SAND2021-12790C

Design Considerations for Horizontal High-Temperature Particle Conveyance Components

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Abstract. A design study was conducted at the National Solar Thermal Test Facility (NSTTF) at Sandia National Laboratories in Albuquerque, NM with the objective of identifying the technical readiness level, performance limits, capital and O&M costs, and expected thermal losses of particle handling and conveyance components in a particle-based CSP plant. Key findings indicated that skips and high temperature particle conveyance technology are available for moving particles up to 615° ± 25° C. This limits the use of mechanical conveyance above the heat exchanger and suggests vertical integration of the hot storage bin and heat exchanger to facilitate direct gravity fed handling of particles. Skip rails and support structures add significant cost and must be factored into cost analysis. Chutes can be a low cost option for particle handling but uncertainties in tower costs make it difficult to know whether they can be cost effective in areas above the receiver.

BACKGROUND

Increased efficiencies and lower levelized cost of electricity may be achievable with in tower-based concentrating solar power systems that can increase the temperature of heat transfer media above 700° C so that it is compatible with a Brayton cycle using sCO2 [1]. Research is being conducted on the use of flowing bulk solid particles as the heat transfer media. Solid particles are inert, do not freeze, and have been heated to temperatures > 1000° C [2, 3] without sintering. The combination of high-temperature and high-mass flow rates required to accommodate commercial scale thermal energy storage and heat transfer pushes the boundaries of solid particle handling equipment that is largely appropriated from other industries such as mining and cement manufacturing.

Research was conducted at the National Solar Thermal Test Facility to identify the technical readiness, performance limits, Capital/O&M costs, and expected thermal losses of horizontal particle conveyance methods and relevant interfaces in commercial-scale CSP systems. Prior research has been conducted on the designs and thermal losses of vertical conveyance systems [4] [5]. Models from this work have been adopted to look at the utilization of skip hoist systems not only as the primary particle lift to the receiver, but as the downcomer from the receiver to the hot storage bin and as the cold storage lift in a two-bin storage configuration. Figure 1 depicts a conceptual 100 MW_e

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particle-based CSP plant used as a basis to evaluate the methods within the system that convey particles horizontally either directly or through a combined interface with another component.

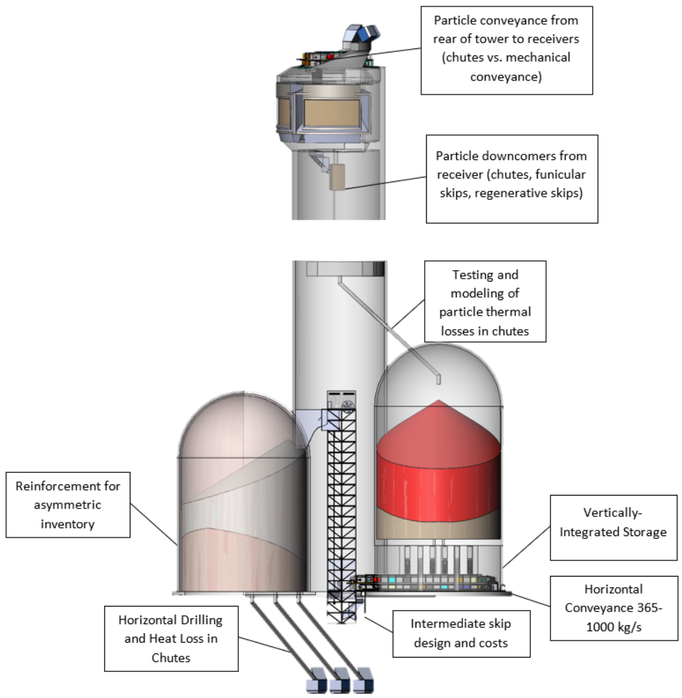


FIGURE 1. Diagram of particle-based CSP system with annotations on horizontal conveyance considerations.

PARTICLE PROPERTIES

The sintered bauxite proppant, CARBO HSP 40/70, is used as the basis of this design study. Flow properties testing was performed on new particles by Jenike and Johanson (J&J). The distribution of particle diameter (d_p) was measured and found to be normal. The 50th percentile particle diameter was 400 µm with the 10th and 90th percentile at 300 and 550 µm respectively. At 200° C (the maximum available temperature), the loose fill bulk density (ρ_b) was 2100 kg/s. Bulk density measurements were fit as a function of vertical consolidating pressure (σ_1) in kPa as

$$\rho_b = 2128\left(1+\frac{\sigma_1}{49.53}\right)^{0.00626}$$

Wall friction testing was conducted over a range of temperatures from 20-800° C and consolidating pressures from ~5-100 kPa. Figure 2 top shows the wall friction angle (ϕ') measurements on refractory and steel surfaces at constant pressure (1-2.5 kPa) vs temperature. Figure 2 bottom shows the same parameter at fixed temperature (800° C) across the range of pressures. The effective angle of friction (δ) decreases dramatically above 550° C and was found to range from 33° at 20° C, to 30° at 550° C to 18° at 800° C which effectively requires higher, steeper discharge cones for mass flow beneath the heat exchanger. Similarly the kinetic angle of internal friction (ϕ) decreases with temperature from 33° at 20° C to 30° at 550° C to 24° at 800° C. Further characterization at high temperature is recommended for future work.

