Proceedings of the ASME 2016 10th International Conference on Energy Sustainability
ES2016
June 26-30, 2016, Charlotte, North Carolina

ES2016-59619

DESIGN AND ANALYSIS OF A HIGH TEMPERATURE PARTICULATE HOIST FOR PROPOSED PARTICLE HEATING CONCENTRATOR SOLAR POWER SYSTEMS

Kenzo K. D. Repole
Georgia Institute of Technology,
Atlanta, GA, USA

Sheldon M. Jeter
Georgia Institute of Technology,
Atlanta, GA, USA

ABSTRACT:

The central receiver power tower (CRPT) with a particle heating receiver (PHR) is a form of concentrating solar power (CSP) system with strong potential to achieve high efficiency at low cost and to readily incorporate cost-effective thermal energy storage (TES). In such a system, particulates are released into the PHR, and are heated to high temperature by concentrated solar radiation from the associated heliostat field. After being heated, the particles will then typically flow into the hot bin of the TES. Particulates accumulated in the hot bin can flow through a heat exchanger to energize a power generation system or be held in the hot TES storage bin for later use such as meeting a late afternoon peak demand or even overnight generation. Particles leaving the heat exchanger are held in the low temperature bin of the TES.

A critical component in such a PHR system is the particle lift system, which must transport the particulate from the lower temperature TES bin back to the PHR. In our baseline 60 MW-thermal (MW-th) design, the particulate must be lifted around 70 m at the rate of 128 kg/s. For the eventual commercial scale system of a 460 MW-th design the particulate must be lifted around 138 m at the rate of 978 kg/s. The obvious demands on this subsystem require the selection and specification of a highly efficient, economical, and reliable lift design.

After an apparently exhaustive search of feasible alternatives, the skip hoist was selected as the most suitable general design concept. While other designs have not been dismissed, our currently preferred somewhat more specific preliminary design employs a Kimberly Skip (KS) in a two-skip counterbalanced configuration. This design appears to be feasible to fabricate and integrate with existing technology at an acceptably low cost per MW-th and to promise high overall energy use efficiency, long service life, and low maintenance cost. A cost and performance model has been developed to allow optimization of our design and the results of that study are also presented. Our developed design meets the relevant criteria to promote cost effective CSP electricity production.

INTRODUCTION

A particle heating receiver (PHR) system is a form of the familiar Central Receiver Power Tower (CRPT) plant for concentrator solar power (CSP) production that employs a particulate rather than a fluid for energy collection. In a generic CRPT, concentrated solar radiation is, by definition, absorbed by the collection medium in the central receiver; and in a PHR system, the collection medium is a falling curtain or layer of particles rather than a fluid. A major advantage that PHR systems have over other forms of CSP is the ability use low cost materials to collect and store sensible thermal energy over a longer period of time in a Thermal Energy Storage (TES) subsystem. Storage allows off-sun and nighttime generation, and the resulting extended use of the energy plant reduces the levelized cost of energy (LCOE) [1].

As shown in Figure 1, the particulates are heated in the PHR and then will then typically flow into the hot bin of the TES and be accumulated and possibly stored for some time in this bin. These particulates can next flow through a heat exchanger, the Particle to Working Fluid Heat Exchanger (PWFHX), to transfer energy to a thermal power generation system. The power conversion system will likely be a steam cycle, supercritical carbon dioxide Brayton cycle, or some other more conventional Brayton cycle. Stored hot particles can be used to compensate for the random variations in demand or solar input experienced during the day or is held to meet an expected late afternoon peak or even be held long enough for nighttime or even overnight generation. Particles leaving the heat exchanger are held in the low temperature bin of the TES until needed for energy collection. To have several hours of TES, the TES and the PWFHX will occupy a considerable vertical distance.

Larger capacity PHR systems with substantial TES will, with reference to Figure 1 for illustration, necessarily occupy much of the power tower for the vertical array of PHR, hot bin, PWFHX, and the low temperature bin. Consequently, the particle lift system must be highly efficient to minimize the parasitic load on power generation and be cost effective to facilitate a competitive price for the electric energy produced.

Currently, there are several potential candidate methods for elevating fine particles to the PHR. However, to increase the capacity of the power plant and its efficiency, the particles entering the PHR need to be at higher temperatures ranging from 300°C (572°F) up to 600°C (1112°F). At such temperatures some conventional methods of delivery of large amounts of working particles are not viable since the operating environment is outside their feasible operating range.

This paper discusses the development of a conceptual design of a lift systems for first a mid-sale 60 MW-thermal (MW-th) demonstration unit and secondly the preliminary design of a commercial scale 460 MW-th design which is on the scale of other commercial thermal conversion plants. The result of these design processes is a candidate design, which is described in some detail in the following. It notable, that the lift investigated herein is distinctive for two features (1) high temperature operation and (2) a lift midway between an industrial application and a mining application, wherein the skips travels on the order of 100 meters rather than multiple 1000 meters as in conventional mining applications.

