Numerical Investigation of a Helically Coiled Solar Cavity Receiver for Simultaneous Generation of Superheated Steam and Air

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- Research questions/Problems
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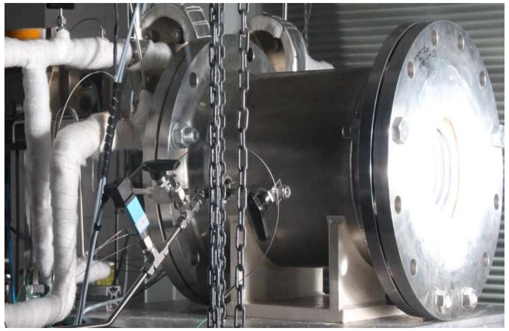
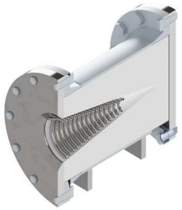
Background

- Concentrated solar thermal technologies have enormous potential for a carbon-neutral society.
- The high-grade thermal energy can be used for electricity generation or process heat.
- One promising application is hydrogen generation in a solid oxide electrolyzer (SOEC) which requires high-temperature air and steam (~850°C).



Solar cavity receiver (DLR FF)

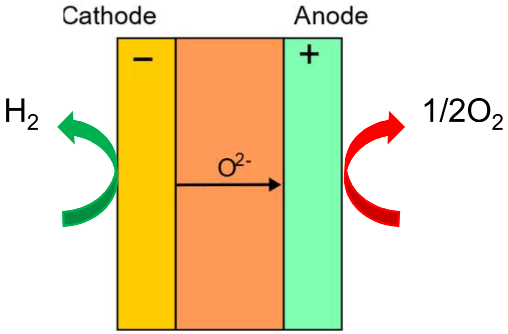
SOEC (DLR TT)



QSP Future Fuels

Steam: 5 kg/h, 700℃
SOEC: 2 kWel
Hydrogen: 1600 L (4h)

SOEC (700-1000°C)

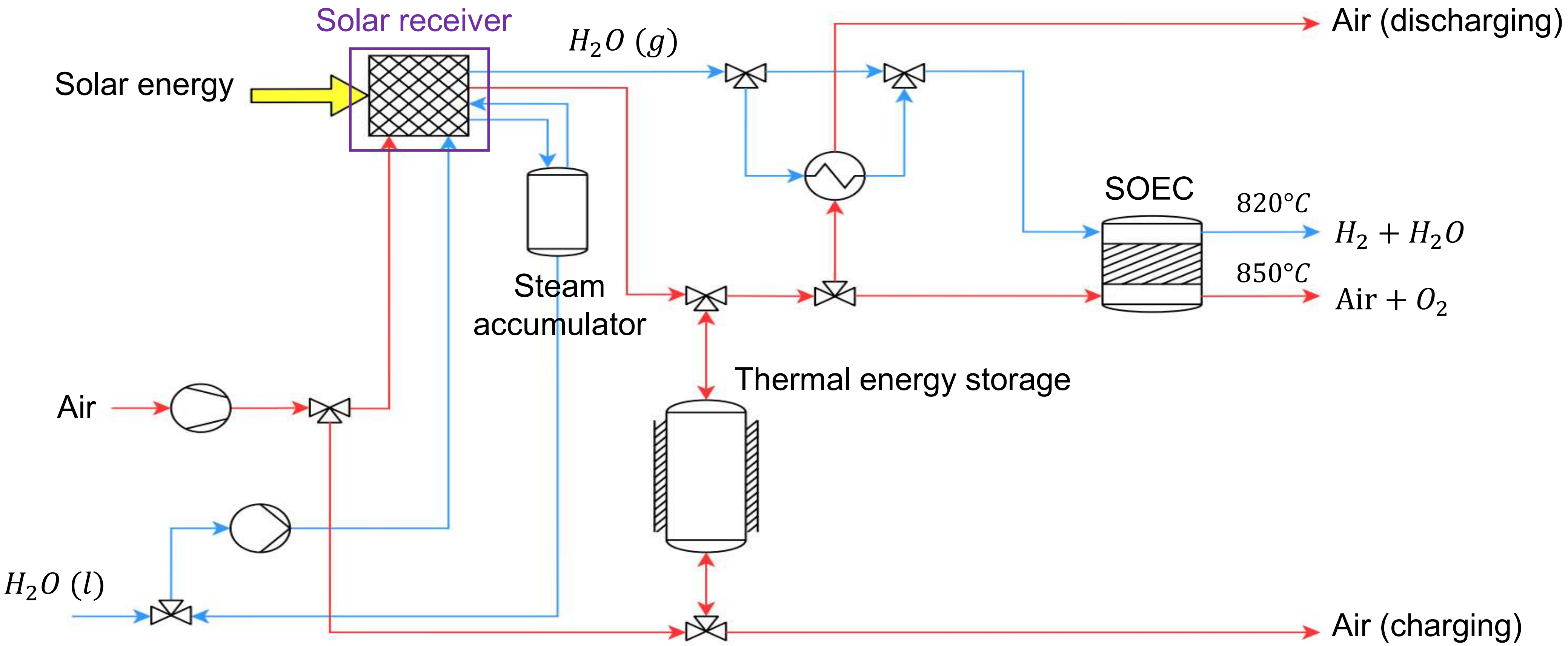


Anode: $O^{2-} \rightarrow 2e^- + \frac{1}{2}O_2$

Cathode: $H_2O + 2e^- \rightarrow H_2 + O^{2-}$

Overall reaction: $H_2O \rightarrow H_2 + \frac{1}{2}O_2$

Background

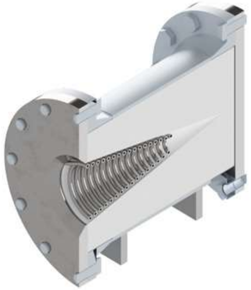

Solar receiver
Solar energy
$H_2O\ (g)$
Air (discharging)
SOEC
820°C
$H_2 + H_2O$
850°C
Air + O_2
Steam accumulator
Thermal energy storage
Air
$H_2O\ (l)$
Air (charging)

Research questions/Problems

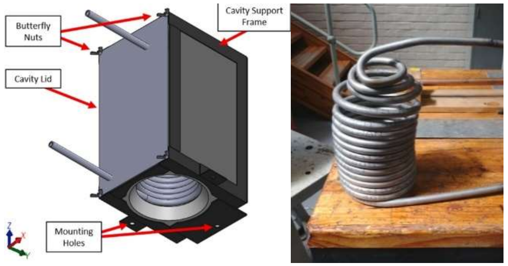
*DS: direct steam generation
SA: superheating air

- How to produce the superheated steam and air for SOEC (cavity receiver design)?
- Is it possible to simultaneously produce superheated steam and air with a single cavity receiver?
- If possible, what would be the receiver’s performance?

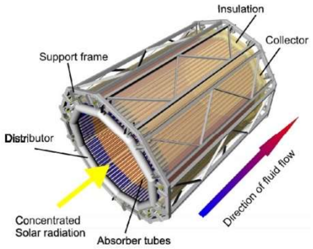
Only few studies are done with the above mentioned receiver!



QSP Future Fuels (DS)



Swanepoel et al. (DS) [1]



SOLUGAS (SA) [2]

SOLGATE (SA) [3]

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[1] Swanepoel, J. K., *et al.* (2021). Helically coiled solar cavity receiver for micro-scale direct steam generation. *Appl. Therm. Eng.*, *185*, 116427.
[2] R. Uhlig, *et al.*, Energy Procedia, 69 (2015) 563-572.
[3] Heller, P., *et.al.* (2006). Test and evaluation of a solar powered gas turbine system. *Solar energy*, *80*(10), 1225-1230.

