Simulations of heat transfer to solid particles flowing through an array of heated tubes

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Abstract. A novel solar receiver that uses solid particles as a heat transfer fluid is being developed at the National Renewable Energy Laboratory for use in concentrating solar power plants. The prototype considered here is enclosed and contains arrays of hexagonal heat transfer tubes that particles flow between. Discrete element method (DEM) simulations were completed for a laboratory-scale solar receiver for different geometric configurations, hexagon apex angles, particle sizes, and mass flow rates. The heat transfer strongly depends on the particle size, where increased heat transfer is obtained using smaller particles. At higher solids mass flow rates, more particles contact the heat transfer surfaces and the overall heat transfer increases. When a sharper apex angle was used, the particles flow through the receiver at a faster velocity, but the heat transfer decreases because the solids concentration decreases slightly at higher velocities. The DEM simulations show that the heat transfer strongly depends on the solids concentration near the heat transfer surfaces as well as particle size. A new continuum model has recently been developed (A.B. Morris et al., 2015) that accounts for both of these effects, and it was previously tested for simple systems. In the current effort, the continuum model was applied to the complex solar receiver and validated via comparison to DEM data. The results indicate that the new continuum model accurately predicts the local heat transfer coefficient and yields an overall heat transfer coefficient with an average error less than 5%.

Nomenclature

Variables

A	= surface area [m ²]
C_p	= specific heat [J/kg-K]
D_p	= particle diameter [μm]
e	= normal coefficient of restitution
$\mathbf{g}$	= gravitational acceleration [m/s ²]
H	= thermal conductance [W/K]
h	= heat transfer coefficient [W/m ² -K]
H_g	= interphase thermal conductance [W/K]
k_n	= collision spring constant [N/m]
l	= gap distance between the particle surface and the wall [m]
m_p	= particle mass [kg]
P	= pressure [Pa]
$\mathbf{q}$	= heat flux (thermal or granular) [W/m ²]
Q	= heat transfer rate [W]
R	= radius [m]
r	= radial distance [m]
s	= minimum conduction distance [m]
t	= time [s]
T	= thermodynamic temperature [K]
$\mathbf{v}$	= velocity [m/s]

Dimensionless groups

Bi	= Biot number, hD_p/κ_p
Nu	= Nusselt number, hD_p/κ_g
Nu _g	= Nusselt number for interphase transfer
Pr	= Prandtl number
Re	= Reynolds number

α	= Hexagon apex angle
β	= gas-solids drag coefficient
δ	= particle-wall overlap [m]
ε	= volume fraction
ϕ_g	= gas-solids granular energy dissipation term
γ	= dissipation term due to inelastic collisions
κ	= thermal conductivity [W/m-K]
Θ	= granular temperature [m ² /s ²]
ρ	= density [kg/m ³]
τ	= shear stress tensor
μ_g	= gas viscosity [Pa-s]

Subscripts

c	= contact
g	= gas phase
$lens$	= fluid lens
p	= particle
pp	= particle-particle
pfp	= particle-fluid-particle
pw	= particle-wall
pfw	= particle-fluid-wall
ref	= reference value
s	= solids phase
w	= wall
Θ	= granular temperature

1. Introduction

Concentrating solar power (CSP) plants operate by collecting heat from concentrated solar energy and then converting that thermal energy to electricity via a turbine and electrical generator (Ho and Iverson, 2014). This process is similar to conventional power plants where instead of combusting fossil fuels or using nuclear fuel rods, many mirrors covering a large area focus solar energy onto a smaller-area solar receiver. For large-scale power plants capable of producing hundreds of megawatts of electrical power, a

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4 power tower design for the solar receiver is standard. For this receiver, a field of mirrors concentrate
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6 solar radiation onto a central tower that contains a heat transfer fluid (HTF). The HTF can act directly as
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8 a working fluid to turn a turbine, i.e., steam, or indirectly to heat a working fluid.
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11 The HTF is a critical component for the thermal and economic performance of a CSP facility
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13 (Vignarooban et al., 2015). In general, desirable characteristics of the HTF include: ability to operate at
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15 high temperatures ($>650^{\circ}\text{C}$), ability to flow at low temperatures, high heat capacitance for thermal
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17 storage, efficient heat transfer, low cost, and low corrosion with metal alloys containing the HTF. The
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19 ability to operate at high temperatures is critical because the thermodynamic efficiency can be increased
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21 by raising the temperature of the working fluid that enters the turbine. The ability for thermal storage
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23 significantly reduces costs associated with transient operation due to diurnal cycling or cloud cover. Solid
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25 particles (e.g., sand) are a candidate HTF that satisfies the desired characteristics and solar receivers that
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27 utilize solid particles as a HTF are under development at the National Renewable Energy Laboratory
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29 (NREL) and Sandia (Ho et al., 2009; Martinek and Ma, 2015; Siegel et al., 2010). Solid particles can
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31 improve upon existing molten salt technologies because solid particles such as sand are inert at high target
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33 temperatures (900°C) and are inexpensive. Similarly, solid particles can improve upon HTF's such as
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35 steam or oil because solid particles have high heat capacity and perform well as a thermal storage media.
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37 There are many challenges associated with particle receivers, such as abrasion of heat transfer surfaces or
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39 attrition of particles, but the scope of this work is limited to heat transfer and hydrodynamic performance.
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45 The prototype, solid-particle receiver designed at NREL, shown schematically in Figure 1, is a power
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47 tower that contains many hollow heat transfer tubes. The concentrated solar radiation enters the end of
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49 each tube and is internally absorbed with near black-body efficiency. Cold particles are lifted via a
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51 bucket elevator and poured over the heat transfer tubes. These particles are heated as they fall over the
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53 tubes and eventually stored in an insulated hot silo. When needed, the particles from the hot silo are
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55 transferred to a fluidized bed heat exchanger and the energy is transferred to the working fluid for power
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57 generation.
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Figure 1. A schematic showing the enclosed particle receiver with hexagonal heat transfer tubes.