OVERVIEW OF RELEVANT LITERATURE

The scope and operating conditions for the PHR particle life are rather unique with a vertical lift greater than a typical industrial lift but shorter than the usual mine hoist. The elevated operating temperature, since even the “cold” particles are likely to be at least 300 C if not much hotter for high efficiency thermal conversion and industrial process heat applications. The efficiency, reliability, and cost effectiveness demands are also challenging. Consequently, the relevant literature and practical technology directly addressing this application is sparse. Nevertheless, as detailed and demonstrated below the topical sections the basic technology and even adaptable designs are reasonably well described, at least at the fundamental level, in the existing literature. For example, as described in [2] suitable design alternatives exist and one traditional design was found to be especially appropriate. Furthermore as outlined in [14], systematic design procedures were helpful in selecting the preferred design approach from consideration of the design requirements and the technological alternatives. Cost estimation, is particularly important in this application, since the unique application requires and essentially new design with no current commercial alternatives with known costs. Nevertheless, existing data (such as [17] through [20]) and newly acquired and adapted information (such as [22]) was found adequate to support a preliminary cost analysis. Finally, no suitable efficiency model is published in the current literature, so as described in the following section a model was developed based on available information cited in the pertinent section. The other relevant literature is also cited as necessary in the following topical sections.

FUNCTIONAL REQUIREMENTS

For our initial design a 60 MW-th capacity was considered. This thermal capacity should be adequate for a mid-sized demonstration facility using a commercially available gas turbine engine thought to be compatible with an external heat exchanger replacing its normal combustor. For such a demonstration plant, a particle transport solution was needed to deliver particulates up to a receiver height of 138 m in the CRPT with a design lift of about 70 m. The height of the lift is determined by the size of the TES bins used to store the particulate for use during the day for the assumed 9 hour off-sun period. Based on this information and other constraints, the first level of Functional Requirements (FR) was developed, and these requirements are shown in Table 1. These FR were used as selection criteria for the initial round of concept generation and concept selection. Subsequently, after defining these requirements, the design process moved on to the generation and consideration of the design options as discussed in the next section.

Table 1 First Level Functional Requirements (FR)

FR#	FR Description
FR01	Capability to transport large mass of small particulates
FR02	Compatible with shaft temperature 150 to 200°C
FR03	Adaptable to design with minimal heat loss
FR04	Capability to minimize particulate spillage.
FR05	Ability to resist expected erosion and wear
FR06	Potential to achieve overall energy efficiency > 75%
FR07	Compatible with rail and or ship containers
FR08	Adaptable to reasonable structural safety factor

PARTICLE TRANSPORTATION OPTIONS

Some of the options that are available to meet this challenge are bucket elevators, Olds Elevators, conveyer belts, and skip hoists similar to those used in the mining industry. A preliminary investigation [2] identified and investigated several other alternatives, but these could be eliminated as evidently unsuitable or under developed.

Suitable bucket elevators may have the ability to operate at moderately high temperatures, greater than 200°C (392°F) [3]. However, it would be very difficult to insulate the individual buckets in such an elevator. Consequently, this technology would typically require the entire shaft for particle transport to be kept at the same high temperature of the particles for a reasonable heat loss, which is already 300°C in a baseline design but potentially much hotter in an advanced design. Moreover, direct observation and pertinent research, such as [4] and [5], confirms that the typical bucket conveyor would likely experience a high spillage rate during operation.

Olds Elevators (OLDS) [6], which employ a rotating outer drum to pull particulates up an internal helix by friction, have the ability to deliver the working particles in a continuous flow and at high temperature, and such units are currently in use in prototype scale PHR systems [7],[8]. However, as the height of the tower increases the cost of the OLDS probably increases at least linearly due to the nature of its design. Furthermore, the energy efficiency of such a friction based system is too low [9] to be competitive with any bucket or skip system; therefore, this option can be omitted from further consideration.

Conveyer belts can have little spillage but are difficult to integrate into a tower and not suited to convey high temperature particles without huge heat loss.

Evidently the most suitable design option, which can address the current and future needs of larger capacity PHR systems and maintain high thermal efficiency and low exergy degradation, is the skip hoist [10]. Therefore, with considerable confidence, the design process focused on the various options for efficient and cost effective skip hoists. The skip hoist system includes two obvious major subsystems, (1) the skip and (2) the winder or hoisting system. Alternatives for both can be considered independently as described in the next section.

SKIP AND HOIST ALTERNATIVES

Skips for particle lifts are similar to those used in the mining industry come in different forms. The main designs currently used in the mining industry are Bottom Dump skips, Front Dump skips, overturning or Kimberly skips, and Arc Gate skips.

The main tradeoffs between the different skips types are (1) the extra height required during operation, (2) ease of operation at high temperature, (3) spillage during use, (4) maintenance requirements, and (5) compatibility with adequate and effective thermal insulation.

Bottom Dump skips are charged from the top and discharged by a trap door forming part of the bottom of the skip. This design does not require excessive extra height for its operation in comparison to the other types of skips. This skip design is light weight and rugged; but due to the fine size, expected to be around 250 nm, of the particles used in the PHR, spillage may be large during the transport and especially during the discharge of the particles.

Front Dump skips, are charged from the top and discharged through a sliding gate forming part of the lower section of the front side of the skip, meaning the vertical side facing the particulate bin. Such skips are reportedly able to carry large volumes of particles and to put the least amount of stress on the head frame [11]. However, the spillage rate may still be high in comparison to other types of skips.

Arc Gate skips are considered safe and rugged [11]. They are charged from the top and discharged through a pivoted or “arc gate” on the side. This feature accounts for the descriptive name. As with the Bottom Dump skip, the Arc Gate skip may experience significant spillage especially during discharge. In addition, this skip has many moving parts implying an increased risk of failure or fouling under high temperature and challenging environments that would be experienced in transporting fine particles in the PHR system.

