Heat Transfer Characteristics in a Cylindrical Duct Packed with Solid Spheres

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ABSTRACT: The present paper investigates the heat transfer characteristics in a cylinder packed with porous medium of solid spheres for various parameters such as mass flow rate, sphere diameter, length of the porous medium, and gas temperatures. Pressures and temperatures at the inlet and outlet regions were measured by using static pressure gages and R-type thermocouples. The modified relationship based on the Ergun equation is suggested for the estimation of pressure drops. In addition, the useful empirical correlation for thermal efficiency is obtained in the current study. Thermal efficiency is expressed in terms of non-dimensional time, sphere diameter, porosity, and pressure drops. It is also found that the pressure drop through the cylinder becomes larger as the gas temperature does higher at the inlet region, whereas it substantially decreases when the inlet flow rate decreases.

Nomenclature

A, B	: empirical constants
C_w	: empirical constant
D_p	: diameter of solid sphere [m]
D_t	: diameter of cylinder duct [m]
D^*	: dimensionless diameter [m]
E	: energy
k, l, m, n	: empirical constant
L_p	: length of a packed bed with sphere [m]
P^*	: dimensionless pressure
Q	: volume flow rate of hot gas at the inlet region [m^3/h]
T	: temperature [K]
t^*	: dimensionless time
u	: area-averaged velocity through the medium [m/s]

Greek symbols

ε	: porosity
η	: thermal efficiency
ω^*	: dimensionless parameter

Subscripts

in	: inlet region
out	: outlet region

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1. Introduction

In general, the porous material plays an important role in a variety of applications including the transport of pollutants in aquifers, the inter-seasonal storage of heat, the transport of fertilizers, and the design of packed-bed reactor. For enhancing thermal efficiency and using waste heat in more efficient way, some experimental and theoretical efforts[1-3] have been continuously made. Their results helped us in analysing the regenerative combustion systems using the porous media. Suzukawa et al.[4] reported that highly pre-heated air for combustion would reduce the fuel consumption substantially. Noh et al.[5] performed extensive experiments for thermal performance in the combustion system involving both heating and regenerating processes. Their work used the ceramic honeycomb as the orthotropic porous medium. The empirical correlation of thermal efficiency was suggested in terms of switching time, cell size, and length of honeycomb.

Most of combustion systems involving the regenerative process use the porous medium to utilize the waste heat from the hot gas efficiently. Since porous media such as solid spheres or honeycomb interact with turbulent flows strongly, the heat transfer characteristics between porous medium and hot gas exhausted from the burner are very complex. The fundamental understanding of heat transfer characteristics would be thus necessary for optimal design for combustion systems in real industries.

Varahasamy and Fand[6] studied the flow and heat transfer characteristics in a pipe packed with solid spheres and Atamanov et al.[7] proposed a new correlation between the heat transfer and the flow energy dissipation inside a packed bed of solid spheres. Furthermore, Kimura[8] investigated the forced and natural convection heat transfer characteristics for a vertical cylinder including a porous medium. Although the above researches give us useful information on the flow and heat transfer mechanisms between porous media and flows from the academic viewpoint, their relationships are difficult to be used in real industries easily because most of them are using the local thermo-fluid properties and parameters, which are difficult to be obtained. It would be true that a lot of system designers require such correlations in terms of certain bulk parameters that can be obtained in real industry fields as easily as possible.

To satisfy this requirement of system designers, this study is planned to figure out the heat transfer characteristics in the cylindrical duct packed with the ceramic solid spheres, and to offer more useful and acceptable empirical correlation for estimating thermal efficiency of system in terms of burner capacity, dimensions of porous medium, and so forth, which can be obtained easily in industries of interest. Besides, we make the modified Ergun's relationship for pressure difference from experimental data. To validate the modified Ergun equation, CFD simulations are performed using the well-known code, STAR-CD.

2. Experimental and theoretical method

2.1 Experimental apparatus and method

Figure 1 shows the schematic diagram of the experimental apparatus for a cylindrical duct of 60 mm packed with ceramic solid spheres made out of cordierite of 50% and mullite of 50%. To figure out size effects of solid particle on

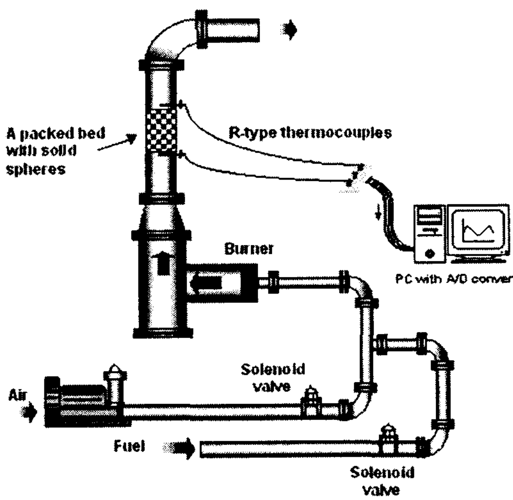


Fig. 1 Experimental apparatus.

