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OPTIMIZATION OF STROAGE BIN GEOMETRY FOR HIGH TEMPERATURE PARTICLE-BASED CSP SYSTEMS

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ABSTRACT

Solid particle receivers provide an opportunity to run concentrating solar tower receivers at higher temperatures and increased overall system efficiencies. The design of the bins used for storing and managing the flow of particles creates engineering challenges in minimizing thermomechanical stress and heat loss. An optimization study of mechanical stress and heat loss was performed at the National Solar Thermal Test Facility at Sandia National Laboratories to determine the geometry of the hot particle storage hopper for a 1 MWt pilot plant facility. Modeling of heat loss was performed on hopper designs with a range of geometric parameters with the goal of providing uniform mass flow of bulk solids with no clogging, minimizing heat loss, and reducing thermomechanical stresses. The heat loss calculation included an analysis of the particle temperatures using a thermal resistance network that included the insulation and hopper. A plot of the total heat loss as a function of geometry and required thicknesses to accommodate thermomechanical stresses revealed suitable designs. In addition to the geometries related to flow type and mechanical stress, this study characterized flow related properties of CARBO HSP 40/70 and Accucast ID50-K in contact with refractory insulation. This insulation internally lines the hopper to prevent heat loss and allow for low cost structural materials to be used for bin construction. The wall friction angle, effective angle of friction, and cohesive strength of the bulk solid were variables that were determined from empirical analysis of the particles at temperatures up to 600°C.

1. INTRODUCTION

The DOE Solar Energy Technology Office has invested in de-risking particle technologies to make them more commercially viable as part of the Generation 3 Concentrating Solar Power (CSP) initiative [1]. As the particle receiver system is proven, it has the opportunity to replace the current CSP "standard" for power plants. It can take the place of the nitrate salt or steam systems and be the backbone of future CSP power plants with increased efficiency and reduced LCOE. Existing commercial CSP systems utilize heat transfer fluids that freeze when cold, wick/leak when hot, and are limited to heat flux limitations that could result in receiver failure. Particle receiver systems are innovative and solve most problems that result from molten salt/steam receivers (Table 1). The main improvements include: no flux limitations on direct particle absorption, higher operation temperatures, no freezing, and are inert with direct thermal storage possible. These benefits enable improvements over the current state of the art, but additionally present a solution for the supply of heat above 800°C or even 1000°C that is not achievable by other scalable CSP technologies currently.

Table 1. Comparison of CSP technologies.

Solid Particle Technology	**Nitrate Salt Technology**	**Steam Technology**
Operation Temps >1000°C	Limited to 600°C	Limited to 600°C
No flux limitations on particles	Limited to tube wall fluxes of 800-1200 kW/m^2	Limited to tube wall fluxes of 800 kW/m^2 or less
No freezing of the media	Freezing of salt below 300°C	No freezing of the media but requires high pressure
Inert materials, non-corrosive	Corrosive to the containment materials	Corrosive to non-stainless steels
Direct thermal storage	Direct thermal storage	No direct thermal storage

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The proposed Gen 3 Particle Pilot Plant (G3P3) consists of a top hopper that drops CARBO HSP 40/70 sintered bauxite spherical particles into the receiver where they are heated to ~800°C by concentrated solar irradiance from the heliostat field. The hot particles are then delivered by force of gravity into a hot particle storage tank, a heat exchanger in which they impart their heat to supercritical CO_2, a cold storage tank, and eventually into a bucket/Olds elevator system which delivers the particles back to the top hopper.

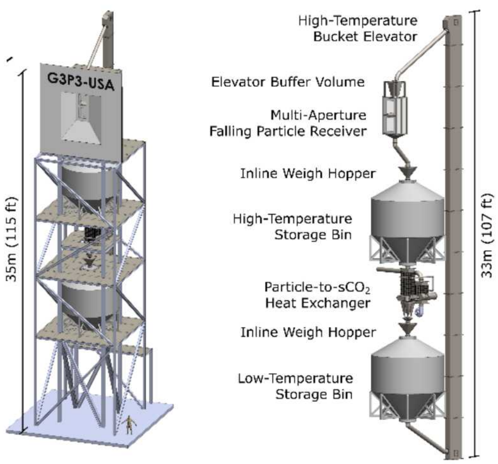

Figure 1. Proposed Gen 3 Particle Pilot Plant [1].

1.1. Definition of Terms

Table 2 defines terms and variables used in the paper.

