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Effect of particle ingestion on the fouling reduction and heat transfer enhancement of a No-Distributor-Fluidized heat exchanger

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Abstract

To overcome the fouling problem that is common in heat exchangers for waste heat recovery, a new type of fluidized heat exchanger was devised and tested. Fluidized bed heat exchangers are considered to be a good candidate for waste heat recovery flue gases due to their demonstrated ability to avoid fouling or to clean out deposition on heat transfer surfaces, but have a major drawback with significant pressure losses. These pressure drops typically associated with the distributor plate, which is a key component in constructing any conventional fluidized bed system, limit the applicability of fluidized bed heat exchangers for use as an energy saving device. In a new design, however, dilute gas-solid particulate is maintained without having a distributor plate. The main feature of this no-distributor-fluidized (NDF) heat exchanger is the self-cleaning action by ingested circulating particles at minimal additional pressure loss. In the present study, a multi riser NDF heat exchanger of 7,000 kcal/hr capacity was built to evaluate its heat transfer performance and fouling reduction characteristics. To experimentally simulate the fouled condition, fuel rich combustion gas with soot was introduced to the heat exchanger, then a cleaning test was performed by introducing glass bead particles (600μm) inside the gas passage of the heat exchanger unit. Through the present experimental study, the performance degradation due to fouling was successfully demonstrated and the cleaning role of particle circulation was identified. It was also demonstrated that small amounts of circulating particles contribute not only to the fouling reduction on the gas side, but also to the heat transfer enhancement. Experimental operation data for 50 hours including accelerated fouling are obtained to simulate the long-term behavior of the system.

Keywords: No distributor fluidized bed (NDFB); Heat exchanger; Fouling; Pressure drop; Particulate flow

1. Introduction

Waste heat may be defined as the energy associated with waste streams of air, exhaust gas, and/or fluids that leave the boundaries of a plant or building and enter the environment [1]. Energy recovery from the waste heat not only reduces the consumption and expenses related with the purchased energy, but also is considered as one of the most direct measures for rational use of energy resources and reduction of the energy usage, which eventually contributes to the CO_2 reduction.

Sources of waste heat can be grouped into three temperature ranges: high (600℃≤T≤1650℃), medium (200℃≤T≤600℃), and low (27℃≤T≤200℃) temperatures. High temperature sources are typically from furnaces and incinerators, while medium temperatures are from boilers, heat-treating furnaces, ovens and reciprocating engines, and low temperatures are from process steam condensates and cooling waters from various processes [1]. In Korea, about 80% of the annual estimated total waste heat, which amounts to about 9,000,000 TOE, is from the waste gas, of which only 50% is being recovered [2].

Exhaust gas, such as from incinerators, contains

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significant amounts of corrosive fly ash, which contributes to the contamination (fouling) of the heat transfer surface. Fouling causes degradation of the efficiency of the heat exchanger and also corrosion related problems. In practice, periodic cleanings are conducted in some cases, while heat recovery is abandoned in smaller systems. The needs and measures to effectively recover heat energy from the waste gas without fouling problems, therefore, have been raised and sought by many investigators.

Fluidized bed heat exchangers are well known to give enhanced heat transfer performance. However, its application to low grade waste gas heat recovery has been challenged by the cost of pressure drop, operational instability and complications with particle handling near dew temperature. For these reasons, different types of heat recovery devices are being studied depending on their specific application needs [3-10].

Rodriguez et al. [3] performed an experimental study on the affecting parameters such as particle size, mass flow rate, and the gas flow rate on the heat recovery performance in the case of a shallow fluidized bed, while Wang et al. [4] reported the development of a new type of external heat exchanger for a circulating fluidized bed boiler and its successful application to a 12MW boiler. Park et al. [5] measured pressure drop and heat transfer performance in the case of a single vertical riser and compared the data with a one-dimensional pressure loss model. Park et al. [6] performed numerical study on the gas-solid direct contact heat exchanger that is to be used for relatively high temperature waste gases, in which they considered the effect of radiation due to hot gases. Jung et al. [7] reported an increase in the heat transfer coefficient of up to 93% due to the particles in a horizontal multi-tube type circulating fluidized bed (CFB) heat exchanger. Lee et al. [8] investigated the effects of design parameters of vertical CFB, such as the diameter ratio of the riser pipe-to-baffle plate hole and height of mixing chamber on the heat transfer characteristics. Recently, Jun et al. [9, 10], devised and reported the heat transfer performance of a new type of CFB which does not utilize baffle plates for particle fluidization to minimize the pressure drop through the heat exchanger.