Heat transfer models and Nusselt number correlations exist for traditional liquids or gases flowing through a solar receiver, but such relations are not readily available for solid particle flows. Particle flows also exhibit unique behavior not observed in molecular fluids, such as clustering instabilities that arise due to inelastic (or dissipative) collisions (Goldhirsch, 2003). In addition, the discrete nature of particles can give rise to phenomena contradictory to that of molecular fluids. For example, although vibrating or agitating a molecular fluid can enhance heat transfer by increased mixing, such agitation can have the opposite effect on particle flows (Al-Ansary et al., 2012). Such behavior makes it difficult to extrapolate heat transfer models for solids flow in simple geometries to flow through a complicated solar receiver.

Quantities such as the local heat transfer coefficient and particle temperature fields are desired for improving the receiver design, but such data are difficult to obtain non-intrusively via experiments. Simulations will thus play a key role in the design of the solar receiver. Two common methods for simulating a granular flow are the discrete element method (DEM) (Cundall and Strack, 1979) and a two-fluid model (TFM) (Gidaspow, 1884; Jackson, 2010; Pannala et al., 2010). The DEM is a Lagrangian method that is highly accurate and solves force and energy balances for every particle. The TFM represents the solid phase as a continuum of particles (analogous to the Navier-Stokes equations for Newtonian fluids) and is described in terms of macroscopic variables such as the solids concentration and bulk velocity. Both two-fluid and DEM simulations treat the gas phase as a continuum, and interphase

coupling terms for momentum and energy exchange are included. Although DEM simulations require less closures and are thus more accurate than their continuum counterparts, the use of DEM for industrial-scale systems is limited by computational costs. Continuum models are more computationally efficient and can be used for an industrial-scale solar receiver. Nonetheless, the closures for particle-wall heat transfer required for continuum models (Kuipers et al., 1992; Molerus, 1997; Schlünder, 1984; Tsotsas, 2010) have not been validated for complex flows such as that through a solar receiver. In the current effort, DEM simulations are performed for a laboratory scale solar receiver. The dominant conductive heat transfer mechanisms are identified and the impact of design alternatives on the heat transfer are evaluated. Radiative heat transfer is expected to be an important mechanism. These simulations, however, neglect radiation because the aim of this work is to isolate and test the validity of the conductive heat transfer models. In the future, validation should include radiative heat transfer and comparisons should be made to experimental data.

2. Methods

2.1. Heat Transfer Mechanisms in Prototype Receiver.

The solar receiver transfers energy to the falling particles by conduction from the tube walls to contacting particles, interphase coupling with heated background gas, and radiative heat transfer. Conduction can occur directly across the contact area between touching bodies (Batchelor and O'Brien, 1977; Yovanovich, 1967) and is commonly referred to as particle-particle or particle-wall conduction. Conductive heat transfer can also occur through the thin gap of interstitial gas between two bodies that are in contact or nearly in contact, a mechanism referred to as particle-fluid-particle or particle-fluid-wall conduction (Tsotsas, 2010). At high temperatures expected for normal operation of the receiver, radiation is expected to significantly affect the heat transfer. However, radiation is neglected in this work because the first aim is to assess the validity of continuum models for conduction and convection, which are also expected to play a role at desired receiver temperatures.

Conductive heat transfer to granular media is controlled by the contact resistance between particles and a heated surface as well as the penetration of heat into the bed. The contact resistance arises through particle-wall and particle-fluid-wall heat transfer between the wall and the first layer of particles adjacent to the wall. The contact resistance with the wall depends on the particle size as well as the orientation (or arrangement) of particles near the wall (A.B. Morris et al., 2015; Van Antwerpen et al., 2010). For dense flows with a high solids concentration near a wall, many particles contact the wall and the contact resistance is less than that of moderately dense or dilute particle flows. The penetration resistance controls how heat is transferred into the bulk of the granular media and depends on the effective thermal conductivity of the bed, convective motion of particles, and interphase coupling with the background gas. For stagnant or very slow moving particle beds, the penetration resistance dominates and the heat transfer is limited by how quickly heat can be absorbed by the bulk of the granular media (Tsotsas, 2010). In the falling-particle solar receiver, convective mixing reduces the penetration resistance and the heat transfer is dominated by the contact resistance between particles and the hexagonal tubes.

2.2. DEM Model

The discrete element method simulates a gas-solid flow by solving force and energy balances for all particles while the gas phase is modeled via the Navier-Stokes equations. The equations are for the particle and gas phases are summarized in Table 1. The simulated are performed using 2013 release of Multiphase Flow with Interphase Exchange (MFIx) (Syamlal et al., 1993), which is an open source simulation tool developed at the National Energy Technology Laboratory. The particle-wall and particle-fluid-wall heat transfer models will be available in a future release. The discrete element solver in MFIx (Garg et al., 2012; Li et al., 2012) using ‘soft sphere’ contact models – the deformation throughout a collision is resolved as opposed to the collision being modeled as an instantaneous event. The contact force between touching bodies is modeled as a spring force that is linearly proportional to the overlap between two bodies. A dashpot term that is proportional to the relative velocity of colliding bodies is also

included to allow for inelastic collisions. The gas phase is solved using finite volumes while the particle phase is modeled in the Lagrangian framework.