Table 2: Terms and symbols

β_h	β in the angled hopper part of the silo
β_v	β in the parallel part of the silo
angle of internal friction (ϕ_t)	Internal friction is a result of particles moving past each other. This is typically expressed as an angle.
bulk solid	material made of discrete solid particles that behave as a collective mass
cohesive strength	the strength of the bulk solids to resist shearing when acted upon by a force
cylinder	vertical part of bin. It may be round or rectangular but has a constant cross section (Figure 2)
effective angle of internal friction (δ)	A property derived from linear regression of shear steps that is used in calculating more tangible properties such as effective wall friction
effective wall friction	Angle of incline at which a bulk solid begins to slide over itself
funnel angle	angle of flow channel in funnel flow measured from vertical
funnel flow	flow pattern inside a bin where the bulk material only moves in a flow channel above the outlet when withdrawn
geometry function $H(\theta')$	
hopper angle (θ')	angle from vertical to slope of hopper
hopper	the converging section of a storage vessel. The entire vessel is also commonly referred to as a hopper. (Figure 2)
mass flow	flow pattern inside a bin where all material is in motion when withdrawn
wall friction angle (ϕ')	Angle of incline at which a specific bulk solid slides past a given surface
β	angle between direction of major principal stress at the wall and normal to the wall

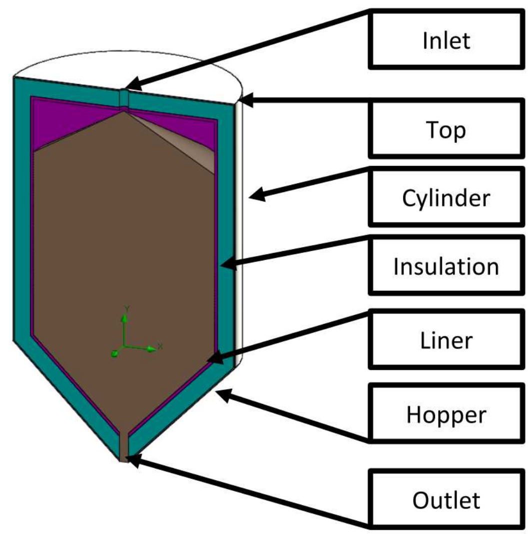

Figure 2: Nomenclature of hopper features

2. PRINCIPLES OF HOPPER DESIGN

Hopper designs vary in geometry for use in handling bulk solids or liquids. Contrary to fluids which have a linear relationship between hoop stress on the vessel and depth, bulk solids take up some of the stress through particle to particle stacking and by shear friction along the walls.

Generally, hoppers designed for solids are elongated (often referred to as silos) to take advantage of the supportive walls and minimize stress while fluid hoppers adopt a more equal height to diameter dimension ratio.

There are also certain flow impediments related to the friction and cohesive properties that are unique to bulk solids including *arching*, a complete clog that forms when cohesive materials form an arch whose strength is greater than the load force from materials above and *ratholing*, a condition whereby flow only occurs through a small irregular hole above the outlet but is otherwise packed along the sidewalls where it may or may not collapse during flow. The theories of Jenike define the necessary slope of the hopper and diameter of the outlet to prevent these impediments. Stable flow can occur as *mass flow*, where all particles in the hopper move at essentially uniform velocity in the vertical direction, or *funnel flow*, where vertical velocity only occurs in the center, funnel, portion of the hopper. There is zero vertical velocity along the walls as particles first move horizontally away from the walls and into the funnel as the hopper drains.

The design of a hopper with predictable flow characteristics requires knowledge of the cohesive strength of the bulk solids. That is, the force required to shear the bulk as a function of an applied normal force. This function, called the *flow function* can be defined by packing solids into a form with an applied packing force downward and then applying a horizontal force that increases until the packed solids shear. The process is repeated at different packing forces and a curve is drawn through the shear points to define the function. Cohesive materials can exhibit ratholing and arching. Hopper angles, the angle of the tapered funnel portion of a hopper from vertical, can be informed by knowing the wall friction angle, the incline angle from horizontal at which the solids will begin to slide along the surface material, and internal friction, a derived property related to the ability of particles to flow over other particles. These properties are tested in a similar manner by applying normal force and measuring the force at which particles shear from the surface material. Internal friction is derived from the cohesion test outputs.

