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Design and performance evaluation of a high-temperature cavity receiver for a 2-stage dish concentrator

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Abstract: Here a new design of a cavity heat-pipe receiver for a 2-stage dish concentrator is proposed. Both optical and thermal simulations are used for the design and for performance evaluation of the cavity. The receiver was fitted to a conventional 2-stage and an improved (overlapped) 2-stage dish. The latter system configuration shows superior performance compared to the conventional one, in particular in terms of compact structure, uniformity of the incident flux and temperature distribution, and solar-to-thermal efficiency. The variance of the irradiation distribution at the cavity decreased by 25% and the largest adjacent temperature difference decreased by 54%. In total, the conversion efficiency increased from 61.3% to 68.6%. Moreover, the new receiver with the improved 2-stage dish concentrating system has less limits of scales (e.g. weight and volume) compared to the traditional single dish design.

Keywords: 2-stage dish, cavity receiver, solar thermal simulation, Monte-Carlo ray tracing method

31 **Nomenclature**

32

33 *Symbols:*

34	A	area, m ²
35	C	concentration ratio
36	F	view factor
37	Gr	Grashof number
38	h_{apt}	heat transfer coefficient of the convection through the aperture, W/(m ² ·K)
39	h_f	heat transfer coefficient at the insulation enclosure of the receiver, W/(m ² ·K)
40	h_{rad}	equivalent radiation heat transfer coefficient, W/(m ² ·K)
41	L	thickness, m
42	Pr	Prandtl number
43	Ra	Rayleigh number
44	q	irradiation, W/m ²
45	$\bar{q}$	average irradiation, W/m ²
46	R	radius, m
47	S^2	relative sample variance
48	T_w	temperature at cavity walls, K
49	T_∞	surroundings temperature, K
50	α	volumetric expansion, 1/K
51	δ	Dirac delta function
52	δT	largest adjacent temperature difference, K
53	ΔT	global temperature difference, K
54	ε	emissivity
55	η	efficiency, %
56	θ	inclination angle, deg.
57	Θ	dimensionless inclination angle
58	λ	thermal conductivity, W/(m·K)
59	ν	viscosity, m ² /s

60	ρ	density, kg/m ³
61	σ	Stefan-Boltzmann constant, W/(m ² ·K ⁴)
62	σ_{slope}	slope error, mrad
63	$\sigma_{tracking}$	tracking error, mrad
64	ϕ_{rim}	rim angle, deg.
65	<i>Subscripts</i>	
66	<i>absor</i>	absorbed
67	<i>act</i>	irradiated active region
68	<i>app</i>	apparent
69	<i>apt</i>	aperture
70	<i>cav</i>	cavity
71	<i>conv</i>	conventional
72	<i>inc</i>	incident
73	<i>layer</i>	insulation layer
74	<i>loss</i>	loss
75	<i>max</i>	max
76	<i>net</i>	net
77	<i>nov</i>	novel
78	<i>optical</i>	optical
79	<i>side</i>	side
80	<i>sky</i>	sky
81	<i>stag</i>	stagnant
82	<i>surf</i>	cavity surface
83	<i>thermal</i>	thermal
84	<i>top</i>	top
85	<i>total</i>	total
86		
87	<i>Abbreviations:</i>	
88	ANU	Australian National University
89	CSP	concentrated solar power system

DNI directly normal irradiance
SNL Sandia National Laboratory
PDC paraboloidal dish concentrator

1. Introduction

Due to a high solar concentration and good optical efficiency, the paraboloidal dish concentrator (PDC) is regarded as a promising option for future Concentrated Solar Power systems (CSP). There has been consistent evolution and improvement in parabolic dish designs since 1970s (Coventry and Andraka, 2017). The concentration ratio (C) of commercial PDC systems can be as high as 3,000 suns (Mancini et al., 2003), which is at least an order of magnitude higher than with parabolic trough systems. Some key challenges with PDC have been the high costs, mechanical constraints, and tracking inaccuracies with traditional large dishes, verified e.g. by the SG3 and SG4 dish of the Australian National University (ANU) (Lovegrove et al., 2011; Lovegrove et al., 2003) and the PETAL in Israel (Biryukov, 2004). To address these issues, an improved 2-stage dish concept providing more flexibility and stable structures has been proposed (Wang et al., 2017). Thanks to the new dish concept with a unique hollowed design, the receiver including the power conversion unit can be shifted to the bottom of the concentrator making the whole configuration more stable, flexible, and easier to install with thermal storage systems. In our previous work, this novel 2-stage dish configuration could reach a higher optical efficiency and concentration ratio than a conventional 2-stage dish concentrator (Wang et al., 2017; Yang et al., 2018a).