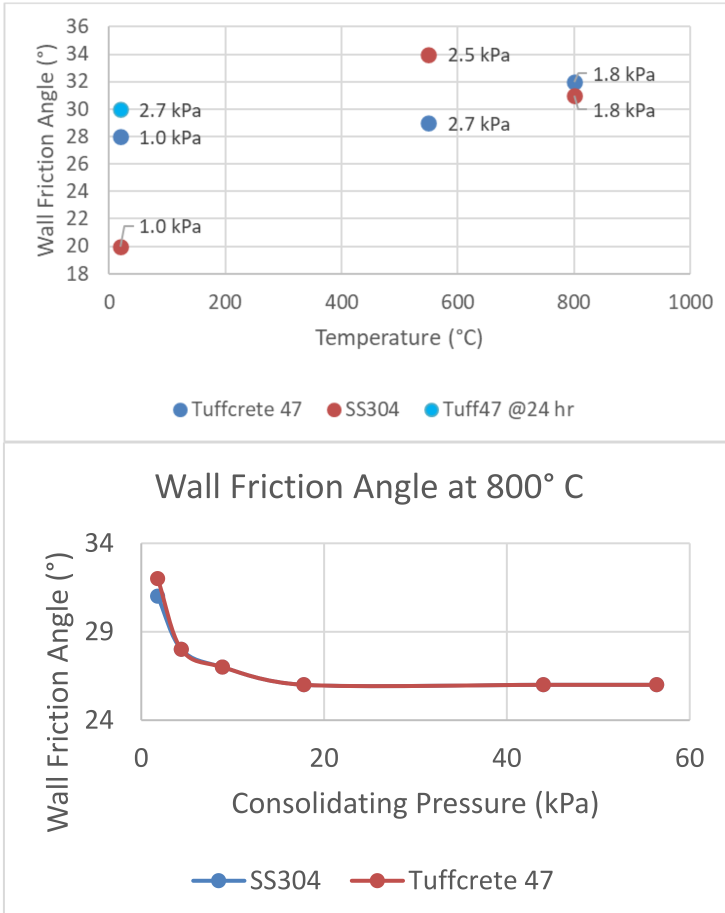


FIGURE 2. Top: Wall friction angle measurements of CARBO HSP 40/70 on a refractory and steel surface. Bottom: Wall friction angle at constant 800° C temperature at increasing consolidating pressure.

CHUTES

The acceleration (a) of particles in chutes of incline angle (α) from horizontal is a function of the wall friction angle $a = g(\sin(\alpha) - \cos(\alpha)\tan(\phi'))$. Thus α must always be greater than ϕ' to maintain non-zero velocity but shallow enough to limit height and maximize the horizontal component over the travel distance. Particle velocity was calculated as $V = \sqrt{V_o{}^2 + 2aS}$ where S is the length of each segment and the resultant forces on connecting chute members was calculated as $F = m\Delta V$. Thus in a 25 m segment (diameter of a tower) at 45° would have a resultant force of ~80kN which is less than the shear strength of a 5/8th bolt.

Erosion rates also increase with normal force, favoring faster thinner flow [6]. Figure 2 shows how wall friction increases with temperature and decreases with consolidating pressure. J&J reported a time dependent effect at low temperatures such that in the case of cool particles inside a refractory chute, an increase wall friction may be expected if particles remain stagnant over a 24 hour period. Furthermore, wall friction angles decrease with increasing consolidating pressure with the highest angles corresponding to a flow depth of ~5 cm. Chute designs herein are assumed to have an angle of 40° to provide margin. Future work should evaluate the flow properties of degraded particles that have undergone attrition.

A chute diameter (D_{ap}) can be selected using the principles of Beverloo et al [7]. A closed form of the Beverloo equation for a given mass flow rate $\dot{m}$ where C_1 and C_2 are dimensionless constants related to the material properties and g is the gravitational acceleration constant can be used to estimate the required diameter of an aperture through which particles of a certain type can flow.

$$D_{ap} = C_2 d_p + \sqrt[1/n+0.5]{\frac{\dot{m}}{C_1 \rho_b g^{0.5}}}$$

When the velocity of the sliding particles change as they flow down the chute, the cross-sectional area will change. Jenike and Johanson advised that a chute only be 1/3 full at the point of minimum velocity to ensure unobstructed flow [8]. The chute diameter is estimated herein as $D_{chute} = \sqrt{3}D_{ap}$. Thus at 2400 kg/s a chute diameter of 1.2 m would be sufficient.

Figure 3 shows as detailed section of the hot particle chute above the hot storage bin in the G3P3 system design. The length must be divided into several segments connected by thermal expansion joints. While the details of each chute design are unique, minimizing the moving overlap of each gap is necessary to avoid openings in the chute and avoiding collisions of expanding segments.

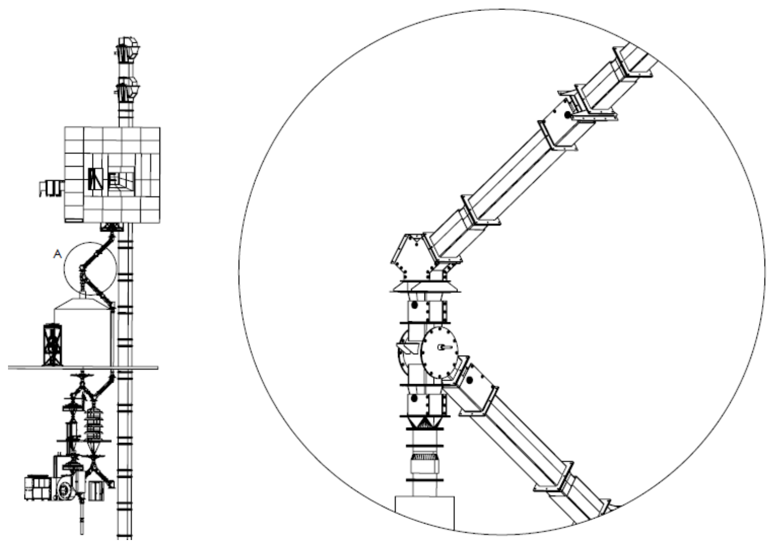


FIGURE 3. Detail drawing of G3P3 ductwork segment above hot storage bin (K. Albrecht)

Cost models in this work assume custom fabrication and assembly of a similar number of segments per vertical meter. The cost of steel chutes is based on a quote for custom fabrication provided to the G3P3 project and a professional estimate of labor and assembly for the section shown in Figure 3. Material costs in G3P3 were 1500-3400 $\$/m_{vertical}$. The high variation is a result of differing complexity for a given section throughout the system. Scaling of material costs can be relative to a 0.2 m diameter schedule 40 pipe. Installation costs were estimated at 2500 $/m and include the assembly and fixturing of steel members, cutting and placing insulative fiber gaskets between members, wrapping with fiber wool, and cladding with aluminum. These estimates are for a single pilot plant and do not reflect any cost reductions for subsequent builds. Future work should look at parametric guidelines for chute joints and segment design.

