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ANALYSIS OF HEAT TRANSFER BETWEEN TWO PARTICLES FOR DEM SIMULATIONS

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ABSTRACT

The purpose of the present study was to develop a model of heat transfer between particles that can be incorporated into the discrete element method (DEM). The flow around a particle was measured by particle imaging velocimetry (PIV) and the temperature of the particle was measured using a thermocouple and an infrared camera. The experimental data of heat transfer were classified according to the heat transfer mechanism, namely convection, conduction and contact. These values for heat transfer were compared with those calculated using previously derived estimation equations. From these results, we adopted the thermal contact resistance model, which is related to the surface roughness and contact force. Experiments were also carried out to examine the validity of the model. The contact heat transfer increased as the surface roughness increased. This is not a general trend because a large surface roughness causes a large thermal resistance, resulting in a small heat transfer. This trend is considered to be due to the increase in the contact area that accompanies an increase in surface roughness. The contact heat transfer calculated by considering the effect of the surface roughness on the contact area was found to show better agreement with the experimentally obtained values.

INTRODUCTION

Recent improvements in computers have allowed researchers and engineers to conduct numerical experiments for the purpose of verification. Owing to environmental problems and the increased need for energy conservation, highly accurate analyses of heat transfer characteristics in fluidized beds are required.

The discrete element method (DEM) is a numerical method for particulate systems and is useful for simulating heat transfer in fluidized beds. The DEM has been widely utilized because it is able to account for factors that cause problems associated with agglomeration, sintering, attrition and/or erosion. Rong and Horio [1] simulated char combustion in a fluidized bed using the DEM and obtained remarkable results. For example, they found that the characteristic temperature fluctuates at a frequency of 5

- 7 Hz and that the maximum characteristic temperature was 50±5°C higher than the average bed temperature. In their simulation, the contact heat transfer between two particles was calculated by assuming the heat conduction through a gas film between the two contacting particles. However, this assumption has not yet been validated, and there exist few decisive constitution equations of heat transfer between particles that can be incorporated into the DEM.

Accordingly, the authors [2] performed visualization of heat and flow and determination of the heat transfer between two particles in order to establish a model of the heat transfer between particles for DEM simulation. Various heat transfers were estimated using the correlations in the literature, and these were compared with the experimental values. The contact heat transfer calculated using the thermal resistance model showed better agreement with the experimental value than that calculated using the Rong and Horio equation [1]. The thermal resistance model will give reasonable values if the thermal contact resistance can be estimated more accurately. As far as the thermal contact resistance is concerned, various investigations have been done. However, there are a few investigations on the contact thermal resistance between non-flat surfaces. Lambert and Fletcher [3] made analytical and graphical investigations on the macroscopically non-uniform thermal contact conductance and added the contact between two non-flat rough spheres into thermo-mechanical model. Kumar et al. [4] predicted the thermal contact conductance between curvilinear surfaces under a certain range of contact pressures which was developed based on the Monte-Carlo simulation model.

The purpose of the present investigation is to examine the applicability of the thermal resistance model to the contact heat transfer between two particles.

EXPERIMENT

Two stainless steel spheres (SUS304) having diameters of 19.8 mm were utilized as particles. One particle was heated to 50, 100, 150 or 200°C, and the other particle was maintained at 21°C. This state was set as the initial condition, and the two particles were then brought into contact. The two particles were then compressed with some forces, $F=100$ to 500N, in order to simulate the force occurring during a collision between spheres.

The temperature of each particle was measured using a K-type thermocouple that was inserted into the center of each particle. The total heat transfer from the heated particle was obtained based on the time-variation in temperature:

$$Q_{total} = mc \frac{dT}{dt} \quad (1)$$

where m is the mass of the particle and c is the specific heat of the particle.

The flow around the two contacting particles was measured using a particle imaging velocimetry (PIV) system (TSI Inc, USA). The PIV system consists mainly of a

double-pulse Nd:YAG laser (BM industries, SERIE 5000; wavelength: 532 nm, 10 mJ – 3 J/pulse) and a CCD camera (TSI Inc, PIVCAM10-30, Model 630046) capable of recording at 30 fps with a resolution 1,000x1,016 pixels.