Table 1 Experimental conditions

	D_p [mm]	L_p [mm]	Q [m^3/h]	T_{Gas} [K]	ε
Case 1	8	100, 200	20, 40	303, 873, 1073, 1273	0.38
Case 2	10	100, 200	20, 40	303, 873, 1073, 1273	0.39
Case 3	13	100, 200	20, 40	303, 873, 1073, 1273	0.40
Case 4	15	100, 200	20, 40	303, 873, 1073, 1273	0.41

heat transfer characteristics, as listed in Table 1, various solid spheres with different diameters are used at various inlet temperatures of 303 K, 873 K, 1073 K, and 1273 K. Temperatures and pressures are measured at different times by the R-type thermocouples and the static pressure gages, respectively. A ceramic insulator is also used for a cylindrical duct to minimize the heat loss.

The mass flow rates of air and fuel are adjusted before the hot gas enters into the inlet region of the duct to control the temperatures of exhausted hot gas from the burner. Chungnam town gas is used as a fuel, which consists of 88% methane and 10% buthane. It is difficult to measure the pressures and temperatures inside the porous medium directly. In Fig. 1, temperatures and static pressures are thus measured at the center position of the inlet and outlet cross-sections, not inside the porous medium region. The data signals for a measuring time are obtained by collecting five data of temperature per second using A/D converter (DT Vee).

2.2 Theoretical approach

A classical method for the pressure drops in the porous medium is to use the Ergun's relationship[9] based on the Darcy's law where the pressure drop is proportional to the velocity.

$$\frac{dp}{L_p} = -\frac{A\mu(1-\varepsilon)^2 u}{\varepsilon^3 D_p^2} - \frac{B\rho(1-\varepsilon)u^2}{\varepsilon^3 D_p} \quad (1)$$

where, L_p is the length of packed bed, u the mean velocity, D_p the diameter of solid sphere, and ε is the porosity. Both A and B values are empirical constants, typically taken as 150 and 1.75, respectively. The first term on the right-hand side represents a viscous loss term and the second an inertial loss, which should be considered for turbulent flows. However, the original Ergun's equation is difficult to predict the axial changes of the pressure effectively for porous media flows whose temperature is relatively high because it does not include any term explicitly which describes the effects of temperature gradient on the pressure drops. In order to involve the temperature gradient effect explicitly, a new function $f(D_p, \varepsilon, Q, T)$ is inserted into the original form of the Ergun's equation. This function takes into accounts the additional pressure drops in the duct varying with temperatures. It is assumed that the pressure drops by the heat transfer can be described through the convection effects. The modified Ergun equation is expressed by inserting a new function f into the convection part of the original Ergun equation, Eq. (1) as follows:

$$\frac{dp}{L_p} = -\left[\frac{A\mu(1-\varepsilon)^2}{\varepsilon^3 D_p^2} + f(D_p, \varepsilon, Q, T)\right]u - \frac{B\rho(1-\varepsilon)u^2}{\varepsilon^3 D_p} \quad (2)$$

Porosities of pipes packed with spheres are obtained by the following relationship proposed by Fand and Thinakaran:[10]

$$\varepsilon = \frac{0.151}{(D_t/D_p - 1)} + 0.36 \quad (3)$$

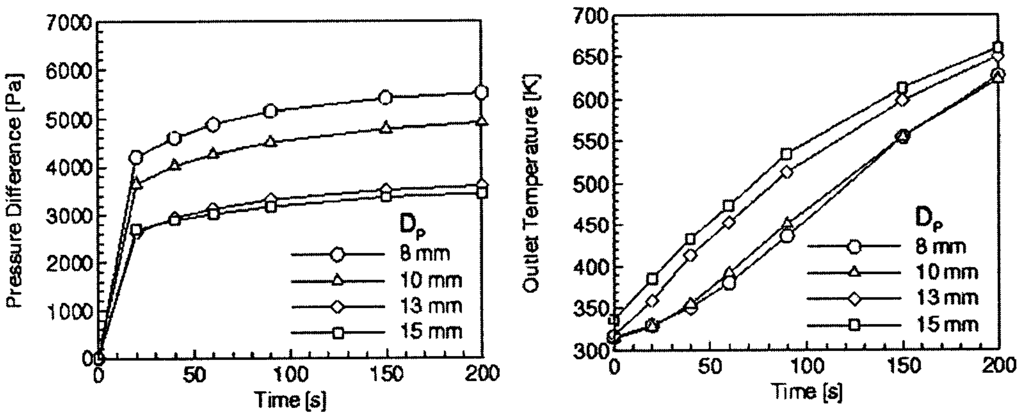
where D_p and D_t represents the diameters of solid spheres and cylinder, respectively. The above equation is applicable only for D_p/D_t larger than 2.033. In Eq. (2), the function $f(D_p, Q, T)$ is found from experimental data and a final form of this function will be introduced later.