Once these parameters are understood the hopper angles can be calculated for mass flow and for funnel flow. The outlet diameter can be calculated as a minimum diameter to prevent arching and a minimum diameter to prevent ratholing. In non-cohesive materials such as the sintered bauxite used in falling particle receiver applications the minimum diameter is only a few particle diameters wide. Thus the driving considerations are not based on avoiding flow obstructions but rather accommodating flow rate and interfacing with downstream components such as chutes, valves, or rotary feeders.

3. G3P3 HOT PARTICLE STORAGE VESSEL DESIGN

The design approach for the hot particle storage hopper involves three parts: 1, determine the geometric features of the hopper region including hopper angles and outlet diameter based on cohesive strength, internal friction and wall friction properties of the particles on the liner at high temperature. Ensure design does not impede flow or cause irregularities. 2, determine the geometric features of the cylinder region including volume needed for heat and mass transfer requirements of the system. 3, evaluate design alternatives to control heat loss, stress, and improve durability of the hopper.

3.1. Materials

The current design configuration utilizes CARBO HSP 40/70 proppants made of sintered Bauxite as the heat transfer medium. The hot and cold particle storage vessels are lined with three layers of insulation designed by G3P3 partner, Allied Mineral. The innermost 63.5 mm layer is a high density smooth refractory material, Tufcrete 47. The purpose of the inner layer is to minimize friction and erosion from hot particle flow. The next layer is a lower cost 381 mm thick solid, Insulmix 19L. The third layer is a 25.4 mm microporous insulation, Elmtherm 1000 MP.

Table 3: specific heat (c_p), conductivity (λ), and thermal expansion (α) coefficients of storage hopper materials

Material	$c_p\left(\frac{J}{kg\cdot K}\right)$	$\lambda\left(\frac{W}{m\cdot K}\right)$	$\alpha\left(\frac{1}{K}\right)$
Tufcrete 47	1175	1.53	2.25
Insulmix 19L	1386	0.15	1.75
Elmtherm 1000 MP	1050	0.05	2.25
Carbon Steel ANSI32	440	43	11.7
Inconel 625	590	22.8	15.8
CARBO Accucast IDK-50			
CARBO HSP 40/70	1282	0.7	*
Air	1006	0.024	0.007
*coefficient unavailable			

High temperature particle storage tanks demand special consideration of erosion and abrasion. In lower temperature applications a hopper made for handling very hard and fine particles might be lined with a hard smooth metal surface that would reduce overall height by allowing mass flow over steeper hopper angles (measured from vertical) due to lower wall friction, and withstand the impact of falling particles (abrasion) and the sliding friction along the walls (erosion). Falling particle storage tanks can undergo swings in temperature from cold ambient air to over 800°C making stresses caused by mismatches in thermal expansion of the metallic lining and the refractory problematic (Table 3).

In order to avoid a metallic liner the storage tank design team considered a smooth hard refractory material, *Tufcrete*. Wall friction testing was underway at the time of writing so the design is preliminarily based on the empirically tested wall friction values of CARBO *Accucast* IDK-50 particles on a Mild Carbon Hot Rolled Steel surface.

3.2. Hopper Angles and Outlet Geometries

Many fundamental flow properties that inform the hopper geometry change at high temperatures. Jenike & Johanson provided testing at 22°C and 600°C temperatures. At the time of writing the particle flow properties of CARBO HSP 40/70 on Tufcrete 47 refractory insulation is unknown and all bulk solid properties and wall friction properties are assumed to be similar to results from earlier testing performed on CARBO Accucast ID50-K particles on a mill finished hot rolled carbon steel surface at 600°C.

3.2.1. Mass Flow Geometry

Given the material properties of the particles and the interior substrate, maximum hopper angles for mass flow (θ') as measured from vertical can be calculated in terms of the wall friction angle (ϕ') and angle of internal friction (δ) whose values are derived empirically from shear cell testing as

$$\theta' = 90 - \frac{1}{2}\cos^{-1}\left(\frac{1-\sin(\delta)}{2\sin(\delta)}\right) - \beta$$

where β is the angle between the principal plane and the plane normal to the hopper wall and can be expressed

$$2\beta = \phi' + \sin^{-1}\left(\frac{\sin(\phi')}{\sin(\delta)}\right) \text{ [2]}$$

The minimum outlet diameter to prevent arching can be calculated as a function of critical stress (σ_{crit}) which is the value at which the material's unconfined yield strength (f_c) is equal to the external arch stress. The value of f_c is derived from shear cell testing. The stress state of an arch at the outlet has been expressed analytically by Jenike and is a function of the materials bulk density and geometry function $H(\theta')$.