The authors [2] found that the thermal resistance model gives reasonable values by which to estimate the contact heat transfer between two particles. Because the surface temperatures are needed for the calculations using the thermal resistance model, the temperature distribution should be measured for a large particle, such as the particle in the present study. Accordingly, the temperature distribution in a particle that has a ditch was measured using an infrared (IR) thermal imager (NEC San-ei Instruments Ltd., Japan, TH9100 PMV). Moreover, experiments were carried out for various surface roughnesses. The surface roughness of the particle was measured using a confocal optical microscope (KEYENCE Co., Ltd., Japan, VF-7500).

In a previous study [2], the particles were set in a vertical array. However, natural convection causes a large stagnant region near the contact part of two vertically arrayed particles. The heat transfer through this stagnant region is classified as conduction heat transfer in the present analyses (see the next section). The accuracy of the analysis is reduced when there are several types of heat transfer. Accordingly, the particles were set in a horizontal array in order to eliminate or reduce the effect of the conduction heat transfer.

The temperature distribution in particles was measured using particles with a groove as shown in Fig. 1. The temperature on the bottom surface of a groove was observed using the IR camera. In order to examine the effect of a groove, the comparison of the temperature at the center of a particle was made between two particles with and without the groove. The maximum temperature difference is 2.43 °C and the average difference is 1.43 °C until $t = 1598$ s ($T_h = 200$ to 34.6 °C). As a result, the effect of the groove is small and can be ignored.

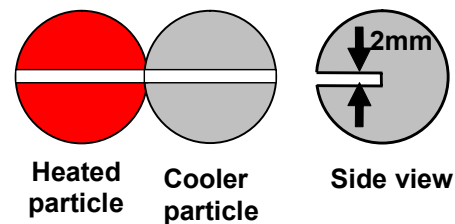


Fig. 1 Grooved particles for temp. distribution measurement

ANALYSES

Classification of heat transfer

In order to incorporate the model of heat transfer between particles into the DEM, the measured heat transfer was classified as shown in Fig. 2. The calculation procedure for the various heat transfers is the same as that described in the previous report [2]. The contact heat transfer, Q_{cont} , was obtained by subtracting the convection heat transfer, Q_{conv} , and the heat loss to a thermocouple and an insulator board, Q_{loss} , from the total heat transfer from a heated particle, Q_{total} . Both Q_{conv} and Q_{loss} were measured separately. Here Q_{conv} was obtained from Q_{total} for a single heated particle

on a prorated basis of the surface area, assuming that the heat flux on the surface was uniform. In the present system, the stagnant region is expected to be small. Thus, Q_{conv} can be ignored. As such, the area of the convection heat transfer can be obtained by subtracting the contact area from the total surface area. This was confirmed by flow visualization by PIV.

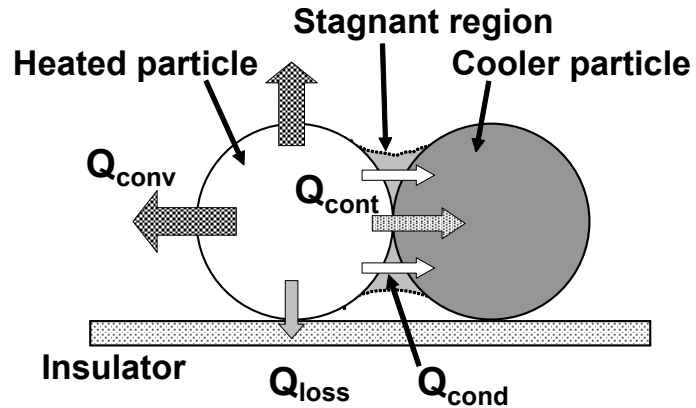


Fig. 2 Classification of heat transfer between two contacting particles.

Thermal resistance model

The heat transfer through the contact area can be modeled based on the thermal resistance. Accordingly, the contact heat transfer was also estimated using the thermal resistance model, as follows:

$$Q_{\text{cont}} = -\frac{1}{R_{th}}(T_h - T_c)\pi(r_p \sin \theta_{\text{cont}})^2 = -\frac{1}{R_{th}}(T_h - T_c)A_{\text{cont}} \quad (2)$$

where R_{th} is the contact thermal resistance, and A_{cont} is the contact area calculated using Hertz's contact theory [5]. In the present analyses, R_{th} was estimated using the equation of Zhang et al. [6]:

$$\frac{1}{R_{th}} = 12\sigma^{0.4} \quad (3)$$

where σ is the contact stress between two particles.