The receiver is an integral part of a concentrator system to reach a high-performance value. The focus of this paper is in designing a novel receiver for the 2-stage dish concentrator to together provide an outstanding novel concentrator system. The receiver couples the dish concentrators to the power conversion unit, typically with a Stirling or Brayton cycle. Stirling engines can reach a high power conversion efficiency (Karabulut et al., 2009; Mancini et al., 2003), whereas Brayton engines are more

flexible for simplified hybrid operation (Li, Y. et al., 2015; Mills, 2004). Regardless of engine-type used, the receiver always plays a crucial role in the solar-to-heat conversion of a PDC. Cavity receivers containing liquid-metal reflux components are ideal options for dish systems due to several advantages, such as blackbody effect, high thermal transfer ratio, and isothermal heat source for the engine (Moreno et al., 1991). A range of designs and thermal models for cavity receivers in dish systems have been developed using numerical and/or experimental methods (Adkins et al., 1995; Andraka et al., 1994; Bader et al., 2015; Daabo et al., 2016; Loni et al., 2018; Loni et al., 2017; Moreno et al., 1991; Paitoonsurikarn and Lovegrove, 2006a, b; Pavlovic et al., 2017; Pye et al., 2016; Reddy and Kumar, 2009; Reddy and Nataraj, 2018; Shuai et al., 2008; Taumoeofolau et al., 2004; Wu et al., 2011; Zou et al., 2017). In early 1990s, the Sandia National Laboratory (SNL) (Moreno et al., 1991) demonstrated a 75-kW sodium heat pipe receiver in Sandia's nominal 75-kW parabolic-dish concentrator. Several studies have focused on the heat loss and temperature distribution modeling and surpassing the convective losses for different type of receivers (Bader et al., 2015; Loni et al., 2017; Paitoonsurikarn and Lovegrove, 2006a, b; Reddy and Kumar, 2009; Shuai et al., 2008; Taumoeofolau et al., 2004). Other studies have presented improved receiver configurations for solar dish systems, e.g. based on heat pipes (Wu et al., 2011) and receivers with special cavity geometries (Pye et al., 2016; Shuai et al., 2008). Also different design and optimization methods for solar cavity receivers (Zou et al., 2017) and performance analyses for different working fluid (Loni et al., 2018; Pavlovic et al., 2017) have been presented. Other studies, although applied to other type of CSP plants, have contributed to hybrid multi-dimensional models (Li et al., 2017a, b) as a multi-level analytical methodology which are also applicable to solar dish systems.

The concept of 2-stage concentrating receivers has been widely used in so-called beam down solar tower systems (Hasuike et al., 2006; Li, X. et al., 2015), but not in dish systems. Previous studies on solar dish receivers have employed traditional dish configurations, which are not as such applicable to the improved concentrator design of interest here, because it has quite different solar concentration effects and patterns.

The novel 2-stage dish concentrator in this study (Fig. 1) has a unique hollowed design of four mirrors employing the overlap method yielding a clearly better optical performance than the conventional 2-stage dish concentrator (Wang et al., 2017). The novel concentrator has the potential to produce a more uniform radiation flux and temperature distribution at the cavity surfaces, which as a whole could lead to a higher solar-to-thermal conversion rate than with the conventional 2-stage dishes. However, to capture such improvements, the concentrator will need to be equipped with a tailor-made receiver, which has not yet been discussed to our best knowledge in the current literature. Our aim is to fill this gap by proposing a new design of a liquid-sodium wick heat pipe receiver attached to the 2-stage dish configuration. Both the optics and heat transfer aspects of the receiver are comprehensively analyzed. For this purpose, in-house developed heat transfer models are employed accounting for radiative, convective, and conductive losses, coupled with ray tracing simulations for the optics part of the analyses.