Abrasive wear, defined as damage to a surface from the relative sliding motion of a second surface has been studied for CARBO HSP 40/70 on SS316 and results indicate a specific wear rate of 1.99E-2 $mm^3\ N^{-1}\ m^{-1}$ [6]. This could indicate a need to replace chute segments approximately every 3 years for continuous operations at commercial scale capacities. Future work should study the abrasive wear of actual chutes used in hot particle systems such as the falling particle receiver at the NSTTF or in G3P3 to see whether actual surface interaction behaves similarly to the experimental configuration. Real chutes have stratification of temperatures, normal force, and particle velocity that may lead to different results. The specific cost of chutes is similar that of skips when capital replacement every 3 years is factored. The use of skips in lieu of chutes is discussed below.

Underground chutes are necessary to deliver particles from the cold storage bins to the primary receiver skips. These chutes were not investigated in depth but cost estimates for excavation and horizontal drilling were obtained from the Society of Mining Engineers. In the oil industry, shafts of similar diameter are common and are cased with high-grade steel to prevent the annulus from collapsing. The installation cost was estimated to be 1300 $/m. In addition to the unit cost, there is a preproduction or set-up cost. This cost is documented at ~$500k for large scale mining operations but it is unknown whether small chutes would cost proportionally less. As an alternative, the cost of excavation is ~150 $\$/m^3$. However, excavation costs may require not only the chute volume but the volume of required for the egress of excavation equipment. In addition to the cost of casing, refractory liners may be necessary to minimize heat loss. Refractory material costs have been quoted at 2700 $\$/m^3$.

While specific costs are low for chutes, heat losses may offset those savings relative to other options such as skips. Testing was performed at the NSTTF with two objectives: 1, provide model calibration data of bulk particle temperature drop over a length of chute. 2, evaluate the stratification of particle temperatures over the depth of the flowing body. The test chute was 2 m in length and located between the outlet of the Old's elevator and the falling particle receiver at the Falling Particle Receiver test module at the NSTTF (Figure 4).

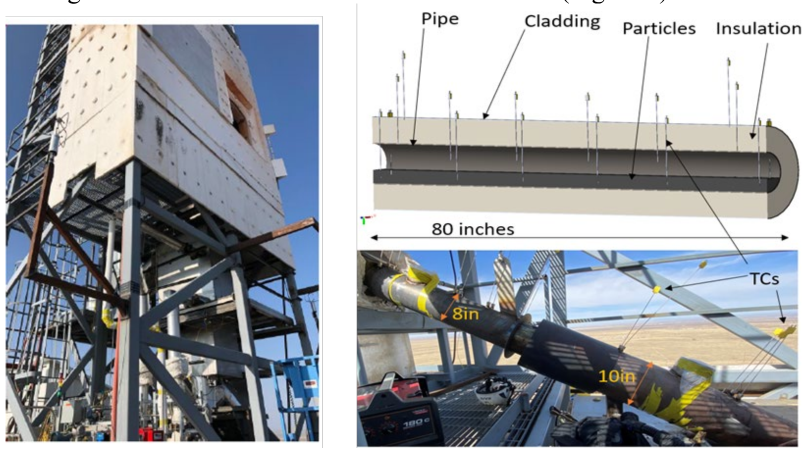


FIGURE 4. Right. Falling Particle Receiver and SuNLaMP test facility at the NSTTF. Left-top. Diagram of chute with thermocouple locations. Left-bottom. Image of test chute.

Figure 5 shows the results from the chute test. The blue and yellow lines represent the TCs at the inlet and outlet of the chute, respectively. The green curve on the right ordinate shows the measured ΔT. The dashed red line (right ordinate) shows the predicted temperature drop from the model. Because the particles were coming from the particle elevator, the temperature was not independently controlled. In the most stable temperature region, there was a ~0.4 °C drop over 2m. There is a 0.75% error associated with K-type TCs which is greater than the measured delta making model calibration inconclusive, but the consistency of the delta between the inlet and outlet gives confidence that the drop is real and informs the approximate scale of the expected drop.

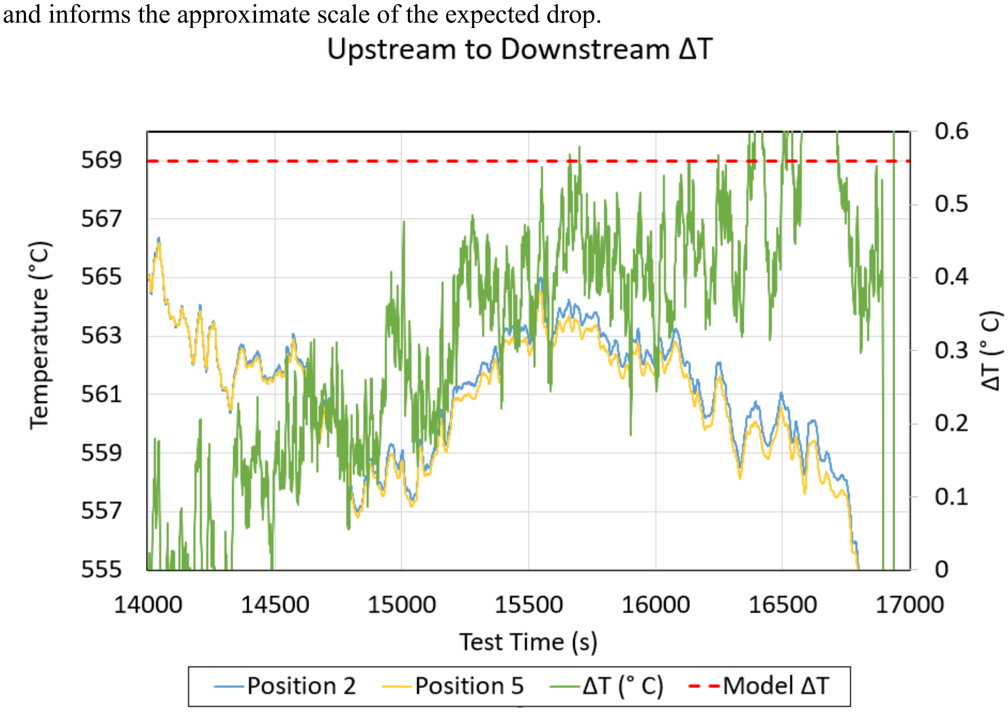


FIGURE 5. Temperature measurements in chute.