Kimberly skips (KS), are charged and discharged from a single door at the top of the skip. The particles are loaded into the skip from the top with the skip body vertical. The skip then travels in this vertical configuration until it reaches its discharge location. As it reaches the discharge location, a set of scroll wheels on the skip engages scroll guides on the shaft walls. These wheels guide the skip through the dump zone and force the skip to rotate to about 120° from its vertical position. This action discharges the particles from the top of the skip. The KS is expected to have the lowest initial cost, the lowest maintenance cost and highest service life in comparison to the other skip types [11]. KS also should have the lowest amount of spillage occurrence during use. However, KS requires larger headroom and width clearance than any other skip design. They may also exert the largest amount of stress on the head frame since the skip itself must be rotated [12], [19].

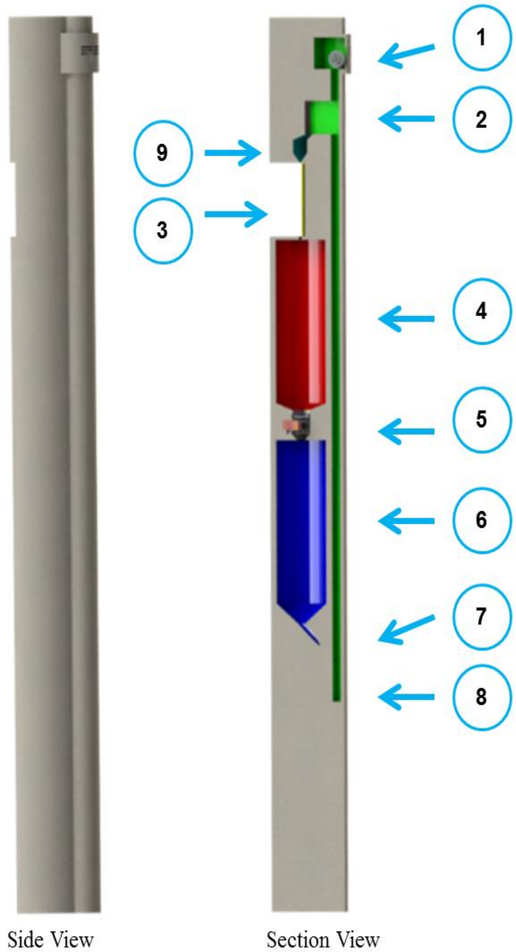
In our development, the pivoted Arc Gate dump and Kimberly skips appeared to be most promising, so scale models were developed for qualitative comparison. Operation and observation of the scale models indicated that the KS would be easiest to effectively insulate and most likely the most reliable to operate at the expected high temperatures.

The other important subsystem in the particle lift is the hoist. There are two main types of hoists [12] in common service: (1) the direct-winding drum type hoist and (2) the friction dependent Koepe hoist. In the drum hoist a single rope or multiple ropes are fixed to and wound onto a drum turned by an electric drive motor. A preferred version of the drum hoist is the Blair hoist [12] which is a design with pair of drums connected by a clutch. The other main type is the Koepe friction hoist. In this design, the rope is not connected to the drum. Instead two skips or one skip and a counter balance are operated by a rope passing around but not connected a drum, and the friction between the rope and the drum supports and lifts the skip.

The Koepe hoist is reported to be the most common hoist system used in the mining industry today [12]. It is based on using the friction between the drum and the wire rope to enable the drum to drive the skip operation. Despite its attractive low cost and wide use it was not initially selected in this project due to concerns about reliable friction-dependent high temperature operation. For example, we cannot yet discount the possibility that the heat generated from the friction in addition to the high surrounding temperature of the shaft could disrupt the operation as well as greatly reduce the life of the wire rope.

Finally, for this design a Blair Drum hoist [12] was considered. This a design with pair of drums connected by a clutch. One of the advantages of the Blair Drum is its ability to run the skips independently of each other in cases of emergency thus giving a contingent means to continue operating if one skip becomes nonfunctional.

The final more detailed selection and specification of the skip and hoist combination is described in the next section.



No.	Name
1	Lift Machine Room
2	Lift Discharge Chute
3	Particle receiver
4	High Temperature TES Bin
5	PWF Heat Exchanger
6	Low Temperature TES Bin
7	Lift Charge Chute
8	Lift Shaft
9	Top hopper

Figure 1 Overall Schematic Showing Integrated Lift [13]

REFINING PARTICLE LIFT SELECTION

To support the final selection and specification of hoist and skip for particle lift, a more detailed array of Design Parameters (DP) was generated as shown in Table 2.

. The complete design process resulting in this selection and specification is further explained in detail in an upcoming paper under preparation [14].

Table 2 Detailed Design Parameters

DP#	Description
DP01-01	Kimberly Skip Design
DP01-02	Blair Drum Type
DP01-03	AC Variable Frequency Electric Drive
DP02-01	Metal for skip is of SS 316 alloy or equal
DP02-02	Rope Lubricant with flash point over 200°C
DP03-01	Skip insulated for acceptable heat leak
DP04-01	Olds Elevator in sump to recover spillage
DP05-01	Loading and Unloading angles greater than 30 degrees
DP06-01	Overall Lift efficiency greater than 75%
DP07-01	Skip Maximum dimensions LxWxH (2m x 2m x 11m)
DP08-01	Safety Factor between 3 and 5
DP08-02	Rope Diameter 37 mm to 75 mm
DP08-03	Rope Core is SS316 alloy or equal

Several options and combinations were considered. Design analysis considering the listed DP identified the best choice of hoist and skip. The counterbalanced Blair Drum hoist is the most promising hoist technology based on efficiency, cost, and reliability. Two generic skip types were considered most promising: (1) the Arc Gate skip and (2) the Kimberly skip. The Arc Gate skip is evidently favored in traditional mining since its layout is compatible with a relatively small cross section and longer length. The smaller cross section is highly desirable in mining where the vertical shaft can be hundreds to thousands of meters deep. In contrast, the simplicity of the KS promotes a low initial cost, low maintenance cost and high service life. All these features are important in CSP applications; therefore the KS was selected for this application.