3. Results and discussion

3.1 Pressure drop and heat transfer characteristics

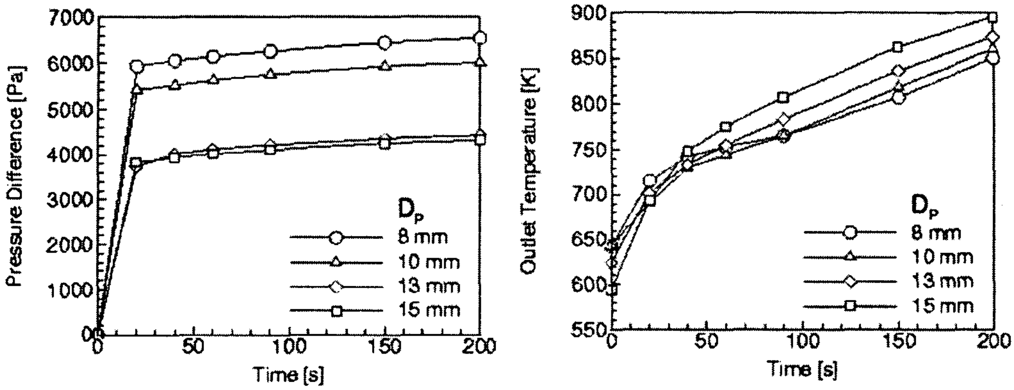
Figure 2 represents the influence of sphere sizes on pressure drops and outlet temperatures for T=873 K when Q=40 m^3/h and L_p=200 mm. The pressure drops increases between the inlet and outlet regions with decreasing the sizes of solid spheres. This increase of the pressure drop is because the friction losses occurring between flows and solid spheres are augmented for small porosities. In addition, it is observed that the pressure drops maintain constantly after t=50 s. As expected, the outlet temperatures increase gradually with the time. In particular, the outlet temperature for D_p=15 mm is about 100 K higher than that when D_p=8 mm. This means that the heat transfer from the hot gas to the solid spheres becomes more vigorous as the porosity does smaller. This is because the heat transfer area increases considerably as the sizes of sphere decrease. Meanwhile, no significant difference of outlet temperature is observed between the results of D_p =8 mm and 10 mm as seen in Fig. 2(b). Comparing Fig. 2(a) with Fig. 3(a), it is found that for higher gas temperature, the pressure drops increase because of the substantial changes in densities over time for higher gas temperature, relative to the case of T=873 K. We can see the similar trends with the case of 873 K, except for the fact that the measured temperatures at the outlet region are closely collapsed at the early stage of time as shown in Fig. 3(b).

Figure 4 shows the effects of the volume flow rates on pressure drops and heat transfer when L_p=100 and 200 mm at T=1073 K and D_p=10 mm. As the length of a packed bed is longer, the pressure drops becomes larger. In addition, when the length of the packed bed varies from 100 mm to 200 mm, the pressure drop increases by about 67% at Q=40 m^3/h, whereas it does by about 34% at Q=20 m^3/h, as seen in Fig. 4(a). This trend indicates that



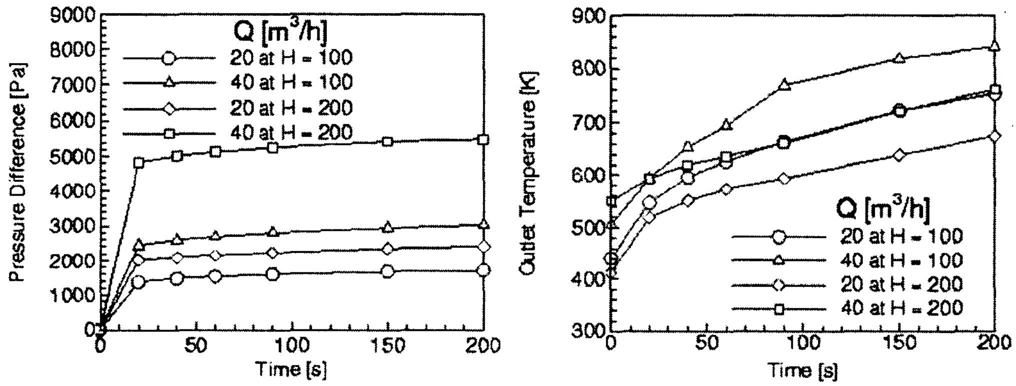
(a) (b)