In the present study, the fouling reduction characteristics of the no-distributor-fluidized-bed (NDFB) heat exchanger are studied and demonstrated through experiment along with the performance data at clean and fouled conditions, respectively. In this study, operational data for total of 50 hours are obtained to simulate the long time behavior of the system. Performance data are presented in terms of the overall heat transfer coefficient with the corresponding terminal temperature difference (TTD) values.

2. Experimental apparatus

Fig. 1 shows the schematic diagram of a multi-riser type, no-distributor-fluidized (NDF) heat exchanger test apparatus. The apparatus is composed of a combustor that supplies high temperature exhaust gas, water preheater, and heat recovery heat exchanger. Fuel flow rate of the combustor burner is 12liter/hr (9.6kg/hr with S.G. of light oil being 0.8 at 15℃), which delivers about 99,000kcal/hr. The shell-and-tube type water preheater provides the warm water to the CFB heat exchanger and at the same time reduces the gas temperature down to 300-350℃, which may represent the temperature range of medium waste heat sources. The feed water for the system is supplied by a water pump from a 1 ton water tank and is controlled and measured to yield the specified temperature of 60℃ from the water preheater (the first heat exchanger). This feed water is then further heated in the second heat exchanger (NDFB heat exchanger) before being sent for possible usage. The tested heat exchanger is composed of 4 vertical risers (STBH 50A: I.D. =54mm), with 16 wound downcomer tubes (copper tubes with I.D. =10mm), with which actual heat transfer occurs from the particulate gas to the cooling water. The gas flow rate is measured by the orifice installed at the downstream of the heat exchanger and is controlled by a damper (See Fig. 2). Particles are carefully introduced through the particle inlet port installed in the upper part of the heat exchanger and fall down through the particle downcom-

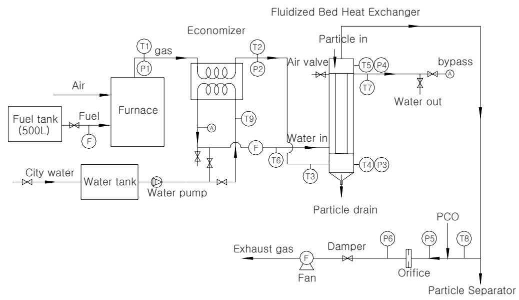


Fig. 1. Test apparatus for 7000 kcal/hr No-distributor-fluidized-bed heat exchanger.

ers to be introduced to the up-rising gas stream through a slit along the circumference of the riser. This gas-particulate two phase flow is considered to be mainly responsible for the heat transfer. After going through the riser, particles are inertially separated from the gas stream and introduced to the downcomers, which completes the circulation cycle of particles.

3. Test conditions and procedures

3.1 Operating conditions and measurements

The gas flow rate was determined by considering both stable operation of the burner and good particle circulation. Gas flow rate of 0.066-0.071kg/s was maintained, while the feed water flow rate was varied in the range of 0.23kg/s and 0.28kg/s, to consider the effect of temperature differences on the heat transfer performance. The gas flow rate from the orifice against the gas analyzer showed a maximum deviation of less than 3% [11]. The gas flow rate was measured by using an orifice at the downstream of the heat exchanger unit with temperature correction. The water flow rate is measured by using a digital flow meter (OVAL) and controlled by a valve right upstream of the heat exchanger inlet. For the evaluation of the heat exchanger performance with and without particles ingestion, analyses were conducted under a steady state condition. To secure the steady state data, the system was operated without any change for a minimum of 30 minutes before taking data for analysis. The temperature measurement for the feed water and the exhaust gas was done with K-type thermocouples and was stored on a PC via a recorder (Yokogawa DA-100). Temperatures were measured at 9 locations for gas and water and the measurement locations are shown in Fig. 1 and Table 1, respectively.

Table 1. Temperature measurement locations.

