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OPTIMIZATION OF STORAGE 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

The proposed Gen 3 Particle Pilot Plant (G3P3) consists of a top hopper that drops a stream of ~400-500 μm spherical sintered bauxite 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 as shown in Figure 1.

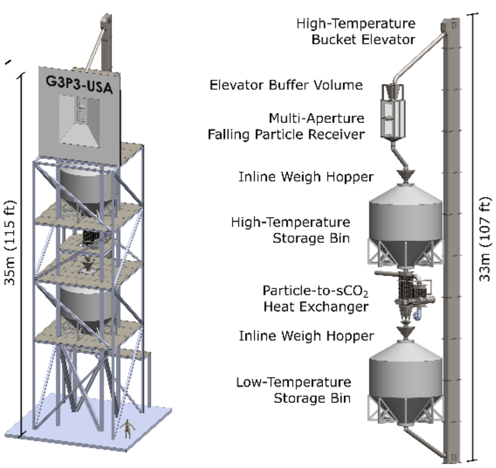

Figure 1. Proposed Gen 3 Particle Pilot Plant [1].

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
ϕ_t	angle of internal friction in incipient flow
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)
drawdown angle	the angle formed by bulk solids around the outlet of a flat-bottomed container
δ	effective angle of internal friction of a bulk solid during flow
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
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
ϕ'	Wall friction angle: 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
rathole	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

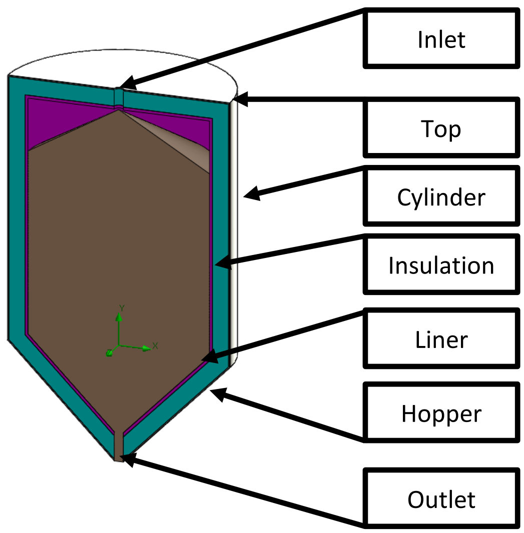

Figure 2: Nomenclature of hopper features

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 HSP 40/70	1235	0.35	*
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 of *Accucast* IDK-50 at 22°C and 600°C and HSP 40/70 and 22° C, 550° C, and 800° 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 $B_{out} = \frac{2(1+m)\tan(\theta)}{g}\left(\frac{\dot{m}}{A\rho_b}\right)^2$ [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 $\sigma_1 = \frac{\rho_b g D}{k_c \tan(\phi')}$ 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].

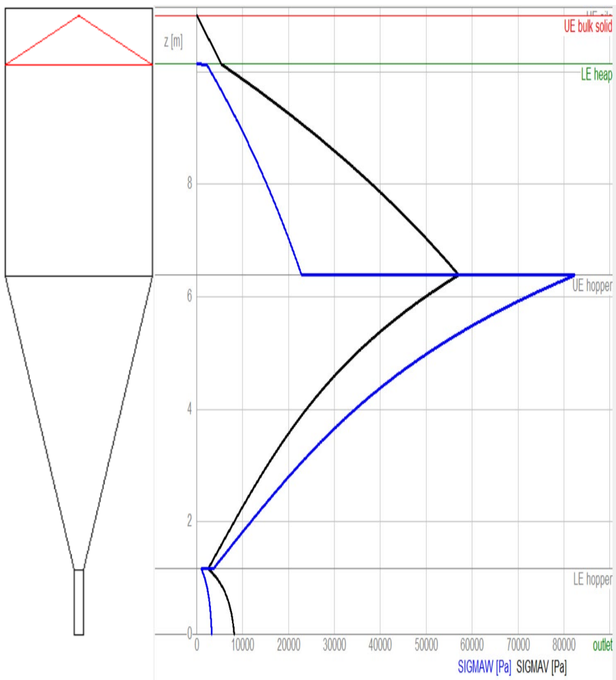

Figure 3: Graph of vertical (black) and horizontal (blue) stress profiles on a notional mass flow design as a function of effective head and hopper geometry.

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

When hopper angles are shallower than a determined mass flow angle, they provide enough vertical support so there is no flow along the walls. Particle flow only occurs through a funnel section in the middle wherein particles move in a mass-flow-like fashion. As the level of the tank drops, particles flow away from the walls and into the center portion.

