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HEAT TRANSFER COEFFICIENT BETWEEN FLAT SURFACE AND SAND

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ABSTRACT

Thermal energy storage (TES) systems are of interest in solar thermal power applications as an effective means of retaining energy. One of the primary issues with this type system is the exchange of thermal energy coming off the power field. In a heat exchanger, the effective heat transfer coefficient between the exchange mediums plays a crucial factor in determining the sizing of the heat exchange unit. A concept utilizing sand as a cheap particulate thermal medium was recently proposed for an alternative thermal energy storage system. The overall system will be described in some detail; however, the primary focus of this research report will be to present the experimental results measuring the heat transfer coefficient between flowing sand and a representative heat exchanger surface.

To measure the heat transfer coefficient a horizontal rotating drum is used to continuously deposit sand over a centrally positioned test article. The heat transfer coefficient in this case was calculated by taking the power input divided by the known area of the test article covered by the sand as well as the measured temperature difference between the article surface and sand temperature. Calibrated thin film thermocouples attached to the test article surface as well as thin film thermocouples suspended into the sand pooling in drum satisfy the needed temperature measurements. Then, by electrically heating a known area of the test article, a heat transfer coefficient between the sand and surface can be determined. Insulation of key end surfaces and errors such as heat leak due to air as well as measurement inaccuracies were also accounted for in the experimental setup and are included in the report's error propagation analysis. The overall results compare heat transfer coefficients measurements for a range of different sands and sizes, as well as model comparisons with known literature on the subject.

1. INTRODUCTION

The following paper covers the results of exploratory research on a proposed Thermal Energy Storage (TES) system which intends to compliment solar thermal power generation. Specifically the focus of this paper is the evaluation of the heat transfer performance of sand as suitable thermal storage medium. Standard concentrator solar thermal power plants typically employ a heat transfer fluid (HTF) that is heated in the collector field then routed to the power generators or TES unit. A fairly clear option for a TES system would be to utilize the existing HTF as the working storage medium. Using conventional HTF's comes with some unfortunately stiff economic drawbacks. These fluids are quite costly as quantities needed for storage are quite significant, in addition their associated high vapor pressures require expensive highly reinforced storage facilities. The proposed storage system seeks to use sand as the storage medium; greatly reducing the expenses involved for both medium and storage costs. TES designs using sand or other solids in a fixed or non-kinetic state for thermal exchange suffer significant losses due to charge/discharge temperature drops. The proposed TES system will instead move the sand to drive a general counter flow thermal exchange. Note that some of the descriptions of the TES system and development in this paper have been presented previously as [1] and [2]; however since development has been ongoing and since these publications had limited circulation, some descriptive material is repeated herein. This counter flow design allows for a much closer temperature of approach as compared to a fixed bed. As cost and performance are the primary goals to tackle of the proposed system, the evaluation of the sand's thermal exchange effectiveness in a flowing state is necessary. Ongoing experiments are being conducted to measure the effective heat transfer coefficient between the sand and representative solid surfaces used as the heat transfer conduits. Key investigational aspects of these experiments involve the sand grain size as well as shape of the thermal exchanger conduits.

Thermal Energy Storage Concept

Heat transfer to flowing particulates, such as sand, is not commonly encountered in thermal engineering applications. Nevertheless, one emerging interest is that sand could be used as the storage medium in a proposed thermal energy storage system. The proposed TES will be incorporated into an overall concentrator solar power (CSP) system. In operation, a conventional heat transfer fluid is heated in the collectors. This heat must then be transferred to and from sand acting as the storage medium. The overall TES design concept, illustrated in Figure 1.1 for the heat storage process, is to utilize a very inexpensive and benign storage medium, specifically ordinary silica sand or similar fine grained material. In operation, the sand is transported between separate insulated storage containers as the sand is heated and then the transport is reversed as stored heat is recovered from the sand. For a typical CSP trough system, the hot HTF from the solar field is around 370 C (700 F), and the fluid is typically returned at temperature around 270 C (520 F). In such an energy storage system, one container would contain only moderately warm sand close to 270 C that is available to be heated and store energy, and the other would contain the hot sand at around 370 C after it has been heated with HTF from the solar field to store energy.

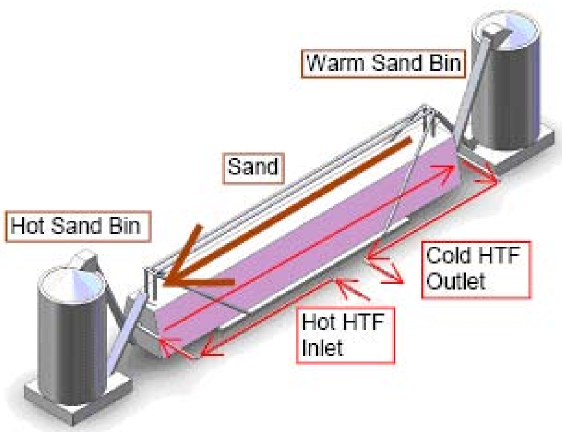

Figure 1.1 Conceptual Design of the Proposed TES System Shows during heat storage with high and low temperature containers connected by Sand Shifter heat exchanger/conveyor.

The Sand Shifter

The HTF will both heat sand for storage during solar energy collection and recover heat from storage. Obviously, heat exchange is needed in any such indirect TES concept, in which different heat collection and storage media are used. The innovative enabling technology in this system is the combined sand conveyor and heat exchanger identified by our development team as the Sand Shifter.