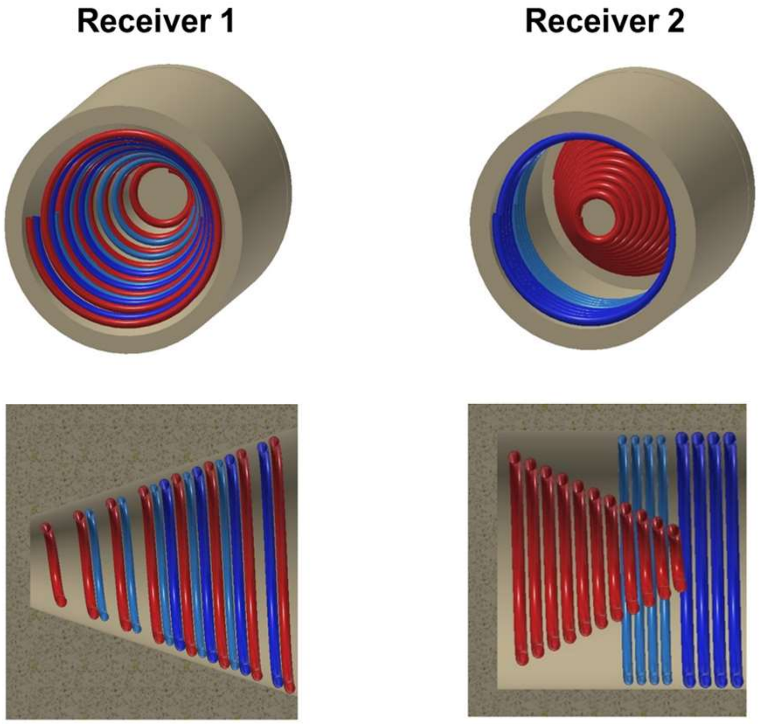
Objective

- Develop and numerically investigate a single cavity receiver that can conduct the following three processes: (i) water evaporation, (ii) superheating steam and (iii) superheating air.

Approach

Overall

1. Determine the cavity receiver’s geometry
2. Conduct ray-tracing analysis to obtain a distributed heat flux map inside the proposed receiver
3. Conduct steady-state thermal analysis by using 1D two-phase fluid flow-3D cavity heat transfer combined numerical model



Red: Superheating air Light blue: Superheating steam Blue: Evaporating water

Approach

1. Determine the cavity receiver's geometry

- Absorber tube diameter and length
- Insulation cavity diameter and length

SOEC Operation Conditions
(Thermoneutral Operation, 50 kWe SOEC)

	Steam Flow (Cathode)	Sweep Gas Flow (Anode)
Temperature	820°C	850°C
Mass Flow	20 kg/h	80 kg/h
Pressure	1 atm	1 atm

Other conditions

- ΔT between fluid and tube wall < 150℃
- Fluid average velocity < 50 m/s
- Aspect ratio (cavity diameter/length) = 1

Approach

2. Ray-tracing analysis

FEMRAY (***F***inite ***E***lement ***M***esh ***Ray*** Tracing)

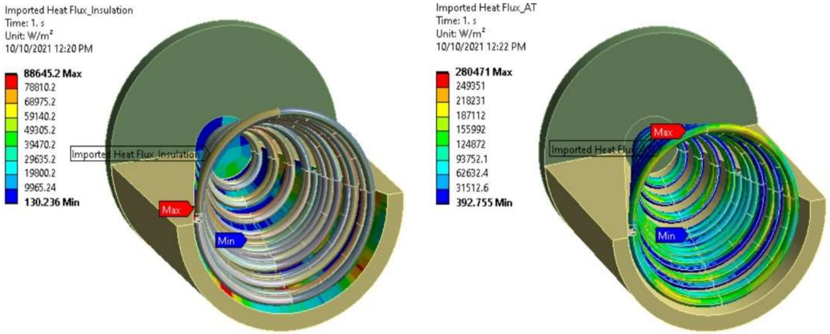
-> The code uses the geometric data and the optical properties of each finite element of the mesh to calculate the expected heat flux distribution using a simple ray-tracing algorithm.

Synlight
(100 kW)

SPRAY
(Solar Power
Raytracing Tool)

FE Model

ANSYS Workbench

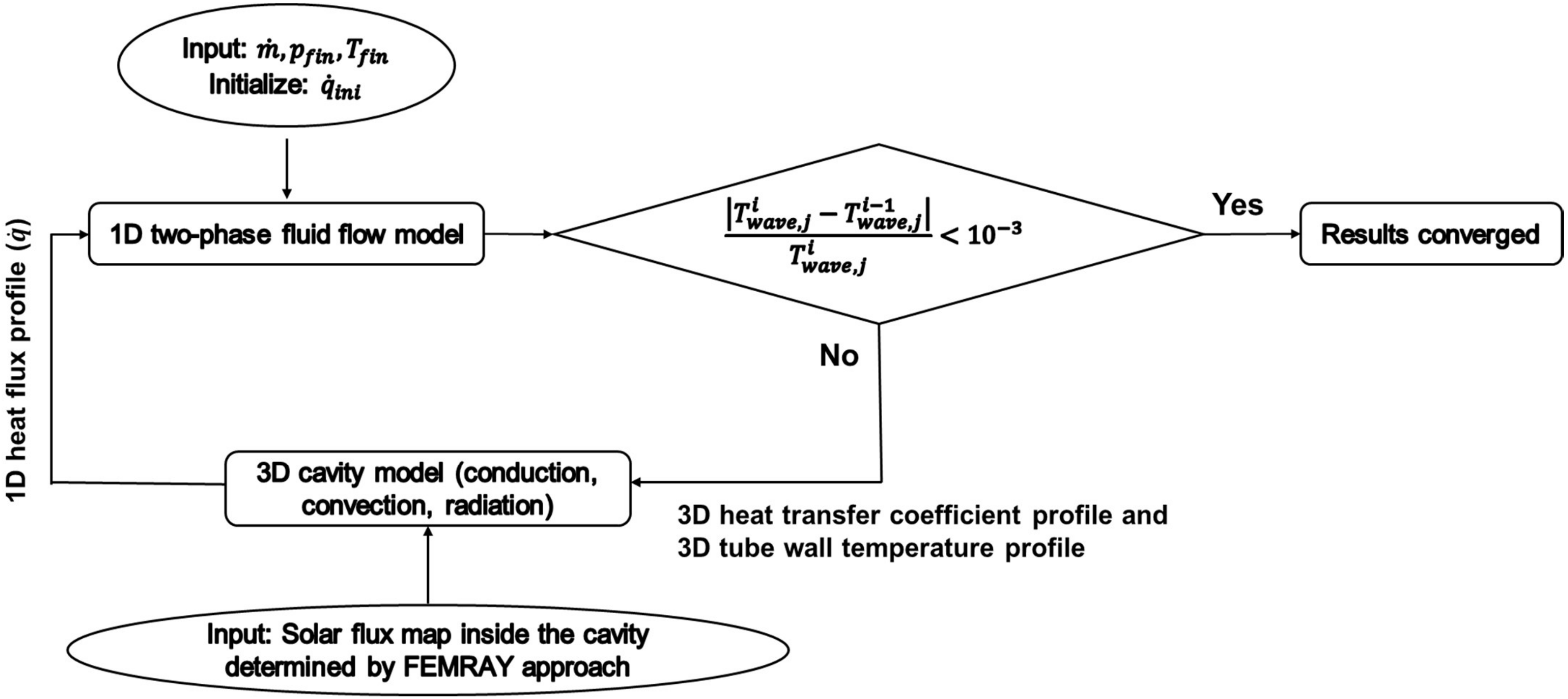


Solar heat fluxes

Temperature distribution

Approach

3. Steady-state thermal analysis (entire flow)



Input: $\dot{m}, p_{fin}, T_{fin}$
Initialize: $\dot{q}_{ini}$

1D two-phase fluid flow model

$\frac{\left|T^{i}_{wave,j} - T^{i-1}_{wave,j}\right|}{T^{i}_{wave,j}} < 10^{-3}$

Yes

Results converged

No

3D cavity model (conduction, convection, radiation)