$$H(\theta') = \left(\frac{130° + \theta'}{65°}\right)^{i} + \left(\frac{200° + \theta'}{200°}\right)^{1-i}$$

where i= 0 for rectangular outlets and 1 for axisymmetric hoppers [2].

$$B_{arch} = \frac{H(\theta')\sigma_{crit}}{\rho_b g}$$

The minimum diameter to prevent ratholing is expressed in terms of the empirically determined unconfined yield strength (f_c) time angle of internal friction (ϕ_t) as

$$B_{rat} = \frac{G(\phi_t) f_c}{\rho_b g}$$

where $G(\phi_t) \approx -5.066 + 0.490\phi_t - 0.112\phi_t^2 + 0.000108\phi_t^3$ [2].

The minimum outlet diameter to accommodate the necessary flow rate of 5 kg/s can be calculated as

$$B_{out} = \frac{2(1+m)\tan(\theta)}{g}\left(\frac{\dot{m}}{A\rho_b}\right)^2 \text{ [3].}$$

The interface between the hot storage tank and the heat exchanger is a 0.2 m chute. Therefore, the 0.2 m outlet diameter is adequately sized and will not impact the possibility of mass flow.

3.2.2. Peak Stress

Mass flow hoppers exhibit a concentrated stress discontinuity at the intersection of the cylinder and the hopper section (Figure 3). This peak horizontal stress can be estimated as

$\sigma_1 = \frac{\rho_b g D}{k_c \tan(\phi\prime)}$ where the values of k_c, the ratio of horizontal stress to vertical stress on a particle in the vertical cylinder portion of the vessel, can be assumed to be 0.4-0.6 [2] where $k_c = \frac{1+\sin(\delta)cos2\beta}{1-\sin(\delta)cos2\beta}$ [4]. At the transition, there is a discontinuity. The ratio k_c the horizontal to vertical stress ratio at the transition between the parallel and slanted portions of the vessel can be expressed as $k_c = \frac{\sqrt{\tan^2(\Delta)+\cos^2(\delta)}+\sin(\delta)\tan(\Delta)}{\sqrt{\tan^2(\Delta)+\cos^2(\delta)}-\sin(\delta)\tan(\Delta)}$ where $\Delta = \beta_v - (\beta_h + \theta)$ [4]. σ_1 at the transition can be assumed to be 3 times the parallel value just before for conical and wedge-shaped hoppers and 1.3 for expanded-flow hoppers [2].

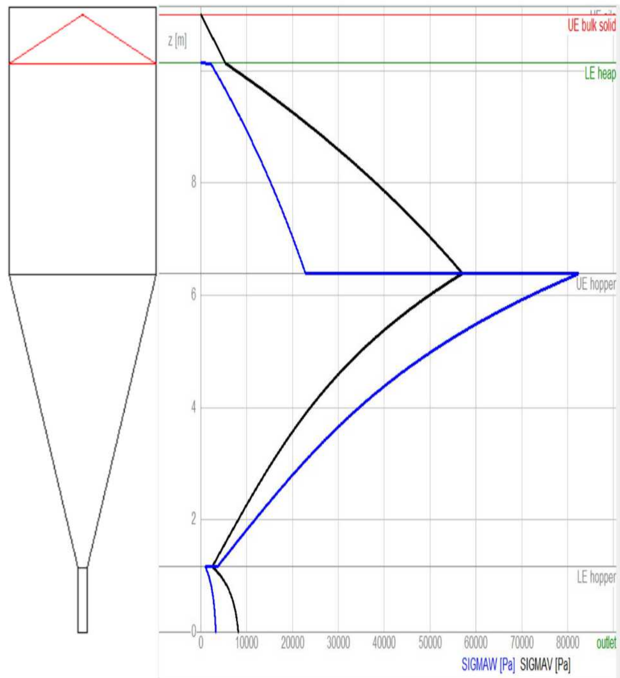

Figure 3: Graph of vertical and horizontal stress profiles on a notional mass flow design as a function of effective head and hopper geometry.

In funnel flow hoppers the flow channel may expand enough to reach the container walls and form a peak stress. The location of this intersection is difficult to predict and so funnel flow containers should be designed for the same peak stress as those of mass flow [5].