Analysis conditions

The analysis conditions are shown in Table 1. The Wen-Yu [7] correlation indicates that the minimum fluidization velocity, u_{mf} , is 7.48 m/s under the present conditions. This is due to the large particles (sphere) used in the present experiment.

Table 1 Analysis conditions

Particle:	SUS304 stainless steel
d_p [mm]	19.8
ρ_p [kg/m ³]	7.93×10^3
c [J/kgK]	0.5×10^3
R_A [μm]:	0.3, 1.1, 2.2, 5.5, 8.5
F [N]	100, 200, 300, 400, 500
Gas:	Air

RESULTS AND DISCUSSION

Visualized flow near the contact region

In the present experiments, two particles were set in a horizontal array in order to

eliminate or reduce the effect of conduction heat transfer. Figure 3 shows the visualized flow near the contact region obtained by PIV. The flow caused by natural convection can be observed near the contact region. Since the stagnant region is small, the conduction heat transfer was ignored in the present analyses.

Influence of contact force on heat transfer

Figure 4 shows the variation of the contact heat transfer with the contact force for various initial temperatures. The broken line indicates the value calculated with Eq. (2). The calculated results agreed well with the experimental results when the contact force is small. However, the difference between the estimated and experimental values becomes large as the contact force becomes large. Although the maximum difference is 2.3 times, this difference is much smaller than that which is calculated with the method of Rong and Horio [1].

Thermal resistance and heat flux between particles

Figure 5 shows the temperature distribution in two particles at $T_h=197.1\text{ }^\circ\text{C}$ and $t = 7\text{ s}$. The contact point is at $x=0$. The temperature gradient near the contact point is not so steep because the space resolution of the IR camera is 0.5 mm in this case. Accordingly, the temperature on the contact surface was defined as one at an inflection point around $x = 0$.