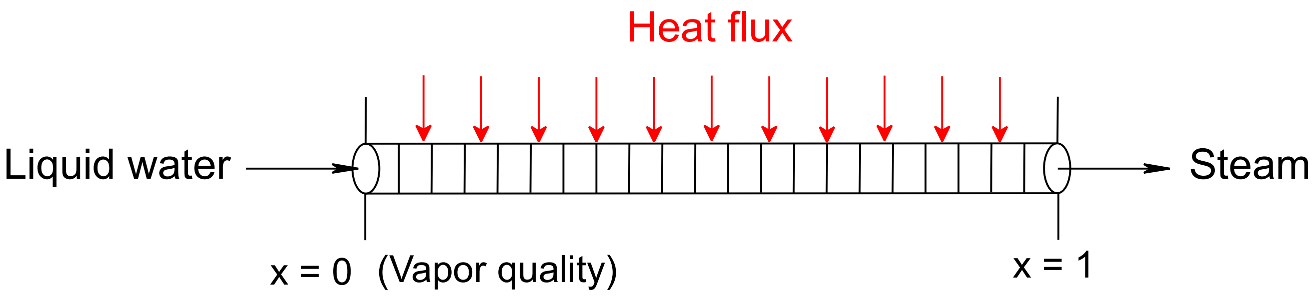
3D heat transfer coefficient profile and
3D tube wall temperature profile

1D heat flux profile ($\dot{q}$)

Input: Solar flux map inside the cavity determined by FEMRAY approach

Approach

3. Steady-state thermal analysis (1D two-phase fluid flow model)


Heat flux

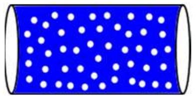
Liquid water

Steam

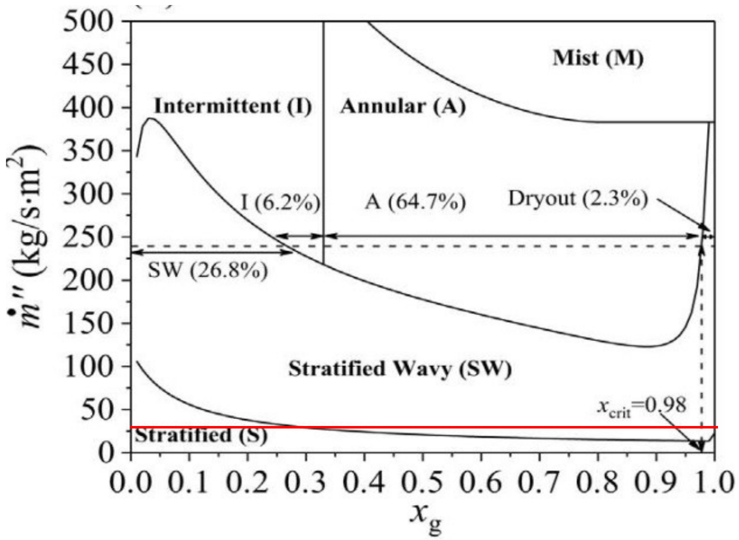
x = 0 (Vapor quality)

x = 1

1. Nucleate boiling region (bubbly flow) [1]



2. Forced convection evaporation region (stratified, stratified wavy flow) [1]



Thome-El Hajal maps

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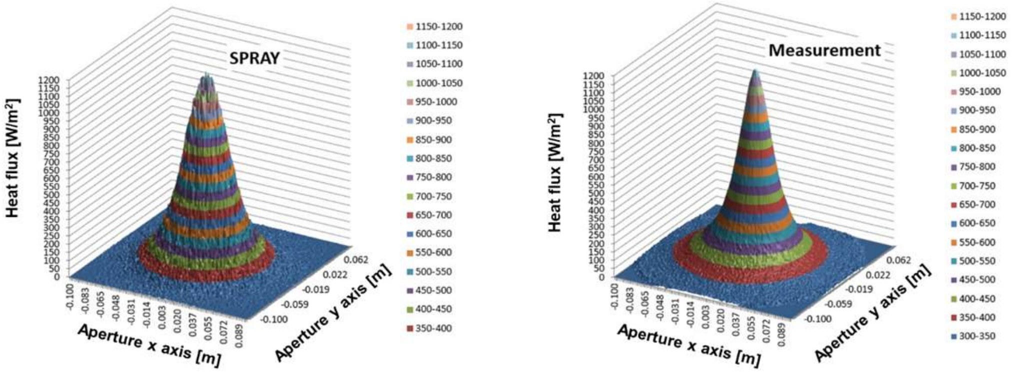
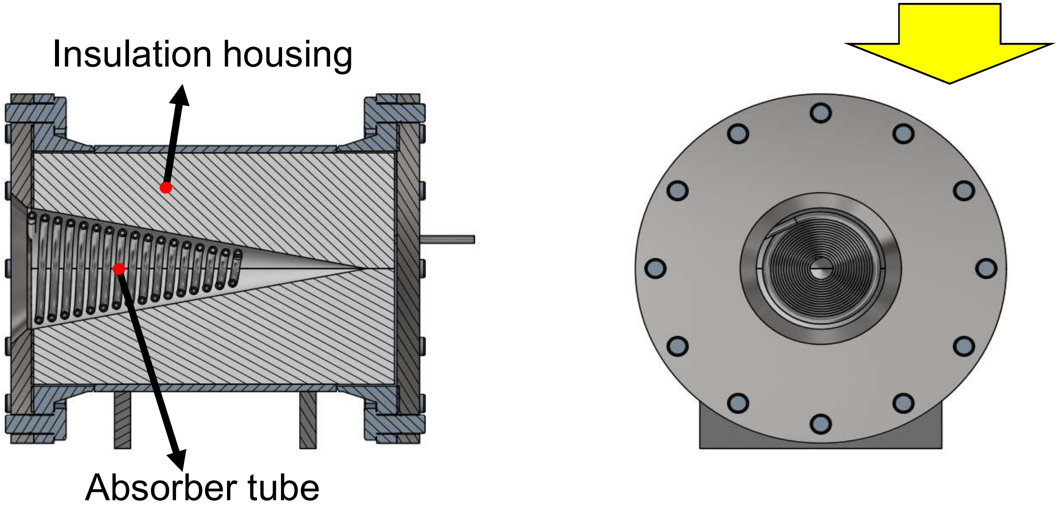
[1] Fsadni, A. M., *et.al*. (2016). A review on the two-phase heat transfer characteristics in helically coiled tube heat exchangers. *Int. J. Heat Mass Transf*, *95*, 551-565.

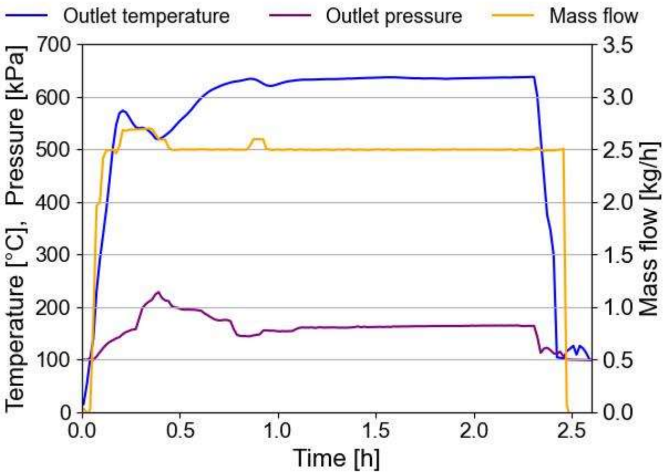
Validation, Conical helical tube and non-uniform heat flux

Solar simulator in Cologne



3D heat flux map at receiver's aperture (4.18 kW)