Table 1. Summary of DEM equations

Gas-Phase (Continuum) Equations	
Continuity	$\frac{\partial}{\partial t}(\varepsilon_g \rho_g) + \vec{v}_g \cdot \nabla(\varepsilon_g \rho_g) = 0$
Momentum	$\frac{\partial}{\partial t}(\varepsilon_g \rho_g \vec{v}_g) + \nabla \cdot (\varepsilon_g \rho_g \vec{v}_g \vec{v}_g) = -\varepsilon_g \nabla P_g + \nabla \cdot \bar{\bar{\tau}}_g - \beta_{g,s}(\vec{v}_s - \vec{v}_g) + \varepsilon_g \rho_g \vec{g}$
Energy	$\frac{\partial}{\partial t}(\varepsilon_g \rho_g C_{p,g} T_g) + \vec{v}_g \cdot \nabla(\varepsilon_g \rho_g C_{p,g} T_g) = -\nabla \cdot \vec{q}_g - H_g(T_g - T_s)$
Constitutive relations for gas phase	
Interphase Heat Transfer (Gunn, 1978)	$H_g = \frac{6\kappa_g \varepsilon_s Nu_{gs}}{D_p^2}$ $Nu_{gs} = (7 - 10\varepsilon_g + 5\varepsilon_g^2)(1 + 0.7 Re^{0.2} Pr^{1/3}) + (1.33 - 2.4\varepsilon_g + 1.2\varepsilon_g^2) Re^{0.7} Pr^{1/3}$
Gas Conduction	$\vec{q}_g = -\varepsilon_g \kappa_g \nabla T_g$
Interphase drag	$\beta_{g-s} = \begin{cases} \frac{3}{4} C_d \frac{\rho_g \varepsilon_g \varepsilon_s \vec{v}_g - \vec{v}_s }{D_p} \varepsilon_g^{-2.65} & , \varepsilon_g \geq 0.8 \\ \frac{150 \varepsilon_s (1 - \varepsilon_g) \mu_g}{\varepsilon_g D_p^2} + \frac{1.75 \rho_g \varepsilon_s \vec{v}_g - \vec{v}_s }{D_p} & , \varepsilon_g < 0.8 \end{cases}$ $C_d = \begin{cases} \frac{24}{Re} (1 + 0.15 Re^{0.687}) & , Re < 1000 \\ 0.44 & , Re \geq 1000 \end{cases}$ $Re = \frac{\rho_g \varepsilon_g \vec{v}_g - \vec{v}_s D_p}{\mu_g}$
Solids-Phase DEM Equations	
Momentum	$m_{p,i} \frac{d\vec{v}_{p,i}}{dt} = \sum_j \vec{F}_{c,ij} + \vec{I}_{g,s} - m_{p,i} \vec{g}$
Thermal Energy	$m_i C_{p,i} \frac{dT_i}{dt} = \sum_j (\dot{Q}_{ij}^{pp} + \dot{Q}_{ij}^{pfp}) + H_g(T_g - T_i) + \dot{Q}_{pw} + \dot{Q}_{pfw}$
Constitutive relations for solids phase	
Contact model	$F_{c,ij} = -k_n \delta - \eta_n \frac{d\delta}{dt}$

Particle-particle
conduction
(Batchelor and
O'Brien, 1977)

$$\dot{Q}_{pp}^{ij} = H(T_p^j - T_p^i) = 2\kappa_p R_c (T_p^j - T_p^i)$$

Particle-fluid-
particle conduction
(Rong and Horio,
1999)

$$\dot{Q}_{pfp}^{ij} = H_{pfp} (T_p^j - T_p^i) = (T_p^j - T_p^i) \int_{r_{in}}^{r_{out}} \frac{2\pi\kappa_g}{\max(s, l^{ij})} dr$$

$$l^{ij} = 2 \left[R_p - \delta - \sqrt{R_p^2 - r^2} \right]$$

Particle-wall
conduction

$$\dot{Q}_{pw} = H_{pw} (T_w - T_p) = \frac{4\kappa_p \kappa_w}{\kappa_p + \kappa_w} R_c (T_w - T_p)$$

Particle-fluid-wall
conduction

$$\dot{Q}_{pfw} = H_{pfw} (T_w - T_p) = (T_w - T_p) \int_{r_{in}}^{r_{out}} \frac{2\pi\kappa_g r}{\max(l, s)} dr$$

$$l = R_p - \delta - \sqrt{R_p^2 - r^2}$$

MFIX uses a structured grid with quadrilateral cells for the gas phase, but the geometry of the solar receiver is complex and contains many hexagonal tubes. To simulate the complicated geometry, a cut-cell method is used to represent the hexagonal tube boundaries (Dietiker et al., 2013). The hexagon boundaries are first defined by a set of facets and boundary surface normal vectors. Next, a pre-processing routine intersects the tube geometries with the Cartesian grid. Cells that intersect the tube boundaries are truncated and the cell volumes and face areas are updated accordingly. Particle-wall detection routines are implemented to check for contacts that particles may have with the hexagonal tubes. If a particle collides with a tube boundary, a repulsive linear spring dashpot (LSD) collision force is applied and particle-wall as well as particle-fluid-wall heat transfer terms are included in the particle energy balance. The LSD spring constant is chosen to be identical to particle-particle contacts.

Particle-particle and particle-wall conduction are modeled in DEM according to (Batchelor and O'Brien, 1977; Yovanovich, 1967). This contact conduction model has been implemented in other DEM solvers and used to study heat transfer in rotary kilns (Figueroa et al., 2010; Shi et al., 2008). The particle-particle (wall) heat transfer is proportional to the contact radius between two touching bodies as

well as the duration of contact. The contact radius and duration depend on model parameters for the linear spring dashpot (LSD) contact model. For example, a softer material (or spring constant) will have greater deformation and collision times compared to a harder material. Correction terms have been developed to capture the true deformation and collision times for realistic material properties (A. B. Morris et al., 2015). Such terms remove the dependence of particle-particle (or wall) heat transfer on the numerical parameters for the contact model and are used in this work.

Particle-fluid-particle and particle-fluid-wall conduction are modeled using the approach of Rong and Horio (Rong and Horio, 1999). This approach assumes that each particle is surrounded by a lens of stagnant interstitial gas, a value that selected as 1.4 times the particle radius in this work. The thermal conductance increases as the gap distance between the bodies decreases. At contact, the thermal conductance becomes singular because the gap distance is zero. Physically, this singularity does not occur because the particle has microscopic roughness that prevents the gap distance from becoming singular. The particles are assumed to be very smooth and the microscopic surface roughness height is approximated as 2.75×10^{-8} m in this work. For atomically smooth particles, the minimum conduction distance is limited by the mean free path of gas molecules and the gas thermal accommodation coefficient (Molerus, 1997). The selected value of 2.75×10^{-8} m is also comparable to the gas mean free path and standard conditions and is therefore a limiting value for smooth particles.