Fig. 2 Pressure drops and outlet temperatures for various diameters of spheres when Q=40 m^3/h and L_p=200 mm at T=878 K.



(a) (b)

Fig. 3 Pressure drops and outlet temperatures for various diameters of spheres when Q=40 m^3/h and L_p=200 mm at T=1273 K.

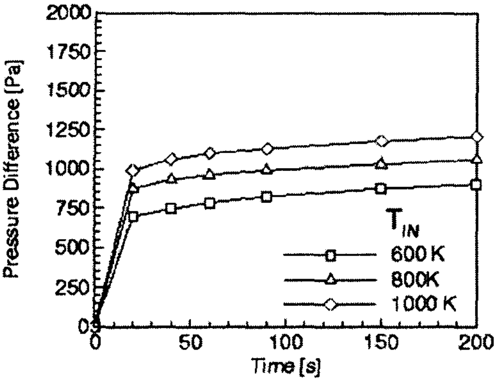


(a) (b)

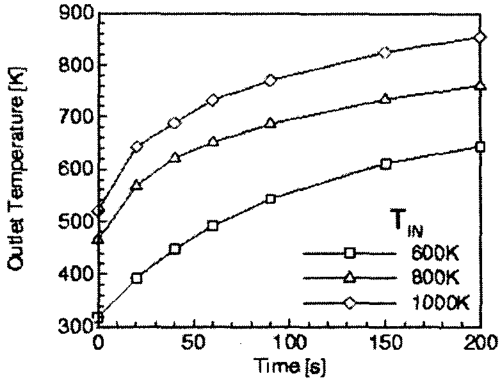
Fig. 4 Effects of volume flow rates on pressure drops and outlet temperatures when L_p=100 and 200 mm at T=1073 K with solid spheres of 10 mm diameter.

the changes in volume flow rates have a large effect on the pressure drops across the packed bed. The decrease of the outlet temperatures means the augmentation of the thermal energy accumulated in the solid spheres. The changes of temperature in the outlet region are thus related with the heat transfer characteristics. As seen in Fig. 4(b), the heat transfer rates decrease with increasing mass flow rates at the inlet region. In the case that Q=20 m^3/h and L_p=200 mm, for instance, the outlet temperature is lower than that of other cases. This shows that the thermal energies accumulated in the porous medium are controllable by changing some parameters such as the volume flow rates, length of a packed bed, and burner capacities. Actually, the estimation of the thermal energy accumulated in solid spheres is very important because it is closely associated with the regenerating process in a combustion system.

Figure 5 shows the influence of the burner heat capacities on the pressure drops and heat transfer for solid spheres of a 13 mm diameter. Both the outlet temperatures and pressure drops increase as the inlet temperature becomes higher. This shows that the changes of gas temperature may influence on the pressure drops significantly.



(a)



(b)

Fig. 5 Effects of hot gas temperatures on pressure drops and outlet temperatures when $Q=20\,m^3/h$ and $L_p=100\,mm$ with solid spheres of 13 mm diameter.

3.2 Empirical correlations for pressure drops and thermal efficiency

The present study provides some empirical correlations of pressure drops and thermal efficiency as a function of some important parameters, such as burner capacity, length of a packed bed, diameter of solid sphere, which are easily obtained. These relationships are useful in understanding the flows and heat transfer characteristics inside a cylinder with a packed bed of solid spheres and in designing the optimal conditions for higher efficiency of a combustion system. From the experimental data, an empirical relationship for pressure drops is obtained. As referred the above, a new function f used in the modified Ergun equation is divided into two parts as follows:

$$f(D_p, Q, \varepsilon, T) = f_1(D_p, Q, \varepsilon)g(T, D_p) \quad (4)$$

A main idea for finding the above function is in considering the difference between the experimental data and the predictions by the original Ergun equation. If the original Ergun equation were valid, the difference would be zero. Otherwise, such discrepancies between them would be appeared. The following equation expresses the discrepancy between measured data and predictions using Eq. (1) can be expressed as

