer efficiency of the processes should lead to greater productivity, efficiency, and in some cases reduced equipment size.

In the past, heat transfer enhancement was done by means of installing either fins to increase the heat transfer area, or installing a turbulent promoter at the heat transfer area so as to increase the heat transfer coefficient at the surface. In high temperature applications, particularly, a gaseous core nuclear reactor, plasma and high temperature heat exchanger radiation heat transfer from high-temperature combustion products play an important role in increasing the rate of heat transfer. Much effort has been made to enhance the rate of heat transfer by using radiative heat transfer as different from convection heat transfer and conductive heat transfer. This can be done by using a multiphase medium (consisting of fluid phase (gas) and particulate phase (solid or liquid)) [1-4]. Due to the better characteristics in emitting and absorbing thermal radiation of the particulate phase than those of the gases, the particulate phase receives thermal energy from the heating surface by a direct thermal radiation. Then the thermal energy is transferred to the fluid phase by thermal convection resulting in a higher heat transfer coefficient than that of the single phase system. Moreover, heat transfer in the multiphase medium is further enhanced at the heating surface due to the random motion of the particles in the turbulent flow, leading to a reduction of the thickness of the viscous sublayer. However, heat transfer enhancement by multiphase media is limited in such a case that the carrier medium (fluid phase) is gaseous. If the carrier medium is liquid (such as in this study), the heat transfer in the multiphase medium would no longer be enhanced.

This study proposes an efficient means of enhancing heat transfer to a cooling water pipe in a combustion system by using a surface combustor-heater (SCH) technology equipped with a convection-radiation converter (CRC) or porous medium. Exploratory studies of the SCH concept were done in Russia in the 1970's [5]. Both laboratory and bench scale units were studied, but the development of the concept was abandoned because of some operational difficulties encountered. The SCH is a novel combustion heat transfer device involving relatively cold heat exchange surfaces (tube) embedded in a stationary bed of refractory material of porous matrix in which a gaseous fuel is burned. As the bed is heated by the combustion products, the heat liberated is extracted by the embedded heat exchanger and the porous matrix. Due to a higher temperature of the porous matrix than that of the tube heat exchanger, thermal radiation emitted from the porous matrix is then transferred to the heat exchanger and the air/fuel mixture leading to an increase in the heat transfer coefficient at the heat transfer surfaces and the combustion temperature [6, 7]. In a conventional combustor, the presence of large areas of cold heat exchange surfaces within the combustion zone quenches the flame, thereby producing carbon monoxide and hydrocarbon emissions. Unlike the conventional combustor, the SCH technology can provide a heat transfer simultaneously with the combustion process which has the advantage of reducing the combustion temperature and suppressing formation of nitrogen oxides [8, 9]. The natural gas-fired SCH has a high combustion efficiency and firing density, high heat transfer to the load, high thermal efficiency and a moderate turndown ratio. A two-dimensional model has been developed [10] to predict the convective-radiative heat transfer in a single module of a surface combustor heater. The numerical analysis shows that the convective heat transfer coefficient between the gas and the porous matrix is a very important parameter and that, even at relatively high temperatures of the bed, convection is a factor of about two times greater than radiation. The analysis also revealed a large number of independent parameters such as operating conditions, heater geometry, tube diameter and arrangement, particle diameter as well as the radiation properties of the solid bed that control the performance of SCH.

The primary purpose of this experimental study is to understand the heat transfer characteristics within the SCH equipped with the CRC as well as to study how the operating conditions and radiation properties of the porous bed can be expected to control the performance of the SCH.

2. PRINCIPLE OF THE CONVECTION-RADIATION CONVERTER (CRC) AND ITS APPLICATION TO ENHANCEMENT OF HEAT TRANSFER AND AUGMENTATION OF COMBUSTION

The principle of a porous medium is relatively simple [11]. It consists of two heat transfer processes namely convection and radiation. The porous medium is capable of converting a part of the enthalpy of the hot gas flowing through it to thermal radiation or vice-versa as shown in Fig. 1 and Fig. 2, respectively. In Fig. 1, when hot gas is flowing through the porous medium, heat is transferred from the gas to the porous medium by convection which causes the gas temperature to decrease. Due to the facts that the porous medium has high surface area to volume ratio and that solids have a higher emissivity than gas, the porous medium would then emit thermal radiation into both the upstream and the downstream directions of the flowing gas. However, the main portion of the thermal radiation is directed toward the upstream side. The porous medium is then said to be working as an emitter. Alternatively, upon receiving an incident thermal radiation $F(L_p)$, the porous medium is able to increase the enthalpy of the flowing gas by convective heat transfer leading to an increase in the gas temperature as shown in Fig 2. The porous medium is then said to be serving as an absorber. These prominent characteristics of the porous medium can enhance the heat transfer and augment the combustion by installing that in a combustion system in which an arrangement of the porous medium could simultaneously perform as both emitter and absorber characteristics. This allows heat recirculation from the exhaust gas back to the mixture of air and fuel by thermal radiation in the combustion system as shown in Fig. 3. The right-hand porous medium works as an emitter by which the enthalpy of the flowing exhaust gas is converted