These funnel flow geometries are beneficial in tower applications as they allow steeper hopper angles that can reduce overall tower height. This is of concern to CSP applications because the particles at the walls are cooler than the center. The dynamic mixing of particles at different temperatures as the tank drains requires additional research.

A formula for the form of the flow funnel angle (θ_f) is given as $\theta_f = 45° - 0.5\cos^{-1}\left(\frac{1-\sin(\delta)}{2\sin(\delta)}\right)$ [6]. Figure 4 illustrates the expected material specific funnel flow channel. At a colder 22° C, the heat transfer solids are expected to form a drawdown angle of ~34° estimated to be the average of the angles of internal friction in flow and incipient flow.

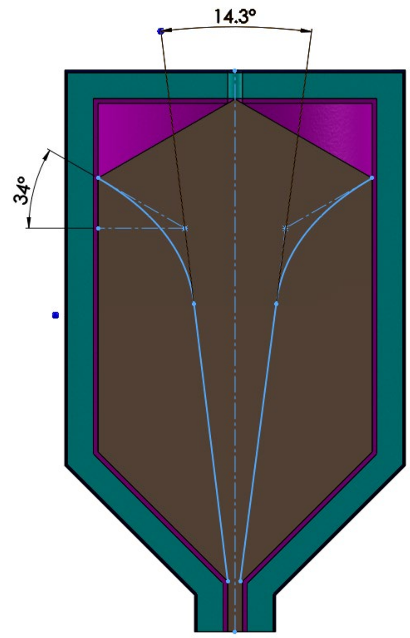

Figure 4: The expected funnel flow channel of specified particles.

Another consideration in the use of funnel flow geometry in CSP applications is the formation of ratholes when outlet diameters are too narrow. For non-cohesive materials such as CARBO HSP, guidance on minimum outlet size recommends 10 diameters of the particles [5]. However, material properties of non-cohesive particles change at high temperature. CARBO HSP was shown to vary slightly at very high temperatures near 800° C and was measured to be slightly cohesive such that at the large heights and radii used in solar applications, the particles can form a shallow rathole to the height at which the funnel angle reaches the critical ratholing diameter.

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. However, bulk solids also produce a peak stress proportional to bin diameter. Elongated geometries are preferred for minimization of stress (Figure 5, right).

Figure 5: 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. 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 with a bulk density (ρ_b) tested to be to 2146 $\frac{kg}{m^3}$ at 200° C (max tester temperature), an average specific heat (c_p) determined empirically to be 1235 $\frac{kJ}{kg*K}$, and the temperature differential between the hot and cold tanks (ΔT) of 195° C. Thus, for a 1 MW$_t$ capacity (t) over 6 hours, the nominal mass and volume required could be calculated as $\dot{m} = \frac{\dot{Q}}{c_p \Delta T} = 4.15 \frac{\text{kg}}{\text{s}}$ which requires approximately 90,000 kg of total flowing mass with a volume of 42 m^3 plus any additional margin or ullage required.

The form of the flowing particles in an axisymmetric conical bin has three parts: the heap on top whose geometry is dictated by the diameter and the angle of repose, the funnel portion on the bottom whose geometry is dictated by the hopper angle and the diameter, and the parallel cylindrical portion. In the case of flat bottom hoppers, additional mass is required to account for the stagnant particles forming the drawdown angle. Thus, the dimension to be optimized is the diameter of the cylindrical portion as shown in Figure 6.

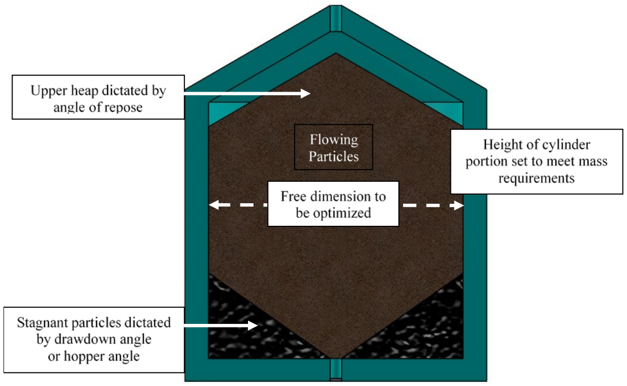

Figure 6: Geometric regions of bulk solid formation.

Given that all dimensions can be defined as a function of diameter, the diameter that produces the minimum surface area can be solved iteratively for the figure above as is shown in Figure 7.