The shifter moves the sand and oil in overall counterflow as heat is exchanged. The Sand Shifter itself is an innovative design that combines the functions of conveying sand and exchanging heat in one device. This system is currently the subject of intellectual property applications, so its details will not be disclosed at present. Nevertheless, to understand the overall system and the engineering issues, it is only necessary to understand the general operation of the device. In the Sand Shifter, sand will be moved longitudinally between high and low temperature storage containers while the sand is simultaneously lifted and poured over finned tubes (or other metal conduits) that contain the counter-flowing HTF. The primary issue in this design is the performance of the heat exchange process between the flowing sand and metal surfaces. The achievable heat flux density largely dictates the overall sizing of the sand shifter and, as a result, its economic feasibility. The balance of this paper addresses measurements of the coefficient of heat transfer between flowing sand and a heat exchange surface.

2. LITERATURE REVIEW

When looking at current literature on particle interaction there are several subsets of interest: fluidized bed heat transfer, heat transfer in an immobile bed of particles, particle flow, and heat transfer in a moving bed of particles. Because convection from the gas to the particle plays a much higher role in a fluidized bed it would be expected that these would have little relevance to the current project, however, there are a number of papers modeling the heat transfer between particle-particle and particle-surface interactions [3-5]. These mostly use a kinetic/collision theory approach to determine the heat transfer. The papers on immobile beds focus on the effective conductivity of the particles using thermal particle dynamics (TPD) [6-8]. The particle flow papers can be broken up into two different types, kinetic theory based flow for low particle density (falling particles/fluidized beds) [9] and shear based flow for denser particle flow [10-11]. The heat transfer in a moving bed of particles can be further broken up into heat transfer in a rotary vessel using discrete element method (DEM) modeling [8,12-13] which are numerical methods for computing the motion of a large number individual of particles of micrometer-scale size and above. The remaining set are direct experiment based heat transfer models for a moving bed of particles [14-20].

Particle Interaction

Süle et. al. [3] used a population balance model which takes into account the particle-particle and particle-wall collisions to describe heat transfer processes in fluid-solid systems and employs a compartment model to describe the spatial distribution of the temperature in a unit. Both the particle-particle and particle-wall heat transfer are modeled by collisions with random parameters. The results of this showed that the intensity of inter-particle collisions play a significant role in reducing the temperature dispersion of particles while, increasing fluid-particle heat acts inversely. In Sun et. al.'s [4] investigation, the particle impact heat conduction through time varying contact area during impact was examined for the purpose of quantizing the direct conductive contribution of heat transfer between particles and surfaces in suspension flow and fluidized beds. Sun et. al. found that this heat transfer mechanism does not appear to be dominant in fluidized bed under typical conditions. Natale et. al. [5] report experimental results on the heat transfer between a fluidized bed of fine particles and a submerged surface. Their results show that the heat transfer coefficient increases with particle Archimedes number and is almost independent from particle thermal conductivity for $K_p/K_g>30$.

Thermal Particle Dynamics

Vargas, et. al. [6] investigated the heat conduction in a packet bed of cylinders both experimentally and computationally using the discrete element method. By explicitly modeling individual particles within the bulk material, bed heterogeneities are directly included and dynamic temperature distributions are obtained at the particle level. They found that stress chains in a particle bed tended to augment heat flow along a particular axis while hampering heat transfer in the perpendicular of that axis. Vargas, et. al. [7] also extended the numerical technique, the thermal particle dynamics method, to study heat conduction in granular media in the presence of stagnant interstitial fluids. Vargas, et. al. determined when $K_p/K_f>>1$, TPD provides good qualitative and quantitative agreement between measured and calculated values of the effective conductivity for a wide variety of materials and for packed beds at different loads in the presence of both liquid and/or gases. Furthermore Vargas-Escobar's [8] thesis addressed heat conduction in granular systems both under static and slow flow conditions with and without the presence of a stagnant interstitial fluid using the TPD method. For a rotating drum flow Vargas distinguished the mechanisms by which the heat transport process takes place: at low shear rates (small mixing factor) conduction through particle contacts dominates due to lasting contacts; as the shear rate increases (larger mixing factor), convective mixing caused by an increased granular temperature enhances the transport of heat and therefore the effective conductivity increases proportionally. This latter analysis also included DEM modeling.

Particle Flow

Lun, et. al. [9] studied the flow of granular material using statistical methods analogous to those used in the kinetic theory of gases. Two theories are developed: one for the Couette flow of particles having arbitrary coefficients of restitution (inelastic particles) and a second for the general flow of particles with coefficients of restitution near 1 (slightly inelastic particles). Approaching from a different angle using a dense particle arrangement, Thompson, et. al. [10] described molecular-dynamics simulations of non-cohesive granular assemblies under shear. Low shear rates exhibit stick-slip dynamics while steady-state motion occurs at larger shear rates with a static and a flowing layer. Also for dense particle flow, Baxter, et. al. [11] described dynamic measurements of the stress obtained during a sand flow. The data showed a large noise component.

Discrete Element Method Modeling

Shi, et. al. [12] employed a computational technique that couples the DEM, computational fluid dynamics (CFD), and heat transfer calculations to simulate realistic heat transfer in a rotary kiln. They found that at low particle conductivities, the heat transfer is dominated by gas–solid conduction; however, at higher particle conductivities solid–solid conduction plays a more important role. This is similar to the results found in Vargas's work. Chaudhuri, et. al. [13] used the discrete element model to simulate the dynamic behavior of cohesive and non-cohesive powder in a rotating drum (calciner) and double cone (impregnator). The granular material was considered as a collection of frictional inelastic spherical particles. Each particle was able to interact with its neighbors or with the boundary only at contact points through normal and tangential forces. The model simulated flow, mixing, and heat transport in granular flow systems for the rotary calciners and impregnators. Their simulations showed that as rotation speed decreases, both heat transfer and temperature uniformity of the granular bed for both calciner and impregnator increase.