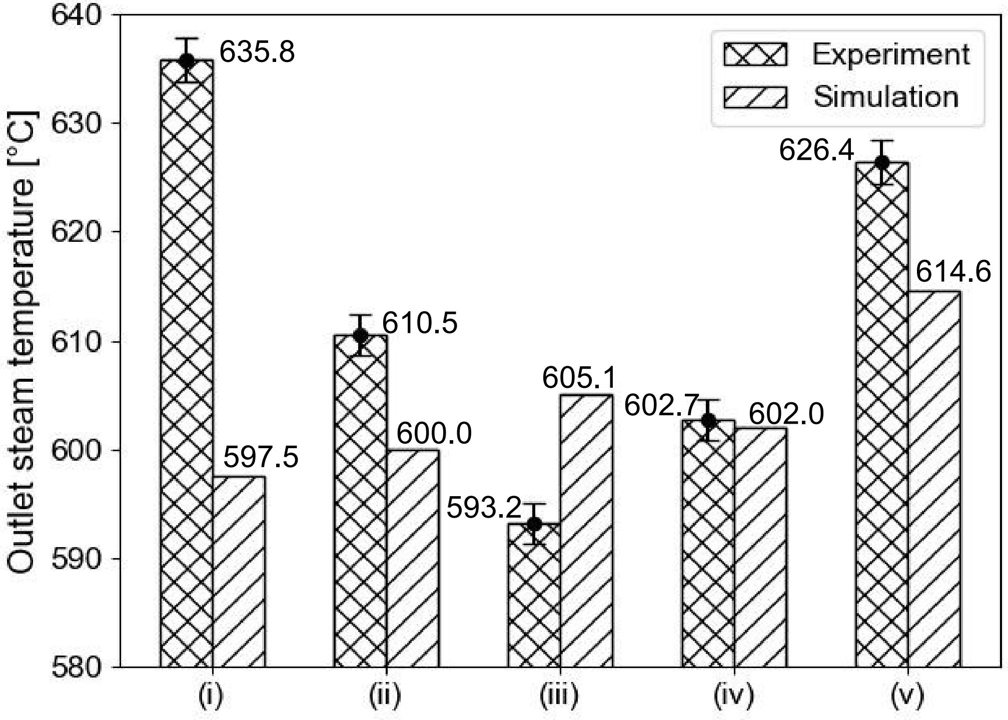




Solar steam generator used in DLR project QSP Future Fuels **Experimental results**

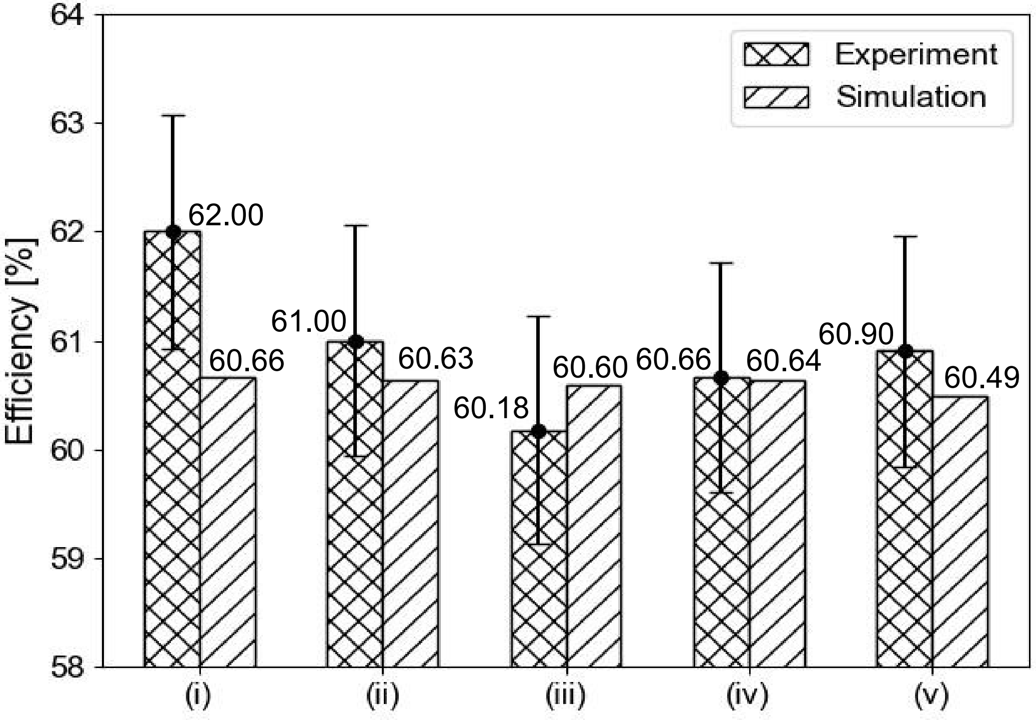
Validation, Conical helical tube and non-uniform heat flux

Outlet steam temperature



Efficiency

(Energy absorbed by fluid/Energy from solar simulator)



Deviation < 6%

Inlet temperature: 15-30℃
Inlet pressure: 1.5-2.0 bar
Mass flow: 2.5 kg/h

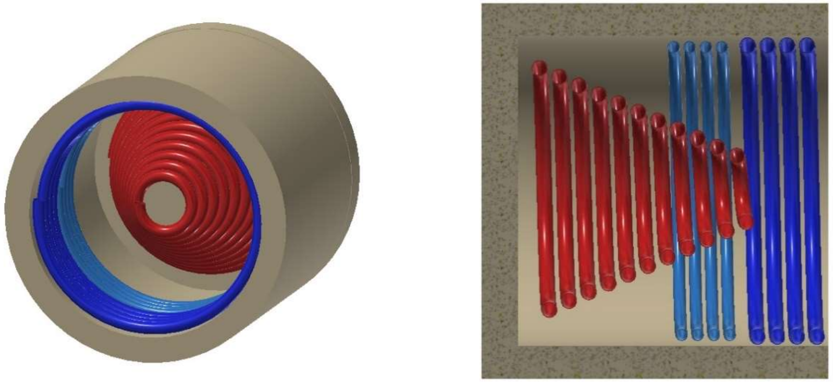
Summary

- Concentrated solar thermal technologies are combined with SOEC to produce green hydrogen
- We are numerically investigating a single solar cavity receiver that can conduct the following three processes: (i) water evaporation, (ii) superheating steam, and (iii) superheating air
- FERMRAY approach is used to analyze the heat flux map inside the cavity receiver
- 1D two-phase fluid flow model and 3D cavity heat transfer model are combined for the steady-state thermal analysis
- Validation was conducted with project QSP Future Fuels’ experimental results and showed deviation < 6%

Outlook

- Complete the numerical investigation of proposed solar cavity receiver (results will be published in the future).
- Experimental investigation of Receiver 2 by using Synlight.

Receiver 2



Red: Superheating air Light blue: Superheating steam Blue: Evaporating water

Thank you very much for your attention! (Q & A)

Acknowledgments

Institute of Future Fuels, DLR: Timo Roeder, Vamshi Krishna Thanda, Kai Risthaus, Nathalie Monnerie, Christian Sattler

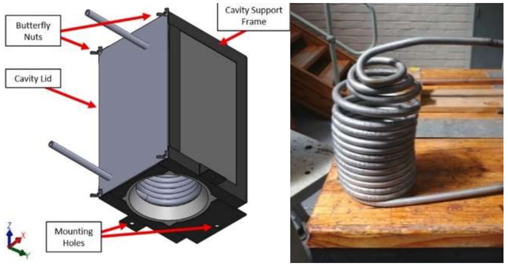
Institute of Solar Research, DLR: Reiner Buck, Ralf Uhlig

Background

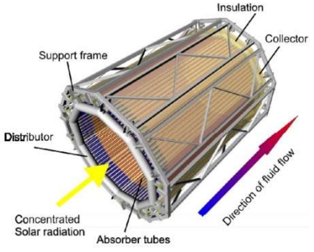
*DS: direct steam generation
SA: superheating air

- Two basic receiver types exist: (1) external receivers and (2) cavity receivers.

Ivanpah (DS) [1]



Swanepoel *et al.* (DS) [2]



SOLUGAS (SA) [3]

(1)

(2)

- Cavity receivers are considered highly efficient receivers.
- Most cavity receiver designs consist of a helically coiled tube due to its compact design, easy manufacturing, and higher heat transfer performance.