3.2.3. Funnel Flow Geometry

Funnel flow geometries allow steeper hopper angles that can reduce overall height. Particle flow only occurs through a funnel section in the middle wherein particles move in a mass flow like fashion. Formulas for the form of the flow funnel are shown in Figure 4. Conical and pyramidal funnel angles for 600°C *Accucast* are approximately 3.6° outward from a vertical cylinder inscribing the outlet and normal to the plane of the outlet. Funnel angles for slotted hoppers are approximately 23.6°.

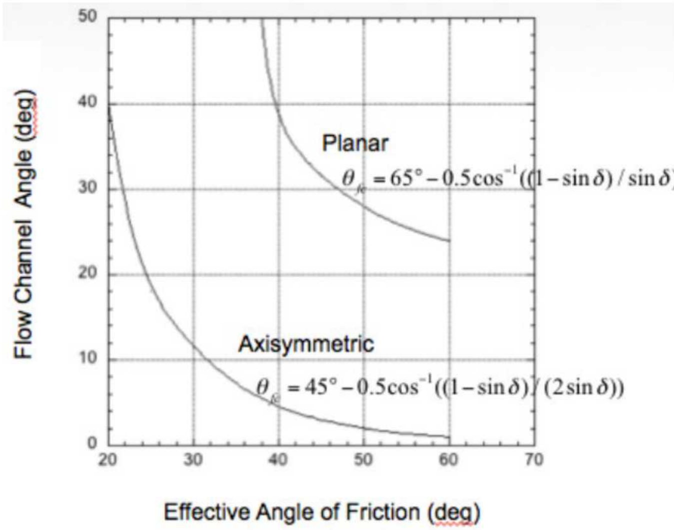

Figure 4: Flow channel angles

Regions outside the funnel angle have the opportunity for greater cooling. Figure 5 presents a notional cooling profile comparing the distribution of temperatures across a simplified cylindrical insulated tank at 10 and 24 hours. The blue angle represents a 3° funnel from a conical concept while the white angle represents a slotted hopper of notional diameter. After 10 hours (top) the temperature of the particles flowing into both sized funnels are consistent. After 24 hours (bottom) lower temperature particles from the bottom corners begin to flow into the slotted hopper funnel.

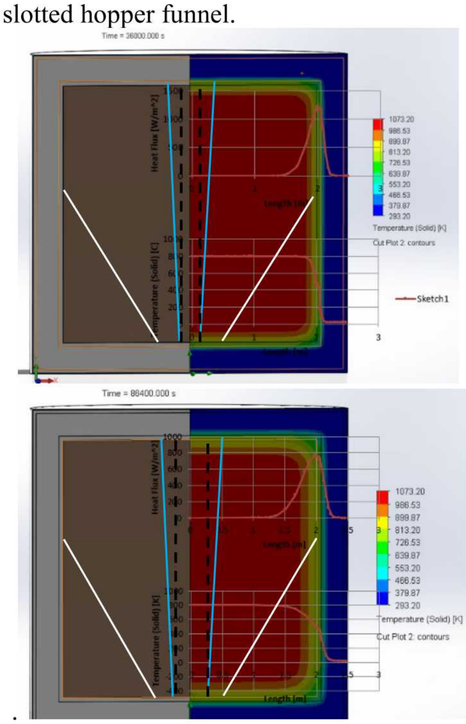

Figure 5: Cross-sectional thermal flux and temperature profiles of a hot storage tank with approximate funnel angles

The hopper angles for a funnel flow hopper can vary. Hopper angles that exceed the wall angle of friction enable complete discharge. For a cohesive material the minimum outlet diameter needed to prevent ratholing or arching in a funnel flow type hopper can be calculated as $D_f = \frac{G(\phi_t) f_c}{\rho_b g}$ (see 3.2.1) [2].

For non-cohesive materials, guidance on minimum outlet size recommends 10 diameters of the particles [5]. Outlets larger than 7 mm will prevent arching of CARBO particles. The preferred 0.2 m outlet chute diameter is ample.

3.3. Cylinder Geometry and Size

3.3.1. Geometric Considerations

The ideal configuration to minimize heat loss in a storage bin is to that which minimizes surface area. Thus a height to diameter ratio of the parallel cylinder section of 1 is preferred (Figure 6, left). However, bulk solids also produce a peak stress proportional to bin diameter. Elongated geometries are preferred for minimization of stress (Figure 6, right).

Figure 6: Hopper designs with identical volume configured with increasing height/diameter ratios.