Heat Transfer in Moving Bed of Particles

Molerus [14] reported that the contact resistance for the heat transfer between adjacent particles was the limiting factor for heat transfer in moving beds of particles consisting of rather hard solid materials and filled with a stagnant interstitial gas. Brinn, et. al. [15] measured slug flow of sand flowing through a pipe. They found better transfer coefficients in smaller inner diameter pipes, with increased flow rate further improving the heat transfer. The heat transfer coefficient for slug flow in the pipe cases were found to range between 40~120 W/m^2-K. The overall purpose of the experiments was to create a theoretical model of the heat transfer mechanics of the pipe flow. Measurements done by Denloye, et. al. [16] noted that the flowing packed bed to surface heat transfer coefficient increases with decreasing residence time, with decreasing particle size, and with increasing gas thermal conductivity. Regarding the residence time, they indicated the increase in heat transfer coefficient was more pronounced for smaller particles. In the experiments using air/sand, Denloye, et. al. (1977) measured maximum heat transfer coefficients of ~125 W/m^2-K for 2370 µm sand, ~310 W/m^2-K for 590 µm sand, and ~475 W/m^2-K for 160 µm sand. Hyde, et. al. [17] measured a range of materials heat transfer coefficients using a fluidized bed. Bubbling air through sand they measured ~300 W/m^2-K for 560 µm sand, ~360 W/m^2-K for 450 µm sand, ~405 W/m^2-K for 295 µm sand, and ~450 W/m^2-K for 225 µm sand. Determination of heat transfer between the fluidized bed and the heating or cooling elements submerged in the bed was carried out by calculating the maximum heat transfer between a sphere and a wall under brief contact, as well as taking into account in addition the void fraction of the stationary packing and bed expansion. Patton, et. al. [18] proposed a model relating the

Nusselt number to a Péclet number and a Froude number. The predicted results of the model were compared with the experimental data from heat transfer over a flat plate. The experimental model was able to estimate a heat transfer coefficient for the flowing sand over a surface using a well-defined set of parameters. Another report by Babcock and Wilcox in 1981 [19] looked into a range of TES options including sand. They predicted based on Denloye, et. al.'s (1977) work that with a moving bed of fine grained sand they would achieve a heat transfer coefficient on their charging heat transfer elements for steam to the sand of ~930-1160 W/m^2-K and a coefficient of ~803-1308 W/m^2-K on the discharge side. Green, et. al. [20] did further exploration into thermal energy exchange systems and using a shell and tube design reported expecting heat transfer coefficient of ~1470 W/m^2-K for the sand side in that configuration. This result, though, was based in part from the flowing sand predictions in the Babcock and Wilcox (1981) report. The range of heat transfer coefficients from the various pieces of literature have been tabulated below in Table 2.1.

Table 2.1 Heat Transfer Coefficients for Moving Sand Reported in Literature

Authors	Description	Heat Transfer Coefficients
Brinn, et. al. (1948)	Slug flow of sand in pipes.	40~120 W/m^2-K
Denloye, et. al. (1977)	Packed bed flow over heated surface.	125 W/m^2-K~475 W/m^2-K Sand: 2370 μm-160 μm
Hyde, et. al. (1980)	Heat transfer for air/sand fluidized bed.	300 W/m^2-K~450 W/m^2-K Sand: 560 μm-225 μm
Patton, et. al. (1986)	Heat transfer model for flowing granular material.	Olivine: 350~1100 W/m^2-K Silica: 200~600 W/m^2-K
Babcock and Wilcox (1981)	Steam to flowing bed of sand heat exchanger.	Charge: 930~1160 W/m^2-K Discharge: 800~1310 W/m^2-K
Green, et. al. (1986)	Shell and tube heat exchanger using air and sand	~1470 W/m^2-K

Heat Transfer Model

Of the literature on particulate flow, Patton et. al.'s results were the best match for a comparative model analysis with the Sand Shifter's flow concept. The convective heat transfer coefficient of flowing sand was modeled in Engineering Equation Solver (EES) [21] based on the relations found in Patton, J. S., et. al.(1986) to compare it to the convective heat transfer coefficient determined experimentally. The heat transfer coefficient was determined for two types of sand. Olivine sand was experimentally measured to have a mean diameter of 78 μm with a standard deviation 30 μm. Silica sand was measured to have a mean diameter of 0.55 mm with a standard deviation of 0.32 mm. The conductivity, heat capacity, and density were obtained from the EES materials property package and the properties were assumed to be similar for both olivine and silica. The velocity of the sand layer was estimated to be between 0.15~0.3 m/s, the thickness of the layer to be between 1~4 mm, and the packing ratio between 0.42 and 0.74. Using these parameters the convective heat transfer coefficient was estimated to be between 350~1100 W/m^2-K for the olivine and 200~600 W/m^2-K for the silica. The flat surface experimental results, found in this paper, of ~500-600 W/m^2-K for olivine and ~300-500 W/m^2-K for silica fall well within the models predicted range. These values also were near those found in Denloye, et. al.'s [16] packed bed of flowing sand measurements and Hyde, et. al's [17] fluidized bed measurements. Of the factors that affect the heat transfer, particle size had the dominant effect on the heat transfer coefficient followed by velocity, and finally the layer thickness and packing ratio. On the higher range of reported results, Babcock and Wilcox in 1981 [19] indicated expecting a heat transfer coefficient, on their charging heat transfer elements, of ~930-1160 W/m^2-K using a moving bed of sand. The sand utilized in Babcock and Wilcox report had grain size of 44-77 μm, a packing ratio of 0.40, and a sand velocity ~0.15-0.3 m/s, right around the range of the olivine sand of the current experiment. Their results and those of Green, et. at. [20] suggest possibility for higher effective thermal transfer through the shape of the transfer surface. Furthermore Muchowski [22], Wunschmann, et. al. [23] reported that vibrating the vessel could yield a small improvement in the heat transfer coefficient for a particulate such as sand. These studies indicated that small vibrations had a positive effect on the heat transfer coefficients (an enhancing effect is observed), but the trend did not continue at larger vibrational accelerations. Another important side effect of vibration is that it allows sand to flow on much shallower slopes, allowing the angle of repose limit for natural gravity driven flow to be circumvented.