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[1] Ho, C. K. (2017). Advances in central receivers for concentrating solar applications. *Solar energy*, *152*, 38-56.

[2] Swanepoel, J. K., *et al.* (2021). Helically coiled solar cavity receiver for micro-scale direct steam generation. *Appl. Therm. Eng.*, *185*, 116427.

[3] R. Uhlig, *et al.*, Energy Procedia, 69 (2015) 563-572.

Radiation, convection and conduction

- **Radiation inside the cavity (surface-to-surface model)**

$$J_i = E_i + R_i \sum_{j=1}^{N} F_{ij} J_j$$

J_i: radiosity [W]
E_i: emissive power [W]
R_i: reflectivity
F_{ij}: view factor

- **Radiation outside the cavity**

$$\dot{Q}_{rad,out} = \varepsilon_{cav,out}\sigma\left(\bar{T}_{cav,out}^4 - T_{amb}^4\right)A_{cav,out}$$

- **Natural convection outside the cavity**

$$h_{nc,out} = 5.0\ \mathrm{W/m^2\ K}$$

- **Natural convection inside the cavity**

$$h_{nc,in} = \frac{k_{air}}{L_{rel}} Nu_{nc}$$

$$Nu_{nc} = 0.0196 Ra_{nc}^{0.41} Pr^{0.13}$$

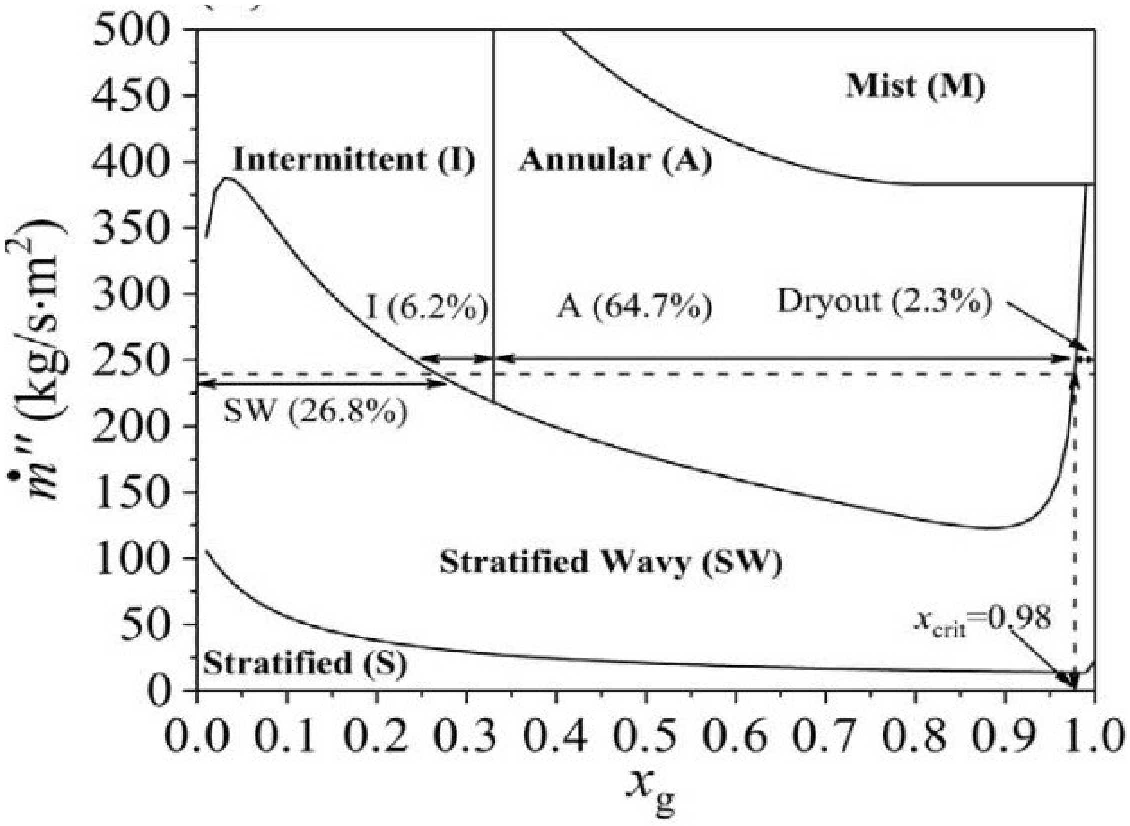
$$Ra_{nc} = \frac{g\beta_{air}\left(\bar{T}_{cav,in} - T_{amb}\right)L_{rel}^3}{\nu_{air}\alpha_{air}}$$

$$L_{rel} = \left|\sum_{i=1}^{3} a_i cos(\phi + \varphi_i)^{b_i} L_i\right|$$

S. Paitoonsurikarn, *et.al.* (2011). Numerical Investigation of Natural Convection Loss From Cavity Receivers in Solar Dish Applications. *J. Sol. Energy Eng.*, 133, 021004.

β_{air}: thermal expansion coefficient [1/K]
α_{air}: thermal diffusivity [m²/s]
k_{air}: thermal conductivity [W/m K]
ν_{air}: kinematic viscosity [m²/s]

Thome-El Hajal maps



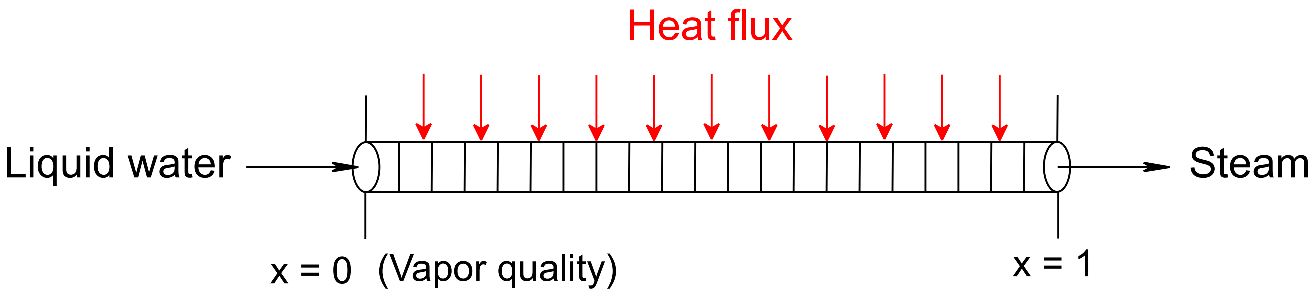
- Mass flux in our study: < 50 kg/s m^2

Stratified or stratified wavy flow is a dominant flow

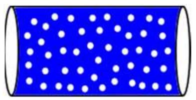
*Thome-El Hajal map is for the horizontal tube, but the same flow pattern is applied for helical tube due to the centrifugal force.

Approach

3. Steady-state thermal analysis (1D two-phase fluid flow model)



1. Nucleate boiling region (bubbly flow) [1]



$$\frac{h_{fc,tp}}{h_{fc,sp}} = 1 + 2.21\left(\frac{1}{\mathrm{X}_{tt}}\right)^{0.3} \quad (1/\mathrm{X}_{tt} < 1.2)$$

2. Forced convection evaporation region (stratified, stratified wavy flow) [1]

$$\frac{h_{fc,tp}}{h_{fc,sp}} = 3.24\left(\frac{1}{X_{tt}}\right)^{0.165} \quad (1/\mathrm{X}_{tt} \geq 1.2)$$

Martinelli parameter

$$\mathrm{X}_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}$$

Vapor quality

$$x = \frac{\dot{Q}_{tot}}{\dot{m}L_{vap}}$$

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[1] Fsadni, A. M., *et.al*. (2016). A review on the two-phase heat transfer characteristics in helically coiled tube heat exchangers. *Int. J. Heat Mass Transf*, *95*, 551-565.