3.3.2. Size Considerations

The size of the overall hot storage hopper tank is defined by the power requirements of the system. The G3P3 pilot is a $1MW_t$ system with scalability to $10MW_t$ and $100MW_t$. The system performance requirements demand that the hot particle tank must have 10 hours of deferred storage before discharging for 6 hours (21,600 seconds) to the heat exchanger without recharging. The heat ($\dot{Q}$) required to be provided to the heat exchanger is a function of mass flow ($\dot{m}$) of particles (not to be confused with mass flow hoppers) with a bulk density (ρ_b) tested to be to 1,870 $\frac{kg}{m^3}$ at a consolidating pressure of 10.0 kPa, specific heat (c_p) of

$1.82 \frac{kJ}{kg*K}$, and the temperature differential between the hot and cold tanks (ΔT) of (750°C – 570°C) where $\dot{Q} = \dot{m}c_p\Delta T$.

$$\dot{m} = \frac{1.00E6\,\frac{J}{s}}{1280\,\frac{J}{kg*C}(180°C)} = 4.34\,\frac{kg}{s} \times 21{,}600\,s = 93{,}800\,kg$$

$$\frac{93{,}800\,kg}{2000\,\frac{kg}{m^3}} = 51.0\,m^3$$

The total volume will be divided between the bottom hopper portion, the cylinder, and the mound on top, whose geometry is determined by the bulk solid's angle of repose (~30°). Additional volume should be available to accommodate 10% ullage space, additional bulk mass for particle attrition, heat loss, margin for thermal capacity.

Further modelling is proposed to determine whether additional volume is necessary to ensure cold stagnant regions do not flow into the heat exchanger upon full discharge.

4. DESIGN ANALYSIS

Thermal and stress analysis were performed to understand performance in the steady state use cycles where the insulation temperature has come to a cyclic equilibrium over the course of several days and to understand how geometry impacts thermal performance in the first use condition over 10 hours. The G3P3 goal is to deliver $1MW_t$ to the heat exchanger for 6 hours after 10 hours of deferred storage.

4.1. Ten hour deferred storage at -25°C ambient

The 10-hour model looked at the thermal profile of a conical funnel flow hopper sized for enough mass to provide 1MW heat transfer with a full hopper section of particles remaining. This model also considered the adjoining chute, a 0.2 (8 in) diameter 0.003 m (1/8 in) thick steel pipe, welded to the steel hopper shell, and wrapped in a single layer of 1" insulation. The ambient air conditions were based on a worst case -25°C air. The inlet was assumed to be open to ambient conditions. All solids were assumed to be at -25°C and the bulk particles were assumed to be at 800°C. Material properties were informed by manufacturer data sheets as shown in Table 3. Three height-to-diameter ratio configurations were analyzed to determine how heat loss would vary as bin geometry elongated.

Modelling was performed in SolidWorks Flow Simulation. Figure 6 shows the hopper model. A 3D model with assumed symmetry was used to improve runtime.

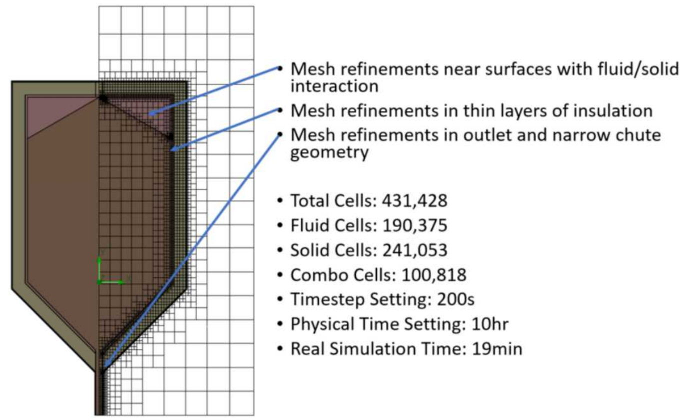

Figure 7: Mesh and model settings

Mesh convergence studies were performed to determine the level of refinement necessary. Results were independent of mesh size above 100,000 cells.

Table 4 summarizes the model results. The total heat lost was calculated as a function of the difference between the initial 800°C temperature of the bulk mass and the average volumetric temperature after 10 hours of deferred storage. The principal stress was calculated using. The modeled geometry results in funnel flow. While there is not an analytical formula for funnel flow stress, mass flow stress is being considered as an upper bound in the absence of discrete element modeling. Stress increases by approximately 40% for a fixed volume relative to the H:D=1.