The qualitative design analysis (including construction and operation of two scale models) outlined above identified the KS with Blair Drum hoist as the promising design. Furthermore, design analysis shows that from the perspective of the system designer the combination is a highly suitable design. Nevertheless, according to some expert opinion, this may not the best possible choice and alternatives discussed below will be considered in the future.

The design also envisions an lift shaft allowed to stay at 200°C (392°F), which further minimizes incidental heat leaks. Altogether the proposed design ensures a minimal heat leak that will have negligible effect on the overall system efficiency.

CONCEPTUAL AND PRELIMINARY DESIGNS

The analysis and concept selection described above resulted in the general selection of KS with Blair hoist. The next task was to develop this concept into first a conceptual design for a 60 MW-th demonstration system and secondly a preliminary design for a 460 MW-th commercial power plant. More detailed design and engineering of the commercial particle lift system was therefore completed. Conceptual design drawings and energy efficiency and heat loss modeling have also been completed. Detailed efficiency modeling based on reliable published component efficiencies resulted in an energy efficiency of nearly 80% which exceeds the 75% energy efficiency target for this program. This target was selected to be attractive to potential investors. With this efficiency, the parasitic power consumption should be less than 1% of the rated electrical output.

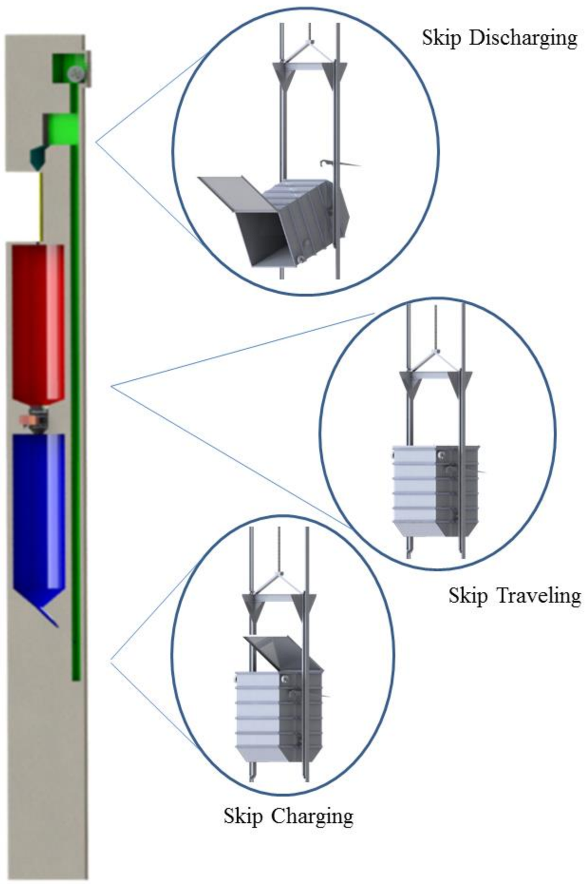


Figure 2 Conceptual Insulated Kimberly skip charging, travel position and discharging [13].

The selected skip design, which is the KS type, is seen in Figure 2 showing the skip its charging, travel and discharge configurations. The design specifications for demonstration system were then developed. With this experience, the specifications of the commercial particle lift, which were similar except for scale to the demonstration, were developed with results as seen in Table 3.

The selected KS is both filled and discharged from the top and does not have a complicated and leak-prone bottom hatch. This arrangement facilitates a design that is very simple structurally and mechanically. The single top hatch, which is opened and closed by motion of the skip thereby eliminating any mechanical or hydraulic actuators, is critical to this simplicity. Importantly, this design appears to be almost leak proof and should easily achieve much less than 0.1% target for temporary spillage of particulate during filling and discharge.

Table 3 Commercial Particle Lift Design Specifications

Design Specification	Value
Power Capacity of Tower	460 MWth
System Mass Flow	979 kg/s
Skip installed	2 skips
Estimated Skip Dimensions (LxWxH)	2m x 2m x 6m
Ropes in use per skip	1 rope per skip
Rope Type	SS 316 6x37 IWRC
Rope Diameter	76 mm (3 in.)
Drum to Rope Diameter Ratio	60
Rope Layers on Drum	3
Rope wraps per layer	5
Electric Motor	AC Induction Motor
Gear Reduction Ratio	46
Overall Safety Factor	3

All such spillage will be accumulated in a sump built into the lift shaft, which can be emptied as necessary; therefore, there will essentially zero net loss of particulate from the system. Minimal heat leak is also an objective, and simplicity of the proposed skip design makes it easy and inexpensive to install adequate internal insulation to keep the heat leak from the skip well under 0.1% of the rated capacity of the system.

The first set of drawings and specifications have been completed, and consultation one company familiar with steel fabrication and industrial lift manufacture was consulted. This company has commented that the design will be easy to fabricate. After incorporating some minor modifications based on this review, a skip-hoist component supplier was also informally consulted. With helpful input from these initial reviews from smaller companies, we contacted one of the major manufacturers.

This manufacturer commented that our design should be generally feasible to fabricate and install; however, they have also responded that the Koepe hoist and a bottom discharge skip should be considered as well. In practice, the Koepe hoist may be simpler and less expensive, and it should also have lower drum inertia, which would reduce dynamic loads and deceleration losses. Going forward, these alternatives will definitely be considered.