Approach

3. Steady-state thermal analysis (1D two-phase fluid flow model)

Heat transfer coefficient in a single-phase flow [1]

$$h_{fc,sp} = \frac{k_{fluid}}{d_{in}} Nu_{fc}$$

Frictional pressure loss in a single-phase flow (Darcy-Weisbach equation)

$$\Delta p_{sp} = f_D \frac{L}{d_{in}} \frac{1}{2} \rho V^2$$

Frictional pressure loss in a two-phase flow [2]

$$\Delta p_{tp} = K(x) \frac{G^{1.91}}{d_{in}^{1.2} \rho_m} L$$

Outlet fluid temperature

$$T_{fout,j} = T_{fin,j} + \frac{\dot{q} A_{T,out}}{\dot{m} c_{sp}}$$

Tube wall temperature (Newtons law of cooling)

$$\dot{q} = h_{fc} \left(T_{wave,j} - T_{fave,j}\right)$$

Two-phase frictional multiplier

$$K(x) \\ = -0.0373x^3 + 0.0367x^2 \\ - 0.00539x + 0.0108$$

Flow rate mixture density

$$\rho_m = \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l}\right)^{-1}$$

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[1] VDI e. V. (2010) *VDI Heat Atlas (2 ed.)*. Springer, Berlin, Heidelberg.

[2] Santini, L., *et. al*. (2008). Two-phase pressure drops in a helically coiled steam generator. *Int. J. Heat Mass Transf*, *51*(19-20), 4926-4939.

Nusselt number for single-phase flow

(i) If $Re \leq 6300$ (laminar flow), then,

$$Nu_{fc} = 3.66 + 0.08\left[1 + 0.8\left(\frac{d_{in}}{D}\right)^{0.9}\right] Re^{m} Pr^{\frac{1}{3}}\left(\frac{Pr}{Pr_{w}}\right)^{0.14}$$

where

$$m = 0.5 + 0.2903\left(\frac{d_{in}}{D}\right)^{0.194}$$

(ii) If $Re > 10000$ (fully developed turbulent flow), then,

$$Nu_{fc} = \frac{\left(\frac{\xi}{8}\right) RePr}{1 + 12.7\sqrt{\frac{\xi}{8}}\left(Pr^{\frac{2}{3}} - 1\right)}\left(\frac{P_{r}}{P_{rw}}\right)^{0.14}$$

where

$$\xi = \left[\frac{0.3164}{Re^{0.25}} + 0.03\left(\frac{d_{in}}{D}\right)^{0.5}\right]\left(\frac{\mu_{w}}{\mu}\right)^{0.27}$$

(iii) If $6300 < Re \leq 10000$ (transition region), then,

$$Nu_{fc} = \gamma Nu_{lam,6300} + (1 - \gamma) Nu_{turb,10000}$$

where

$$\gamma = \frac{10000 - Re}{10000 - 6300} \qquad (0 \leq \gamma \leq 1)$$

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[1] VDI e. V. (2010) *VDI Heat Atlas (2 ed.)*. Springer, Berlin, Heidelberg.

Darcy friction factor for single-phase flow [1]

(i) If $Re \leq 500$ (low laminar flow), then,

$$f_D = 4 \times 38 \times Re^{-1} \left(\frac{d_{in}}{D}\right)^{0.15}$$

(ii) If $500 < Re \leq 6300$ (laminar flow), then,

$$f_D = 4 \times 5.25 \times Re^{-\frac{2}{3}} \left(\frac{d_{in}}{D}\right)^{0.15}$$

(iii) If $6300 < Re \leq 10000$ (transition flow), then,

$$f_D = 4 \times 0.31 \times Re^{-\frac{1}{3}} \left(\frac{d_{in}}{D}\right)^{0.15}$$

(iv) If $Re > 10000$ (turbulent flow), then,

$$f_D = 4 \times 0.045 \times Re^{-\frac{1}{8}} \left(\frac{d_{in}}{D}\right)^{0.15}$$

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[1] Ali, S. (2001). Pressure drop correlations for flow through regular helical coil tubes. *Fluid Dyn. Res.*, *28*(4), 295.

Validation, Cylindrical helical tube and uniform heat flux

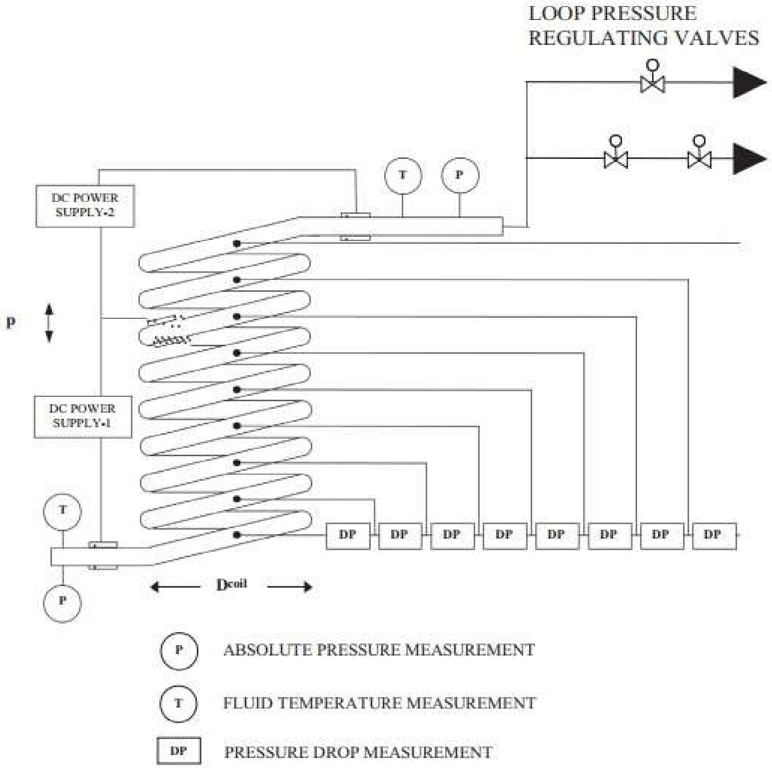

Table. Parameters and conditions used in Santini *et al.* [1, 2].

Parameters	Value
Helical tube inner diameter	0.01249 m
Helical tube outer diameter	0.01723 m
Helical coil diameter	1.00 m
Total length of the tube	32.0 m
System pressure	2 MPa (a)
Temperature at inlet	181.61° C
Averaged flux	51.0 kW/m^2
Mass flux	206.0 kg/m^2 s

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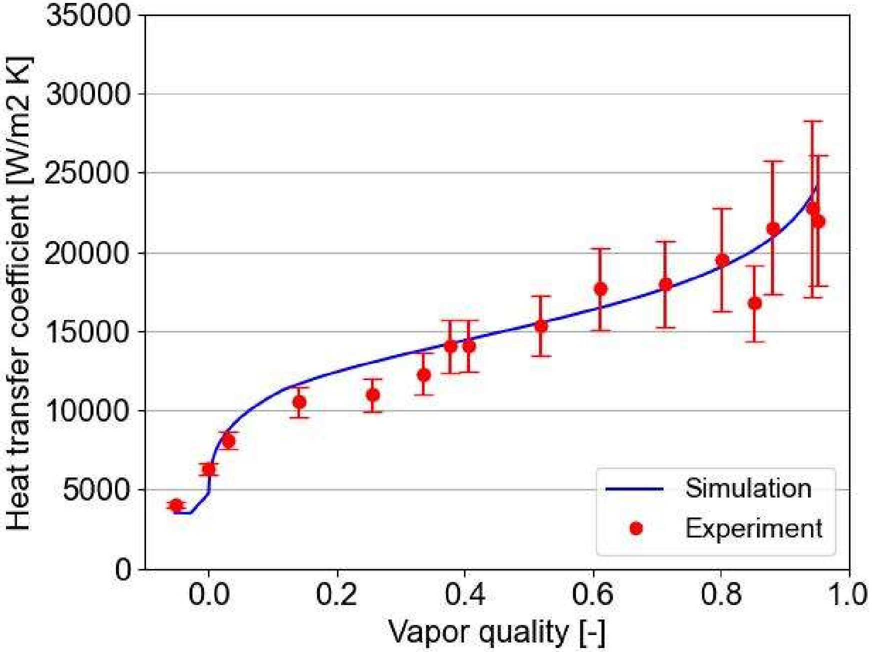
[1] Santini, L., *et.al*. (2016). Flow boiling heat transfer in a helically coiled steam generator for nuclear power applications. *Int. J. Heat Mass Transf.*, *92*, 91-99.