Table 4: Results of Ten hour deferred storage modeling

H:D Ratio	Ave Heat Flux W/m²	Ave Bulk Temp (C)	Total Energy Lost (kJ)	Wall Stress (kPa)	mass (kg)	% Heat Loss Relative to H:D=1	% Stress Relative to H:D=1
1	0.127	756.9	196.5	86.0	128107	0%	0%
2	0.130	747.7	238.5	68.2	128038	21%	21%
4	0.124	735.7	293.3	54.2	128126	49%	37%

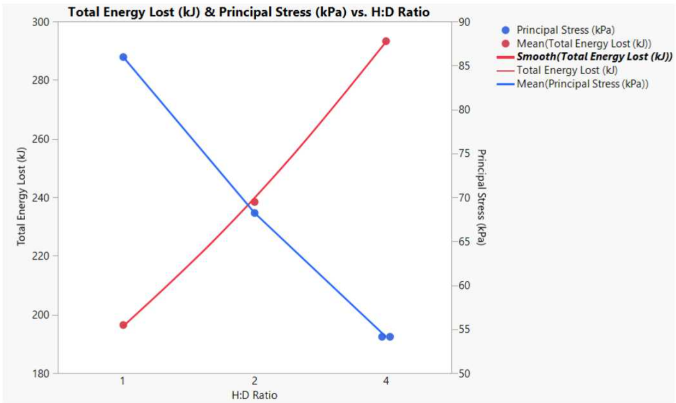

Figure 8: Heat loss (blue) and peak stress as a function of height/diameter ratio

Figure 8Figure 10 show the temperature gradients across the center of the cylinder in the horizontal and vertical directions for the H:D sizing of 1, 2, and 4 respectively. Symmetry is assumed. The temperatures begin to taper below the desired heat exchanger input temperature at 1.5 m, 1.3 m, and 0.83 m respectively. Minimum temperatures and peak heat loss occur at the outlet. Temperatures of particles near the cold air at the inlet do not drop significantly as was expected given the magnitude of the differences between specific heat. Temperatures in the outlet chute are hotter in the middle due to heat losses at the union between the chute and the exterior shell of the bin.