Designation	Measurement Locations
T1	Combustor Outlet
T2	Economizer Outlet
T3	H.E. Inlet
T4	Riser Inlet
T5	H.E. Outlet
T6	Water Jacket Inlet
T7	Water Jacket Outlet
T8	Orifice Upstream
T9	Incoming Water

Differential pressures were measured for the orifice and for the heat exchanger by using two micro manometers (FCO-12; 200 and 2,000 mmAq), respectively, while absolute pressure level was also monitored at the gas line with a U-tube manometer (Dwyer Mark II) for reference.

3.2 Particles ingestion and fluidization

Particles for circulation are preheated and weighed ahead of feeding and are fed through the ball valve on the upper part of the heat exchanger. These particles first fall in the upper region of the heat exchanger unit and then fall through the 16 down comers. Being collected in a hopper under the down comer tubes, particles are introduced to the hot gas stream at the lower part of the riser though an annular slit shaped passage. Once the particles are entrained in the main gas stream, they move upward along the riser in a suspended condition and finally inertially separate in the upper region of the unit from the flue gas. In this way, particles circulate inside the unit, while the flue gas passes through the riser and is exhausted. Glass bead particles of size range of 500-600μm are ingested through a port to the upper region of the heat transfer section, by 100grams per dose, up to 2.5kg. The amount of particle mass circulated inside is self-determined by the flow conditions through the riser.

3.3 Fouling and cleaning test

To simulate the fouling condition in the gas side experimentally, fuel rich combustion was used, through which fouled condition of the heat ex changer could successfully be realized. The exhaust gas with

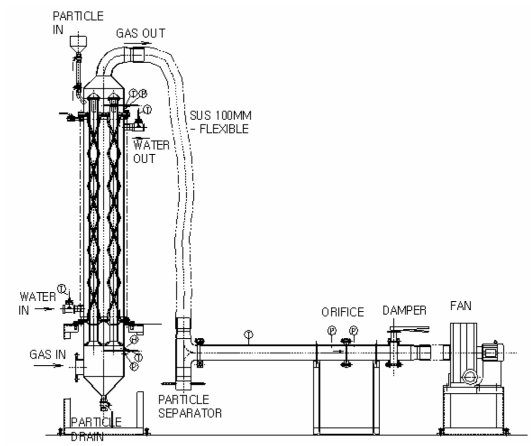
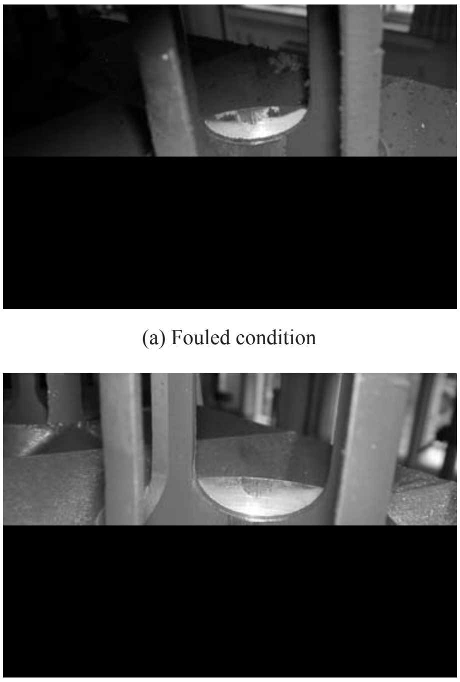


Fig. 2. Schematic of the tested heat exchanger unit.



(a) Fouled condition

(b) Cleaned condition

Fig. 3. Upper region of riser.

unburned soot develops a soft deposition layer on the inside of the riser tubes. The extent of fouling was indirectly monitored through the pressure difference between the riser inlet and outlet and in the present case, fuel rich combustion was maintained about 30 minutes. In order to monitor the effect of fouling from soot and the cleaning from ingested particles, temperatures and pressures were monitored in a clean (before fouling), fouled, and cleaned conditions, respectively. Fig. 3 shows the views of upper region of the riser section (a) in a fouled condition and (b) in a cleaned condition by particles.