The significance of particle-wall heat transfer relative to particle-fluid-wall conduction depends on the ratio $\kappa_p R_c / \kappa_g R_p$. When that ratio is much less than one, particle-fluid-particle (or wall) is the dominant heat transfer mechanism (Vargas and McCarthy, 2001). For sand, a material expected to be used in the solar receiver, the contact radius between touching bodies is significantly less than the particle radius. Our simulations show particle-fluid-wall heat transfer is approximately two orders of magnitude larger than particle-wall conduction.

2.3. Continuum Model.

In a continuum, or two-fluid, model, the solids and gas phases are modeled as interpenetrating continua and are described using macroscopic, or averaged, variables such as solids concentration, bulk velocity, thermal temperature (gas and solids phases), and granular temperature (solids phase). The granular temperature is analogous to the temperature of a dilute gas, and is a measure of average energy associated with velocity fluctuations about the local mean velocity. The granular temperature (or energy) balance is solved for the solids phase, and includes sink terms due to the dissipative nature of inelastic particle contacts. The continuum equations for the solids phase are summarized in Table 2 and the closures can be found in (Garzó and Dufty, 1999).

Table 2. Summary of continuum governing equations for the solids phase

Gas-Phase (Continuum) Equations	
Continuity	$\frac{\partial}{\partial t}(\varepsilon_s \rho_s) + \vec{v}_s \cdot \nabla(\varepsilon_s \rho_s) = 0$
Momentum	$\frac{\partial}{\partial t}(\varepsilon_s \rho_s \vec{v}_s) + \nabla \cdot (\varepsilon_s \rho_s \vec{v}_s \vec{v}_s) = -\varepsilon_s \nabla P_g + \nabla \cdot \bar{\bar{\tau}}_s + \beta_{g,s}(\vec{v}_s - \vec{v}_g) + \varepsilon_s \rho_s \vec{g}$
Energy	$\frac{\partial}{\partial t}(\varepsilon_s \rho_s C_{p,s} T_s) + \vec{v}_s \cdot \nabla(\varepsilon_s \rho_s C_{p,s} T_s) = -\nabla \cdot \vec{q}_s + H_g(T_g - T_s)$
Granular Energy	$\frac{3}{2} \left[\frac{\partial}{\partial t}(\varepsilon_s \rho_s \Theta) + \nabla \cdot (\varepsilon_s \rho_s \Theta \vec{v}_s) \right] = \left(\bar{\bar{\tau}}_s + P_s \bar{\bar{I}} \right) : \nabla \vec{v}_s - \nabla \cdot \vec{q}_\Theta - \gamma_\Theta + \phi_g$

Nusselt number expressions for the thermal contact resistance have been developed previously for closely packed beds (Molerus, 1997) and are not valid for systems where the solids concentration near the surface is dilute or moderately dense. The Nusselt number is defined as $Nu = hD_p / \kappa_g$, where h is the conductive heat transfer coefficient defined as the heat flux divided by the temperature difference between the wall and first layer of particles adjacent to the wall. For dynamic systems, the contact resistance strongly depends on how particles are arranged near the wall, i.e., the number of particles in contact with the wall and the proximity of others that are very close to the wall. Previous work shows that for dynamic systems the particle packing ‘loosens’, the solids concentration decreases, and the contact resistance increases significantly, and such effects must be accounted for when modeling dynamic systems (A.B. Morris et al., 2015). Morris et al. developed a new continuum model that accounts for

arrangement effects, and it was found to agree with DEM data for particles flowing down an inclined ramp. The arrangement of particles near the wall is modeled via a particle-wall distribution function, which is analogous to the classical radial distribution function (Lebowitz, 1964). The particle-wall distribution function describes the probability of a particle existing within a given distance from the wall. Similar to the radial distribution function, the particle-wall distribution function was measured via DEM simulations and was found to be reasonably expressed as only a function of solids concentration near the wall. Unlike past works, the new model thus incorporates the effects of solids concentration via the particle-wall distribution function into a Nusselt number correlation. The new continuum model was tested for simple systems for particle flow down an inclined ramp and closely matched DEM data for dilute to dense flow regimes. Further details of the continuum model can be found in (A.B. Morris et al., 2015), and the Nusselt number correlation is

$$\frac{\text{Nu}}{\text{Nu}_{\max}} = c_0 + c_1\epsilon_s + c_2\epsilon_s^2 + c_3\epsilon_s^3 + c_4\epsilon_s^4 + c_5\epsilon_s^5 + c_6\epsilon_s^6 + c_7\epsilon_s^7$$

$$\begin{aligned} c_0 &= 1.0838 \times 10^{-3} & c_1 &= -2.1709 \times 10^{-2} \\ c_2 &= 2.4268 \times 10^0 & c_3 &= 1.9101 \times 10^0 \\ c_4 &= -1.2243 \times 10^2 & c_5 &= 6.1504 \times 10^2 \\ c_6 &= -1.1706 \times 10^3 & c_7 &= 7.9093 \times 10^2 \end{aligned} \quad (1)$$

The maximum Nusselt number depends on particle size and morphology and is

$$\text{Nu}_{\max} = \frac{\pi}{2\hat{s}} \left[1 - (1 - \hat{s})^2 \right] + \pi \left[\sqrt{2 - \hat{R}_{\text{lens}}^2} + \hat{s} - 1 + \ln \left(\frac{1 - \sqrt{2 - \hat{R}_{\text{lens}}^2}}{\hat{s}} \right) \right], \quad (2)$$

where s is the average microscopic roughness height and R_{lens} is the radius of the fluid lens surrounding each particle. The fluid lens is a model parameter that quantifies the area through which conduction through the gas can occur, and the value used in this work is 1.40 times the particle radius. A similar value of 1.41 is used in (Vargas and McCarthy, 2002) for rotating drum simulations. The hats indicate

lengths that are non-dimensionalized by the particle radius. The new continuum model has been incorporated into MFIX and will be available in a future release.