The vertical stratification test set-up and results are shown in Figure 6. TCs were spaced in a vertical array from the chute wall to the top of the expected flowing bed at intervals of 1.25 cm (0.5 inch). A measurement was also taken in the air pocket above the bed and at the upper wall of the chute. Results show that the steel chute system took over 2 hours to approach equilibrium. At the quasi-equilibrium point, there was a 32 °C difference between TCs at the chute interface and top of the flowing bed which were spaced 38 mm (1.5”) apart. There was an assumed 3 mm accuracy in TC placement.

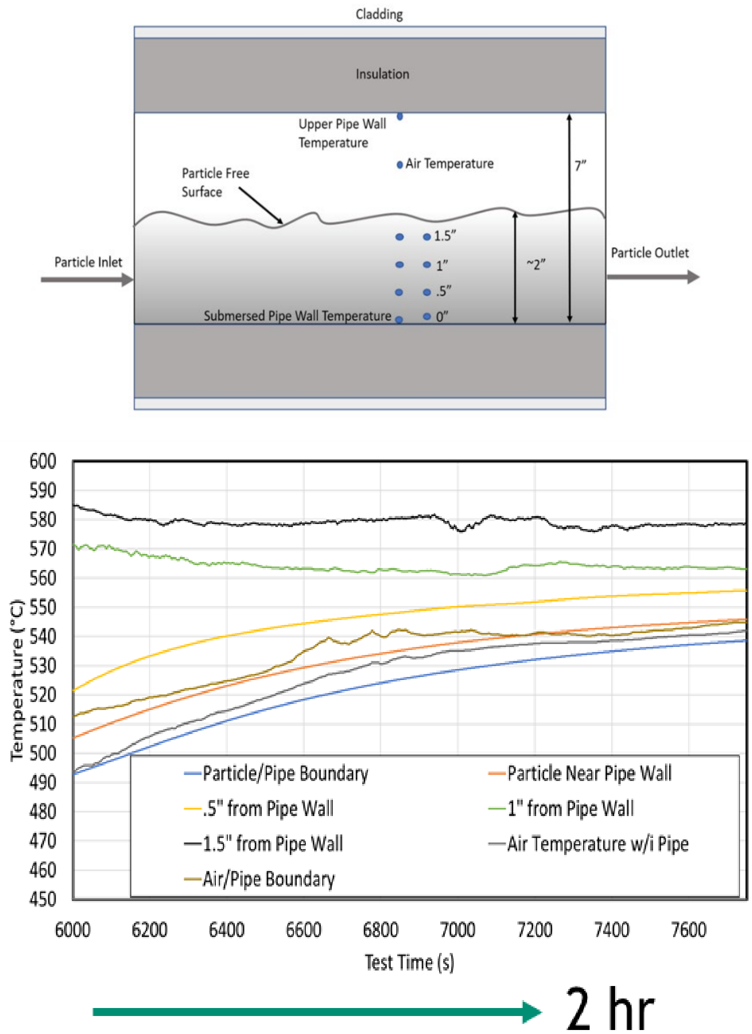


FIGURE 6. Top: position of thermocouples inside chute. Bottom: Vertical stratification of particle temperatures in a flowing bed

A Lumped Capacitance Thermal Model (LCTM) was used to predict heat losses in chutes from the tower to the hot storage bin and from the cold storage bin to the main receiver skips (Figure 1). The length of the model chute was divided into ten control volume cells. The LCTM was applied to each cell such that energy was conserved, and the enthalpy, kinetic energy and potential energy were recalculated at each control volume cell.

The particles were divided into two vertical layers. The top layer was assumed to have a flat surface area profile parallel to the flow of the particles. This assumption simplified the radiation and convection analysis between the particles, the air, and the convex surface of the upper part of the chute. The second or lower layer of the particle was assumed to have a concave surface area parallel to the flow of the particles and in full contact with the lower portion of the chute. A cross sectional view of this layout is shown in Figure 7. The volume of air within the chute was assumed to be isobaric. Test data helped inform that given the magnitude of the temperature drop, the dominant mode of heat transfer from particle to air in the chute is more likely to be conductive than convective.