[2] Santini, L., *et. al*. (2008). Two-phase pressure drops in a helically coiled steam generator. *Int. J. Heat Mass Transf*, *51*(19-20), 4926-4939.

Validation, Cylindrical helical tube and uniform heat flux

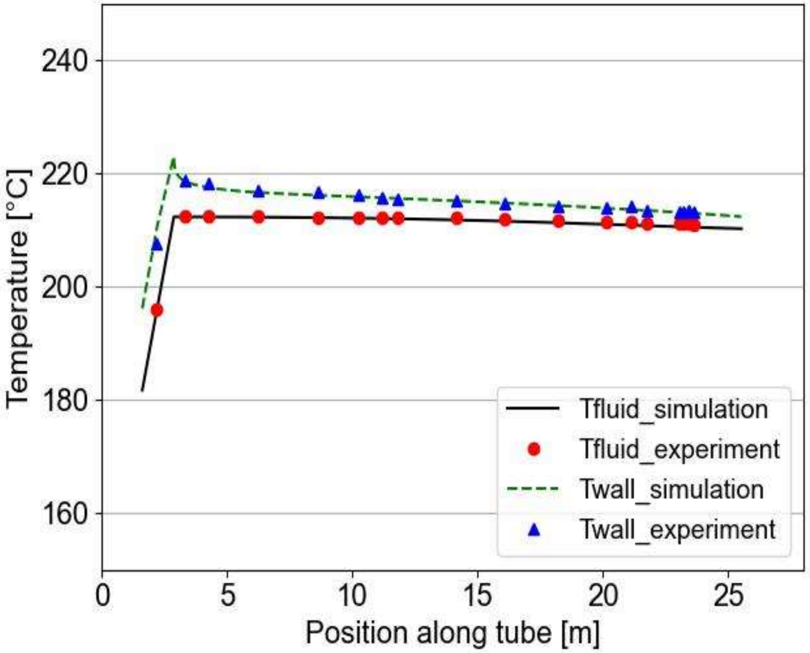
Heat transfer coefficient



Ave. deviation: 8.213%

Max. deviation: 31.91%

Fluid temperature and tube wall temperature

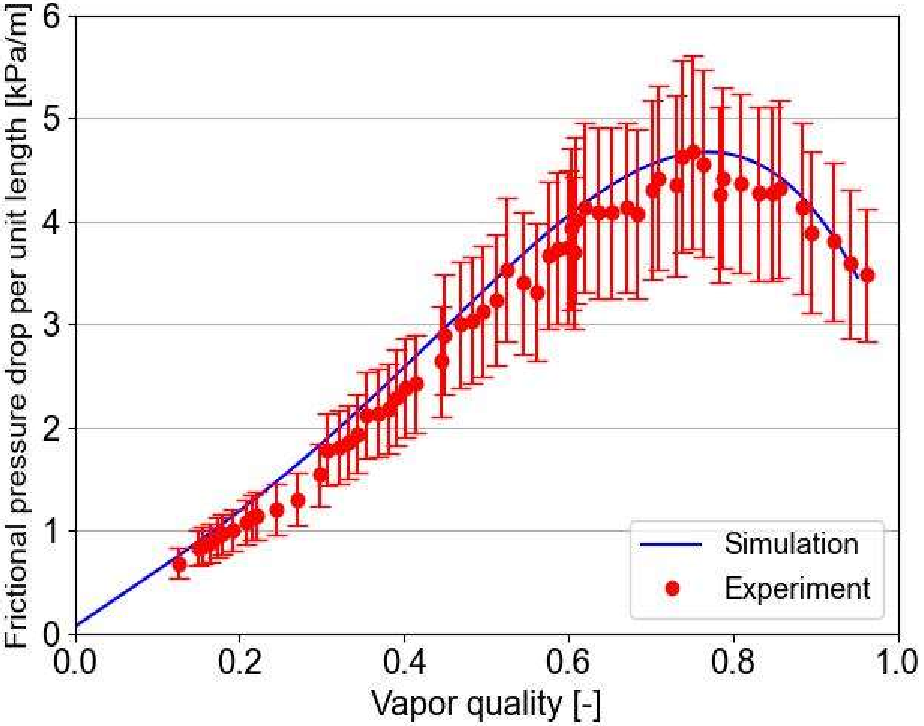


Ave. deviation (T_{fluid}): 0.2279% Ave. deviation (T_{wall}): 0.3362%

Max. deviation (T_{fluid}): 0.4310% Max. deviation (T_{wall}): 1.952%

Validation, Cylindrical helical tube and uniform heat flux

Two-phase frictional pressure drop



Ave. deviation: 6.954%

Max. deviation: 20.19%

Average deviation < 10%
Good agreement!!

Validation, Conical helical tube and non-uniform heat flux

Specifications of solar steam generator

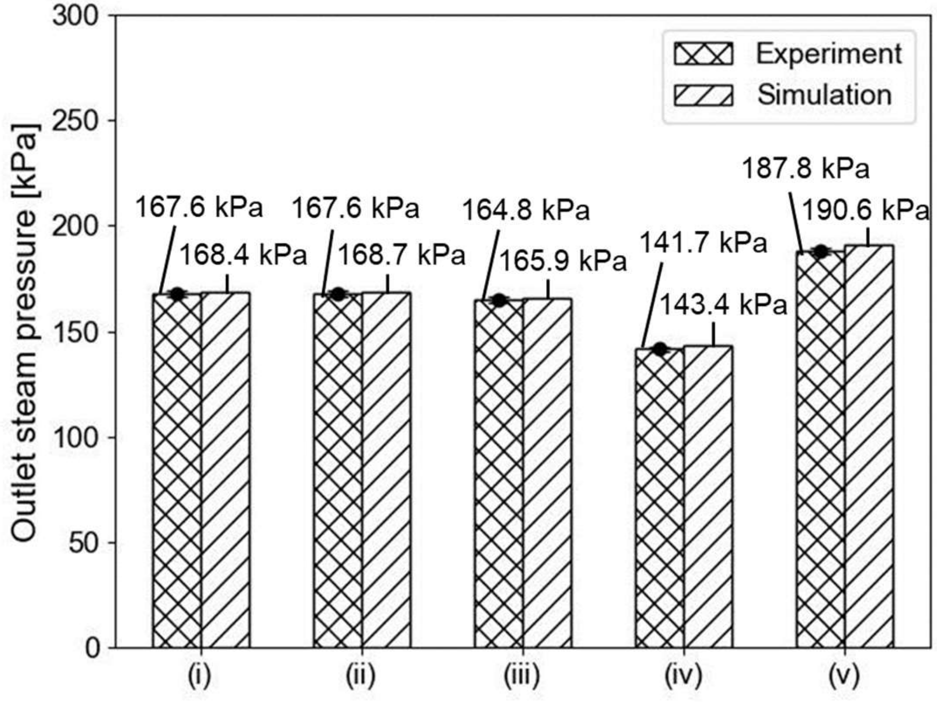
Parameters	**Value**
Cavity inner diameter	0.160 m
Cavity outer diameter	0.305 m
Cavity length	0.450 m
Helical tube inner diameter	0.006 m
Helical tube outer diameter	0.010 m
Helical tube length	4.428 m
Helical tube pitch	0.012 m
Helical tube taper angle	14°
Helical tube winding number	14
Surface emissivity of tube	0.930

Average value of experimental data

Experiment	**Inlet temp. [℃]**	**Inlet pressure [kPa (a)]**	**Mass flow [kg/h]**
(i)	16.5	176.2	2.498
(ii)	18.6	176.5	2.499
(iii)	21.8	174.0	2.500
(iv)	20.3	152.2	2.501
(v)	27.5	198.0	2.498