$$\Delta = \left[\frac{dp}{L}\right]_{EXP.} - \left[\frac{dp}{L}\right]_{Ergun} = \left[\frac{dp}{L}\right]_{EXP.} + \frac{A\mu(1-\varepsilon)^2 u}{\varepsilon^2 D_p^2} + \frac{B\rho(1-\varepsilon)u^2}{\varepsilon^3 D_p} \quad (5)$$

where u is assumed to be the inlet velocity because the local velocity is difficult to be obtained. It is a known value which can be easily obtained from the volume flow rate. Using Eqs. (2) and (4), Eq. (5) can be expressed as follows:

$$\begin{aligned}\Delta &= f(D_p, \varepsilon, Q, T)u \\ &= f_1(D_p, \varepsilon, Q)g(T, D_p)u\end{aligned} \quad (6)$$

Dividing the above Eq. (6) by f_1 and rearranging it, the following expression is rewritten as

$$\frac{\Delta}{uf_1} = g(T, D_p) \quad (7)$$

where Eq. (7) means that the discrepancies divided by a function f_1 can be expressed in terms of a function $g(T, D_p)$ and a known value of μ. Assuming that $f_1(D_p, \varepsilon, Q) = KQ^l \varepsilon^m D_p^n$,

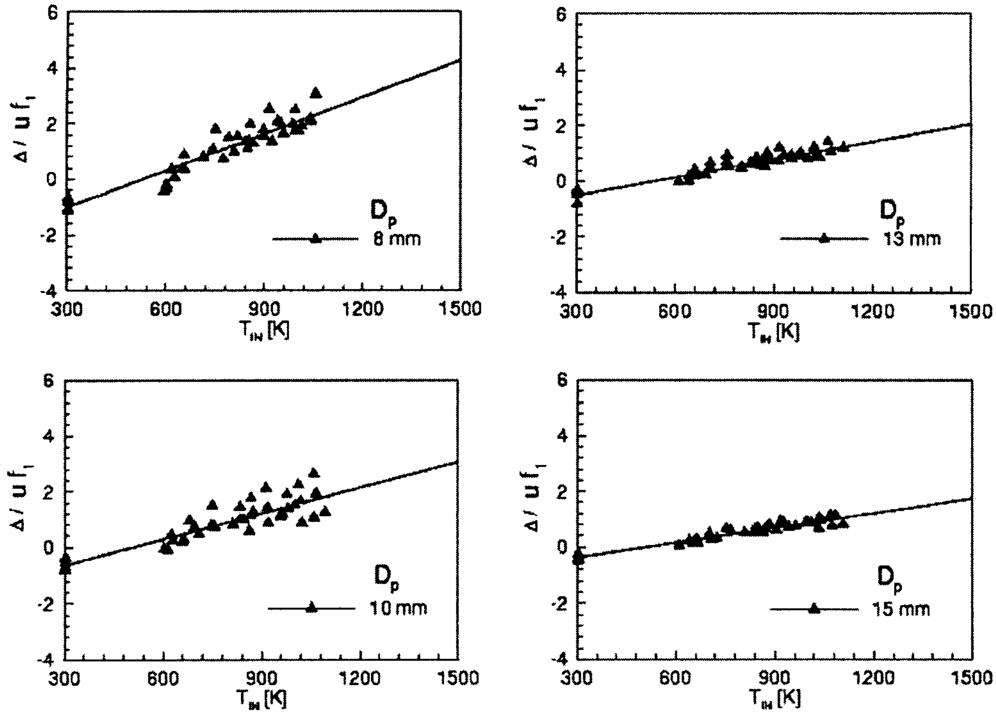


Fig. 6 Empirical correlations between the hot gas temperature and the deviations Δ/uf_1 for various diameters of solid spheres.

the function f_1 is found from experimental data as

$$f_1(D_p, \varepsilon, Q) = 580Q^{2/3}\varepsilon^{1/7}D_p^{2/45} \tag{8}$$

Figure 6 shows the empirical correlations between hot gas temperature at the inlet region and the deviations $g(T, D_p)$ for various diameters of solid spheres when Eq. (8) is used. Good fits linearly are observed, especially for larger diameters.

$$\begin{aligned} f(D_p, Q, \varepsilon, T) &= f_1(D_p, Q, \varepsilon)g(T, D_p) \\ &= 580Q^{2/3}\varepsilon^{1/7}D_p^{2/45}g(T, D_p) \end{aligned} \tag{9}$$