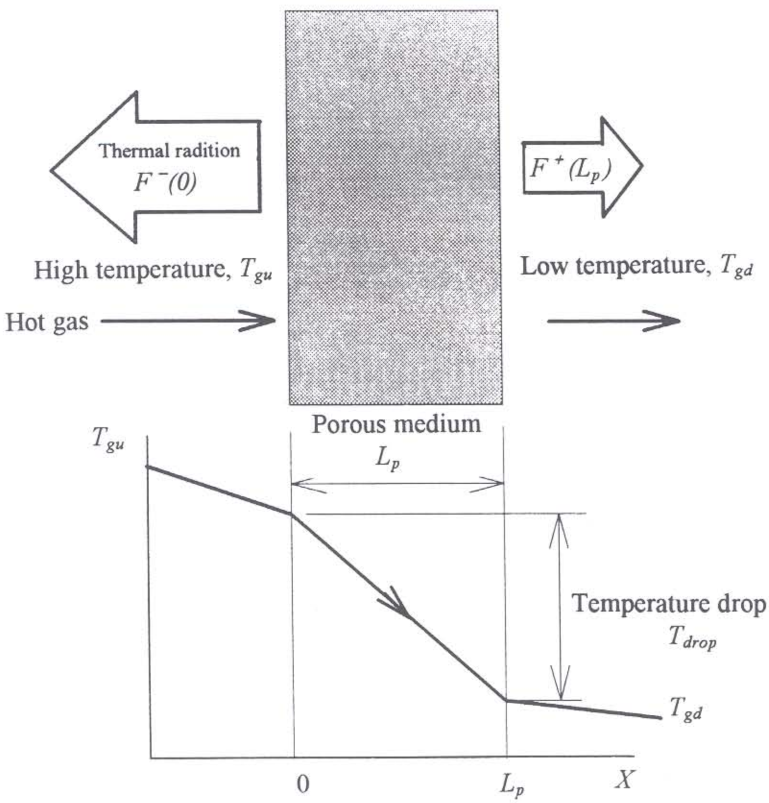

Fig. 1. Principle of porous medium working as an emitter.

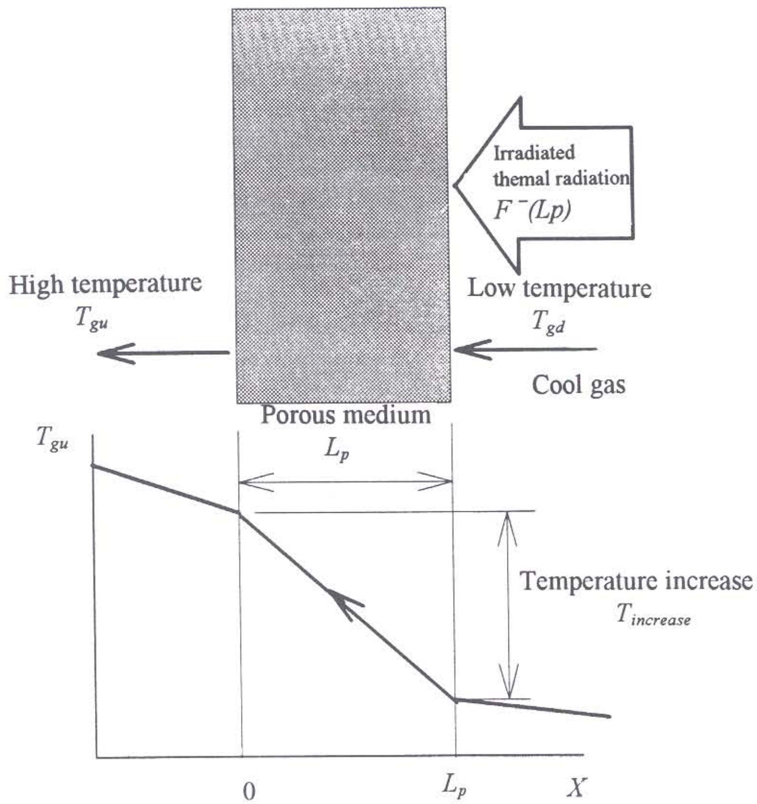

Fig. 2. Principle of a porous medium working as an absorber.

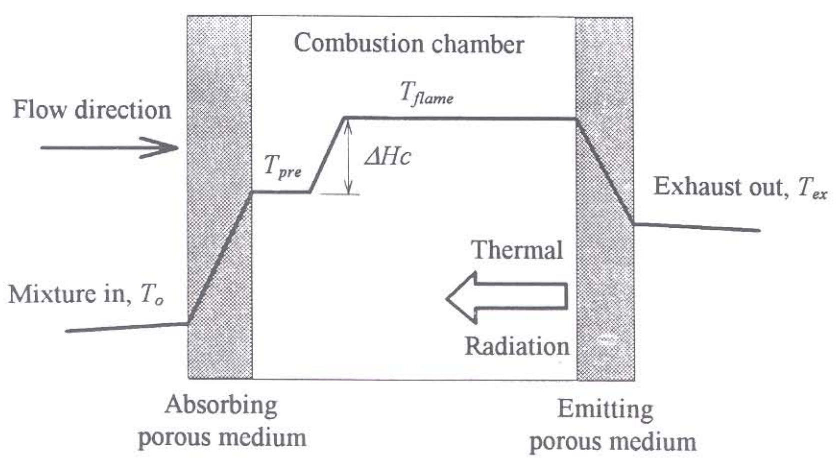

Fig. 3. Heat recirculation from exhaust gas to mixture.

to thermal radiation propagating toward the left-hand porous medium (absorber). At the left-hand porous medium, heat is transferred to the mixture by its function as the absorber. This increases the temperature of the combustible mixture from T_0 to T_{pre} prior to entering the combustion chamber. As a result, high combustion temperature T_{flame} after combustion heat released ΔH_C and good combustion augmentation are obtained.