Figure 1. 3-D sketch of the novel 2-stage dish concentrator (Yang et al., 2018a).

2. Receiver concept

In this section, the technical details for the receiver system are given, which will then be analyzed in detail in Section 3 and 5. Since the novel 2-stage dish concentrator of 20 m diameter can intercept the incident irradiation up to 312 kW, the receiver is designed for a 200-kW power rate.

2.1 Receiver prototype design for 2-stage dish system

Typical geometries for a solar dish receiver include cylinders, semi-sphere surfaces (or partial spheres), and truncated cones (Daabo et al., 2016). Other designs with special geometries are not considered in this paper. As spherical receivers show the best radiation performance in the irradiation areas (Shuai et al., 2008), the bottom of the inner walls is designed as a partially spherical surface to intercept solar rays as uniformly as possible. To allow a compact design, for the rest of the inner walls, a cylindrical geometry is chosen. Then a gravity-assisted wicked heat pipe, with a shape of a crescent chamber containing liquid sodium is attached to the spherical dome of the cavity. A channel stretches out from the topside of the chamber connecting to the condenser. The chamber, the channel, and the condenser surface form together an enclosed space. Figure 2 depicts the schematic and details of the receiver prototype.

187 Figure 2. Schematic of a gravity-assisted wick heat-pipe solar dish receiver (not in
 188 scale). ①insulation wall ($\text{Al}_2\text{O}_3\text{--SiO}_2$; ② spherically absorbing surface attached with
 189 wicks inside; ③ crescent chamber containing liquid sodium; ④ other insulation and
 190 bracing components; ⑤ steam channel; ⑥ condenser; ⑦ thermal engine and
 191 generator.

192

193 2.2 Selection of materials and parameters

194

195 The size of the receiver mainly depends on the area of the bottom spherical wall, which
 196 varies with the local flux density absorbed. Here it is set as the average value of the

incident flux density $\sim 3.8 \times 10^5 \text{ W/m}^2$ in accordance to the designs of the Sandia pool-boiler receiver (Moreno et al., 1991). The rim angle of the bottom part is fixed where the incident irradiation intercepted has dropped down to one tenth of its peak value. For the heat pipe structure, a so-called gravity-assisted felt-metal-wick (Adkins et al., 1995; Andraka et al., 1994) was chosen since the upward inclined cavity limits the applicable options. The concept has been proven to be an effective approach in solar dish receivers (Adkins et al., 1995; Andraka et al., 1994). The optimal parameters of the 2-stage dish configurations corresponding to the novel and the conventional cases are given in Ref (Wang et al., 2017). Table 1 lists the optimal setting for the main parameters of the receiver in Fig. 2. The detailed calculating process will be demonstrated later in Section 3.

Table 1. Main parameters of the heat-pipe solar dish receiver.

Parameters	Novel 2-stage dish concentrator	Conventional 2-stage dish concentrator
Aperture radius (R_a)	180 mm	200 mm
Cavity radius (R_c)	374 mm	398 mm
Dome radius (R_d)	408 mm	540 mm
Side wall thickness (R_l)/ Top disk thickness (L)	40 mm	40 mm
Dome rim angle (ϕ_{rim})	66.4°	47.5°

The absorber (② in Fig. 2) is made of stainless steel to achieve a uniform temperature distribution. The insulation (① in Fig. 2) consists of a top annular disk and a side cylindrical wall. For insulation, $\text{Al}_2\text{O}_3\text{--SiO}_2$ is used and its temperature-dependent thermal conductivity is calculated with a correlation from (Furler and Steinfeld, 2015). To estimate the radiative exchanges of the surfaces, the coated inner walls are modeled as opaque gray-diffuse surfaces with an absorptivity (i.e. the emissivity) of 0.2 for the top and sides, and 0.85 for the bottom, respectively. All insulations are enclosed outside

with Inconel 600 with an emissivity in the range of 0.6-0.7 depending on temperature (Furler and Steinfeld, 2015). The main physical properties of air including density (ρ), thermal conductivity (λ), viscosity (ν), volumetric expansion (α), and Prandtl number (Pr) are described as fitted functions versus temperature functions using standard data (ToolBox, 2005).