The simple design of the KS (basically a bucket with a hinged lid and a lifting bail) allows effective thermal insulation with mere layers of continuous suitable rigid insulation such as firebrick inside the skip and the lid with no complicated bottom hatch to insulate and no mechanism components (such as links and latches) to act as thermal short circuits. In contrast, the bottom hatch of an Arc Gate skip is likely to leak during lifting, which is not an issue when handling typical raw materials but important when hoisting the fine TES medium. Our experience with the two small-scale models was convincing with regard to these issues.

As is evident in Figure 2 the Blair hoist and KS combination is easy to integrate into the CSP system. The two separate drums of the Blair hoist are particularly attractive for integrating into this design. Note that the lift shaft will be kept at elevated temperature between 150°C (302°F) and 200°C (392°F) to minimize heat losses, but the electrical and mechanical equipment (other than the lift drum) will be kept at near ambient temperature for efficiency and economy.

Typically, stainless steel such as SS316 wire rope is selected for durability, corrosion resistance, and excellent high temperature strength. The rope size of 0.076 m (3 inch) based on the above calculated stress and on vendor tensile strength of SS316 using the factor of safety (FS) of 5 as required by OSHA[15],[16] for mining application. This FS is somewhat high for a less sensitive industrial application but has been taken to be appropriate for this scoping study.

COST ANALYSIS

Capital cost estimates are necessary to support conceptual and preliminary designs and eventual optimization. Some helpful cost formulas that would be useful at the planning level were found in literature. Nevertheless, no cost data or models directly applicable to the designs in this study were found. One especially useful higher level formulation is given by Sayadi et al. [17]. In this approach, the important system level parameters can be entered into a regression formula, to generate the overall system level cost. This approach is an improvement over earlier less detailed formulations such as [18] and [12].

The Sayadi cost model could definitely be used at the planning level, but more detail was thought necessary for this application especially because higher temperature operation is necessary and because the lift in this application is much shorter than in most mining applications. Therefore, two cost estimate procedures appropriate in turn to conceptual and preliminary designs were employed in this study. One based on generic subsystem costs was adequate to confirm from a conceptual design that the skip hoist can be cost effective. The second approach uses modular costs for each major component, and this approach allows for a presumably more accurate result and its adaptability allows it to support future optimization in the preliminary design phase.

Cost estimates for the conceptual design of the 60 MWth demonstration were developed rather quickly using a highly regarded source of generic subsystem costs for the hoist system, and analysis of a conceptual; design for skip subsystem. The results are summarized in Table 4. This analysis was conducted with an initial and conservative FS of 5 as justified above.

Table 4 Estimated Cost Analysis for 60 MWth Particle Lift with assumed Safety Factor of 5.

Component	Cost
2 Skips without Hoist System	$198,000
Hoist System: mechanical and	$295,000
OLDS Elevator Particulate Recovery	$30,000
Total Estimated Cost per System	$523,000
Total Particle Lift cost per MWth	$8,700

This improved subsystem-based cost analysis was conducted based on a conceptual design of the insulated skip meeting on our preliminary design requirements and aided by generic guidance from an appropriate handbook [19]. Necessary data was generated including dimensions and other specs important to skip and hoist costs. The typical values were found in cost databases for fabrication of the skips other necessary subsystems including the auxiliaries and the drive and control systems from reference [20] assuming subsystem costs for conventional freight elevators.

The total estimated cost per system was determined by this somewhat more detailed analysis to be $523,000. This cost was then compared to the cost calculated by the empirical cost estimation formula developed by Sayadi [17], which gives a cost of $539,000 or $9,000 per MW-th. These two independent and quite distinct cost estimates for similar sized and rated systems are very close. For a further interesting but rough confirmation, handbook values for representative mine hoists are available in [12]. The cost a for a relatively small practical mine hoist system is reported to be around $1,172,000; however, even this smallest reported mining system uses skips much larger than those in the preliminary PHR system design thereby inflating all the associated costs. Consequently, this handbook cost is useful only as a far upper bound, and it does comfortably exceeds our design estimates but not by an expected but not absurd margin.

Accordingly, the mid-scale demonstration lift system is expected to cost around $8,700 per MWth, which agrees with a previous independent cost estimate calculated using both generic and technology-specific cost engineering research results, which was around $9,600 per MWth [2].

Next the cost for the commercial system was considered. For this more detailed study, a modular cost approach was followed. In such an approach, a flexible cost model is developed for each major component or functional module. An individual modular cost is typically based on important features and ratings of each component including cost premiums for special features such as higher temperature operation. Such modular costs models can be easily incorporated into a computer model to evaluate the overall cost and performance of the system, and the overall cost can be computed for any set of system requirements such as height of lift and mass flow rate of particulate. The resulting total estimated particle lift cost per MWth as seen in Table 5 was found to be $5,500. As expected, costs per unit are improved at larger size and overall efficiency is only negligibly changed with larger system size and longer lift.

Table 5 Current Cost Analysis for 460 MWth Particle Lift designed with Safety Factor of 3.