4. Heat exchanger performance analysis

Total heat transferred can be expressed as

$$q = \rho_g Q_g c_{p_g} \Delta T_g \text{ (Gas side)} \tag{1}$$

$$q = \rho_w Q_w c_{p_w} \Delta T_w \text{ (Water side)} \tag{2}$$

The flow rate of the flue gas which is measured by using a D and D/2 tap orifice downstream of the heat exchanger unit can be obtained by

$$Q = C_d A_t \left[\frac{2(P_1 - P_2)/\rho}{1-\beta^4}\right]^{1/2} \tag{3}$$

with the measured pressure difference between the upstream and downstream of the orifice. In the present case, a pipe of 103mm inner diameter (D) and an orifice plate with diameter ratio (β=d/D) of 0.50 were used. The density at high temperature is obtained from the ideal gas law. Here the dimensionless discharge coefficient C_d for D and D/2 taps orifice is known to be

$$C_d = f(\beta) + 91.71\beta^{2.5}\mathrm{Re}_D^{-0.75} + \frac{0.09\beta^4}{1-\beta^4}F_1 - 0.0337\beta^3 F_2 \tag{4}$$

with

$$f(\beta) = 0.5959 + 0.312\beta^{2.1} - 0.184\beta^8 \tag{5}$$

where the values of empirical constants F_1 and F_2 are 0.4333 and 0.47, respectively. The overall heat transfer coefficient U of the heat exchanger is related as

$$q = UA\Delta T_m \tag{6}$$

where ΔT_m is the log mean temperature difference defined as

$$\Delta T_m = \frac{(T_{g,e} - T_{w,e}) - (T_{g,i} - T_{w,i})}{\ln\left[(T_{g,e} - T_{w,e})/(T_{g,i} - T_{w,i})\right]}. \tag{7}$$

For the present case, $T_{g,e} = T_5$, $T_{g,i} = T_1$, $T_{w,e} = T_7$, $T_{w,i} = T_6$.

Thermal resistance due to the fouling of heat transfer surface can be represented by the fouling factor R_f as

$$R_f = \frac{1}{U_f} - \frac{1}{U_c} \tag{8}$$

where U_f and U_c represent the fouled and clean states, respectively [12].

The indication of particles circulation can be quantified by introducing the so-called suspension density ρ_m that is defined as

$$\rho_m = \frac{\Delta P_{riser}}{gL} \tag{9}$$

where L is the length of the riser and g is the gravitational acceleration. In the present study, the length of the riser is 1 meter.

5. Results and discussion

Heat exchanger performance tests have been per-

Table 2. Heat exchanger performance before fouling.

Designation	Case I	Case II	ratio =II/I
hot fluid	gas only	particulate	
cold fluid	water		
m_w (kg/s)	0.232	0.232	1.0
m_g (kg/s)	0.068	0.068	1.0
m_p (g)	-	83	
T_3 (℃)	261	267	
T_6 (℃)	87.4	88.7	
ΔT_{3-5} (℃)	90.9	96.2	1.06
ΔT_{7-6} (℃)	3.7	4.5	1.22
ΔT_m (℃)	120.2	121.4	1.01
q_g (kcal/hr)	6,318	6,697	1.06
U(W/m^{2}K)	26.6	27.9	1.05
TTD(℃)	79	77.9	
Test Serial No.	57	59	

Table 3. Heat exchanger performance after fouling.

Designation	Case III	Case IV	Case V
hot fluid	gas only (fouled)	particulate (cleaning)	gas only (cleaned)
cold fluid	water		
m_w (kg/s)	0.282	0.283	0.245
m_g (kg/s)	0.066	0.067	0.067
m_p (g)	-	49	-
T_3 (℃)	284.6	288.8	292.
T_6 (℃)	73.7	75.3	80.9
ΔT_{3-5} (℃)	91.9	111.2	96.9
ΔT_{7-6} (℃)	4.53	6.04	5.48
ΔT_m (℃)	173.7	159.2	154.4
q_g (kcal/hr)	6,280	7,636	6,687
U(W/m^{2}K)	18.3	24.2	21.9
TTD(℃)	129.8	107.7	108.9
Test Serial No.	94	96	101

formed at clean, fouled, and cleaned states, respectively in sequence, while maintaining a virtually constant gas flow rate. In each state, the performance with and without particles ingestion are presented and compared.