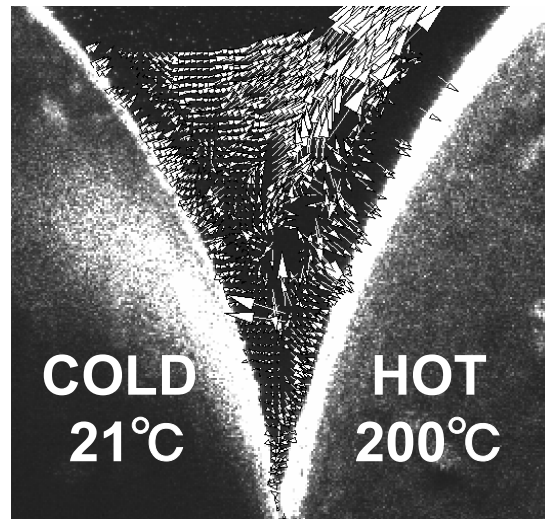


Fig.3 Visualized flow near contact region

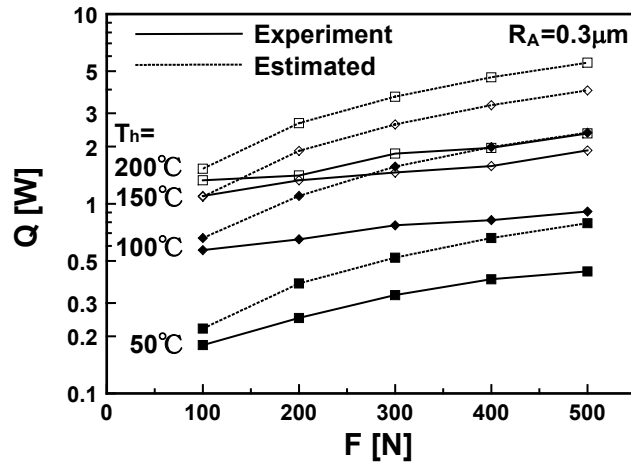


Fig. 4 Influence of contact force on contact heat transfer

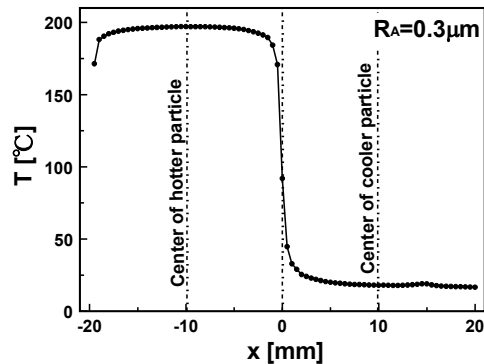


Fig. 5 Temperature distribution at $T_h=197.1\text{ }^\circ\text{C}$

Figure 6 shows the time-variance of the thermal resistance value. In order to calculate the thermal resistance, the surface temperature at the contact point was measured using the IR camera as shown in Fig. 5. The hotter particle was cooled after contact, which is defined as $t = 0$ s. The thermal resistance increases slightly with time. In other words, the resistance increases with the temperature of the hotter particle. Moreover, the resistance increases with an increase in the contact force up to 300 N, which is a general trend. However, the thermal resistance is approximately constant in the contact force range of 300 to 500N.

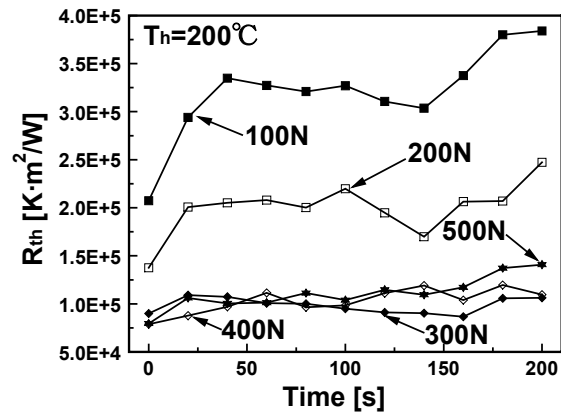


Fig. 6 Time-variance of thermal resistance

The thermal resistance averaged from $t = 0$ to 200 s is plotted with respect to the contact stress in Fig. 7. Here, the contact stress was obtained with the contact area calculated using Hertz's contact theory and with adjustment for the effect of surface roughness [2]. The resistance decreases as the contact stress increases, which is a general tendency. However, the tendency in the contact stress range of 517 to 613 MPa for $T_h = 200^\circ\text{C}$ is different from that in the other range. This may be a result of the calculation of the heat transfer by natural convection. In the present calculation, the convection heat transfer was calculated based on the assumption that the heat flux on the particle surface is uniform. However, the nonuniformity of the temperature distribution becomes large with time.

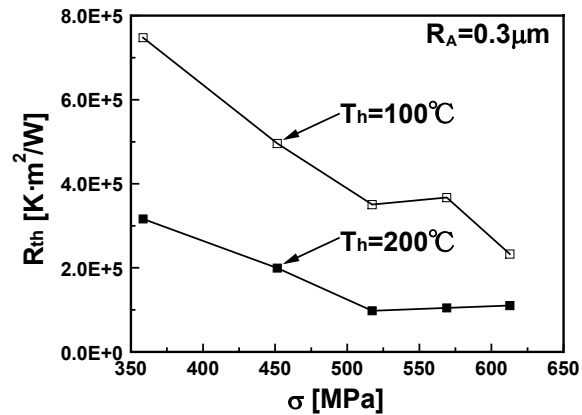


Fig. 7 Thermal resistance plotted against contact stress

Influence of surface roughness on heat transfer

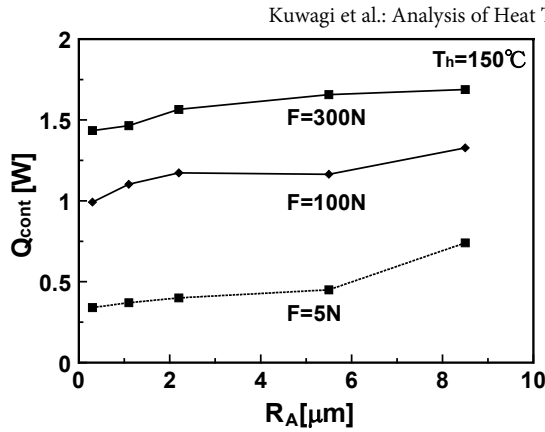


Fig. 8 Influence of surface roughness on heat transfer.