Component	Cost ($)	Reference
Drum x2	$437,000	Advertised Cost
Electric Motor	$278,000	[21]
Gear Reducer	$164,000	Advertised Cost
Variable Frequency Drive	$118,000	Advertised Cost
Brake System	$8,000	[22]
Wire Rope	$21,000	Direct Quote
Skips x2	$390,000	Direct Quote
Bearings x4	$27,000	Advertised Cost
Olds Elevator	$30,000	Direct Quote
Instrumentation	$50,000	Advertised Cost
Sub Total	**$1,523,000**	
Integration Cost (2x14%)	$426,000	[23]
Sub Total	**$1,949,000**	
General Overhead (30%)	$584,700	[22]
Grand Total	**$2,534,000**	
Cost per Skip	**$1,267,000**	
Cost per MWth	**$5,500**	

The values for this cost analysis were based on current data as indicated in the table. Structural related costs in this table were calculated using a FS of 3, which is appropriate for applications such as this in which it is very unlikely that the system could be overloaded and no risk to life is involved during operations. It should also be noted that this is the total erected cost with reasonable provisions for integration and construction overhead.

The integration cost was conservatively taken to be twice the fraction in the cited reference since this system requires on site vertical integration. This cost is well under the target cost for the project supporting this investigation. It is however considerably greater than the cost estimates generated in the important pioneering study of PHR systems by Falcone et al. [10], which was only $2,240 per MWth after correction for inflation. The earlier cost apparently does not include all the details and provisions for higher temperature operation in the current estimate; so the considerable spread between these two estimates is quite reasonable.

The nominal rope size of 0.076 m (3 inch) and other structural sizes are based on the calculated stress and on vendor tensile strength of SS316 using the FS of 3. This FS was used was used since the working area of the skip will not have personnel present. If a higher safety factor is judged necessary or is required in a specific jurisdiction, then a multi-rope system may need to be considered in future analysis since thicker ropes may be difficult to source.

DRIVE SYSTEM BACKGROUND

To estimate the cost of owning and operating the hoist system, operating energy costs are needed to complement the capital costs.

This cost is almost exclusively the cost to operate the drive motor and its power supply. The literature does of course address general application and performance of electric motors as in [21] and [24], and there is extensive literature on the details of electric motor design and theory, some pertinent aspects of which are discussed below.

Motor selection is an important consideration and several informative publications discuss the selection and analysis of hoist motors such as for example [25], [26], [27], [28] and [29]. Development of advanced motor designs such as the doubly-fed induction motor [28], which promises enhanced overall efficiency and other advantages in hoist applications, is of special interest. In general, however, this literature either discusses innovative motors or investigates motor efficiency in isolation from the balance of the hoist system. It does appear that new or emerging motor designs are worth of consideration such as the wound-rotor design discussed in [28]. Nevertheless, at present it appears that the simple and familiar squirrel cage induction motor with variable frequency drive (VFD) is an adequate and conservative choice. While somewhat limited in efficiency, this conventional subsystem should provide high reliability and adequate performance and efficiency. Currently, we are investigating more advantageous choices such as wound-rotor motor designs and systems with advanced soft-start controls. Such alternatives promise both improved efficiency and substantially reduced costs, especially if the VFD can be eliminated. Our approach will be to select the ultimate electric motor drive with due consideration to the efficiency of the entire drive train and the effect of this selection on the design of the mechanical components subjected to the mechanical loads imposed by operation of the drive.

Energy recovery is especially important in lifts and elevators, and this topic is investigated in many publications such as [30] and [31], and in particular [32] addresses the interesting special concept of including an energy storage accumulator. Notably, these publications are focused on cranes or small industrial application lifts or elevators, while our application is considerably larger and more energy intensive. While these electrical technologies will continue to be considered, it does appear at present that the counterbalanced two-skip design is inherently adequate for the recovery of both the potential and kinetic energy stored in the skips. As our design matures, this process of energy recovery will be further investigated by more detailed modeling, which is only now possible since an overall design has now been defined. Special purpose electrical energy recovery and storage, as well as the inherent mechanical energy recovery, may well be an auxiliary feature in the final design.

Further and more detailed dynamic modeling is discussed in the existing literature such as in [33], [34], and [29]; and related useful literature on modeling in available for example in [35] and elsewhere. Smoothness and stability will be important for long life and reliability of the mechanical components, and these features will be emphasized in our future and more detailed dynamic modeling. Research of special importance is investigation of proportional-integral-derivative (PID) controller [29] in hoisting. The familiar PID controller will likely be the principal important component in the overall control system. Reliable control is especially important due to the elasticity of the lifting rope in any skip hoist and the effect of such elasticity on stability. Nevertheless, it appears that the details of the motor control, such as the tuning of the PID controller [29] will have a negligible effect on the overall energy efficiency of the system.

The skip scheduling has been found to be an important issue even in the preliminary design as in the literature [36]. Note however in [36], that the interesting and instructive optimization of hoist scheduling to conserve electric energy is conducted assuming a known and fixed overall hoist efficiency. This approach is appropriate for the consideration of an existing design. In our application, however, it has been and will continue to be necessary to optimize the scheduling along with the mechanical and structural design. The current tentative preferred design appears to be achieved with a maximum acceleration of about 0.2 g which results in a skip of reasonable size without requiring excessive speed.

While important aspects of motor control, scheduling, and efficiency are discussed separately in the literature, there is very little literature on the efficiency of the entire hoist system or presentation of applicable cost modules of each component of the hoist system. Indeed, no suitable published energy consumption models were found. Unlike in most mining applications where the energy consumed during hoisting operations are small in comparison to the value of the payload, in the particle lift energy is considered important in this application For this reason an energy model particular to this application was developed.

DRIVE SYSTEM MODELING

The energy efficiency modeling is based on an energy flow analysis using reasonable lift and recovery efficiency values and the ratio of overall tare to payload (PL) as indicated in Table 6.

Table 6 Estimates of overall efficiency for particle lift design.