3. Optical and thermal models

To study the thermal performance of the receiver, a thermal model was developed to obtain steady-state temperature distributions in the receiver. The value settings and assumptions made are given as follows:

- Temperature of air and sky are set at 293 K and 285 K, respectively (Kalogirou, 2012);
- Working temperature of the absorber is set at 1155.8 K, which is equal to the vaporization point of liquid sodium at atmospheric pressure (the temperature at the airside of the absorber should be slightly higher than the evaporation point due to phase-changing heat transfer. For simplicity, this difference has been ignored here);
- Isothermal boundary conditions on the absorbing surface are assumed, otherwise the third kind of boundary conditions are used;
- All materials are isotropic and the surfaces are opaque gray-diffuse;
- The sky is regarded as a black-body at constant temperature;
- Conductive losses through the insulation are 1-dimensional.

The Monte Carlo ray-tracing method was applied to obtain the incident flux distribution over the inner surfaces and the view factors. After balancing the CPU's time cost and the accuracy of the numerical calculation, 2 *million* photons are generated to simulate the incident radiation and 10 *million* photons are used to determine the view factor matrixes. The inner walls of the receivers were separated into

$N_{surf} = N_r \times N_f + N_{z1} \times N_f + N_{z2} \times N_f + 1$ discrete meshes corresponding to the top, side, and the bottom places plus the aperture. N_r , $(N_{z1}+N_{z2})$ and N_f represent the numbers of nodes in radial, axial and circumferential directions, equal to 9, (16+24) and 40 in the optimal design of the novel 2-stage dish concentrator. The insulation was divided into $N_{layer}=20$ layers for solving the conductive heat transfer discretely. The radiosity method was used to get the net flux distribution at the inner surfaces of the cavities by solving the set of net radiation equations for each assumed gray-diffuse segment (Howell et al., 2010):

$$\sum_{j=1}^{N_{surf}} (\delta_{kj} - (1 - \varepsilon_j) F_{kj}) \frac{q_{net,j}}{\varepsilon_j} = q_{inc,k} - \sum_{j=1}^{N_{surf}} (\delta_{kj} - F_{kj}) \sigma T_j^4 \quad (1)$$

where $q_{inc,j}$ and $q_{net,j}$ represent the incident solar radiation and the net radiative heat flux at the j^{th} segment. ε is the emissivity. σ is Stefan-Boltzmann constant ($5.6704 \times 10^{-8} \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-4}$), δ is the Dirac delta function, and F_{kj} is the view factor from the k^{th} to the j^{th} segment.

The net radiative heat flux in Eq. (1) consists of two parts, the heat flux lost through the cavity (q_{loss}) and the heat flux absorbed by the absorbing surface (q_{absor}). The q_{loss} is caused by the convective loss through the aperture and by the conductive loss from the inner walls of the receiver to the outside. q_{net} can then be written as follows:

$$q_{net} = h_{apt} \cdot (T_w - T_\infty) + k_{app} \cdot (T_w - T_{sky}) + q_{absor} \quad (2)$$

where h_{apt} and k_{app} represent the heat transfer coefficient of the convection through the aperture and the apparent conductivity from the cavity inner walls to the outside. T_w , T_∞ , T_{sky} are the temperature at the cavity walls, of the surroundings, and of the sky. Note that all variables in Eq. (2) are N_{surf} dimensional vectors corresponding to the segments of the cavity surfaces.

The heat loss through the insulation is modelled as 1-D thermal conduction with a temperature-dependent thermal conductivity coupled to convective and radiative losses at the outer surfaces. The apparent conductivity is expressed in Eq. (3) as

$$k_{app} = \begin{cases} \frac{1}{\frac{L}{\lambda} + \frac{1}{h_{f,top}}} + h_{rad} & \text{for the top} \\ \frac{1}{\frac{R_c}{\lambda} \ln \frac{R_l + R_c}{R_c} + \frac{1}{h_{f,side}}} + h_{rad} & \text{for the side} \end{cases} \quad (3)$$