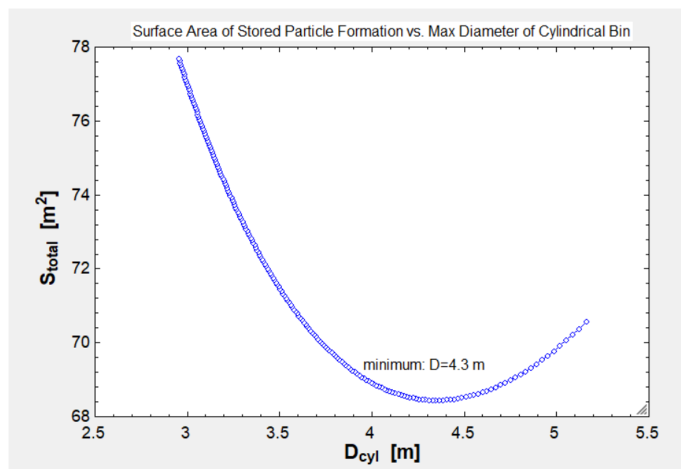

Figure 7: Minimization of Surface Area

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.

4.1. First ten hours with different geometry

The 10-hour model looked at the thermal profile of three hopper designs equally sized for 1MW$_t$. The comparison looked at the effects of partially vs solid filled hoppers as informed by the top geometry and flat bottom vs 45° angled bottom hoppers. Figure 8 displays temperature contours of only three of the configurations for the purposes of conveying the geometric variables of solid filled vs. partially filled top and flat vs. angled bottoms. The ambient air conditions were based on nominal 20 °C air. The inlet was assumed to be open to ambient conditions with natural convection considered. All solids were assumed to be at 20 °C and the bulk particles were assumed to be at 800°C. Three height-to-diameter ratio configurations were analyzed to determine how heat loss would vary as bin geometry elongated as shown in Figure 5.

Figure 8 shows the temperature gradients across the center of the cylinder in the horizontal direction for the H:D sizing of 1 in a filled, partially filled, and flat bottom design, respectively. Symmetry is assumed. The full tank (top) where the design intent is to fill the tank completely with bulk solids shows more heat loss to the refractory insulation than through convective air pockets between the bulk solids and the top in the partially full (middle) tank.