Data	Efficiency
Lift Efficiency	0.85
Recovery Efficiency	0.93
Ratio: Tare/PL	0.24
Overall Efficiency	0.79
Fraction Parasitic	0.0086

The tare fraction is important since the potential energy of the skip and rope cannot be 100% recovered. For this reason, it is important to minimize the mass of the skip compared to the PL since the skip and rope represent the tare mass. Note that the overall energy efficiency is somewhat less than the basic lift efficiency.

An important parameter in the overall efficiency calculation is the lift efficiency, which is taken to be of 85% based on several published standards and models. Furthermore, this value can be confirmed by component modeling shown in Table 7. This parameter is obviously the efficiency of the basic drive system, and will always exceed the overall lift efficiency because not all of the potential and kinetic energy invested in the skip and rope and other components can be recovered. Detailed calculations for an optimized and light weight skip give an expected overall energy efficiency of 79%, which is higher than the target of 75% proposed for the project supporting this study. Some other improvements now being investigated could bring this efficiency above 80%, which is the goal of the design team.

Table 7 Estimates of lift component efficiency for particle lift design.

Component	Efficiency
VF Drive	0.96
Electric Motor	0.95
Gearing, 2 Stage	0.98 x 0.99 = 0.97
Rope/Drum Efficiency	0.98
Overall Product	0.86 to 0.87

Some remaining more detailed aspects that will be investigated in future include design and material selection for all bearings, joints, and ropes. It appears that the particulates being considered for this application will be largely dust free, and the skip shaft can be ventilated to remove dust as necessary. Nevertheless, special care will be taken in the final detailed design to minimize friction and wear and ensure that such effects do not reduce the life or performance of the skip or hoist system.

CONCLUSIONS

In conclusion, the particle lift subsystem of proposed PHR based CSP plants is vitally important for reliable operation, and the cost and efficiency of this subsystem influences the overall energy conversion efficiency, the capital cost, and ultimately the LCOE of the electric energy produced.

A suitable commercial solution is shown to be a KS based particle lift in a counter balanced Blair hoist configuration. This design meets the cost and efficiency targets for potential mid-scale demonstration and larger scale commercial systems. The design promises high overall efficiency, long service life, and low maintenance cost.

While the design presented above is considered adequate for planning purposes, and more mature design is ultimately needed. Conducting more detailed transient modeling and considering other motors and control systems are part of this continuing design development.

ACKNOWLEDGMENT

Part of this work was supported by the US Department of Energy through the Sun Shot Program's project for the "Development of a High Temperature Falling Particle Receiver" (Project ID: DE-AC04-94L85000). The Prime Contractor is Sandia National Laboratories, and the Sandia's PI is Dr. Clifford K Ho. The financial and programmatic support is recognized and greatly appreciated

REFERENCES

[1] C. Ho, J. Christian, D. Gill, A. Moya, S. Jeter, S. Abdel-Khalik, D. Sadowski, N. Siegel, H. Al-Ansary, L. Amsbeck, B. Gobereit, R. Buck, Technology Advancements for Next Generation Falling Particle Receivers, Energy Procedia, Volume 49, 2014, Pages 398-407.

[2] Research Performance Progress Report (RPPR), High Temperature Falling Particle Receiver, DE-EE0000595-1558, 15 July 2013.

[3] "Rexnord High Performance Bucket Elevators", Rexnord Industries, LLC, 4701 W. Greenfi eld Rd., Milwaukee, WI 53214 www.rexnord.com

[4] P Holbrow, G.A Lunn, A Tyldesley, Explosion venting of bucket elevators, Journal of Loss Prevention in the Process Industries, Volume 15, Issue 5, September 2002, Pages 373-383

[5] Rademacher, F. (1979) Non-spill discharge characteristics of bucket elevators. Powder Technology, 22 (2). pp. 215-241. ISSN 0032-5910

[6] Olds Elevator LLC, product brochure, available online at http://www.oldsusa.com

[7] Schramek, P. ; Al-Ansary, H. ; Jeter, S. ; Abdel-Khalik, S. ; Al-Suhaibani Z. ;El-Leathy, A. ; Herzig, S. ; Gaines, G. (2011). High Temperature Solar Gas Turbine Project- Design of Heliostat Field and Particle Receiver. Available in the proceedings-CD of the SolarPaces Symposium 201, Granada, Spain, 20, - 23, Sept. 2011

[8] Golob M,,Jeter S., Abdel-Khalik S., Sadowski D., Al-Anasry H., Elleathy, A., "Development and Design Prototype 300KW-Thermal High Temperature Particle Heating Concentrator Solar Power System Utilizing Thermal Energy Storage", Proceedings of ASME 2014 8th International Conference on Energy Sustainability, ESFUELCELL2104-6679, July 2014, Boston MA.

[9] Sheline W. (2013), "Concentrated Solar Chemistry: Design Stage Theoretical Thermodynamic Analysis of an Iron Ethylene Production Process"(Master's Thesis).Georgia Institute of Technology, Atlanta, GA.

[10] Falcone, P.K., J.E. Noring, and J.M. Hruby, 1985, Assessment of a Solid Particle Receiver for a High Temperature Solar Central Receiver System, SAND85-8208, Sandia National Laboratories, Livermore, CA.

[11] "Ore Skips, Mine Haulage Systems and Other Foundry Products in Ontario." *Wabi Corp*. N.p., n.d. Web. .

[12] "SME Mining Engineering Handbook", Third Edition, 2011, Society for Mining, Metallurgy, and Exploration Inc.