3. EXPERIMENTAL APPARATUS AND PROCEDURES

Figure 4 exhibits the experimental furnace which consists of four main parts, a distributor, a ceramic burner, porous medium *A* and a combustion chamber constructed from porous medium *B*. The furnace is rectangular in shape and its cross-sectional area is a square of 100 mm x 100 mm. It is made of carbon steel sheet, with thickness of 3 mm. The distributor is formed by a stainless steel screen (with a mesh size of 20 meshes/inch) placed in layers, with thickness of 8 mm. The burner is made of a ceramic plate with many small holes on its surface. Due to the high temperature combustion, the wall of the combustion chamber is lined with high temperature cement. A water pipe of 14 mm diameter runs through combustion chamber to serve as a thermal load for the combustion. A mixer is installed at the outlet end of the water pipe so that the water is well stirred before its temperature is measured. The space inside the combustion chamber is filled with a stainless steel wool (porous medium *B*) to serve as a stationary package bed. Two sight glasses are installed at both sides of the combustion chamber walls to observe the combustion regime and heat transfer characteristics inside the furnace. The combustion chamber is lined above by porous medium *A*, the structure of which is similar to that of the distributor.

Type *K* thermocouples with a wire diameter of 0.5 mm were used in measuring the temperatures at various locations in the furnace as shown by dark dots and symbols. Here, T_{gi} is the inlet temperature of the fuel and air mixture. T_{ud} refers to the gas temperature at the upstream direction of the distributor. T_{ub} and T_{db} represent the gas temperature at the upstream and the downstream side of the burner, respectively. T_{ut} and T_{dt}, respectively, represent the gas temperature below and above the water pipe. T_{up} and T_{dp}, respectively, are the gas temperatures at the upstream side and the downstream side of the porous medium *A*. T_{go} is the exhaust gas temperature leaving the furnace. T_{wo} and T_{wi} represent the water outlet and the water inlet temperatures, respectively. All of these thermocouples are connected to a data logger which is coupled with a personal computer. The temperature profiles inside the furnace are monitored and displayed on the computer screen.

Liquefied petroleum gas (LPG) was used in the experiment, consisting of propane (C_3H_8) 40% (by vol.) and butane (C_4H_{10}) 60%, respectively, with a higher heating value of about 26,000 kcal/m^3. An air compressor was used for supplying combustion air to premix with the fuel before entering the distributor.

To start up the experiment, a pilot burner flame was inserted through a side glass hole installed just above the burner. Then the premixed fuel, with an appropriate equivalence ratio Φ (ratio of theoretical air to an actual air) of about 0.8 was gradually introduced into the furnace. After the combustion took place (by observation through another sight glass) the pilot flame was withdrawn from the furnace and the sight glass was fitted. The flow rate of the combustion air was gradually increased (by reducing the equivalence ratio Φ) so as to keep a constant combustion temperature level within the sustainable limit of the material used in forming the furnace. The water was then allowed to flow through the water pipe and the flow rate was adjusted so as to keep a constant temperature difference of 10°C throughout the experiment. It would not be appropriate