5.1 Performance before fouling (clean state)

Table 2 shows tabulated data obtained from the performance tests before fouling, in which a gas flow rate of 0.068kg/s and water flow rate of 0.232kg/s were maintained. Terminal temperature difference (TTD) values are included in the table to reflect the change in heat transfer potential at different test conditions. According to the test results, about 5% increase in the overall heat transfer coefficient was observed in clean state by ingesting particles representing heat transfer enhancement potential due to particles. The amount of circulating particles could successfully be collected and measured after the operation, and was found to be 83grams in this case, and the corresponding increase in the suspension density was found to be 3.1kg/m^3.

5.2 Performance after fouling (fouled and cleaned states)

Table 3 shows the sequence along with the results from each of the tests. After the performance is measured for the test without particles (gas only) under fouled condition (Case III), particles are ingested for the cleaning purpose (Case IV) and finally performance test is conducted again for gas only case after cleaning (Case V).

One of the most appreciable changes in the operating condition after the fouling test was the contamination of the preheater, which caused higher inlet gas temperature and lower water inlet temperature. For these tests, the water flow rate was set to 0.282kg/s for case III and IV, and 0.245kg/s for case V.

Under the fouled condition (case III), a significant decrease (31%) in the overall heat transfer coefficient was observed as compared to the clean state (case I), while the corresponding TTD increased from 79℃ to 129.8℃. Also, there was a slight increase in the riser pressure drop (4Pa), which reflects the fouling effect due to the soot deposition rather than the actual suspension of particles. These changes in the overall heat transfer coefficient and the TTD are believed to be caused not only by the performance degradation of the heat exchanger itself, but also by the change in the operating condition of the entire system due to fouling. It should be mentioned that the effect of different operating conditions due to fouling should be considered.

When particles were ingested to the system for cleaning (case IV), the overall heat transfer coefficient increased by 32%, from 18.3W/m^2 K to 24.2 W/m^2 K. Comparing the gas-only cases at the initial clean state (case I) and at the cleaned state after fouling (case V), the overall heat transfer coefficient was found to decrease by 18% (from 26.6W/m$^2\cdot$ K to 21.9 W/m$^2\cdot$ K), even though the total heat transferred and the gas temperature drop through the riser (ΔT_{3-5}) increased by 5.8% and 6.6%, respectively. The increased value of TTD in case V (Δ in Fig. 4)

Table 4. Pressure drop through riser.

Description	case I	case II	case III	case IV	case V
m_p (g)	-	83	-	49	-
ΔP_{riser} (Pa)	46	76	50	72	46
Vg(m/s)	10.0	10.1	10.6	10.5	10.6
$K*10^3$(-)	0.921	1.50	0.89	1.30	0.82
Tg(℃)	204	208	246	239	239
Remarks	Gas only	Particulate	Gas only	Particu-late	Gas only
	clean		fouled	cleaning	cleaned

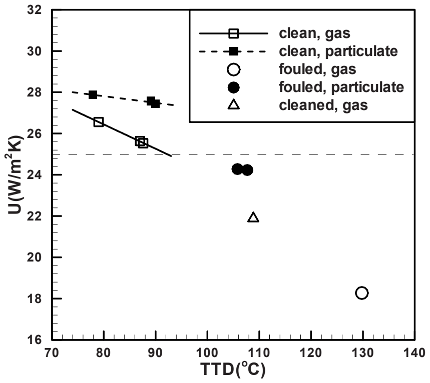


Fig. 4. Overall heat transfer coefficient vs. terminal temperature difference.

is the result of higher inlet gas temperature introduced by fouling of gas passage prior to the heat exchanger inlet. This represents the system fouling effect that shifted the operating condition of the heat exchanger unit, rather than the fouling of the heat exchanger.

The level of fouling reduction in the NDF heat exchanger can be shown by comparing the pressure drop through the riser and by visualization. Virtually no difference was identified in the pressure drop through the riser between the initial clean state (case I) and in the cleaned state (case V) in Table 4. Fig. 3 compares the views of the upper region of the riser (a) in the fouled condition and (b) in the cleaned condition, respectively.

5.3 Pressure drop

The pressure loss of the heat exchanger is represented by the pressure difference between the upper and lower region of the riser pipes, which is designated as ΔP_{riser}. Table 4 shows the measured pressure loss ΔP_{riser}, mean gas velocity through the riser pipes V_g, non-dimensional loss coefficient K, and the average gas temperature in the riser for each tested case. According to the test results, the pressure loss through the heat exchanger ranged from 46 Pa to 50 Pa in the gas-only cases (I and V), while the corresponding pressure drop varied from 72 to 76 Pa in the particulate cases. It could be noticed from the table that the additional pressure losses by introducing particles are comparable to those with gas-only cases, which suggests that additional pressure boosting may not be necessary for circulating particles in the system.