where h_{rad} is the equivalent radiation heat transfer coefficient which equals to $\varepsilon \cdot \sigma \cdot (T_{out}^4 - T_{sky}^4) / (T_{out} - T_{sky})$. $h_{f,top}$ and $h_{f,side}$ represent the heat transfer coefficients at the top disk and the side cylinder of the insulation enclosure. For the bottom part, $h_{f,bottom}$ is not considered in the present work, and is set to 0. They are modeled using Churchill's and Chen's correlations for natural convection on a horizontal cylinder and on inclined topside plates (Chen et al., 1986; Churchill and Chu, 1975). The correlations are as follows:

$$\text{For the outer side wall: } Nu_{side} = 0.579 \left(\frac{Ra}{\left[1 + (0.442 / Pr)^{9/16} \right]^{16/9}} \right)^{1/4} \quad (4)$$

$$\text{For the topside disk: } Nu_{top} = \frac{3}{4} \left[\frac{2 Pr Ra \cos \theta}{5(1 + 2 Pr^{1/2} + 2 Pr)} \right]^{1/4} \quad (5)$$

The convective heat losses through the aperture are estimated by using the correlations of Leibfried's explicit model for convective heat transfer in an empty, hemispherical, inclined upward-facing cavity (Leibfried and Ortjohann, 1995):

$$Nu_{apt} = 0.106Gr^{1/3} \left(\frac{T_w}{T_\infty} \right)^{0.18} \left(4.256 \frac{A_{apt}}{A_{cav}} \right)^s h(\theta, \theta_{max}, \theta_{stag}) \quad (6)$$

where

$$s = 0.56 - 1.01 \sqrt{\frac{A_{apt}}{A_{cav}}} ; \theta_{max} = -42.6^\circ ; \theta_{stag} = 90^\circ$$

$$h(\Theta) = \frac{1}{h_0} \left(1 - \cos(\Theta^{0.85} \cdot \pi) \right) ; \Theta = \frac{\theta - \theta_{stag}}{\theta_{max} - \theta_{stag}} ; h_0 = 1 - \cos(\Theta(\theta = 0)^{0.85} \cdot \pi)$$

The present study is focusing on the upward facing cases only. Here the inclination angle (θ) is set as θ_{max} corresponding to the maximum convective losses calculated from the equation:

$$\theta_{max} = -23^\circ - 260^\circ \frac{A_{apt}}{A_{cav}} \quad (7)$$

The stagnant angle (θ_{stag}) is normally equal to 90° , i.e. facing downward perpendicularly, where the convection is at the lowest level. h is the inclination factor as a function of the dimensionless inclination angle (Θ). s is a constant related to the aspect ratio: $\frac{A_{apt}}{A_{cav}}$, where A_{apt} and A_{cav} represent the areas of aperture and cavity. Nu is the Nusselt number and Gr is the Grashof number.

Finally, the temperature distribution at each layer is determined by simultaneous solution of Eq. (1) - (7), using the following convergence criterion:

$$\sqrt{\frac{1}{N_{layer} \times N_{surf}} \sum_{j=1}^{N_{layer}} \sum_{k=1}^{N_{surf}} \left[(T_{k,j}^i - T_{k,j}^{i-1}) / T_{k,j}^{i-1} \right]^2} < 10^{-6} \quad (8)$$

where T^i means the result of the i^{th} iteration.

4. Validation and limitations

4.1 Validation of models

Prior to the detailed analyses, the validity of the models proposed has been checked. A full experimental validation was out of scope due to missing experimental facilities of this size and geometry. However, validation against other valid models and experiments was used here instead.

The optical simulation model used here has previously been employed for optical analyses of dish concentrators and it has successfully been validated against TracePro® (Yang et al., 2018a). For the thermal analysis models used, detailed validation is difficult as our case is unique and data for validation is very limited. Therefore, we made use of Sandia's on-flux testing results (Moreno et al., 1991) related to another solar dish receiver design, which resembles ours. The Sandia case employs a dome structure as the absorbing surface and a crescent liquid sodium heat pipe as well. The main parameters of the two cases have been listed in Table 2. The temperature in the active region (the air-side dome absorbing surface) is 1,128 K, which is close to our result 1,155 K. The aspect ratio of the cavity is 0.12, which corresponds to $R_{apt}=220$ mm in our novel cases. The thermal efficiency of the cavity receiver system published in Sandia's testing results was 89.0%, which is slightly higher than the 86.6% obtained with our model. The main reason for the small deviation is the upward inclined cavity used in our models, which may increase the convective effect through the aperture compared to the traditional downward cases. Overall, the models used in this study should represent a good standing to be used for the analyses to follow.