3. FLAT PLATE HEAT TRANSFER EXPERIMENT

Since heat transfer to flowing sand is not a familiar process, a preliminary, almost expedient, scoping experiment was conducted. This preliminary investigation of the heat transfer coefficient, defined in Equation (1), was conducted for two candidate materials, olivine and silica sands. Olivine was chosen for its established good performance in some foundry sands. As quantified by Equation (1), the overall experimental concept was to determine the heat transfer coefficient from measurements of the temperature of the free stream flowing sand and the temperature of the heat transfer surface, which was heated with a known power input.

Flat Plate Setup

The preliminary setup used an inclined flat surface. The heat source was a flat plate electric heater 152 mm x 152 mm square mounted beneath an aluminum heat transfer plate. Both flat plates and plates with square fins machined into the surface were studied. The fins were 3.18 mm (1/8 inch) high by 3.18 mm wide with 3.18 mm spacing. As shown in Figure 3.1, the aluminum heat exchange plates are placed on the heater plate. This assembly was inset into insulating board to prevent heat leaks, leaving only the upper surface of the aluminum plate exposed. A 3 x 3 evenly spaced grid of T-type thermocouples were then placed in shallow, small diameter holes drilled into the aluminum plate, and secured in place with a small amount of thermal cement. The surface temperature measurements were averaged to characterize the plate temperature (T_p). A large storage bin for sand with a dispenser nozzle at the bottom was placed above the plate. Gravity, assisted by vibration, allows the sand to easily flow from the bin over the heated surface. A thermocouple placed in the flowing sand just upstream of the dispenser measured the free stream temperature of the sand. With the hot plate operating at a known electrical power input, measurements were made of the temperature of the plate surface and the incoming sand, while a visual estimate was made of the contacted area (i.e. the portion of exposed surface covered by sand) of the flow over the plate. The “contacted” area of the flat plate was taken to be between 95-99% with only the top corners left uncovered. For the finned plate, the contacted area was seen to exclude the lower third of the tops of the finned surface due to insufficient sand submersion. Intentionally, this was a conservative estimate that should not exaggerate the heat transfer coefficient. The heat transfer coefficient is then calculated as follows:

$$h_s = \frac{\dot{W}_{in}}{A_c \cdot (T_p - T_s)} \tag{1}$$

Here the heat transfer coefficient of sand is represented by h_s, $\dot{W}_{in}$ is the electrical power input, A_c is the area contacted by the sand, T_p is the plate temperature measured by the embedded thermocouples, and T_s is the free stream sand temperature measured by the upstream thermocouple.

Flat Plate Measurements

This apparatus and method was applied to both silica and olivine sands, and repeated for both flat and simple square finned plate designs. Figure 3.1 shows sand flowing over a heated, instrumented plate in one of our earliest experiments. As seen in the photo, the sand from the hopper is dispensed over the heated plate. Measurements of the plate surface and sand free stream temperatures were taken when the system reached a steady state, *i.e.* approximately constant difference in temperature. More than 30 experiments of various combinations and configurations were run. The convection coefficient varied with plate temperature and contact area, so it is difficult to assign a specific value to each type of sand.

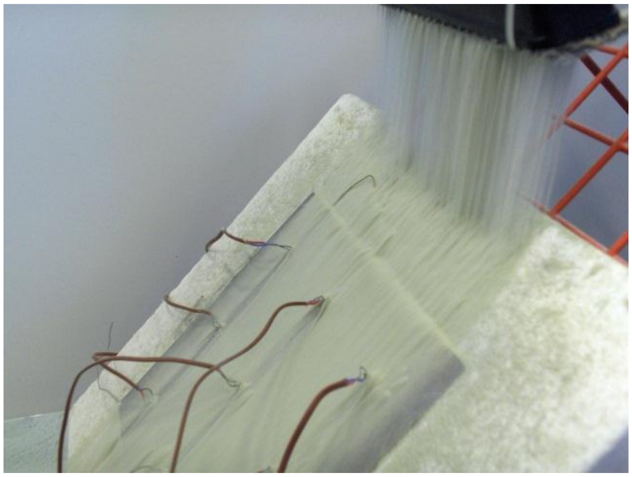

Figure 3.1 Preliminary Heat Exchange Experiment for Flow over Heated Flat Plate. Sand is flowing from nozzle over plate.

However, olivine sand appears to have a higher convection coefficient than silica sand. The ranges of measured heat transfer coefficients are given in Table 3.1.