From Fig. 6, the deviations depending on inlet gas temperatures are fitted using a quadratic form of the temperature with coefficients as a function of the sphere diameter. A final form of the deviation $g(T, D_p) = a(D_p) + b(D_p)T + c(D_p)T^2$ where the coefficients a, b and c are expressed in terms of a diameter of solid sphere as follows:

$$\begin{aligned} a(D_p) &= -7.1 + 723.5D_p - 2.2\times10^4 D_p^2 \\ b(D_p) &= 1.5\times10^{-2} - 1.5D_p + 42.6D_p^2 \\ c(D_p) &= -2.0\times10^{-5} + 5.2\times10^{-3}D_p \\ &\quad -0.46D_p^2 + 13.2D_p^3 \end{aligned} \tag{10}$$

A final form of the modified Ergun's relationship is expressed as follows:

$$\begin{aligned} \frac{dp}{L} = &-\left[\frac{A\mu(1-\varepsilon)^2}{\varepsilon^3 D_p^3}\right. \\ &\left. + 580Q^{2/3}\varepsilon^{1/7}D_p^{2/45}g(T, D_p)\right]u \\ &- \frac{B\rho(1-\varepsilon)u^2}{\varepsilon^3 D_p} \end{aligned} \tag{11}$$

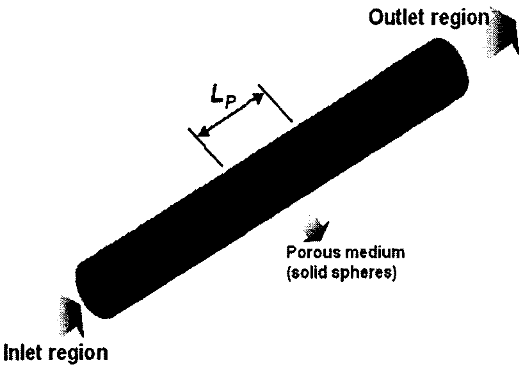


Fig. 7 Computational grids.

For an assessment of Eq. (11), the numerical simulations are carried out using a commercial code, STAR-CD, and the computational grids of 29,000 are generated by ICEMCFD. The modified Ergun equation, Eq. (11), is implemented into STAR-CD which basically includes the original Ergun equation method. As seen in Fig. 7, the calculation domain can be divided into three parts and the porous media is located at the middle of cylinder. The continuity, momentum and energy equations are solved by the finite volume method in a three-dimensional manner. The PISO algorithm and the standard $k-\varepsilon$ model are used to take into consideration the transient and turbulent flows, respectively.

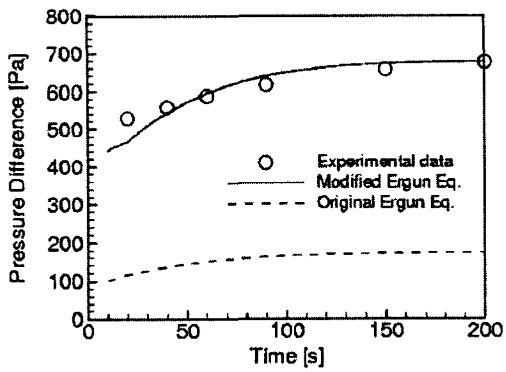


Fig. 8 Comparisons of predicted results with the experimental data when Q=20 m^3/h, D_p=15 mm, L_p=100 mm and T_{in}=873 K.

The predictions by the modified Ergun's equation are compared with those using the original Ergun equation and the experimental data as seen in Fig. 8. The original Ergun's equation underestimates the pressure drops significantly, whereas its modification using Eq. (11) shows good agreements with experimental data. It shows the importance of considering the changes in temperatures over time for better predictions of the pressure drops. It is thus thought that the present correlation may be useful in calculating the pressure drops for high temperature conditions.

The correlation of the thermal efficiency is suggested in the present study. Definition of the thermal efficiency is as follows:

$$\eta = \frac{E_{in} - E_{out}}{E_{in}} \tag{12}$$

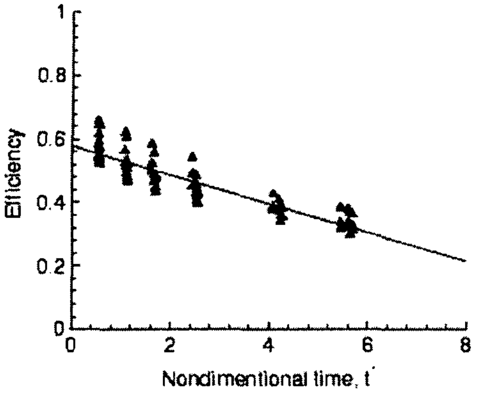
As mentioned previously, the main aim in making the correlation is to estimate the thermal efficiency approximately using the bulk parameters, not the local ones, such as volume fraction of cylinder occupied by solid spheres, mass flow rates at the inlet region, sphere diameters, and porosity. Three parameters in a dimensionless form are considered such as time, length, and momentum scales. Firstly, the characteristic time and length are determined using the inlet velocity, the length of cylinder packed with solid spheres, the diameters of the cylinder and solid spheres as