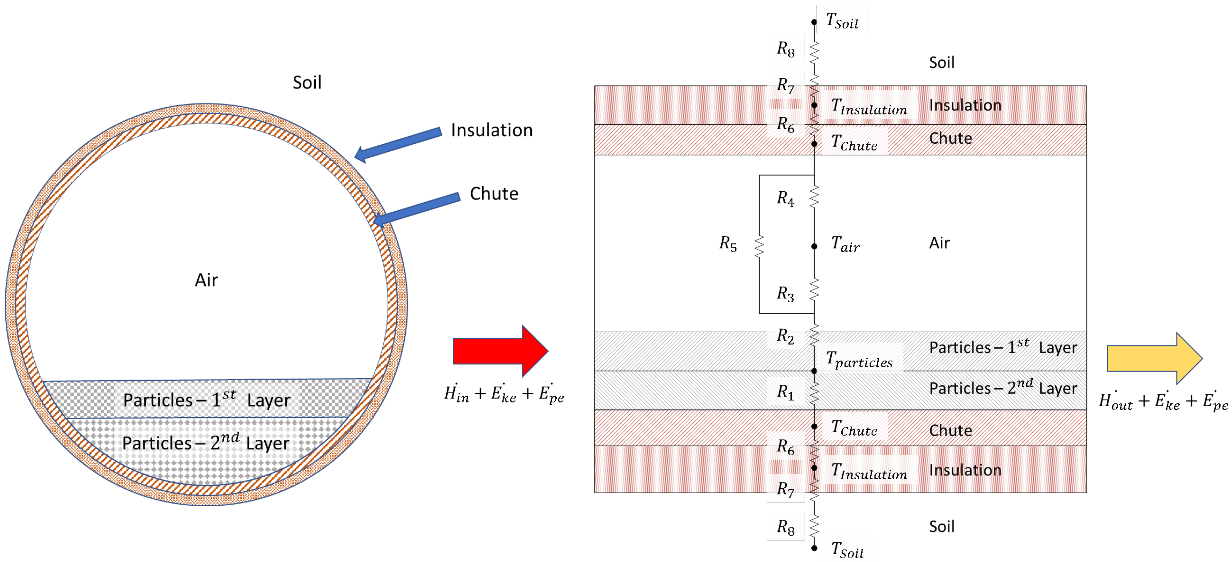


FIGURE 7. Vertical stratification approach used in lump capacitance model for flowing particles in chute.

HORIZONTAL CONVEYORS

This work included a limited survey of mechanical conveyance systems including drag conveyors, belt conveyors, pan conveyors and screw conveyors. High temperature conveyance systems exist in industry, but consultations indicated that it was somewhat novel to convey high-temperature solids without cooling. Many vendors indicated that conveyance of materials >1000° C is common, current technology does not provide for temperatures above 600° C in an insulated environment. Scaling to 1000 kg/s would be a significant technical leap for all systems and numbering of systems in parallel is necessary for the 100 MW_e baseline case. Some vendors mentioned heat loss at the discharge points was higher than expected. Future work is recommended to evaluate heat loss at discharge points and to fully evaluate pneumatic and vibrational methods which were not investigated here.

Magaldi Power S.p.A. was downselected to provide a detailed design for an *Ecobelt* conveyor that uses a steel mesh belt with stainless steel interlocking pans. The

The belt can operate in an insulated chamber with an operating temperature of 615±25° C (maximum 650° C). The largest *Ecobelt* has a normal operating capacity of 500 kg/s (1800 mton/hr, max 2000 mton/hr). Figure 4 left shows the general design of the belt. Only the large rollers at the two ends require cooling thus the intermediate rollers do not contribute as significantly to thermal losses in the particles. The middle image shows the degrees of freedom of the interlocking pans which are bolted to the chain mesh and can translate freely. The mesh belt has an advantage over chains as it can endure significant damage before needing to be replaced. The right image shows a 1000 kg/s 615° C insulated system comprised of two of the largest available units (2 x 500 kg/s). Particles are conveyed in the larger upper portion while the smaller spillage retrieval conveyors run below in the opposite direction.

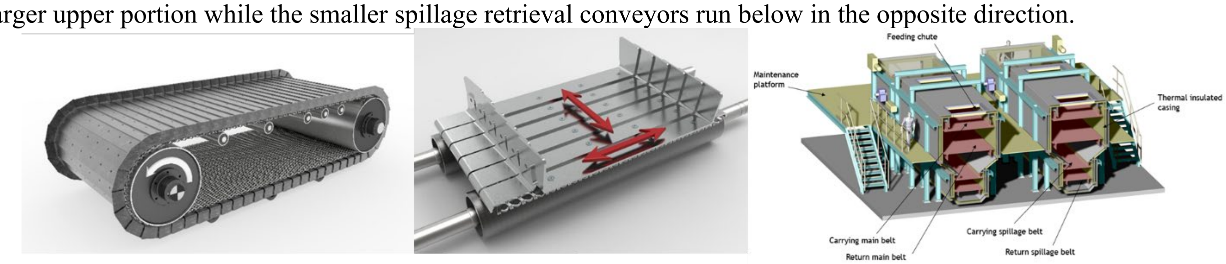


FIGURE 8. (a) *Ecobelt*. (b) detail of thermal expansion degrees of freedom of interlocking pans on mesh belt. (c) design of parallel *Ecobelt* for 1000 kg/s.

The envelope of a single conveyor rated for 500 kg/s is 4.2 m in height by 5 m in width. The belt is 2 m wide. The weight of each conveyor is 300 kN for the head section, 65 kN/m for the middle section and 520 kN for the tail. At this capacity, the system uses 71kW of electrical power. Total heat loss for the 500 kg/s conveyor at 35 m length is 1.09 MW with a particle temperature drop of 1.8° C. A 100 kg/s *Ecobelt* loses 0.58 MW over the same distance with a particle temperature drop of 4.6 ° C. Costs are estimated in Table 1.

TABLE 1. Capital costs from *Ecobelt* study

Model Plant Size (MW_e)	Material Flow Rate (mton/hr)	Double-Decker Ecobelts	CAPEX ($M)	Cost/m (head/tail + mid)
10	365	1 x 365 t/h	2.33	610E3 + 57.3E3/m
50	1800	1 x 1800 t/h	3.69	920E3+81.4E3/m
100	3600	2 x 1800 t/h	6.97	unspecified
150	5400	3 x 1800 t/h	10.3	unspecified

Maintenance schedules require replacement of the whole belt once every 5 years and replacement of the motor and belt uptake system every 4 years with belt plate and idler replacements required once a year. The reliability is 99.5%.

STORAGE BINS

Particle temperatures were limited on horizontal conveyors to <640° C creating a challenge for distribution over the bank of heat exchangers. A raised bin above the heat exchanger could provide a method for direct feed. Cost comparisons were performed for elevating the hot storage bin and using a heat exchanger hoist system with feed hopper. (Skip costs are discussed below). For storage options, an underground bunker was considered which would have provided ground support for a large portion of the bin. In the downselection phase, experts asserted that the retaining wall required would be as substantial as any above ground support structure, the excavation volume of 14250 m^3 would cost over $2M, and contractors preferred the accessibility of an above ground vessel for constructability. The raised storage bin option was chosen for further investigation (Figure 5). Future work to fully evaluate the alternative storage configurations is recommended.