Table 3.1 Heat Transfer Coefficients for Sand Types and Configurations

Configuration	Sand Type	Heat Transfer Coefficient (W/m²-K)
Flat	Silica	310-400
Finned	Silica	212-405
Flat	Olivine	550-925
Finned	Olivine	375-635

It is somewhat surprising to see that sand flowing over a flat plate shows a higher convection coefficient than that flowing over a finned plate, even after a correction for the contact area. It appears this shortcoming is a failure of the sand to continuously submerge all the fin surfaces as shown in Fig. 3.2.

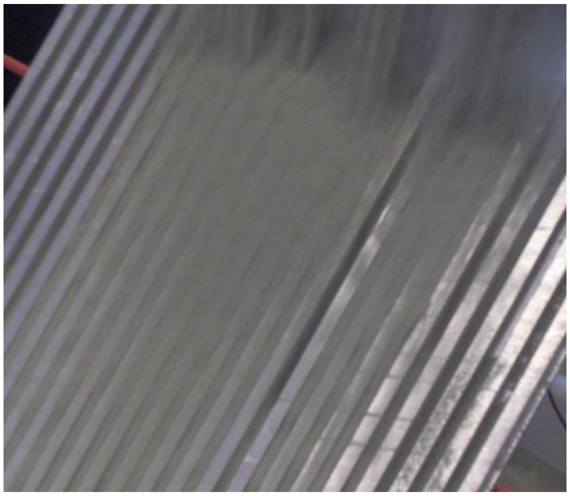

Figure 3.2 Sand Flow on Square Finned Surface.
Note the sand rarely has contact with the tips of the fins.

While this apparatus was helpful for screening experiments, it had several shortcomings: (1) The test articles were effectively limited to flat plates which are not representative of the finned tubes that might be used in practice. (2) The test can be run continuously only with great effort because the reservoir must be refilled manually. (3) The test cannot be run for long periods over a range of temperatures. These shortcomings were resolved by switching to the continuous drum device described in the next section.

4. ROTATING DRUM HEAT TRANSFER EXPERIMENT

This apparatus, seen in Figure 4.2, as well as adjusted methodology, was an improved means to allow continuous heat transfer operation over a larger range of temperatures with more realistic test articles. Here, a rotating drum with an array of internal scoops continuously lifts the sand from the bottom of the drum and pours it over an axially fitted test article.

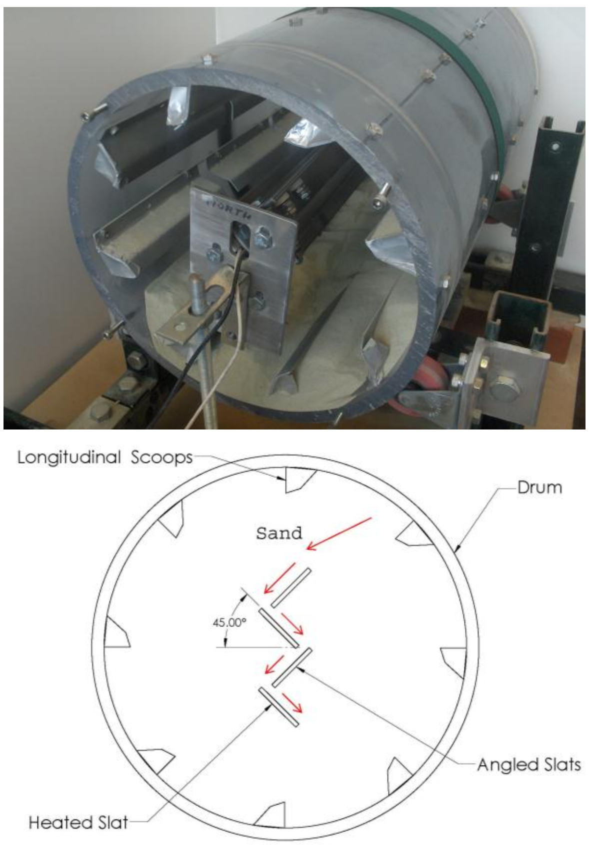

Figure 4.1 Drum Apparatus with Angled Slats
Real and Schematic View

Drum Setup

Intermittent flow was an issue for the design as the scoops only pour the sand at periodic intervals rather than continuously over the subject article. This problem, which must also be overcome in a full size system, was alleviated by testing with an array of angled flat slats. This arrangement created a zigzag flow pattern as shown in Figure 4.1.