3. System Description

Simulations were completed for a laboratory hexagonal tube array with 12 full hexagonal tubes and 4 half-width tubes, as depicted in Figure 1. Particles and entrained gas enter the computational domain from the top boundary with prescribed flow rates. The hexagonal tubes were fixed at 600 K and a no-slip boundary condition was used for the gas phase. The material properties for the solids and gas phases are summarized in Table 3. The baseline hexagon geometry and configuration are summarized in Table 4. The baseline simulation parameters are summarized in Table 5. For all simulations, the particles and gas enter the domain at 300 K and the hexagonal tubes are kept at 600 K. Domain boundaries are adiabatic in all of the simulations.

Table 3. Material properties for the gas and solids phases

Gas Phase – Air			
Thermal conductivity	$\kappa_g = \kappa_{g,ref} \sqrt{T_g / T_{g,ref}}$, $\kappa_{g,ref} = 0.0252$ W/m-K, $T_{g,ref} = 300$ K		
Viscosity	$\mu_g = \mu_{g,ref} \left(\frac{T_g}{T_{g,ref}} \right)^{3/2} \frac{T_{g,ref} + S}{T + S}$ $S = 110.4$ K, $\mu_{g,ref} = 1.72 \times 10^{-5}$ Pa-s, $T_{g,ref} = 300$ K		
Specific heat	1004.2 J/kg-K		
Molar Mass	28.97 kg/kmol		
Solids phase – SiO ₂			
Thermal conductivity	1.402 W/m ² -K	Young's Modulus	70 GPa
Specific heat	794.96 J/kg-K	Poisson ratio	0.17
Friction coefficient	0.0	Restitution coefficient	0.9

Table 4. Baseline receiver geometry

Hexagon Width, W	3.78 cm	Center to center spacing	
Hexagon Height, H	6.42 cm	Horizontal direction	5.04 cm
Apex angle, α	89°	Vertical direction	4.98 cm

Table 5. Simulation parameters for base case configuration

Domain size [cm]	$17.64 \times 23.4 \times 1.5$	LSD Spring constant	100 N/m
Number of cells	$294 \times 390 \times 24$	$\varepsilon_{s,inlet}$	10%
$v_{g,inlet}$	0.1 m/s	$v_{s,inlet}$	1 m/s
Particle diameter	300 μm		

The heat transfer strongly depends on the solids concentration and particle arrangement. To accurately model the packing configuration, the simulations were spatially 3-D because a 3-D packing can be substantially different from 2-D packing. Hydrodynamic instabilities can develop as the particles flow through the receiver. Such instabilities are inherently 3-dimensional and further necessitate spatially 3-D simulations. Initial simulations of the solar receiver were performed in 2-D and led to very different hydrodynamic behavior compared to 3-D simulations. Accordingly, 3-D simulations were used for all of the results presented below.

4. Results and Discussion

4.1. DEM Simulations

Figure 2 shows 300 μm particles flowing through the solar receiver with the baseline conditions. Although radiative heat transfer is an important mechanism, it is neglected because the aim of this work is to validate continuum models for only conduction and convection via comparison to DEM data. Such an approach can test the validity of individual physical models, but in the future, a model that includes radiative heat transfer should be validated via comparisons to experimental data. Individual particles are colored by temperature T_p . Particles that contact the hexagons are warmed while other particles within the central core remain cold through the first several rows of hexagons. The particles adjacent to the wall are warmed by approximately 15 K and the temperature difference between the wall and particles remains

large, approximately 285 K. This observation shows that the heat transfer to the particles is not limited by heat transfer into the bulk of the granular media, because penetration resistance dictates the heat transfer only when the particles adjacent to the wall approach the wall temperature. The flow is fairly steady through the first two rows of hexagons and slight instabilities begin to develop in the third row. In the fourth row of hexagons, much stronger instabilities develop. These instabilities cause increased particle mixing and more uniformly distribute the heat amongst the solids phase. Within the top three rows, particles do not contact the side or bottom faces of each hexagon. In the fourth row, the instabilities cause particles to occasionally contact the side faces. This slightly increases the heat transfer, as will be discussed later.

Figure 2. Solid particles colored by temperature flowing through the solar receiver.

Experiments performed at The Ohio State University also show hydrodynamic instabilities that develop as particles flow through a laboratory-scale receiver, as illustrated in Figure 3 where particles are imaged as they flow through the 3rd row of the receiver (Wang et al., 2013). Similar to our simulations, the particle flow in the first several rows of hexagons is laminar (not shown). Instabilities develop in experimental as the particles flow through the 2nd and 3rd rows of the receiver while our simulations show instabilities forming in the 4th row. The oscillations result in particles occasionally not maintaining

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4 thermal contact with hexagon surfaces, as seen on the top faces of the lower-right hexagon in Figure 3.
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6 The onset of hydrodynamic instability in the simulations is not expected to match experiments because
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8 the receiver geometry, particle properties, are not identical. The simulations do, however, yield the same
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10 qualitative behavior that shows the flow transition from a laminar to turbulent behavior as the particles
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12 fall through the receiver.
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28 **Figure 3.** Image showing hydrodynamic instabilities that develop in the 3rd row of the receiver during
29 experiments performed at The Ohio State University (Wang et al., 2013).
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31
32 To evaluate the effects that tube arrangement, the horizontal and vertical spacing between tubes were
33 systematically varied. Contours of solids concentration are shown for the base configuration with a
34 horizontal spacing of 5.04 cm and a narrow spacing of 4.44 cm in Figure 4. When the horizontal spacing
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36 is small, the flow chokes as it enters the horizontal gap and particles begin to accumulate along the top
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38 surface of the hexagon. This accumulation of particles along the top surface causes a significant increase
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40 of the overall heat transfer. The solids concentration profiles also show that particles contact the side
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42 walls of the hexagons when the gap distance is small. Although particles contact these surfaces with an
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44 approximate concentration of 0.25, the gain in heat transfer along the tube side walls is marginal, as
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46 detailed below. Very little heat transfer is observed along the bottom faces of each hexagon.
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Figure 4. DEM contours of solids fraction when the center-to-center horizontal spacing is 5.04 cm (left) and 4.44 cm (right)