$$t^* = tU_{inlet}/L_p \tag{13}$$

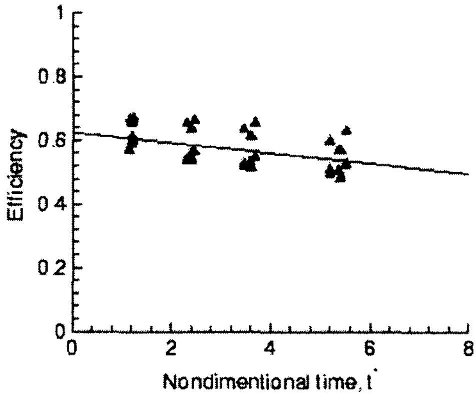
$$D^* = D_t/D_p \tag{14}$$

Dividing the measured pressure drop ΔP by the flow momentum at the inlet region, the dimensionless pressure P^* is determined as follows:

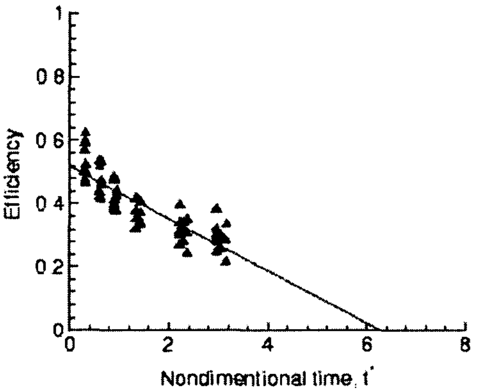
$$P^* = \frac{\Delta P}{(\rho U^2)_{inlet}} \tag{15}$$



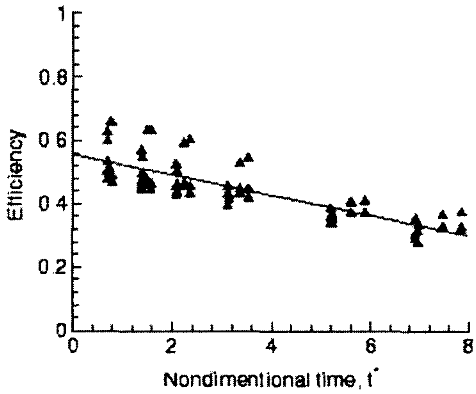
(a) $L_p=100$ mm and $Q=20\ \mathrm{m^3/h}$



(b) $L_p=200$ mm and $Q=20\ \mathrm{m^3/h}$



(c) $L_p=100$ mm and $Q=40\ \mathrm{m^3/h}$



(d) $L_p=200$ mm and $Q=40\ \mathrm{m^3/h}$

Fig. 9 Empirical correlations of thermal efficiencies for the nondimensional times at 873, 1073, and 1273 K.

Figure 9 shows the empirical correlations of thermal efficiencies expressed by the nondimensional time t^*. The dimensionless parameter t^* means that a time duration for which the exhausted hot gas passes through the packed bed with solid spheres. As t^* increases, the thermal efficiencies decrease. Empirical correlations between t^* and η may be acceptable for each case. However, the dimensionless time t^* seems to be inappropriate for representing such a relationship because it reflects only the length effects of a packed bed on the thermal efficiencies. Another way to express the correlations effectively is thus required. Present study assumes that the thermal efficiency is deeply associated with the porosity together with the above three parameters. Thus, we propose a new variable ω^* in a form of $C_\omega t^{*k} D^{*l} P^{*m} \varepsilon^n$ in terms of the porosity, the characteristic time, and the length scales in order to couple with the thermal efficiency and determine the powers as well as the constant C_ω by matching them to the experimental data. The final form is found as

$$\omega^* = \frac{6.207 t^* D^{*13/45} P^*}{\varepsilon^{1/7}} \tag{16}$$

The correlation of the thermal efficiency is found by fitting the thermal efficiencies calcu-