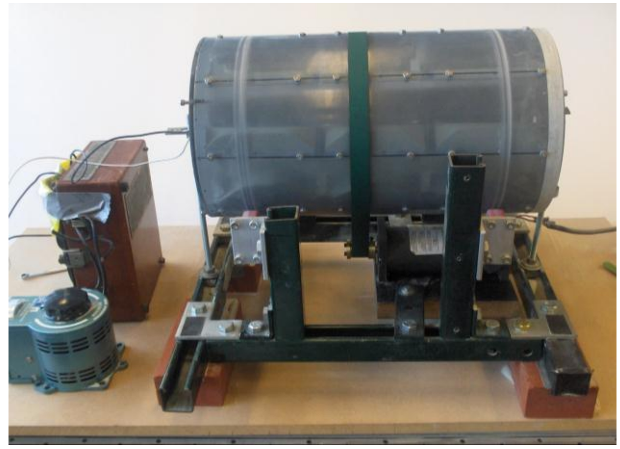

Figure 4.2 Drum Heat Exchange Measurement Apparatus

With this flow pattern, sand was held up by internal drag, and the intermittent deposition of sand as scoops pass over the array was quickly smoothed into a continuous flow over the lower slats. Currently, the slats are electric strip heater plates 38.1 mm wide and 336 mm in length. T-type thermocouples were mounted on the upper and lower surfaces of the lowest slat using thermal cement. Obviously, measuring any true surface temperature is somewhat challenging. In this case, excellent contact between the thermocouples and the metal surface is necessary to give a reliable temperature. Welded bead thermocouples were employed for the first set of measurements. For a second confirmation measurement the experiment was repeated with special purpose thin film thermocouples to improve the surface temperature readings. In two locations flanking the axial midpoint of the drum, rakes were suspended from the plate bundle to support the two thermocouples submerged in the sand to measure its temperature. The bottommost and most representative slat of continuous flow was then heated with a known electrical power input. The flow speed over the slats was estimated by gravity to be ~0.15-0.6 m/s. High speed video and image analysis software [24] looking at the sand flowing over the slat yielded a measured sand speed between, ~0.11 m/s for upper slat to around 0.17 m/s for the lower slats. The thermocouples mounted on the front and back sides measured the temperature of the sand covered surface and air cooled surface respectively.

Heat Leak due to Air and Heat Transfer Coefficient

Experimental runs with no sand, using only air for cooling, yielded a heat transfer coefficient for air cooling of around 10 W/m²-K. This result fell within documented range [25] of buoyant gas convection and was confirmed using the well-known formulas from McAdams [26] for heated plates. The experiment was then repeated with sand, the upper side thermocouple reading surface temperature contacted by the sand, and the lower side thermocouple measuring the temperature of the slat exposed to air. The readings of the two thermocouples suspended on the rakes were averaged to return the medium temperature of the sand. Only the heated portion of the slat was exposed with the top side covered in sand and bottom air cooled. In an experiment, the electrical input power was controlled with an autotransformer and measured with an electronic power meter. The surface and sand thermocouple temperatures were measured with a scanning electronic thermometer, and the heat transfer coefficient can be calculated for the sand as:

$$h_s = \frac{\dot{W}_{in} - \dot{Q}_a}{A_{sur} \cdot (T_{sur} - T_s)} \quad (2)$$

The heat transfer coefficient of sand is represented by h_s, $\dot{W}_{in}$ is the electrical power input (W), $\dot{Q}_a$ is the estimated heat leak rate from the air side (W), A_{sur} the heated area covered by sand (m²), T_{sur} is temperature of the surface exposed to the sand flow (K), and T_s the measured sand temperature (K).

5. CONVECTIVE PARTICLE PERFOMANCE ON A FLAT SURFACE

First Measurement Set: Bead Thermocouples

The newer experimental setup allowed for an improved and more realistic estimation of the heat transfer coefficient of the sand flowing over a controlled heat exchanger surface. Exercising this arrangement with the olivine and silica sands generated measured heat transfer coefficients, and the results computed by regression over a range of temperature differences are shown below in Figure 5. 1.

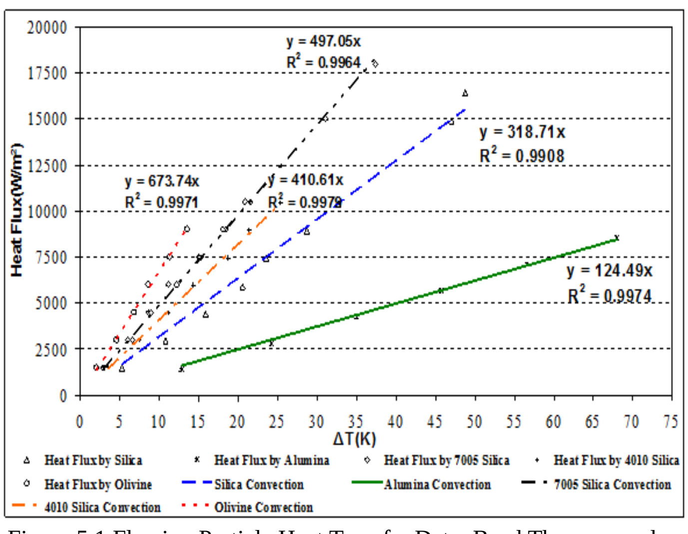

Figure 5.1 Flowing Particle Heat Transfer Data: Bead Thermocouples

The regression line slopes in Fig. 5.1 indicate the average convection coefficient from the data and show a reasonable and consistent trend in temperature difference as the power input is varied. The calculated convection results have been tabulated below in Table 5.1. Examination of the data shows that the particle size played a dominant role in convective performance. This feature is also indicated in the literature.

Table 5.1 Heat Transfer Coefficients by Sand Type

Sand Type	Average Grain Size (µm)	Heat Transfer Coefficient (W/m²-K)
Olivine	80	680
Fine Sifted Silica	140	500
Sifted Silica	290	410
Regular Silica	550	320
Alumina Beads	760	120

As previously indicated, grain size seems to play a major role in the convective transfer performance with smaller grain sizes achieving better flowing surface contact. Observation of the experiment indicates that effective surface contact and flow continuity will be critical factors in the heat exchanger assembly evaluation. A suitable design for heat exchanger elements that effectively direct flow of the sand will be critical to obtaining optimal thermal performance.

Second Measurement Set: Film Thermocouples

Once the experiment had been fully conducted preparations were made to fully repeat the heat transfer measurements after changing out the bead thermocouples for thin film thermocouples. The thin film thermocouples when properly adhered to a flat surface tend to return a more accurate reading of that surfaces temperature. In short, this experiment was repeated to see if the experimental results remained fairly consistent between different means of measurement. Using this arrangement with the olivine and silica sands generated measured heat transfer coefficients. The results were calculated by regression over a range of temperature differences are shown below in Figure 5.2 along with the first set of results.