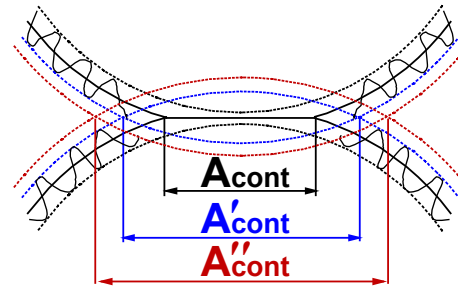


Fig. 9 Relationship between contact area and surface roughness.

Figure 8 shows the variation in the contact heat transfer with average surface roughness, R_A . The contact heat transfer increases as the surface roughness increases for all contact forces. Generally speaking, the contact heat transfer decreases as the surface roughness becomes large because the thermal resistance increases. However, the tendency shown in Fig. 8 is the opposite tendency. This tendency is believed to be caused by the curved surface. When the surface of a particle is completely smooth, the contact area becomes A_{cont} , as shown in Fig. 9. On the other hand, the contact area of a particle with surface roughness becomes A'_{cont} , and that with a larger surface roughness becomes A''_{cont} , as shown in Fig. 9. Thus, the large surface roughness leads to an increase in the contact area for the curved surface. In the present condition, the effect of the increase in the contact area would be larger than the effect of the increase of the thermal resistance, which causes the tendency in Fig. 8.

Figure 10 shows the contact heat flux calculated using A'_{cont} . The heat flux for all contact forces decreases with increasing surface roughness. This result is physically reasonable. However, the heat flux is very large when the contact force is small, $F = 5 \text{ N}$, and the surface roughness is also small. This tendency is different from the other cases. Since the contact area is very small in this case, the conduction heat transfer was generated around the contact region.

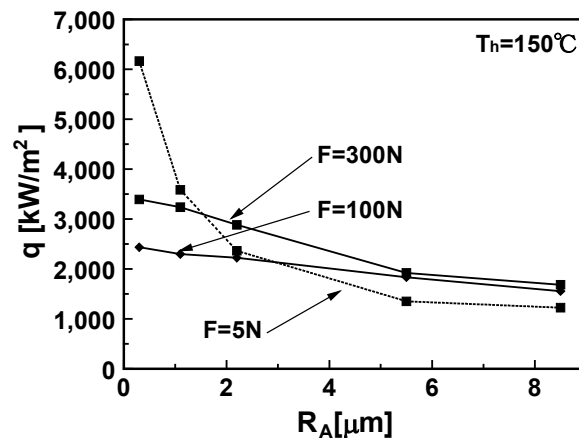


Fig. 10 Influence of surface roughness on heat flux.

CONCLUSIONS

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The contact heat transfer between two particles was analyzed in order to establish the contact heat transfer model, which can be incorporated into the DEM. The thermal resistance model was adopted, and its applicability was examined. From the results, the thermal resistance model agreed well with the experimental values. However, the maximum difference was 2.3 times. On the other hand, the effect of the surface was also examined. The roughness had an important influence on not only the thermal resistance but also the contact area.

In order to obtain higher accuracy in order to estimate the contact heat transfer, the microscopic states in the contact region should be analyzed in further detail.

ACKNOWLEDGEMENT

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NOTATION

A_{cont}	contact area (m ²)	R_A	average surface roughness (μm)
c	specific heat of particle (J/kgK)	R_{th}	thermal resistance (m ² K/W)
d_p	particle diameter (m)	r_p	particle radius (m)
F	contact force between particles (N)	T	temperature (°C)
m	particle mass (kg)	T_c	temp. of cooler particle (°C)
Q	heat transfer (W)	T_h	temp. of heated particle (°C)
Q_{cond}	conduction heat transfer (W)	t	time after contact (s)
Q_{cont}	contact heat transfer (W)	x	distance from contact point (mm)
Q_{conv}	convection heat transfer (W)		Greek letters
Q_{loss}	heat loss (W)	σ	contact stress between particles (Pa)
Q_{total}	total heat transfer from particle (W)		
q	contact heat flux (W/m ²)		

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