Validation, Conical helical tube and non-uniform heat flux

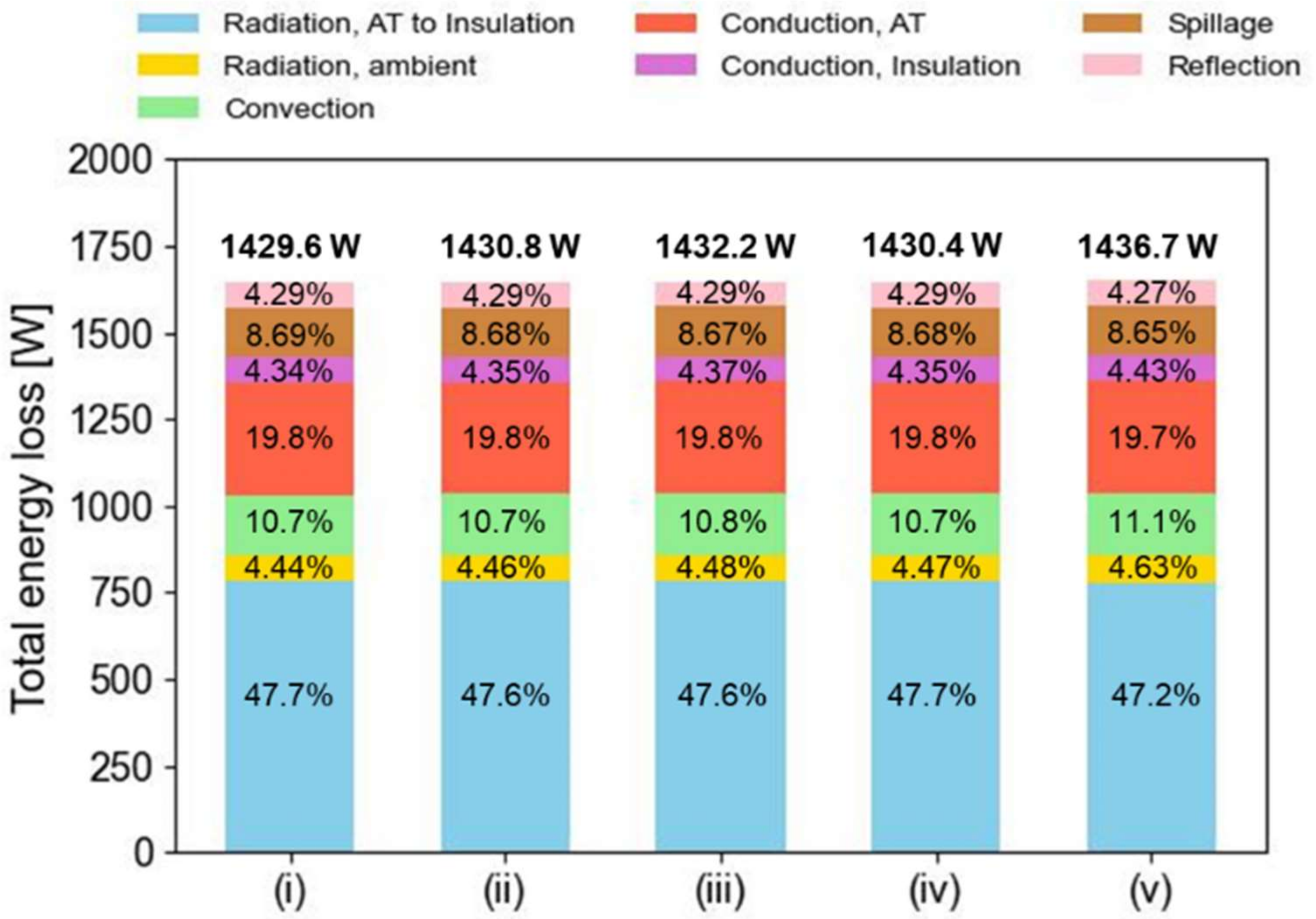
Outlet steam pressure



Deviation: (i) 0.4817%, (ii) 0.6085%, (iii) 0.6966%,
(iv) 0.03909%, (v) 0.6818%

Deviation < 6%
Good agreement!!

Breakdown of energy losses from the solar cavity receiver

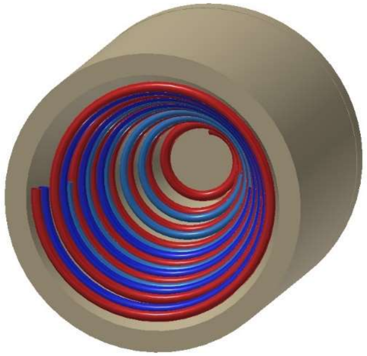


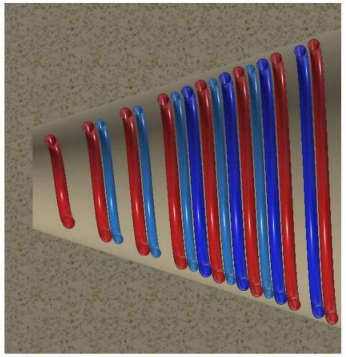
- Since the majority of energy is collected to AT, its energy loss becomes much larger than the other parts

- Since almost all of the solar simulator's heat flux is applied to the helical tube's inner surface, more energy transfers from the inner to outer surface as thermal conduction energy.

Receiver design

Receiver 1





	Water evaporation	Steam superheat	Air superheat
Mass flow	30 kg/h	30 kg/h	120 kg/h
Inlet temperature	20 ℃	151.84 ℃	20 ℃
Outlet temperature	151.84 ℃	820 ℃	850 ℃
Fluid pressure	500 kPa	500 kPa	500 kPa
Inner tube diameter	0.020 m	0.015 m	0.020 m
Outer tube diameter	0.022 m	0.017 m	0.022 m
Winding number	3.5	4.5	7.75
Pitch	0.067 m	0.067 m	0.067 m
Taper angle	20°	20°	20°
Tube length	4.325 m	4.368 m	7.595 m

Red: Superheating air

Light blue: Superheating steam

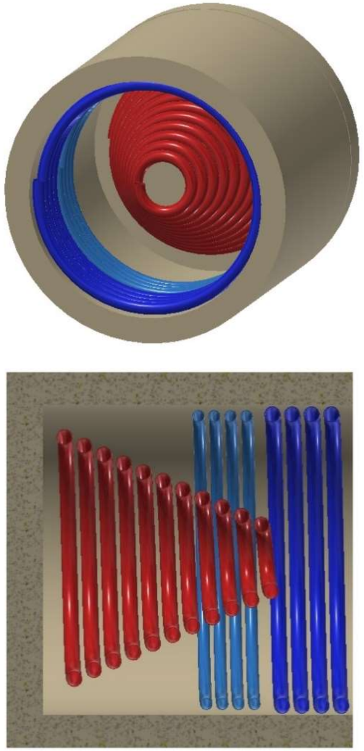
Blue: Evaporating water

Cavity diameter; 0.535 m

Cavity length; 0.550 m

Receiver design

Receiver 2



Red: Superheating air

Light blue: Superheating steam

Blue: Evaporating water

	Water evaporation	Steam superheat	Air superheat
Mass flow	30 kg/h	30 kg/h	120 kg/h
Inlet temperature	20 °C	151.84 °C	20 °C
Outlet temperature	151.84 °C	820 °C	850 °C
Fluid pressure	500 kPa	500 kPa	500 kPa
Inner tube diameter	0.020 m	0.015 m	0.020 m
Outer tube diameter	0.022 m	0.017 m	0.022 m
Winding number	3.5	3.5	11.0
Pitch	0.027 m	0.022 m	0.027 m
Taper angle	0°	0°	24°
Tube length	4.398 m	4.344 m	7.529 m

Cavity diameter; 0.425 m

Cavity length; 0.425 m