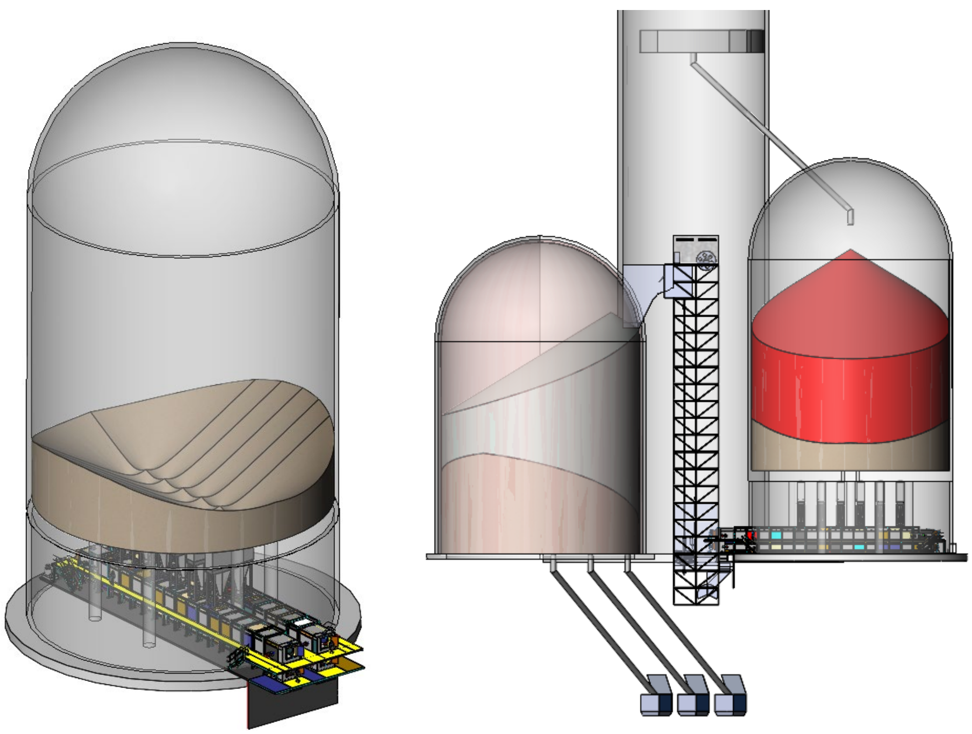


FIGURE 9. (a) Concept model for vertically integrated storage (shown fully discharged), heat exchanger, and horizontal conveyor. (b) System context with conveyor interfacing with cold storage lift.

The height of the heat exchanger is based on the commercial scale shell and plate design. The height of the 16 MW_{th} heat exchanger was 3.5 m arranged in a bank of 16. The heat exchanger design requires mass flow which requires mass flow cones that are approximately 3 m in length given the steep mass flow angles. J&J test results show that a minimum hopper angle of 12° from vertical is required for mass flow in a conical hopper vs a minimum angle of 23° from vertical in a wedge hopper. Future work should evaluate alternative geometry for minimizing the height of mass flow cones in a heat exchangers. The cones must transition from the half the width of the heat exchanger ~1.5 m to the radius of the control valves. These valves are not yet fully designed but may be assumed to be ~0.25 m in diameter. Thus a conical flow cone would need to be ~6m while a wedge would need to be ~3 m. If a conveyor (assumed to be 4.2 m in height) is placed under the heat exchanger to move particles to the cold storage lift then the storage bin must be lifted 11-14 m.

FEA analysis was performed to estimate the material cost of elevating the storage bin. Results to reach a factor of safety of 2 are shown in Table 2. A cost for an equivalent skip hoist is shown as a basis of comparison. An optimization study is recommended for future work to determine the minimum material design for the storage bin. No solution could be found for bin floor surfaces above 100° C so ample flooring material or floor insulation is necessary. Previous studies indicate floor temperatures of 300° C may be expected at cyclic steady-state conditions.

TABLE 2. Support structure for elevated storage bin

Model Plant Size (MWh_{th})	Storage and Flooring Mass (ktonne)	Pillar Diameter (m)	Number of Pillars	Bin Floor Thickness (m)	Cost of Elevated Bin* ($M)	Cost of HX Hoist* ($M)
120	2.2	0.75	4	0.75	0.32	0.36
1400	27.7	1	18	1.52	1.05	1.11
2800	50.7	1.5	21	1.83	2.03	2.70
4200	76.1	1.5	32	2	3.08	4.74

*costs include materials, labor, and installation

In addition to raising the bin, storage bin designs must consider the additional stress and less efficient formation of the bulk particles caused by asymmetric charging and discharging. Figure 5 left shows the asymmetric drawdown cones that would form over the heat exchanger array. Figure 5 right shows the formation of particles that would form if the cold storage bin were to be discharged from the edge closet to the skips vs a central outlet. Furthermore, the hot or cold storage might be located in a position conducive to charge from one side.

ACI 313-16 states that as the number and size of flow channels of multiple-outlet silo increase, the pressures in the static material and the resulting moments increase significantly [9] [10]. The wall design point is generally increased by the eccentricity/radius x 25% of the static pressure. The increase in materials for the 100 MW scale storage bin would be 27,200 kg of rebar ($36,000 material + $21,000 labor) plus 92 m^3 of concrete ($33,700 material + $2,640 labor).

SKIP HOISTS

Skips were evaluated as an alternative to chutes (as is done for both G3P3 systems) for lowering particles from the receiver to the hot storage bin. The use of chutes may be problematic at commercial scale mass flow rates particularly if particle temperatures are at 1000° C. Figure 10 shows two system configurations under consideration for commercial scale particle plants. For systems with tower-integrated storage, a hoist can be placed on the deck behind the receiver and skips can be used to lower particles to the internal hot storage bin. Particles would still be lifted from a primary hoist located outside the tower. This presents an opportunity to add an alternator and flywheel and generate electricity with the downcoming skip. In order to prevent spillage, overturning skips are recommended in CSP applications. However, the internal skip can be a simpler and lower-cost bottom-dump design assuming spilled particles fall onto the storage bin and can be funneled back into the system.