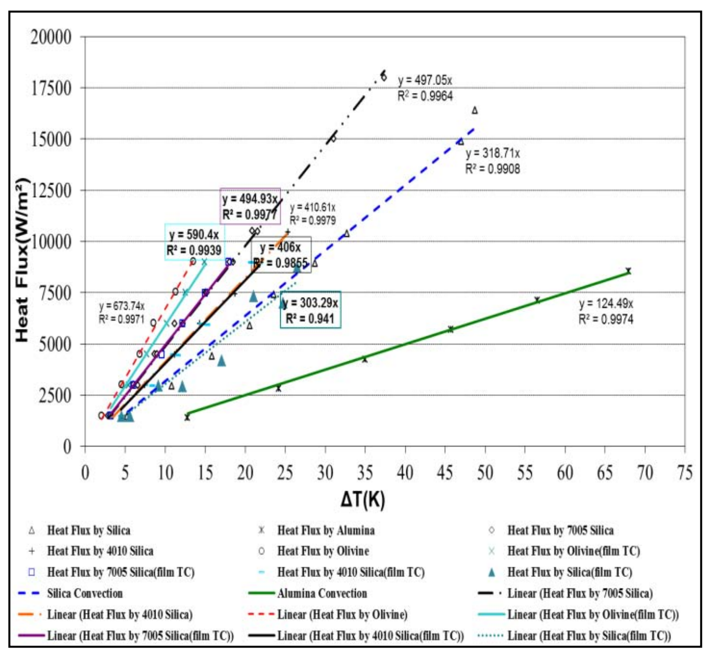

Figure 5.2 Flowing Particle Heat Transfer Data: Film Thermocouples

For this measurement trial, since the Alumina particles had proved to be so destructive, this material was not re-measured with the thin film thermocouples. The regression line slopes in Fig. 5.2 indicate a somewhat lower average convection coefficient from the particles. This was expected as the bead type thermocouples were more protruded into the flow from the surface and as such experience somewhat greater cooling then the surface. The thin film thermocouples due to close surface profile return a more true surface temperature when submerged in the sand flow. There is the same consistent trend in temperature difference as the power input is varied difference as the power input is varied. The calculated convection results with the film thermocouples have been tabulated in Table 5.2. Examination of the data reaffirms the initial set of measurements that showed that the particle size plays a leading role in convective performance.

Table 5.2 Heat Transfer Coefficients by Sand Type: Film Thermocouples

Sand Type	Average Grain Size (µm)	Film Heat Transfer Coefficient (W/m²-K)	Bead Heat Transfer Coefficient (W/m²-K)
Olivine Foundry Sand	80	590	670
Fine Sifted Silica (7005)	140	490	500
Sifted Silica (4010)	290	410	410
Construction Silica	550	300	320
Alumina Beads	760		125

As previously indicated, grain size seems to play a major role in the convective transfer performance with smaller grain sizes achieving better flowing surface contact.

Error Analysis

In conducting analysis of the data it was important to examine its measured reliability. For this analysis, the focus will be on the most refined portion of the experiment, the film thermocouple results. Each component measurement came with a set uncertainty and/or assumptions. In Equation 2,

$$h_s = \frac{\dot{W}_{in} - \dot{Q}_a}{A_{sur} \cdot (T_{sur} - T_s)} \tag{2}$$

$\dot{W}_{in}$ is the electrical power input (W) was measured using an Weston analog watt meter with a scale of 0 to 125 W. The uncertainty obtained from the instrument spec sheet was 1.25 W. This device was hooked up in series from a VARIAC to a heated slat to accurately measure its input power. It was assumed that losses were minimal between the slat and the watt meter, and that the power delivered to the slat heater was all released as thermal energy. For the estimated loss due to air a convection coefficient, $h_{air,}$ of 10 W/m²-K was measured in the rotating drum when run without sand. This $\dot{Q}_a$,

$$\dot{Q}_a = h_{air} \cdot A_{sur} \cdot (T_{back,sur} - T_s) \tag{3}$$

or heat leak due to air was calculated assuming that the underside of the plate was exposed air with a thermocouple located on that surface, $T_{back,sur}$. For the loss, roughly ~5% of the total power input was lost to air cooling. The air temperature as assumed to be the same as the one measured by the sand rake thermocouples, T_s. A hand held temperature probe indicated this assumption to be quite consistent. The area, A_{sur}, was taken as the exposed underside of the slat to the dimensions of the heated portion. The upper surface was exposed to the sand and the ends were wrapped in insulation and assumed adiabatic. A finely indiced ruler was used for determination of the area yielding a measurement uncertainty of 1.85E-07 m². The thermocouples were all calibrated for a range of temperatures using a standardized RTD probe. For the calibrations the T_{sur} had an uncertainty of 0.0244 K. Likewise the T_s was found to have an uncertainty of 0.0167 K. The error propagation for the calculation of the heat transfer coefficient was then preformed. The result for a representative case is shown in Table 5.3 below.