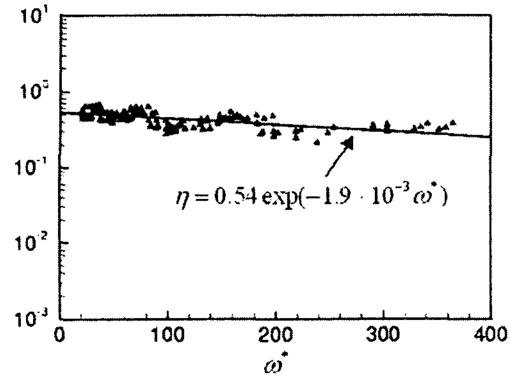


Fig. 10 Empirical correlation of thermal efficiency.

lated from the experimental data for the defined ω^* associated with the pressure drops as seen in Eq. (16). Figure 10 represents the experimental data of thermal efficiencies for a dimensionless parameter ω^* and shows the fitted curve delineated from the data. It is found that the thermal efficiencies increase with decreasing ω^* and its maximum is about 70%. The relative errors are much higher with decreasing thermal efficiencies. From these data, a final form of the thermal efficiency can be suggested as follows:

$$\eta = 0.511\exp(-8.557\times10^{-3}\omega^*) \tag{17}$$

This relationship may be used for prediction of thermal efficiency on the porous media flows in which the heat transfer exists between the solid spheres and the gas flows whose temperature is relatively high. The relative errors between the predictions using Eq. (17) and the measured data are calculated for all cases. Its maximum value becomes about 25% near the high efficiency region, whereas it does about 68% near the low efficiency region. In general, the low efficiency regions are not reflected in designing the combustion systems because the ultimate aim of the combustion system design lies on the fulfillment of the high efficiency. Thus, it is encouraging that the relative errors of the high efficiency region are small from this point of view. It can be also expected that this relationship be helpful for engineering use, regardless of some errors.

4. Conclusions

Heat transfer and flow characteristics are investigated in the cylinder duct including the packed bed with ceramic solid spheres. The conclusions are drawn as follows.

(1) As the porosity becomes smaller, the pressure drop increases between the inlet and outlet regions, and the heat transfer from hot gas to solid spheres becomes more vigorous. At higher temperature, substantial increase in pressure drop can be observed because of significant change in gas density over time.

(2) As the length of packed bed is longer, the pressure drop is larger. Increase of the burner capacities also results in the augmentation of the heat transfer rates and the pressure drops.

(3) A useful correlation is suggested for thermal efficiency with the accuracy less than 25 % in the high efficiency region.

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References

1. Willmott, A. J., 1993, The development of thermal regenerator theory from 1931 to the present, Journal of the Institute of Energy, Vol. 66, pp. 54-70.
2. Akter, S. and Hossain, I., 1997, Waste heat utilization in a ceramic industry, Interna-

tional Journal of Energy Research, Vol. 21, pp. 1215-1221.

3. Kluka, J. A. and Wilson, D. G., 1998, Low-leakage modular regenerators for gas-turbine engines, Journal of Engineering for Gas Turbines and Power, Vol. 120, pp. 358-362.
4. Suzukawa, Y., Sugiyama, S., Hino, Y., Ishioka, M. and Mori, I., 1997, Heat transfer improvement and NOx reduction by highly preheated air combustion, Energy Convers. Mgmt., Vol. 38, pp. 1061-1071.
5. Noh, D. S., Hong, S. K., Ryou, H. S. and Lee, S. H., 2001, Experiment and numerical simulation for heat transfer in a regenerator system with ceramic honeycomb, The Korean Society of Mechanical Engineers (KSME) International Journal, Vol. 15, No. 3, pp. 357-365.
6. Varahasamy, M. and Fand, R. M, 1996, Heat transfer by forced convection in pipes packed with porous media whose matrices are composed of sphere, International Journal of Heat Mass Transfer, Vol. 39, pp. 3931-3947.
7. Atamanov, V. V., Kharitonov, V. V. and Yakutin, N. V., 1996, Correlation between heat transfer and dissipation of flow energy in packed spheres, High Temperature, Vol. 34, No. 4, pp. 585-591.
8. Kimura, S. 1989, Transient forced and natural convection heat transfer about a vertical cylinder in a porous medium, International Journal of Heat Mass Transfer, Vol. 32, No. 3, pp. 617-620.
9. Kaviany, M., 1995, Principles of heat transfer in porous media, 2nd ed., Springer-Verlag New York, Inc.
10. Fand, R. M. and Thinakaran, R., 1990, The influence of the wall on flow through pipes packed with spheres, ASME, Journal of Fluids Engineering, Vol. 112, pp. 84-88.