Table 2. Comparison of simulations in present work and Sandia’s test data.

	Present model	Sandia test data
Cavity aspect ratios	0.12	0.12
Dome rim angle	66.4°	70°
Temperature in the active region	1,155 <i>K</i>	1,128 <i>K</i>
Average incident flux densities in active region	3.79×10 ⁵ W/m ²	3.78×10 ⁵ W/m ²
Cavity thermal efficiencies	86.6%	89.0%

4.2 Limitations

The main motivation of the present work was to verify the performance merits of the novel 2-stage dish concept compared to a traditional 2-stage dish. The focus was on a new receiver design, the liquid sodium heat-pipe dish receiver, which was analyzed in fixed thermodynamic states (atmospheric pressure and temperature range of 1080-1170 K). For this reason, a comprehensive parametric analysis was outside the scope of the present study and left to further work.

The optical and thermal properties and assumptions used in this paper are strictly limited to fixed thermodynamic states given above, i.e. the results are not directly applicable to other conditions. Also, steady-state conditions were assumed meaning that transient conditions, e.g. during start-up, shut-down, cloud shading, or other variations in solar radiance were not considered here.

5. Results

5.1 Radiation distribution in the semi-spherical target

First, we compared the radial distribution conditions at the semi-spherical targets

(R=459 mm) coupled to the novel and conventional 2-stage dish concentrators in same scale. Figure 3 shows the incident flux distributions. A larger irradiated area and more uniform flux distribution are obtained with the novel 2-stage concentrator explained by the so-called overlap effect (Wang et al., 2017). The relative sample variance (S^2) was further used to quantitatively describe the uniformity of the irradiation in the two cases. $S^2=0$ for a totally uniform distribution.

$$S^2 = \frac{1}{N_{act} - 1} \sum_{j=1}^{N_{act}} \left(\frac{q_j - \bar{q}}{q_{max}} \right)^2 \quad (9)$$

where N_{act} is the number of all surface segments within the irradiated active region. $\bar{q}$ and q_{max} represent the average and the maximum of the incident irradiation, respectively. For the novel case $S^2= 0.0769$ can be gotten, which is much better than the one of the conventional case $S^2= 0.1033$. The novel case has a larger active region: 60.9% of the semi-spherical area is covered by irradiation against 37.5% in the conventional case. The novel 2-stage dish concentrator is clearly superior to the conventional one in respect to the uniformity of the intercepted flux distribution and the utilization of the cavity area.

Figure 3. Distribution of incident irradiation at spherical target corresponding to (a) novel 2-stage dish concentrator, (b) conventional 2-stage dish concentrator.

5.2 Analysis of solar-to-thermal conversion

The scale of the cavities depends on the average value of the intercepted irradiation at the absorbing surface (active region), for which $3.8 \times 10^5 \text{ W/m}^2$ is used in this paper. Therefore, the lengths and radii of the cavities coupled to the dish concentrators are fixed (shown in Table 1). The sizes of the apertures should be optimized by balancing the accessible incident irradiances and the convective and radiative losses through the apertures. Figure 4 shows the conversion efficiencies against different opening radii for the novel and conventional 2-stage dish-receiver combinations. Note that $\eta_{optical}$ indicates how large share of the incident irradiation intercepted by the dish configuration can reach the absorbing surfaces. $\eta_{thermal}$ is defined as the ratio of the absorbed net flux to the incident flux reaching the inner surfaces of the cavity. The solar-to-thermal efficiency, or the total efficiency η_{total} , is a product of the two efficiencies above, i.e. $\eta_{total} = \eta_{optical} \times \eta_{thermal}$. The maximum efficiency values are at the point where the optimal aperture sizes are found, $R_{apt,nov}=180 \text{ mm}$ and $R_{apt,conv}=200 \text{ mm}$. The novel case shows a better solar-to-thermal performance with an optical efficiency ($\eta_{optical}$) of 77.8% against 72.0% and a thermal efficiency ($\eta_{thermal}$) of 88.3% against 85.2%, respectively. The maximum solar-to-thermal efficiency (η_{total}) is 68.6% with the novel 2-stage dish concentrator-receiver system. It is worth to noting that the results are affected by the assumptions and physical properties of the selected materials in Section 2 and 3 as well as the optical parameters of the dish concentrators. Here the ‘optimal’ parameters were calculated under imperfect optical conditions as follows: $\sigma_{tracking}=1.0 \text{ mrad}$ and $\sigma_{slope}=1.3 \text{ mrad}$ referring to our previous study (Yang et al., 2018a).