For systems with external storage, the interior of the tower may be empty or at least provide a path for interior skips. A funicular design was investigated where the skip discharges to the receiver, then stops below the receiver to refill, then discharges to the hot storage bin, then refills under the cold storage bin. The additional time for each stop adds to the size the skip must be. Skips maintain a small cross section (3m x 3m) in order to minimize the steel structural support and the hoist reel width (one of the most expensive components). Thus, for a mass flow rate of

2400 kg/s with four stops, a 20 m skip would be necessary. Overturning at this size may be unprecedented and a two-hoist system may be necessary.

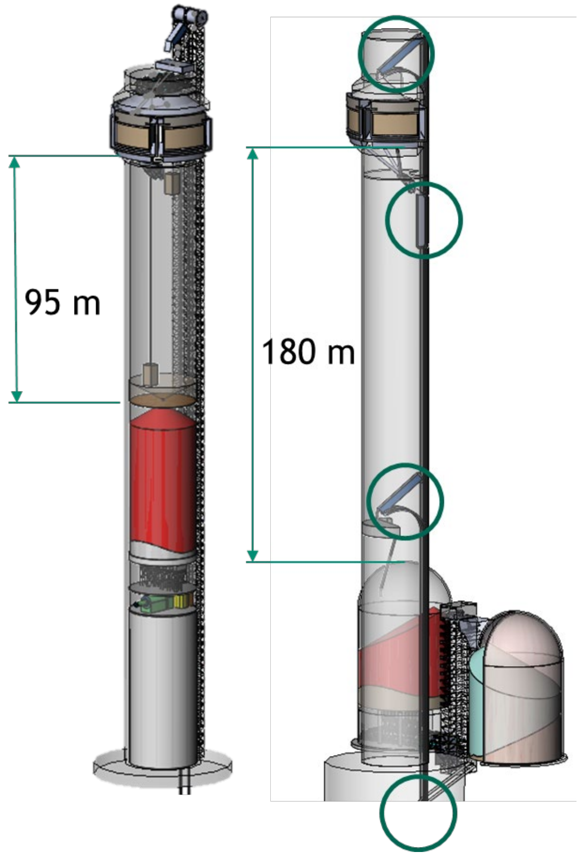
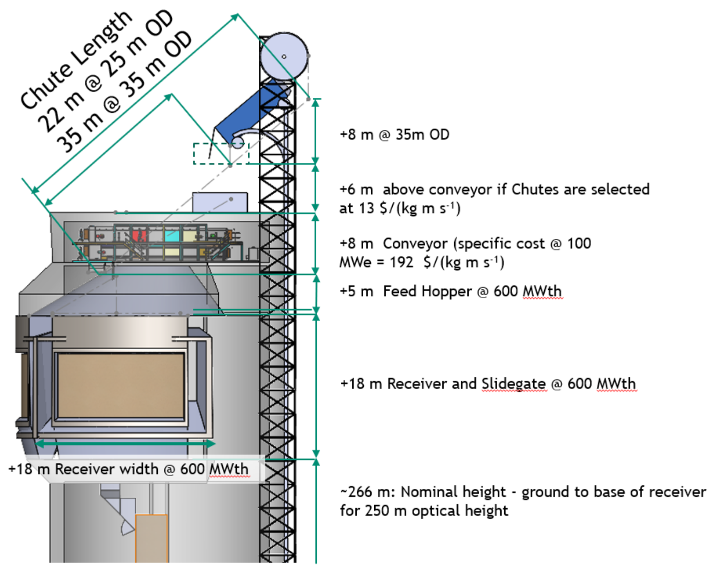


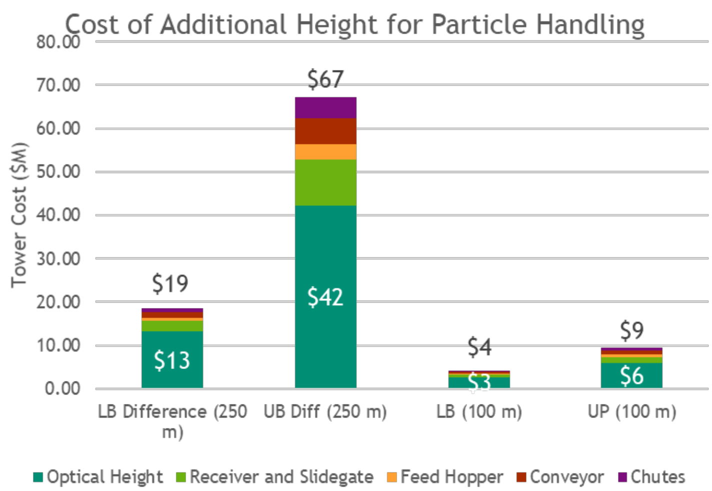
FIGURE 10. (a) Internal bottom-dump skip with hoist located at top of tower. (b) Funicular skip with four charging/discharging stations circled.

TOWERS

Tower cost models developed for molten salt towers are typically a function of optical height (distance from the heliostat elevation axis to the centroid of the receiver aperture). Since there are no known comprehensive studies for particle-based towers, the molten salt models should be adjusted for additional height above the receiver. Figure 11 (a) shows the sources of additional height that should be considered. Firstly, the height of the receiver plus the feed hopper. There must be a method to move particles from the skip discharge point to the receiver. If this is a conveyor a 6-8 m conveyance room (including 0.481 m decks) may be assumed. For receiver mass flow rates greater than 1000 kg/s, there may not be room on the roof to number up the conveyors. Thus chutes may be required. Tower diameters of 25 meters would require 14 additional meters to support a chute incline of 40°. Towers with integrated storage could be 35 meters in diameter if storage capacities were ≥2800 MW_{th} requiring 14 vertical meters to the skip discharge point. Future work to develop a particle-based tower cost model is recommended to avoid the uncertainties in the feasibility and costs associated with superimposing particle-based applications on molten salt-based models.