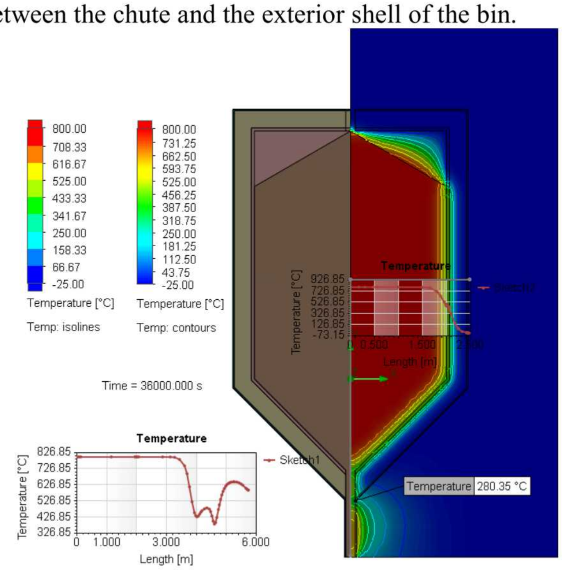

Figure 9: Cut plot of temperature gradient with 1:1 height/diameter size ratio

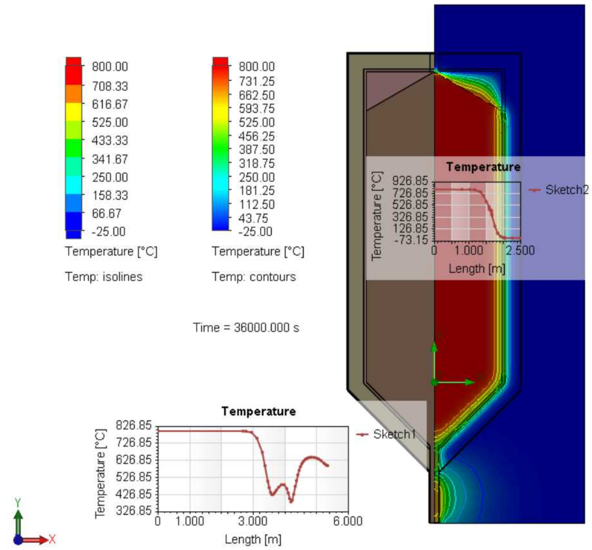

Figure 10: Cut plot of temperature gradient with 2:1 height/diameter size ratio

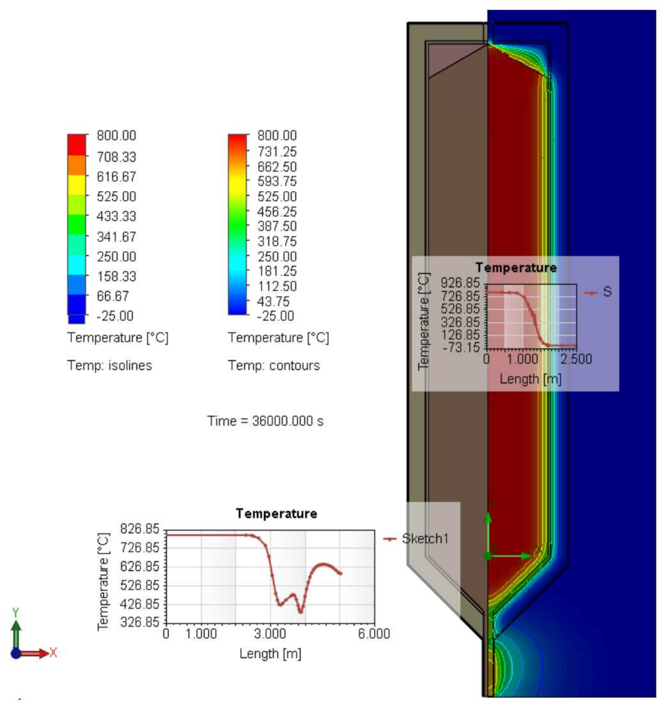

Figure 11: Cutplot of temperature gradient with 4:1 height/diameter size ratio

4.2. Cyclic Analysis

Thermal analysis was performed to predict the behavior of temperature and heat loss over time. A cyclic steady-state simulation was conducted where a cold hopper, initially at 25°C, is filled with 800°C particles (instantly) and held for 10 hours. During the storage period, the particles transfer heat to the refractory layers. The particles are then discharged instantly and the hopper continues to loose heat to the environment for the remaining 14 hours. The cycle is repeated until the cyclic differences between temperature and heat loss deltas between charging and discharging phases are identical (Figure 11).

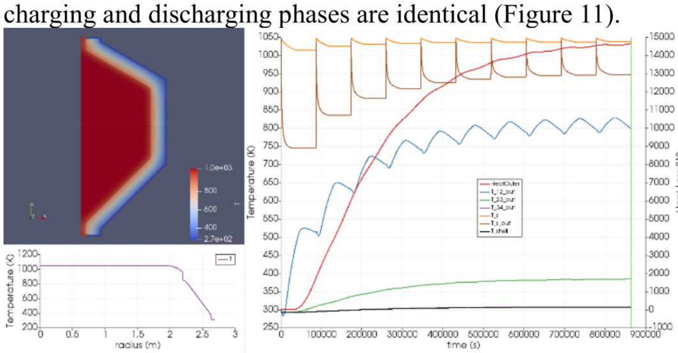

Figure 12: 2D axisymmetric simulation of transient storage bin operation during periodic charging/discharging cycles

5. CONCLUSIONS

The cyclic analysis establishes a good metric for evaluating startup time and steady state heat loss. There is a significant increase in heat loss in the elongated geometries. In advance of a detailed design only qualitative comparative statements can be made. A hopper with a 1:1 height to diameter ratio minimizes heat loss even when a large surface area in the top exposed to extremely cold air at the inlet. Principal stress is linearly related to height but the peak stress was approximately 100kPa which is very small relative to yield strengths of the hopper materials (~15MPa for high density refractory, or 620MPa for the steel liner). In a funnel flow configuration these stresses are expected to be even less as the discontinuity between the sloped and vertical walls is obscured. Furthermore, without mass flow along the walls a metallic liner may be unnecessary as erosion will be minimum and hopper angles will not need to be steep enough for mass flow. Extra particles in the hopper region can prevent abrasion from falling hot particles in the initial fill.

Incorporating a funnel flow geometry will pull particles from the colder regions of the sidewalls into the center flow channel which is insulated from the sidewalls shown to lose little heat. It is possible that this flow profile will reheat particles as they pass through the hot core of the tank on route to the outlet. This concept could also leave cold stagnant regions in the bottom of the tank during cyclic operation which could become particularly problematic if the hopper is not fully discharged at controlled time intervals.

6. AKNOWLEDGMENTS

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