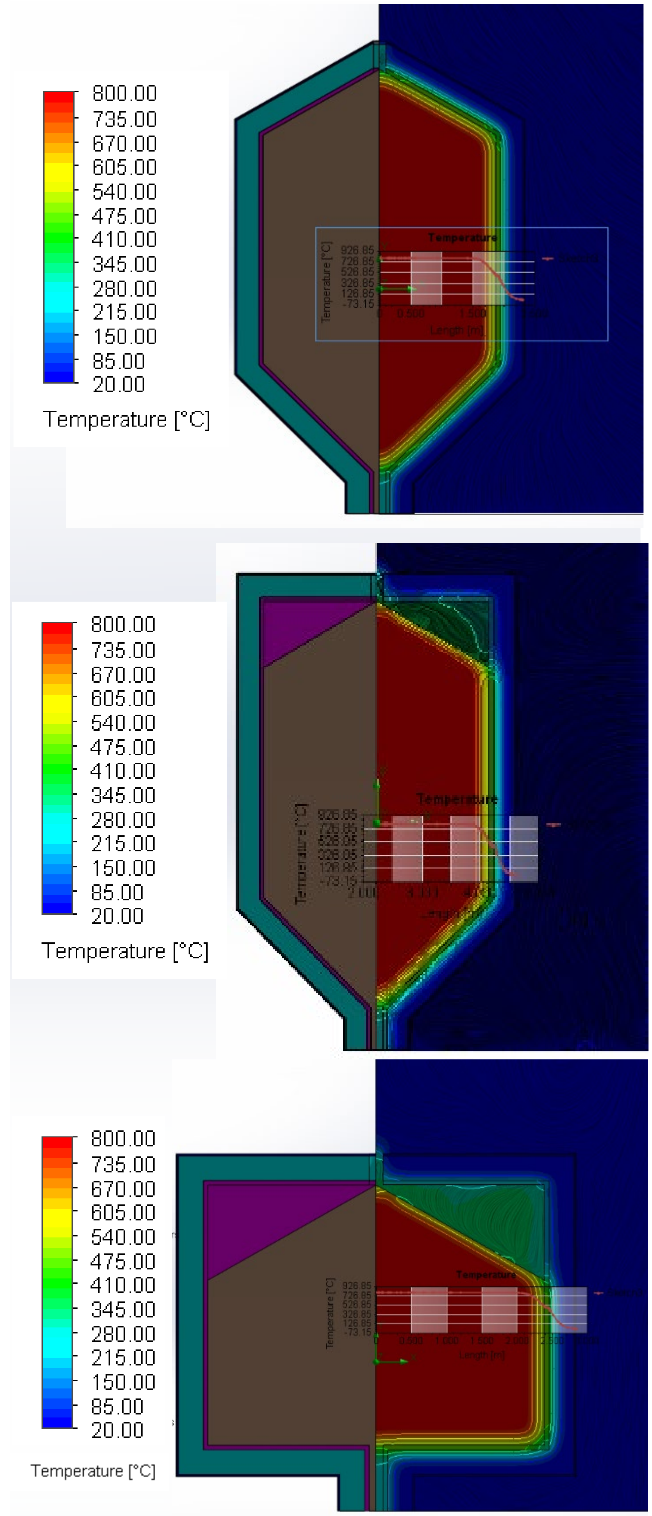

Figure 8: Temperature contours after 10 hours of simulated storage for solid filled (top), partially filled (middle), and flat-bottom (bottom) hopper designs

Table 4 and Figure 9 summarize 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 method discussed in section 3.2.2. 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 computational modeling. In the $1MW_t$ hopper, stresses do not significantly impact the design or material selection. However, commercial scale systems would produce significantly greater stresses. Figure 9 shows the flat-bottomed design may exhibit less stress while a partially filled hopper may exhibit less heat loss. The stress in the flat bottom concept is reduced due to the lack of hopper angles which cause a discontinuous stress peak (Figure 3). The observed reduction in heat loss may be due to the bulk surface that is in contact with air which has a much lower conductivity than the refractory insulations considered. While the refractory is cold it acts more like a heat sink than an insulator.

Table 4: Results of Ten hour deferred storage modeling

H:D Ratio	Design	Surface Area (m^2)	10 hr Ave Bulk Temp (C)	Total Heat Lost (kW)	Wall Stress (kPa)
0.5	Full	72.28	748.15	190.48	16.52
1	Full	70.36	749.70	184.81	19.61
2	Full	78.81	743.59	207.30	13.59
0.5	Partial	72.28	751.53	178.07	16.52
1	Partial	70.36	754.23	168.16	19.61
2	Partial	78.81	745.86	198.96	13.59
0.5	Flat Btm	74.57	750.19	182.95	14.73
1	Flat Btm	75.57	751.22	179.31	15.71
2	Flat Btm	79.83	745.21	201.30	13.58

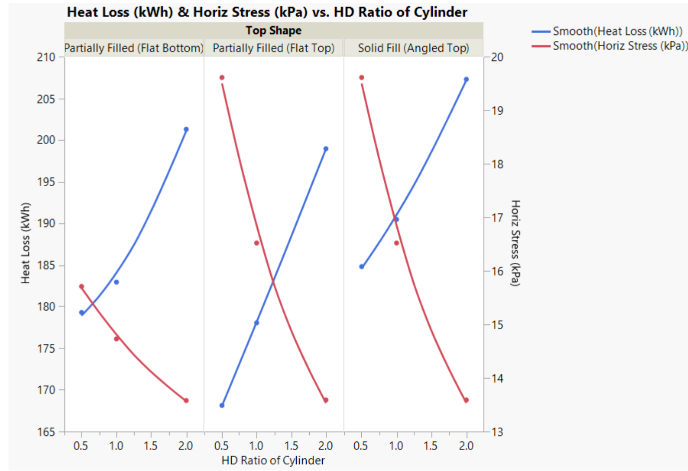

Figure 9: Heat loss (blue) and peak stress as a function of height/diameter ratio

4.2. Cyclic Analysis

Thermal analysis was performed to predict the behavior of temperature and heat loss over time in nominal use conditions where the hopper is filled and discharged daily and the refractory material temperatures have risen to an operational state. 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 lose 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 negligible (Figure 11). Table 5 quantifies the extent to which heat loss increases and consequently rise time decreases as the aspect ratio elongates (Figure 10).