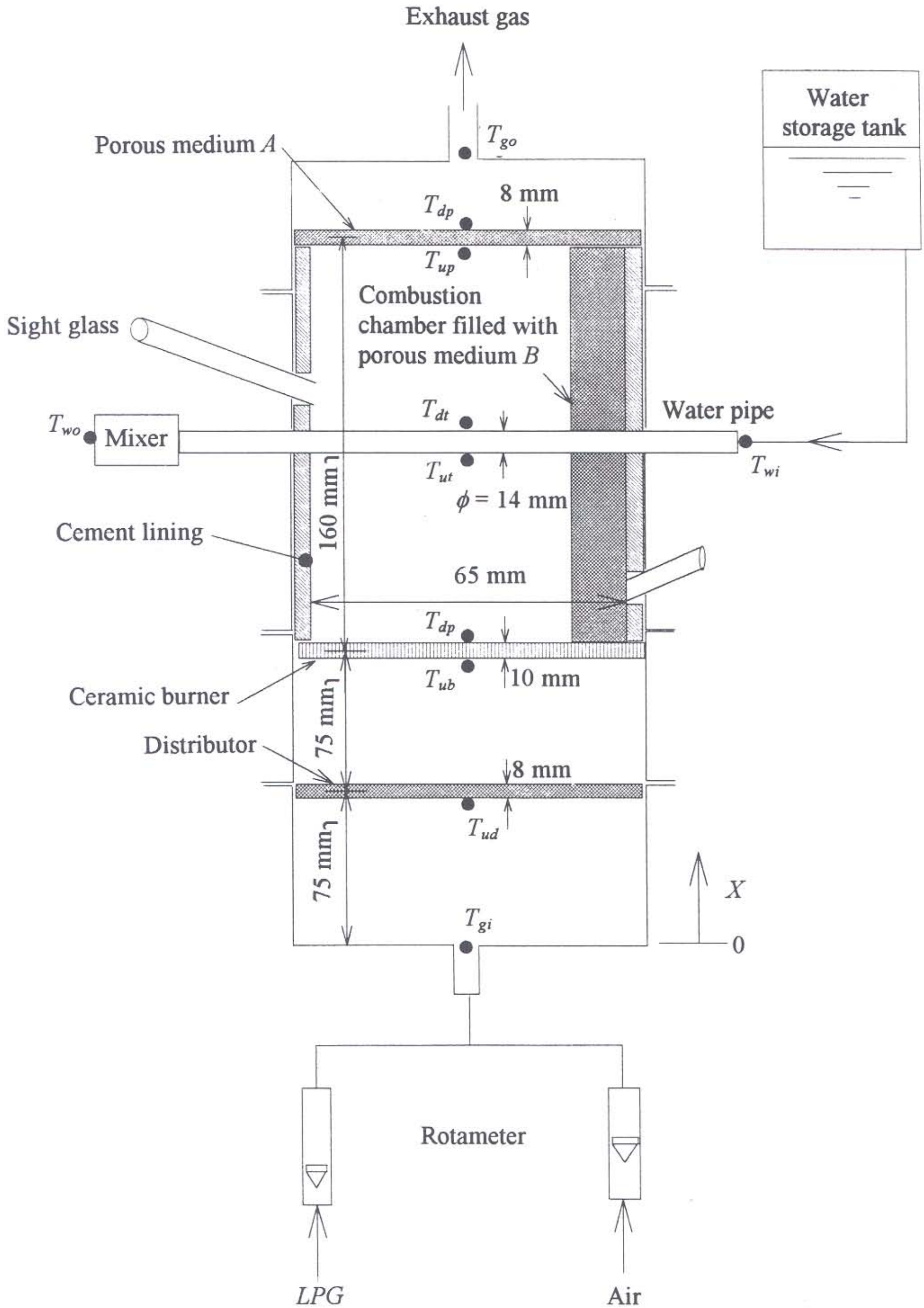

Fig. 4. Experimental furnace.

if the temperature difference T_{WO}-T_{WI} is greater than 10 °C, because T_{WO} will exceed 40°C (T_{WI} is initially at 31°C) leading to partial vaporization of the water and causing a fluctuation in the mass flow rate of water which is then difficult to measure accurately.

The experiment was conducted to study the effect of the rate of combustion L_c, the effect of equivalence ratio Φ and the effect of the optical thickness τ_B of the porous medium B, on the heat transfer characteristics at the water pipe. The effect of τ_B was of great importance in this experiment since the interaction between the gas and the solid phase (fibrous material) in the combustion chamber is expected to play a vital role in enhancing the heat transfer to the cooling water pipe.

4. RESULTS AND DISCUSSIONS

4.1 Effect of Rate of Combustion L_c

Figure 5 shows the effect of the rate of combustion L_c on the thermal structure in terms of the axial temperature distributions of the furnace. The experimental results were obtained at a constant equivalence ratio (Φ = 0.64), an optical thickness (product of absorption coefficient (m^{-1}) and geometrical thickness (m)) of the porous medium B, (τ_B = 10) and a constant cooling water temperature difference (T_{wo}-T_{wi} = 10°C). Three rates of combustion L_c = 1.0 kW, 1.2 kW and 1.4 kW were used in the experiment and they are equivalent to a gas velocity of about 30 mm/s, 36 mm/s and 43 mm/s, respectively. The temperatures at the upstream and downstream side of the water pipe (T_{dt} and T_{ut}) in Fig. 4 and Fig. 5 do not represent the wall temperatures of the water pipe, but the temperatures of the gas in the vicinity of the lower and upper surface of the water pipe.

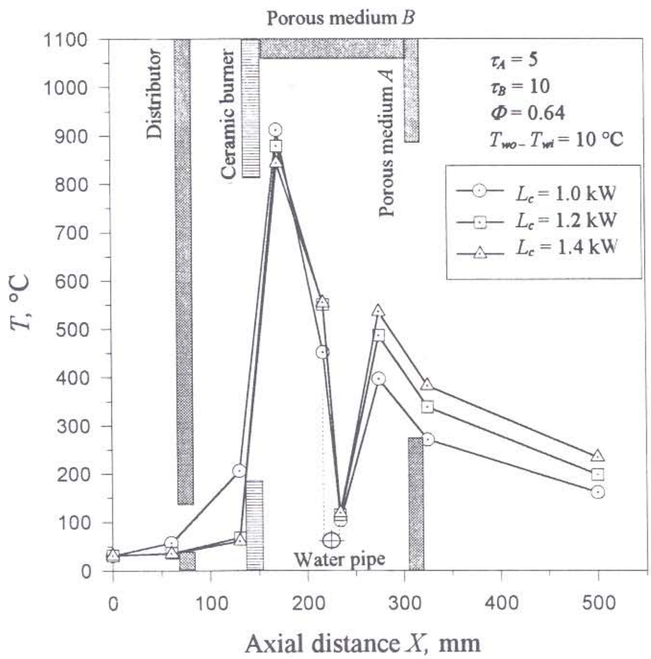

Fig. 5. Effect of rate of combustion L_c on axial temperature distribution in furnace.

The gas mixture flowing through the distributor was preheated in order to increase its temperature prior to entering the ceramic plate burner. It was noticed that, at the lower rate of combustion where L_c = 1.0 kW, there was a significant increase in temperature. This is because the distributor absorbs some of the thermal radiation emitted from the downstream side of the ceramic plate burner and then transfers this to the flowing gas mixture. The gas temperature therefore sharply increases as it flows through the ceramic burner. Since the combustion flame stabilizes on the downstream side of the ceramic plate burner in the porous medium B, the temperature of the mixture reaches a maximum value in this region. Thermal radiation losses from the porous medium B to the upstream direction of the combustion flame and to the cooling water pipe, cause the gas temperature to decrease rapidly while it is flowing towards the cooling water pipe embedded in the porous medium B. The decrease in this gas temperature is enhanced by a forced convection heat loss around the cooling water pipe. However, the gas temperature increases again when the gas has flowed passing the cooling water pipe. This may be attributed to mixing of hot and cold gases and the interaction of the thermal radiative heat transfer between the porous media A and B. The porous medium A is serving as a radiation shield in which a portion of the enthalpy of the hot gas is converted to thermal radiation directing toward the porous medium B. This causes a decrease in the gas temperature while it flows through the porous medium A. By contrast, in the original SCH technology, there is no such porous medium A present. The enthalpy of the high temperature gas is therefore largely lost to the ambience; this situation is improved by the presence of the porous medium A.