Figure 5 shows the share of the different heat loss components of the total heat losses

with different aperture sizes. In general, the radiative heat losses correspond to more than half of the total losses, and slightly increase with the radius. The convective losses vary from 33.3% to 39.4% for the novel concentrator case and from 37.7% to 40.4% in the conventional case. The conductive losses are $< 10\%$ in all cases, and it slightly decrease with increasing the opening radius. The heat losses in the receivers are clearly dominated by the radiative and convective conditions. For the optimal cases, the energy loss of the novel receiver is 18.8% less than that of the conventional one. The share of the convective and conductive heat losses decreases from 48% to 43% with the novel receiver due to a more compact structure, i.e. less heat exchange area. Finally the amount of the net flux absorbed by the heat pipe is 11.9% higher.

Figure 4. Optical, thermal and total efficiencies as functions of aperture radius in novel

and conventional cases. The novel case is marked in read and the conventional case in black, respectively.

435 Figure 5. Percentage distribution with respect to the conductive, convective and
 436 radiative heat losses in different aperture radii. (a) the novel case, (b) the conventional
 437 case.

438

439 5.3 Temperature distribution at the cavity walls

440

441 The temperature distributions at the inner surfaces of two receivers were also compared.

442 Figure 6 shows the results based on the optimal designs. The dome (the bottom of the
 443 cavity) is connected to the heat pipe components, where the temperature is constant at
 444 1155.8 K. The global temperature difference (ΔT) and the largest adjacent temperature

difference (δT) are defined in Eq. (10) and (11):

$$\Delta T = T_{\max} - T_{\min} \quad (10)$$

$$\delta T = \max \left\{ \left| T_{k,j} - T_{k,j-1} \right|, \left| T_{k,j} - T_{k,j+1} \right|, \left| T_{k,j} - T_{k-1,j} \right|, \left| T_{k,j} - T_{k+1,j} \right| \right\} \quad (11)$$

The ΔT of the novel case is only 51 K, whereas the conventional case has a $\Delta T = 79$ K. Moreover, the δT of the novel case is 23.1 K, also less than the conventional case of 51.0 K. A low δT is important for keeping the local thermal stress at a relevantly low level. Therefore, the novel dish receiver is expected to operate more safely with a longer service life. Note that an 1D conductive heat transfer model is used here which means that no heat exchange is considered between the adjacent cells in the same layer. Therefore, in real-world conditions, even a lower δT could be possible.

Figure 6. Comparison of 2D temperature contour images between the novel (a) and conventional (b) cases.

6. Conclusions

A new design of the cavity receiver coupled to a 2-stage dish concentrator has been presented. Comprehensive comparisons on optical and thermal performance between the novel and conventional concentrator cases were done.

The main conclusions are the following:

The novel 2-stage dish concentrator proposed previously enables improving the uniformity of the incident irradiation on the absorbing surface due to the so-called overlap effect. The rim angle increases from 47.5° to 66.4° and the S^2 shrinks from 0.103 to 0.077 compared to a conventional 2-stage dish concentrator.

The novel design can also achieve a higher optical and thermal performance simultaneously. As a result, the solar-to-thermal efficiency is improved. The optimal case has 18.8% lower losses than the conventional system under the same conditions.

The simulations done show that the convective and radiative loss components represent >90% of the total heat losses in the cavity receiver. The convective losses alone represented >30% of the total, but could further be suppressed by advanced designs such as the air curtain (Yang et al., 2018b).

The simulated temperature distributions at the cavity walls shows that the novel dish system has lower global and temperature gradients, which could provide a longer operational life-time than with the conventional dish-receiver system.

The new receiver design is fixed at ground level and it is less limited in weight and volume. Therefore, a large single unit with a power of 200-kW could theoretically be possible with the new design.

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