Table 5: Heat Loss and Rise Time at 3 Aspect Ratios

Aspect Ratio	Heat Loss (kW)	Efficiency	Rise Time days
1	230.4	96.16%	8
2	244.6	95.92%	6
4	278.5	95.36%	5

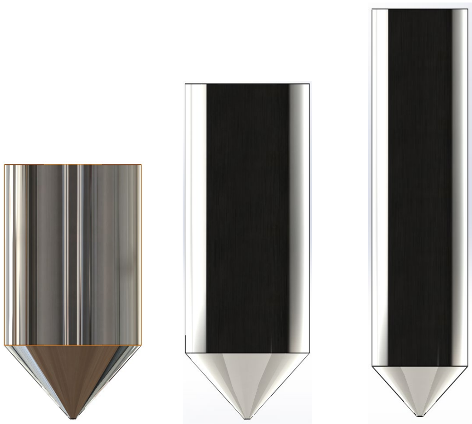

Figure 10: Hopper geometries corresponding to aspect ratios of 1 (left), 2 (mid), and 4 (right)

Figure 11 shows the results of a 2D axisymmetric simulation of transient storage bin operation during periodic charging/discharging cycles with height to diameter ratios of the cylindrical portion of 1, 2, and 4. The left axis is temperature with the following: *T_s* (orange) being the integrated average temperature of the bulk particles. The daily cycles are identifiable as periodic spikes as particles enter at 1073 K and level off. *T_s_out* (brown) is the temperature of the particles where they contact the inner layer of refractory. *T_12_out* (blue) is the temperature between the first and second layer of insulation with *T_23_out* (green) and *T_34_out* (purple) being the temperature between the subsequent layers of insulation. *T_shell* is the average temperature over the surface of the steel shell. On the right axis is heat flux with the *HeatOuter* in Watts (red) indicating the flux out of the system times the surface area and *FluxOuter* in W/m^2 (black) indicating the flux out of the system at each time increment.

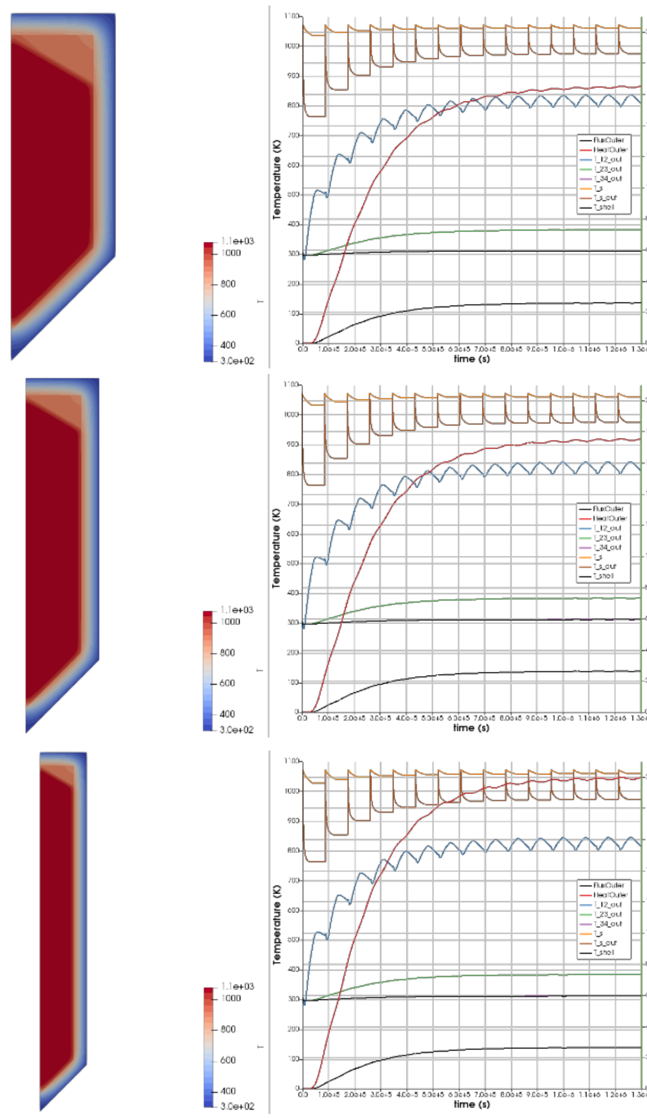

Figure 11: Simulations of transient storage bin operation with height to diameter ratios of 1 (top), 2 (mid), and 4 (bottom).

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 which corresponds to a faster rise time to equilibrium. In advance of a detailed dynamic model only qualitative comparative statements can be made. Principal stress was observed to be 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.

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 problematic as the hopper drains dynamically over several hours.

6. AKNOWLEDGMENTS

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