[13] Repole, K., Jeter, S., (2014) "Preliminary Commercial Design For Transporting Low Temperature Particles for Use in Particle Solar Receivers Using Lifting Skips "Proceedings, Fourteenth Annual Early Career Technical Conference, Birmingham, AL, Paper 63.

[14] Repole, K., Jeter, S., (2015) "Application of a Sequence of a Design Methodologies to the Problem of Transporting Warm Particles in Particle Heating Receiver Solar Energy Systems "Proceedings, Fifteenth Annual Early Career Technical Conference, Birmingham, AL, Paper 89

[15] OSHA "Safety and Health Regulations for Construction" Chapter 29, CFR section 1926.251 (2010)

[16] US Government, Regulation on "Guide Ropes", Chapter 30 C.F.R. section 57.19019 (2014).

[17] Sayadi AR, Lashgari A, Oraee K, Yavari M (2010) Hoisting Equipment Cost Estimation in Underground Mines. SME International Conference on Hoisting and Haulage, Las Vegas, USA.

[18] USBM. 1987. US Bureau of mines cost estimating system handbook, Mining and beneficiation of metallic and nonmetallic minerals expected fossil fuels in the United States and Canada, Open file report 10-87, Denver, Colorado: United States Bureau of Mines.

[19] De la Vergene, J., "Hard Rock Miner's Handbook", Edition 3, 2003, McIntosh Engineering, Ontario, Canada.

[20] R. S. Means, Construction Cost Guides, available on line at: http://www.rsmeans.com

[21]Almedia A., Greenberg S., "Chapter 11: Electric Motor Systems Efficiency", Kreith F., and D. Yogi. Goswami. Energy Management and End Use Efficiency Handbook. Boca Raton, FL: CRC, 2006. Print. page 11-20

[22] Poore R, Lettenmaier , T, Alternative Design Study Report: WindPACT Advanced Wind Turbine Drive Train Designs Study, August 2003, NREL/SR-500-33196 , National Renewable Energy Laboratory, Kirkland, Washington

[23]Dodge Unit Cost Book 1996. New York: McGraw-Hill, 1996. Print.

[24] Wei-Jen Lee; Kenarangui, R "Energy management for motors, systems, and electrical equipment," Industrial and Commercial Power Systems Technical Conference, 2001. Conference Record. Papers Presented at the 2001 Annual Meeting. 2001 IEEE , vol., no., pp.21,26, May 2001

[25] Muhammad, J., Shuraiji, A., “Modeling of DC elevator motor drive for mid-rise building", Engineering & Technology Journal 2013 Volume: 31 Issue: 12 Part (A) Engineering Pages: 2320-2342 ISSN: 24120758

[26] Kumar, N.; Chelliah, T.R.; Thangaraj, R.; Srivastava, S.P., "Economical analysis of induction motor for a mine hoist load diagram," Information and Communication Technologies (WICT), 2011 World Congress on , vol., no., pp.1310,1315, 11-14 Dec. 2011

[27]Repo, A.; Montonen, J.; Sizonenko, V.; Lindh, P.; Pyrhonen, J., "Energy efficiency of hoisting motors," Electrical Machines (ICEM), 2014 International Conference on , vol., no., pp.144,149, 2-5 Sept. 2014

[28] Han Yao-fei, Tan Guo-jun, Li Hao, Ye Zong-bin, Wu Xuan-qin, Back-to-back three-level double-fed induction motor control system for mine hoist, Procedia Earth and Planetary Science, Volume 1, Issue 1, September 2009, Pages 1448-1454, ISSN 1878-5220

[29] Lundin, S. (2007). Load Simulation and Investigation of PID Control for Resonant Elastic Systems. (Master Thesis). Linköpings universitet: Institutionen för systemteknik.

[30] Myungdo Lee, Taehoon Kim, Hong-Ku Jung, Ung-Kyun Lee, Hunhee Cho, Kyung-In Kang, Green construction hoist with customized energy regeneration system, Automation in Construction, Volume 45, September 2014, Pages 66-71, ISSN 0926-5805

[31] Tominaga, S.; Suga, I.; Araki, H.; Ikejima, H.; Kusuma, M.; Kobayashi, K., "Development of energy-saving elevator using regenerated power storage system," Power Conversion Conference, 2002. PCC-Osaka 2002. Proceedings of the, vol.2, no., pp.890,895 vol.2, 2002

[32] Acquaviva S. "Energy Storage and Recovery System for Lift", proceedings of Elevcon 2014, July 2014, Paris , pp 168-178.

[33]Boulter B.,“Elevator Modeling and DC Drive Speed Controller Design”, Applied Industrial Control Solutions ApICS LLC,Report 2000

[34] Knight, C.E.; Becerra, V.; Holderbaum, W.; Mayer, R., "Modeling and simulating the operation of RTG container cranes," Power Electronics, Machines and Drives (PEMD 2012), 6th IET International Conference on , vol., no., pp.1,6, 27-29 March 2012

[35] Zhou Gong-bo, Zhu Zhen-cai, Wang Ying, Sun Xiao-dong, Zi Bin, Sinking-hoists synchronous balance control system in vertical shaft, Procedia Earth and Planetary Science, Volume 1, Issue 1, September 2009, Pages 1503-1512, ISSN 1878-5220

[36] Badenhorst W., Zhang J., Xia X., Optimal hoist scheduling of a deep level mine twin rock winder system for demand side management, Electric Power Systems Research, Volume 81, Issue 5, May 2011, Pages 1088-1095, ISSN 0378-7796