The time-averaged local particle-fluid-wall heat transfer coefficient profiles for solids-phase conduction are shown in Figure 5 for a hexagon in the 2nd row of arrays with different horizontal spacing. The local heat transfer coefficient is defined as the particle-fluid-wall heat flux divided by the temperature difference between the wall and the first layer of particles adjacent to the wall. The integrated heat transfer coefficient for the entire system is largest for the smallest simulated horizontal spacing (4.44 cm) and has a value of 584.16 W/m²-K compared to the base case value of 487.41 W/m²-K when the spacing is 5.04 cm. The integrated heat transfer coefficient is defined as the total heat transfer rate divided by the total surface area of all tubes and the temperature difference between the tube wall and inlet solids temperature. The local heat transfer coefficient profiles show that for the small horizontal spacing, regions of increased heat transfer (relative to the base case) exist where the flow separates from the top face of the hexagon ($\theta \sim 55^\circ$). This increase occurs because the solids flow has a denser concentration when it funnels into the narrower horizontal gap. In addition, some particles contact the side walls ($\theta \sim 100^\circ$) when the horizontal gap distance is narrow. This results in a small contribution to local heat transfer coefficient (only particle-fluid-wall conduction) along the side wall (approximately 100W/m²-K) where other simulations with a larger horizontal spacing show no conductive heat transfer along the side walls). These small additions in the 2nd row of hexagons do not, however, account for the overall increase

of the total heat transfer coefficient. The overall increase occurs primarily because particles fill the troughs between hexagons in the top row, increasing the heat transfer along the top surfaces. The spike in heat transfer near 180° occurs because particle streams collide with each other and cause particles to splash and contact the bottom surfaces of the hexagon.

Figure 5. DEM predictions of the local heat transfer coefficient for particle-fluid-wall heat conduction (h_{pfw}) for a hexagon in the 2nd row when different horizontal spacings are simulated.

The local heat transfer coefficient was not sensitive to the vertical spacing over the simulated parameter space. The integrated heat transfer coefficient over all tube surfaces changed by less than 4.5% when the vertical tube spacing was changed from 4.5 cm to 5.52 cm. Solids concentration fields for simulations with the minimum and maximum vertical spacing are shown in Figure 6. Although the vertical spacing is changed drastically, there is not a significant impact on the heat transfer because the side and bottom walls remain relatively void of particles. The local heat transfer coefficients along the top surface also remain the same. The only significant difference in the local heat transfer coefficient occurs very near the bottom of the tube. For a small vertical spacing, some particles splash before entering the vertical channel and impact the bottom face of the higher row of hexagons. For a large vertical spacing, a similar splash of particles is observed but it does not ever come into contact with the hexagon in the higher row.

Figure 6. Contours of solids concentration when different vertical spacings are simulated.

The tube shape is altered by changing the apex angles and heights of the tubes while keeping the hexagon width constant. The apex angle has a significant impact on the hydrodynamics as well as heat transfer in the receiver. Figure 7 shows the vertical solids velocity for blunt and pointed hexagons. As the hexagons become more pointed, the vertical velocity, as expected, increases because there is less resistance to the flow. For example, the solids velocity asymptotes to approximately 135 cm/s in the horizontal gaps when the apex angle is 89° , while the solids velocity is approximately 180 cm/s when the apex angle is 60° . At higher speeds, there are also stronger instabilities that create contact with the side walls, as seen along the 3rd and 4th rows. For traditional fluids, one would expect the heat transfer coefficient to increase with increased instabilities and higher speeds (i.e., larger Reynolds numbers). The opposite happens for granular flows because the instabilities and higher speeds agitate the flow and loosen the packing of particles (lower solids concentration) near the surface. This behavior is quantified via the total heat transfer coefficient, defined as $\bar{h} = \dot{Q}_{tot} / A_{tot} (T_{wall} - T_{s,inlet})$, which decreases from 487 W/m²-K to 344 W/m²-K as the apex angle decreases from 89° to 60° . It is also anticipated that the thermal performance will also benefit from radiative heat transfer at slower particle velocities due to increased receiver residence times.

Figure 7. Contours of solids vertical velocity when the apex angle is 89° (left) and 60° (right).

The time-averaged local heat transfer coefficient along the surface of a hexagon in the 3rd row of the receiver is shown in Figure 8 for cases with different apex angles. The blunter apex angle shows larger values for the heat transfer coefficient than the more pointed hexagons. The heat transfer decreases because the solids concentration near the surface is smaller when the particles move at a higher velocity (i.e., more pointed hexagons). In addition, particle “splash” occurs when a blunter hexagon is used, thus creating thermal contact with the bottom of the hexagon. This is shown by the relatively large heat transfer near angles of approximately 180 degrees for the blunt hexagon case.

Figure 8. Local heat transfer coefficient for particle-fluid-wall conduction for hexagons in the 3rd row with different apex angles.

The decrease in heat transfer as the solids velocity increases is opposite for solid particles as for molecular fluids. At higher velocities, or Péclet numbers, the heat transfer to a molecular fluid generally increases because of higher temperature gradients near the wall and a thinner thermal boundary layer. The key distinction between molecular fluids and solid particle flows is that the contact resistance is negligible for molecular fluids and the temperature slip (i.e. difference between the fluid temperature evaluated at the wall and the wall temperature) is often negligible. The heat transfer for molecular fluids is thus limited by how quickly the energy can be absorbed into the bulk of the media. For solid particle flows in the solar receiver, the contact resistance is the controlling mechanism for heat transfer. At higher velocities, the particles become more agitated and do not maintain contact with the wall as much as what would occur for slower particle speeds.