It became clear that as the rate of combustion L_c increased, the temperature close to the combustion flame falls uniformly, whereas on the downstream side of the cooling water pipe the temperatures increased. This may be due to the fact that the gas velocity increases with the rate of combustion L_c. The location of the maximum flame temperature therefore tends to move towards the cooling water pipe. However, in comparison to convective heat transfer in which the transferred heat strongly depends upon temperature and velocity, radiative heat transfer (for example in this case, between the porous medium A and the cooling water pipe) depends only on the temperature profile of the porous medium. Therefore, no matter how the rate of combustion L_c is, the combustion flame is well stabilized inside the porous medium B and between the surfaces of the ceramic plate and the porous medium A.

4.2 Effect of the Equivalence Ratio Φ

Figure 6 shows the effect of the equivalence ratio Φ on temperature at a constant optical thickness and T_{WO}-T_{WI} = 10°C. Here, the equivalence ratio Φ means the ratio of the stoichiometric air required to the actual air supplied. Increase in Φ from 0.6 to 0.68 causes a result opposite to that of increase in the combustion rate L_c. At $\Phi = 0.68$, a prominent increase in temperature of the mixture prior to entering the ceramic plate burner was observed. This result is explained using the same discussion given for the effect of L_c on temperature when L_c = 1.0 kW as shown in Fig. 5.

4.3 Effect of Optical Thickness τ_B of the Porous Medium B

Figure 7 shows a typical effect of the optical thickness τ_B of the porous medium B, on the axial temperature profiles T. τ_B was changed from 0, 10 and 20 as the rate of combustion L_c and the equivalence ratio Φ were kept constant at 1.4 kW and 0.64, respectively. Here, τ_B = 0 means no porous medium B at all in the space between the ceramic plate burner and the porous medium A. τ_B was varied from 10 to 20 by doubling the mass of the stainless steel wool in the space.

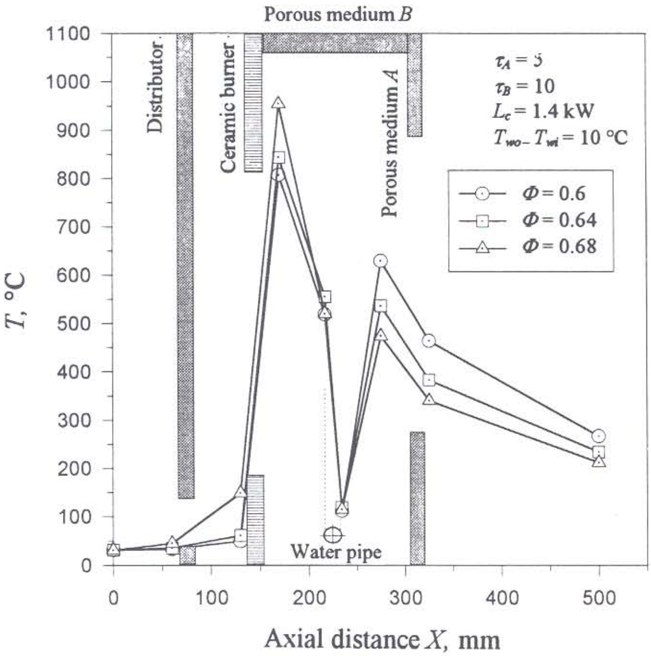

Fig. 6. Effect of equivalence ratio Φ on axial temperature distribution in furnace.

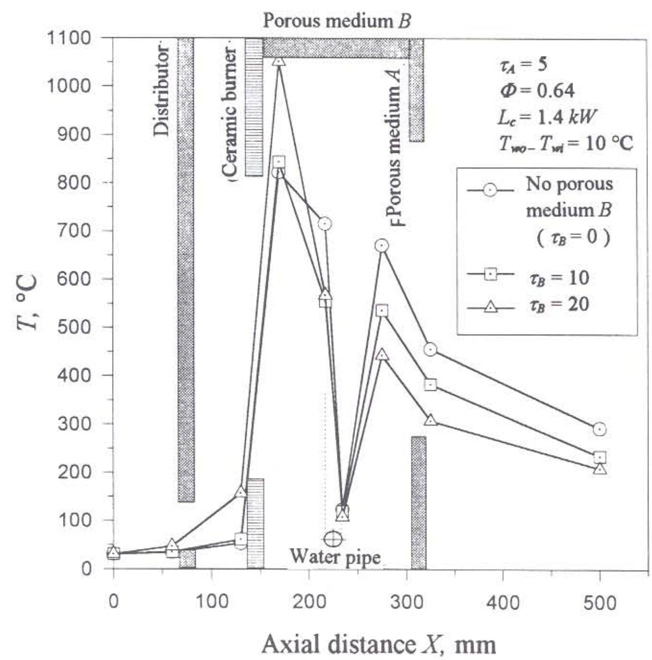

Fig. 7. Effect of optical thickness (τ_B) of the porous medium B on axial temperature distribution in furnace at $L_c = 1.4$ kW.