Table 5.3 Heat Transfer Coefficient Error Propagation

Olivine

Measurement	U_{xi}		$\frac{\partial h}{\partial x_i}$		$U_i^2 = \left(U_{xi}\frac{\partial h}{\partial x_i}\right)^2$		Basis[a]	Source[b]
Power	1.25	W	5.22	(1/K-m²)	42.54	(W/K-m²)²	general specifications	Weston Instruments
Area	1.85E-07	m²	46985	W/K-(m²)²	7.56E-05	(W/K-m²)²	resolution	Indiced Ruler
$T_{surface}$	0.024	K	40.3	(W/m²)	0.967	(W/K-m²)²	calibration	see note (1)
T_{sand}	0.017	K	40.3	(W/m²)	0.450	(W/K-m²)²	calibration	see note (1)
Total $(U_b)^2$					42.55	(W/K-m²)²		

Note(1): Standardized RTD Probe

Table 5.4 Regression Analysis of Heat Transfer Coefficients and Combined Uncertainty

Sand Type	Coefficient (W/K-m²)	U_a (W/K-m²)	U_b (W/K-m²)	U_c (W/K-m²)
Olivine	590	13	6.5	15
Fine Sifted Silica	490	12	5.4	13
Fine Silica	410	20	6.7	22
Construction Silica	300	34	3.9	34

As can be seen in Table 5.3 the primary measured error is from the wattmeter, overall influencing the measured uncertainty to ~43 (W/K-m²)² for the Olivine, an U_b of ~6.5 W/K-m². Regression analysis was performed on the flux/ΔT plot for the heat transfer coefficient to get the random uncertainty, U_a, which was the average of the difference for the upper and lower 95% band. The combined uncertainty, U_c, was calculated by Equation 3, with representative uncertainties for each sand type shown in Table 5.4.

$$U_c^2 = U_a^2 + U_b^2 \tag{3}$$

Overall the heat transfer coefficients measured by this experiment providing the assumptions of adiabatic insulation, cooling by air, and total coverage of only the upper surface by sand, have a high measurement reliability to within ~10% or less of the stated value. This level of uncertainty is well within the range typically expected in heat transfer engineering applications.

6. LITERATURE MODEL COMPARISON

Of the literature on particulate flow that we have reviewed, the work reported by Patton et al. [18] appears to be the most suitable resource for comparing with our empirical results. The convective heat transfer coefficient of flowing sand was modeled with the relations found in that paper [18] to compare the published results with the heat transfer coefficient determined experimentally. These parameters in the equation based model included particle size, conductivity, specific heat, flow velocity, layer thickness, and packing ratio (ratio of air space vs. solid space). From this data a heat transfer coefficient range was determined for several types of particles. For the particle types the following parameters were used: (1) Fine grained olivine sand [27] were experimentally measured to have a mean diameter of 80 µm, with a standard deviation 30 µm. (2) A slightly larger finely sifted silica [28] was measured to have a mean diameter of 140 µm, with a standard deviation of 50 µm. (3) Another sifted silica [28] sand was measured to have a mean diameter of 290 µm, with a standard deviation of 100 µm. (4) A coarser locally purchased construction silica sand was measured to have a mean diameter of 550 µm, with a standard deviation of 320 µm. (5) Finally spherical alumina particles [29] were measured to have a mean diameter of 760 µm, with a standard deviation of 120 µm. Additionally, the velocity of the particle layer was estimated to be between 0.15~0.3 m/s, the thickness of the layer to be 3-5 times particle size, and the packing ratio (volume of solid material vs. total volume of sample) between 0.2 and 0.42. Using these parameters in the EES [21] model the produced convective heat transfer coefficient ranges as listed in Table 6.1.

Table 6.1 Modeled Heat Transfer Coefficient by Particle Type

Sand Type	Model Heat Transfer Coefficients (W/m^2-K)	Heat Transfer Coefficient (W/m^2-K)
Olivine	396-886	590
Fine Sifted Silica	331-640	490
Sifted Silica	310-567	410
Construction Sand	240-551	300
Alumina Beads	389-570	125

Of the materials measured, all except the alumina fell within the modeled range. The model accounts for the enhanced conductivity and specific heat of alumina leading to the higher predicted heat transfer coefficient despite the larger particle size. Experimentally, however, the alumina spheres tended to bounce when dropped on the slat surface. This reduced the period of surface contact and may explain the lower measured convective coefficient by altering the contact area assumption. In addition, it should be noted that even at these low speeds, ~0.17 m/s measured by high speed camera, the alumina beads produced very noticeable wear on the heat exchanger surfaces, a problem that must be avoided in service.

The experimental results found in this paper of around 590 W/m^2-K for olivine sand and around 300-500 W/m^2-K for silica sand generally fall within the range predicted by the model. Of the factors that affect the heat transfer, particle size has the dominant effect on the heat transfer coefficient followed by velocity and finally the layer thickness and packing ratio. Another report also looking into a range of TES options by Babcock and Wilcox in 1981 [19], indicated achieving a convection coefficient on their heat transfer elements of around 930-1160 W/m^2-K using a moving bed of sand. The sand utilized in Babcock and Wilcox report [19] had a grain size of 44-77 µm, a packing ratio of 0.40, and a sand velocity around 0.15-0.3 m/s, which is near the range of the olivine sand of the current experiment. While this earlier report suggests that high heat transfer performance can be achieved with very fine sand, no experimental details were given; and our experimental results do project heat transfer coefficients somewhat under those reported in 1981 [19].

CONCLUSION

This experiment, after some refinement, was able to measure the heat transfer coefficient between flowing particulate sand and a generic heat exchanger surface. The resulting coefficients indicate that smaller particle size plays a major role in improving heat exchange. Of particular interest is that ordinary silica sand, in finer grain size, performs about as well as the slightly more exotic olivine sand. It is also notable that the alumina beads, despite the potential advantage of slightly higher specific heat and significantly greater thermal conductivity, have poor performance and high errosiveness. The poor performance is evidently because of the highly elastic rebound after impact with the heated surface and consequent poor maintenance of contact. These factors and overall higher cost hamper its potential as a viable particulate heat storage medium. For the silica sands, a smaller grain size is highly desirable, and continuous flow with good surface contact will be critical for high performance with any sands or particulate mediums.

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