The inlet mass flux of particles is changed by altering the solids concentration at the inlet. The heat transfer tends to increase, as expected, for increasing inlet solids mass flux. The local heat transfer coefficient is shown in Figure 9 for a hexagon in the top and 3rd rows of the array. The solids concentration near the surface is larger when the mass flux is increased, thus leading to increased heat

transfer. When the inlet solids mass flux is decreased by 50%, the total heat transfer coefficient decreases by approximately 35%. The flow, however, was not choked at the base-case mass flux was 258.2 kg/m²s. Therefore, further improvements to the overall heat transfer is expected by using an even larger inlet mass flux. At too high of a mass flux, particle friction may cause jamming to occur which could be detrimental to the solar receiver. Such effects were not modeled in this work.

Figure 9. Local heat transfer coefficients for hexagons in the top (left) and 3rd (right) rows when the inlet mass flux is 258 kg/m²s (base case) and 129 kg/m²s.

Of all parameters studied, the particle size has the most dramatic impact on the heat transfer coefficient. The new continuum model, shown in Eqns. 1 and 2, predicts that the heat transfer is approximately inversely proportional to particle size. This is only approximate because changing the particle size can affect other model parameters such as the roughness-asperity-height-to-particle-diameter ratio and hydrodynamic variables such as the solids concentration. Simulations were performed for 200, 250, 300 (base case), and 350 μ m particles. As the particle size decreases, the DEM computational time increases because there are larger numbers of particles in the system and the collision time (and thus computational time step) are smaller. Approximately 5 million particles were used for the 300 μ m simulations, and nearly 13.5 million particles were used for the 200 μ m simulations.

The overall heat transfer coefficient increases from 487 W/m²-K to 780 W/m²-K when the particle size is decreased from 300 μ m to 200 μ m. The local heat transfer for a hexagon in the 2nd row of the receiver

is shown in Figure 10 for 200 and 300 μm particles. The heat transfer increases because smaller particles have increased surface-area-to-volume ratio for thermal contact (particle-wall and particle-fluid-wall) with the surface. In addition, the solids concentration is slightly larger for small particles because the granular energy dissipation rate is inversely proportional to particle size. The granular energy is characterized by a granular temperature which describes the energy associated with random particle velocities and is analogous to an ideal gas temperature. The solids concentration is inversely proportional to the granular temperature and increases as the granular temperature decreases.

Figure 10. Local heat transfer coefficient for particle-fluid-wall conduction for a hexagon in the 2nd row when 200 and 300 μm particles are simulated.

4.2. Continuum Model Validation

In addition to assessing the impact of various design parameters on the heat transfer in a solar receiver, another aim of this work is to validate a new continuum heat transfer model which does not contain tunable parameters and was previously found to accurately predict the heat transfer compared to DEM data of particle flow down a ramp (A.B. Morris et al., 2015). Since heat transfer is tightly coupled to hydrodynamics, here we eliminate the effects of hydrodynamic variations between the continuum and DEM models by using the DEM data for hydrodynamic variables (particle concentration and velocity) as input parameters for the continuum model. By doing so, any discrepancies arising between the continuum predictions and DEM data can be attributed solely to the (continuum) heat transfer model.

As previously discussed, the particle size was found to have the most significant impact on the heat transfer coefficient. The local heat transfer coefficients predicted by the new continuum model (Eqns. 1 and 2) are compared to DEM for different particle sizes in Figure 11. The new continuum model very closely matches the DEM data for the local heat transfer coefficient for different tubes within the receiver. Although not shown, similar agreement for the local heat transfer coefficient is obtained for the other portions of the examined parameter space.

Figure 11. Local predictions from the continuum model (A.B. Morris et al., 2015) are compared to DEM data for particles of different sizes flowing over hexagons in the top (left) and 2nd (right) rows of the receiver.

It is worthwhile to note that although the results shown here indicate that the continuum heat transfer model is very accurate in its predictions of the local heat transfer coefficient, the level of agreement is very sensitive to solids concentration near the surface. In other words, accurate prediction of the solids concentration by the continuum model is critical for maintaining the accuracy of heat transfer predictions due to the tight coupling between hydrodynamics and heat transfer. Although previous continuum models for particle-fluid-wall contact conduction depend on the particle size (Molerus, 1997), such models are derived for packed bed regimes and fail in regions where the flow is not quite packed but still moderately dense, as demonstrated by (A.B. Morris et al., 2015) for flow down a ramp.

The total heat transfer coefficient, defined as $h = \dot{Q}_{tot} / A_{hexes} (T_{wall} - T_{s,inlet})$, is computed using the new continuum model and compared to DEM data. The total heat transfer coefficient for hexagonal

arrays with different horizontal and vertical center-to-center spacing are shown in Figure 12. The DEM simulations are in excellent agreement for the larger horizontal spacing and have an error of 13.3% for the smaller separation. The larger discrepancy for a small spacing occurs because the continuum model overpredicts heat transfer along the side walls. Even though the solids concentration near the side walls is large for a small horizontal gap, as indicated in Figure 4, many of the particles are not physically in contact with the side wall. The agreement between DEM and continuum simulations for changes in vertical spacing are excellent.

Figure 12. Integrated heat transfer coefficient predictions from the continuum model are compared to DEM data for different horizontal and vertical tube spacings. Unless explicitly indicated, the model parameters are the same as for the base case configuration (Table 5).

The total heat transfer coefficients predicted by the new model and DEM data are plotted for different apex angles and particle diameters in Figure 13. Both DEM and continuum predictions show that the heat transfer coefficient decreases with decreasing (or more pointed) apex angles. The DEM and continuum models match closely with errors ranging from 1.5% to 12%. The new continuum model also accurately captures particle size effects and closely matches DEM data. The heat transfer is most sensitive to particle diameter. As previously discussed, this indicates that the heat transfer in the solar receiver is limited by particle-fluid-wall contact conduction to the first layer of particles in contact with a wall.