Basically, the temperature profiles are similar to those shown in Fig. 5 and Fig. 6. Figure 7 shows that the temperature T in the porous medium B, between the ceramic plate burner and the cooling water pipe, clearly increases when τ_B increased, whereas the temperature T between the cooling water pipe and the porous medium A decreases when τ_B increased. This is attributed to a more even distribution of heat recirculation from the flame location to the ceramic plate burner and its upstream side as τ_B increases. Increase in the optical thickness τ_B of the porous medium B means increase in its absorption coefficient and the effect of thermal radiation shield of the porous medium B. Thus, more thermal radiation energy arising from the convection of gas enthalpy is obtained, and is directed towards the upstream side of the combustion zone. The combustion zone tends to appear at a narrower region closer to the ceramic plate burner. A prominent preheating effect at $\tau_B = 20$ could be expected compared with at $\tau_B = 0$ and at $\tau_B = 10$. Since the most amount of heat is transferred to the upstream side of the porous medium B, the remaining heat that would be transferred to the cooling water pipe may become less. Figure 8 and Fig. 9 show effect of optical thickness τ_B at the combustion rate L_c of 1.2 and 1.0 kW, respectively. In Fig. 9 data for $\tau_B = 20$ could not be obtained due to flashback. The results show the same trend as in Fig. 7.

4.4 Heat Transfer Characteristics

Heat Flux

In order to clarify the heat transfer characteristics, an average heat flux at the wall of the cooling water pipe H_p is calculated from the temperatures measured at inlet and outlet of the cooling water pipe. Figure 10 and Fig. 11, respectively, show the effect of the optical thickness (τ_B) of the porous medium B, on the heat flux H_p at various rates of combustion L_c and equivalence ratio Φ. From Fig. 10 the equivalence ratio Φ is kept constant at $\Phi = 0.64$, whereas in Fig. 11 a constant rate of combustion $L_c = 1.4$ kW is used in the experiment.

Figure 10 shows an average magnitude of heat flux H_p. Although the temperature of the porous medium between the flame and the cooling water pipe was dramatically modified with τ_B as shown in Fig. 7, the heat flux H_p does not show a significant change by τ_B increases. The heat flux H_p increases only about 10 kW/m^2 when τ_B increases from 0 to 10. At $\tau_B = 20$, the heat flux H_p is reduced below the value obtained when $\tau_B = 0$ and $\tau_B = 10$. This may be attributed to an increase in the radiative heat loss from the flame to the upstream direction of the ceramic plate burner rather than to the cooling water pipe. A larger value of τ_B results in a higher flame temperature between the ceramic plate burner and the cooling water pipe as shown in Fig. 7. Therefore, the combustion flame would be confined in a narrower region close to the downstream side of the porous medium B. As a result, a large portion of the gas enthalpy is being converted to thermal radiation at the above region and is directed towards the upstream side of the ceramic plate burner and less thermal radiation is directed to the cooling water pipe.

Figure 11 shows the same trend as in Fig. 10. The heat flux H_p increases with the equivalence ratio Φ. This is due to an increase in the adiabatic flame temperature when the gas mixture becomes close to the stoichiometric air/fuel ratio.

Average Nusselt Number

Based on the average heat flux H_p, the average Nusselt number Nu_m at the outer surface of the cooling water pipe was calculated and plotted in Fig. 12 and Fig. 13. For this purpose, Dittus-boelter [12] Eq. (1) was used in estimating the heat transfer coefficient at the inner surface:

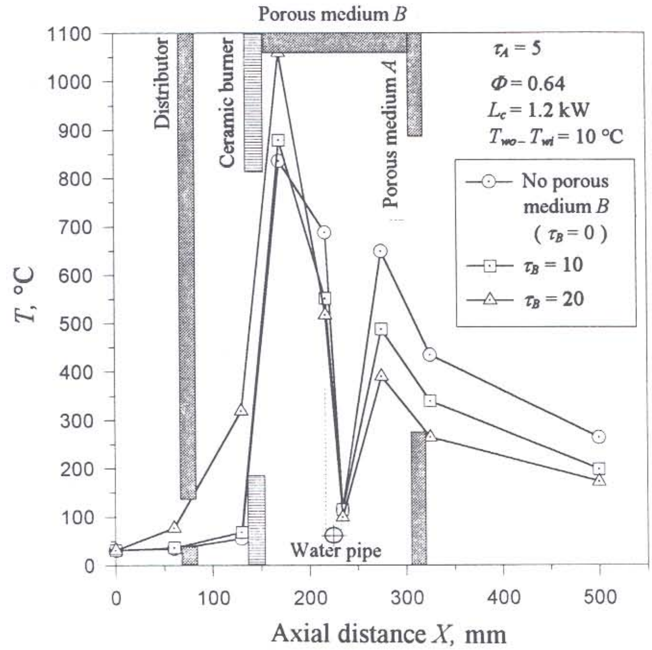

Fig. 8. Effect of optical thickness (τ_B) of the porous medium B on axial temperature distribution in furnace at $L_c = 1.2$ kW.

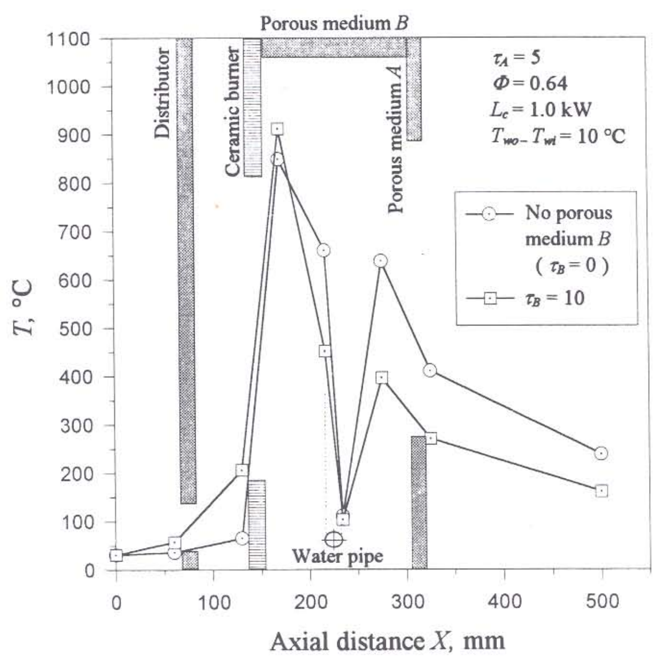

Fig. 9. Effect of optical thickness (τ_B) of the porous medium B on axial temperature distribution in furnace at $L_c = 1.0$ kW.