(a)



(b)

FIGURE 11. (a) Additional height considerations for particle-based receiver towers. (b) Associated cost for increased heights with a lower and upper bound cost model.

COST ANALYSIS

A cost study was performed on a single-tower 100 MW_e system with both external (Table 3) and tower-integrated TES (Table 4). The overall cost is ~10% higher for the external TES configuration due to the opportunity to move particles by force of gravity in components that are tower integrated. However, these savings are largely offset by the substantially higher tower costs that are required to support the weight of the storage and other components. Tower-integrated systems also have lower O&M costs in part because there are less electrical systems and skip with regenerative electrical systems is assumed to offset parasitic losses. Chute replacement costs are not factored pending further investigation of abrasive wear that will be performed in G3P3. Heat losses are also higher in the external TES configuration due to the conveyor and chutes. Future work should investigate heat losses in a tower-integrated system more thoroughly and heat loss models should be refined.

TABLE 3. Cost analysis of a 100 MW_e system with external TES

External TES Configuration						
Route	**CAPEX ($M)**	**O&M/yr ($M)**	**Electricity/yr ($M)**	**Peak Electrical Power (kW)**	**Particle ΔT (° C)**	**Heat Loss (kW)**
Chutes from Primary Skip to Receiver and Tower Height increase	$14.60	$0.00	$0.00	0	1.05	114
Skips to Receiver	$22.57	$1.48	$0.89	18	0.059	196.1
Hot Storage Bin Modification	$2.57	$0.00	$0.00	0	5	213
Ecobelt	$6.97	$0.14	$0.09	142	1.8	2188.0
Cold Storage Lift	$5.78	$0.33	$0.58	3300	0.0494	55.4
Cold Storage Bin Modification	$1.32	$0.00	$0.00	0	3.5	148
Chutes from Cold Storage to Primary Skip	$5.73	$0.00	$0.00	0	0.28	24
Total ($M)	$59.52	$1.95	$1.55	3460	11.74	2577
Grand Total ($M)	$59.52 + $3.51 * yr					

TABLE 4. Cost analysis of a 100 MW_e system with tower-integrated TES

External TES Configuration						
Route	**CAPEX ($M)**	**O&M/yr ($M)**	**Electricity/yr ($M)**	**Peak Electrical Power (kW)**	**Particle ΔT (° C)**	**Heat Loss (kW)**
Chutes from Primary Skip to Receiver and Tower Height Increase	$26.80	$0.00	$0.00	0	0.22	23.78828571
Skips to Receiver	$19.17	$1.13	$1.60	19	0.0395	120
Hot Storage Bin Modifications	$0.00	$0.00	$0.00	0	5	213
Ecobelt	$0.00	$0.00	$0.00	0	0	0
Internal Downcomer Skip	$8.73	$0.69	($0.76)	1745	0.0231	80.2
Cold Storage Bin Modifications	$0.00	$0.00	$0.00	0	3.5	148
Chutes from Cold Storage to Primary Skip	$0.09	$0.00	$0.00	0	0	0
Total ($M)	$47.05	$1.82	$0.85	1764	8.78	224
Grand Total ($M)	$54.79 + $2.67 * yr					

Table 5 shows a breakdown of the costs by major subsystem.

TABLE 5. Cost breakdown of conveyance components by sub-system

Subsystem	CAPEX External TES	CAPEX Tower-Integrated TES
100 MWe All Particle Handling ($/MWe)	$595,000	$548,000
560 MWth Receiver ($/MWth)	$77,000	$82,000
200 MWth Heat Exchanger ($/MWth)	$0	$0
2800 MWh Storage ($/MWhth)	$6,000	$3,000

CONCLUSIONS

Additional tower height is very costly and estimating these costs has high uncertainty given the spread in existing salt tower cost models. Efforts to adapt these models to novel particle-based applications assumes the feasibility for uses that were not originally intended. Future work to develop particle-based tower cost and structure models is encouraged. Chutes offer the lowest cost means of lowering particles. Velocity calculations indicate a 40° incline switch back method would result in controllable forces at wall connections. Erosion may be exacerbated at 100 MW flow rates (3000 kg/s) at temperatures >800° C. Temperatures >1000° C may require refractory chutes. Skip concepts add complexity, but were evaluated as a risk mitigation for chutes. Electricity offsets from funicular and internal skips with alternators were found to be likely to offset higher skip capital costs at the midlife of the plant. Hot bin temperatures exceed the limits of the conveyors explored in this study. The hot storage bin could be raised 12-14 m to accommodate the heat exchangers, funnel flow cones, and conveyor potentially costing less than alternatives that use an additional hoist and chute system. Concrete is significantly weaker at high temperatures. FEA results show a significant quantity of supports would be needed and refined models that factor steel reinforcement are recommended as future work. Direct feed to the heat exchanger results in asymmetric inventory in the bin which requires taller bins to accommodate inefficient packing formation and extra reinforcement in the walls per ACI code. Horizontal conveyors can be used at temperatures <640° C but insulating the payload is not typical for the industry and mechanical cooling systems may be incompatible with CSP heat loss goals. Magaldi provided an insulated design with a specific cost of ~200 $/(kg m s-1). Peak capacities were found to be 1800 tonne/hr and can be numbered up to reach higher system capacities. The specific cost formulas for the cold storage lift should be higher than the primary receiver lift. Specific costs for shorter skip routes are proportionally higher in cost and O&M costs associated with the cold storage lift need to be increased because they operate continuously. Underground refractory chutes may be 50%-100% of insulated steel chute costs if horizontal drilling methods can be adopted. Cost models assumed that automated bore casing technologies could line concrete chutes with steel and insulative refractory materials. Excavation is a more costly alternative.

ACKNOWLEDGMENTS

Sandia National Laboratories is a multimission laboratory managed and operated by National Technology & Engineering Solutions of Sandia, LLC, a wholly owned subsidiary of Honeywell International Inc., for the US Department of Energy's National Nuclear Security Administration under contract DE-NA0003525. SAND No.

This material is based upon work supported by the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) under the Solar Energy Technologies Office Award Number DE-37368.

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