5.4 Interpretation of the fouled performance

Due to its dependence on several design and operating parameters, the use of the overall heat transfer coefficient is guided by HEI (Heat Exchanger Institute, USA [13]) such that the overall heat transfer coefficients should be reported with its temperature potentials for any comparison purpose. In the present study, the terminal temperature difference (TTD) is selected to represent the temperature potential for heat transfer of our heat exchanger. Fig. 4 shows the tested heat exchanger performance in terms of overall heat transfer coefficient with the corresponding TTD values. In this way any change in the operating conditions can be successfully represented by the variations in TTD values. In the figure each symbol represents conditions before fouling (□ and ■), fouled (○), cleaning (●) and cleaned (△) states. According to Fig. 4, it could be mentioned regarding the performance characteristics of the present heat exchanger system that:

(i) The overall heat transfer coefficient decreases as the TTD increases. The TTD increases with the increased feed water flow rate (□ and ■) and due to fouling (○).

(ii) When particles are ingested in clean state, heat transfer performance improves, and the change in the overall heat transfer coefficient with respect to the TTD variation is less sensitive than gas-only cases (□ and ■).

(iii) When the heat exchanger is fouled, significant decrease in the overall heat transfer coefficient is observed which is associated with the increase in TTD (○).

(iv) When the system is cleaned (△ and ●), TTD becomes smaller and the overall heat transfer coefficient is recovered.

5.5 Notes on the fouling factor

The fouling factor defined in Eq. (6) represents the performance deterioration due to the contamination of the heat transfer surface. However, as can be seen in Fig. 4, the value of the overall heat transfer coefficient depends not only on the fouling, but also on other operating conditions such as the water flow rate as discussed earlier. For the desirable evaluation of the fouling factor, it should be emphasized that relevant information needs to be specified. In the present study, simple numeric evaluation is abandoned to avoid further confusion regarding the fouling factor.

6. Conclusion

Through the present study, fouling and cleaning tests were conducted for a newly devised no-distributor-fluidized (NDF) heat exchanger with multiple risers. Major findings from the present study are as follows:

(1) Fouled condition of the heat exchanger was successfully simulated experimentally by using fuel-rich combustion. Performance data under a fouled condition were obtained. From the test, the overall heat transfer coefficient decreased by about 30% after 30 minutes of exposure to the fuel-rich burned exhaust gas out of Kerosene.

(2) Cleaning action and the performance improvement by particle ingestion to the system were clearly demonstrated. With less than 100 g of 500-600μm glass bead particles ingested, overall heat transfer coefficient increased by 5% in a clean condition, while 30% increase were observed under a fouled condition. However, the net amount of circulating particles seems sensitive to flow conditions.

(3) Additional pressure loss due to the particle ingestion was comparable to that with gas only flow case, which suggests that this type of system may not need additional pressure boosting devices and power.

(4) For the analysis of the long-term behavior of the system, the heat transfer performance is described in terms of the terminal temperature difference (TTD).

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Nomenclature

A : Heat transfer area (m^2)
C_d : Discharge coefficient
C_p : Specific heat at constant pressure(J/Kg K)
d : Orifice inner diameter (m)
D : Orifice outer diameter (m)
g : Gravitational acceleration (m/s^2)
K : Thermal conductivity (W/m K)
m : Mass flow rate (kg/s)
P : Pressure (Pa)
q : Rate of heat transfer (W)
Q : Flow rate (m^3/s, m^3/min, m^3/hr)
R_f : Fouling factor
T : Temperature (K or ^oC)
U : Overall heat transfer coefficient (W/m^2 K)

Greek letters

β : Diameter ratio(=d/D)
Δp : Pressure drop along heat exchanger (Pa)
ΔT_m : log mean temperature difference (K)
ρ_m : Suspension density (kg/m^3)
ρ : Density (kg/m^3)

Subscripts

e : Exit
g : Gas side
i : Inlet
p : Particle
w Water side

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