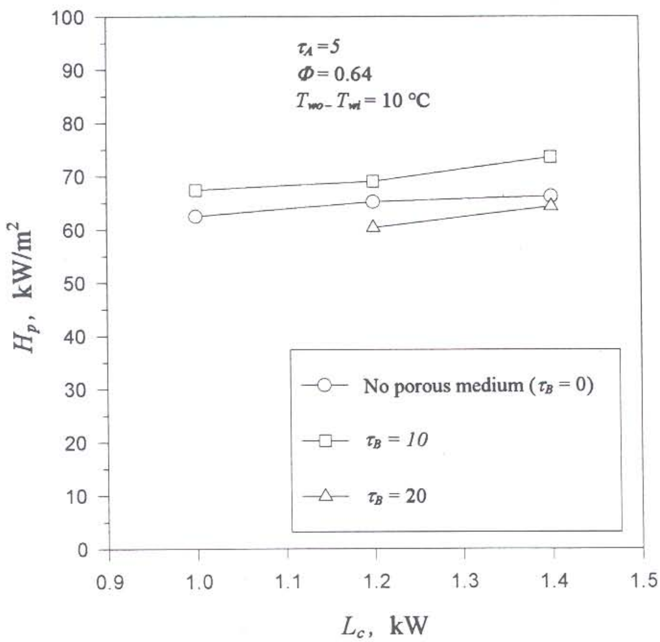

Fig. 10. Variation of the average heat flux at the wall of the cooling water pipe with L_c.

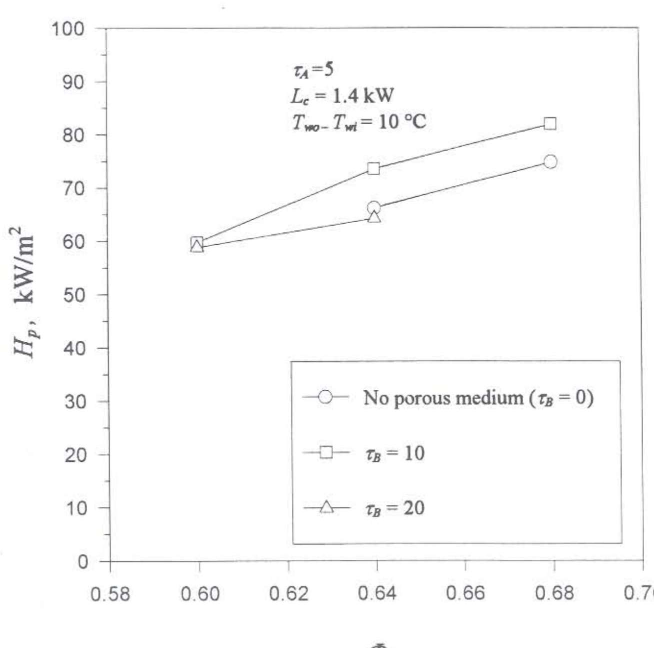

Fig. 11. Variation of the average heat flux at the wall of the cooling water pipe with Φ.

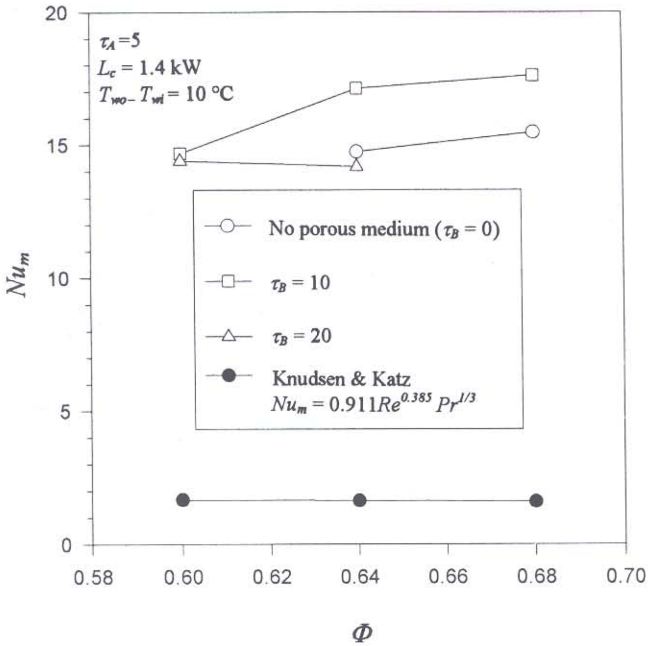

Fig. 12. Variation of the average Nusselt number at the wall of the cooling water pipe with Φ.