Figure 13. Integrated heat transfer coefficient predictions from the continuum model are compared to DEM data for different apex angles and particle diameters. Unless explicitly indicated, the model parameters are the same as for the base case configuration (Table 5).

The new continuum model matches the DEM data for both the low and high mass flux cases, as shown in Figure 14. The heat transfer coefficient decreases by approximately 50% when the inlet mass flux is reduced from the base case value of 258 kg/m²·s to 129 kg/m²·s. As the inlet mass flux increases from the base case value to 323 kg/m²·s, the DEM data show the heat transfer coefficient increases slightly due to increased solids concentration near the surface. The increased discrepancy that occurs at higher mass fluxes may occur because the solids concentration near the surface is larger for a high flux and the heat transfer is increasingly sensitive to concentration near the packed limit.

Figure 14. Integrated heat transfer coefficient predictions from the continuum model are compared to DEM data for different inlet solids mass flux. Unless explicitly indicated, the model parameters are the same as for the base case configuration (Table 5).

5. Conclusions

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4 Simulations that neglect radiative heat transfer were performed for particles flowing through a solar
5 receiver with hexagonal heat transfer tubes. In reality, radiation will enhance the heat transfer for high
6 temperature systems, and the conclusions of this work are limited to non-radiating systems. The
7 simulations elucidate the fundamental mechanisms by which heat is conducted to the particles. Of the
8 various conductive mechanisms, conduction between the thin layer of interstitial gas between the tube
9 walls and adjacent particles is the most significant and accounts for more than 90% of the heat transfer in
10 the system. The simulations also show that the heat transfer is limited by the contact resistance between
11 particles and the wall, not penetration of that heat into the bed. Contact resistance is the dominant
12 mechanism since there is ample mixing and refreshing of cold particles being brought into contact with
13 the hot walls, thus preventing particles from stagnating on the surface and warming to the wall
14 temperature. Since contact conduction, and not penetration of heat into the bed, is the limiting heat
15 transfer mechanism, the thermal performance of the receiver strongly depends on particle size. The
16 simulations show that as the particle size decreases from 300 to 200 μm , the overall heat transfer
17 coefficient increases by approximately 60%. This increase occurs because the gap distance between
18 particles and the wall is smaller, thus leading to less thermal resistance. In addition, more small particles
19 can contact the wall per unit area than large particles, further enhancing the heat transfer.
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40 Radiative heat transfer will be most significant along surfaces where particles do not maintain thermal
41 contact, such as the side and bottom surfaces of each hexagon. Along upper tube surfaces, conductive
42 heat transfer is expected to be the dominant heat transfer mechanism. For an upper tube surface with an
43 anticipated wall temperature of 1200 K, the maximum emissive radiative flux from the wall can be
44 computed by assuming a wall emissivity of unity. The maximum emissive flux is approximately 10% of
45 the conductive heat flux for the baseline case with 300 micron particles. The conductive heat flux
46 increases for smaller particles, and the relative significance of radiation is expected to decrease for
47 smaller particles. For the receiver configurations and anticipated operating temperatures below 1200 K,
48 radiation is not expected to significantly impact our conclusions.
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Changes in the horizontal and vertical spacing of tubes do not have a large impact on the overall heat transfer. The exception occurs when the horizontal spacing is constricted enough such the flow chokes, resulting in particles collecting along the top face of the hexagon, which in turn increases the solids concentration and heat transfer. Although choked flow results in higher heat transfer, it may not be desirable due to jamming effects associated with friction; physics associated with friction and jamming were neglected in this work. Increasing the solids mass flow rate also increases the heat transfer because conduction strongly depends on the solids concentration and the solids concentration increases as the mass flux increases.

The simulations show heat transfer phenomena that are contrary to what would be expected for molecular fluids. When the hexagon apex angle decreases, the solids velocity increases while the heat transfer decreases. For molecular fluids, the heat transfer is expected to *increase* at higher velocities because the thermal boundary layer thins, resulting in larger gradients. The same does not occur for granular flows, because unlike molecular fluids, heat transfer in solid particles is limited by the contact resistance. As the solids velocity increases, the packing loosens slightly and results in a smaller value of solids concentration near the wall. Consequently, fewer particles maintain contact with the wall and the heat transfer rate decreases. This counter-intuitive behavior motivates the need for understanding the fundamental heat transfer mechanisms as well as the need for models that are valid over a wide range of flow regimes.

The other objective of this work is to validate a new continuum heat transfer model that accurately predicts the local heat transfer coefficient in an enclosed solar receiver under a wide range of operating conditions. The validation is performed by comparing the heat transfer predicted by the continuum model to DEM data. Discrete element simulations are highly accurate and model the motion and heat transfer for all particles in the domain within the need for continuum closures, and is thus considered as ‘ideal data’ for purposes of model validation. The simulations show that the continuum model closely matches the local heat transfer coefficient for different center-to-center horizontal and vertical tube spacings,

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4 hexagon apex angles, solids flow rates, and particle size. The average error in the overall heat transfer
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6 coefficient for a laboratory scale receiver was less than 5% when compared to DEM data.
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9 The continuum heat transfer model validation has been completed using hydrodynamic data from
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11 DEM simulations as an input into the continuum heat transfer model. This was done to eliminate any
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13 discrepancies associated with hydrodynamic variations between two-fluid and DEM solvers. To be
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15 practical for large-scale solar receivers, the continuum model must accurately predict the solids
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17 concentration in the receiver. Future work is needed to determine how well the two-fluid (or continuum)
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19 model predicts the solids concentration and hydrodynamics, as the heat transfer in the system is very
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21 sensitive to the solids concentration near the tube surfaces.
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25 The simulations in this work are for a small-scale receiver and neglect radiation. An actual receiver
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27 will contain many more heat transfer tubes and the temperature rise will be much more significant. In
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29 order to test the model accuracy for a realistic receiver, radiative heat transfer mechanisms will also need
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31 to be included and comparisons should be made to experimental data. In addition, the conductive heat
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33 transfer depends on particle size and morphology, and the particles used should be well-characterized for
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35 rigorous model validation.
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