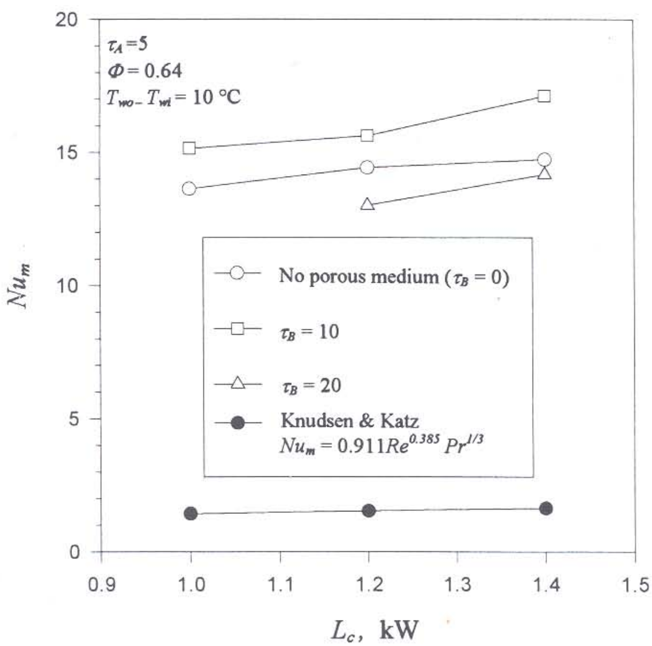

Fig. 13. Variation of the average Nusselt number at the wall of the cooling water pipe with L_c.

$$Nu = 0.023Re^{0.8} Pr^{0.4} \qquad (1)$$

of the cooling water pipe. This heat transfer coefficient was then used in computing the wall temperature of the cooling water pipe. The physical properties of the fluid are represented as an average value between the inlet and outlet temperatures of the cooling water. For the sake of simplicity, corresponding adiabatic flame temperatures of the mixture were used in determining the average Nu_m. In order to compare the heat transfer characteristics, the Nusselt number obtained from the experiment by Knudsen & Katz [13] for force convection only is also plotted in Fig. 12 and Fig. 13. It is clear that the heat transfer characteristics of the SCH with a porous medium system (in which the radiative heat transfer plays an important role in enhancing the heat transfer to the cooling water pipe) is far superior to heat transfer by the force convection alone. The Nusselt number, Nu_m for the system with a porous medium system is consistently about 15 times greater than the Nu_m obtained by force convection alone.

5. CONCLUSIONS

An experimental study of the heat transfer enhancement to a cooling water pipe by a combined surface combustor heater (SCH) technology and the convection-radiation converter (CRC) technology is presented. Important results are concluded as follows:

1. The SCH system with a porous medium system yields heat transfer potential at the water pipe surface which is about 15 times larger than that of the heat transfer obtained by force convection only.
2. The larger the optical thickness τ_B of the porous medium B, the closer the reaction zone moves towards the upstream direction which lowers the heat transfer to the cooling water pipe and causes a high heat loss to the upstream ambience environment. It is suggested that an optical thickness τ_B of the porous medium B less than 20 is sufficient for enhancing the rate of heat transfer to the cooling water pipe.
3. The convection-radiation converter (CRC) or the porous medium A plays an important role as thermal radiation shield. The CRC effectively converts the enthalpy of the high temperature gas coming out from the porous medium B to thermal radiation. Porous medium A, therefore, acts as a source of radiation energy in the upstream direction (towards porous medium B) which results in a large increase in the heat transfer to the cooling water pipe.

6. REFERENCES

1. Echigo, R., and Hasegawa, S. 1972. Radiative heat transfer by flowing multiphase medium-Part I. An analysis on heat transfer of laminar flow between parallel flat plates. *International Journal of Heat and Mass Transfer* 15: 2519-2534.
2. Echigo, R.; Hasegawa, S.; and Tamehiro. H. 1972. Radiative heat transfer by flowing multiphase medium-Part II. An analysis on heat transfer of laminar flow in an entrance region of circular tube. *International Journal of Heat and Mass Transfer* 15: 2595-2610.
3. Tamehiro, H.; Echigo, R.; and Hasegawa, S. 1973. Radiative heat transfer by flowing multiphase medium-Part II. An analysis on heat transfer of turbulent flow in a circular tube. *International Journal of Heat and Mass Transfer* 16: 1199-1213.

4. Hasegawa, S.; Echigo, R.; and Shimizu, A. 1986. *Handbook of Heat and Mass Transfer, Heat Transfer Operation.* S.l.: Gulf Publishing Company.
5. Ravish, M.B. 1974. *Gas and its Utilization*, Nauka, Moskva (in Russian).
6. Echigo, R. 1986b. Heat transfer augmentation in high temperature heat exchangers. *High Temperature Equipment.* Washington, DC: Hemisphere.
7. Echigo, R. 1987. Sophisticated application of radiation heat transfer. *Heat Transfer in High Technology and Power Engineering.* Washington, DC: Hemisphere.
8. Jasionowski, W.J.; Kunc, W.; Khinkis, M.H.; and Zawacki, T.S. 1987. *Combustion Systems: A Porous Matrix Burner and Surface Combustor, Gas Research Institute, Topical Report No. GRI-87/0186.* S.l.: Gas Research Institute.
9. Khinkis, M.H.; Kunc, W.; and Xiong, T.Y. 1989. Experimental evaluation of a high efficiency surface combustor-heater concept with low pollutant emissions, Paper No.20. In *Proceeding of the 1989 Fall International Symposium of the American Flame Research Committee*, 25-27 September, Short Hills, N.J., U.S.A.
10. Mohamad, A.A., and Visckanta, R. 1989. *Combined Convection-Radiation Heat Transfer in a Surface Combustion Process Heater, in Simulation of Thermal Energy System*, Vol. HTD-124 New York: ASME.
11. Echigo, R. 1982. Effective conversion method between gas enhalpy and thermal radiation and application to industrial furnaces. In *Proceedings of the Seventh International Heat Transfer Conference.* Munchen, Germany, Vol. 6, 361-366.
12. Dittus, F.W., and L.M.K. 1930. Boelter: Univ. Calif., Berkeley, Publ. Eng., 2: 443.
13. Knudsen, J.G., and D.L. Katz. 1958. *Fluid Dynamics and Heat Transfer.* New York: McGraw-Hill Book Company.