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Development of a Direct Supercritical Carbon Dioxide Solar Receiver Based on Compact Heat Exchanger Technology

Saeb M. Besarati, D. Yogi. Goswami, Elias K. Stefanakos

Overview

- Background information
- Computational model for a direct s-CO_2 receiver
- Design a of a 3 MW_{th} cavity receiver
- Thermal performance evaluation
- Summary and conclusions

Solar power tower

Ref (5)

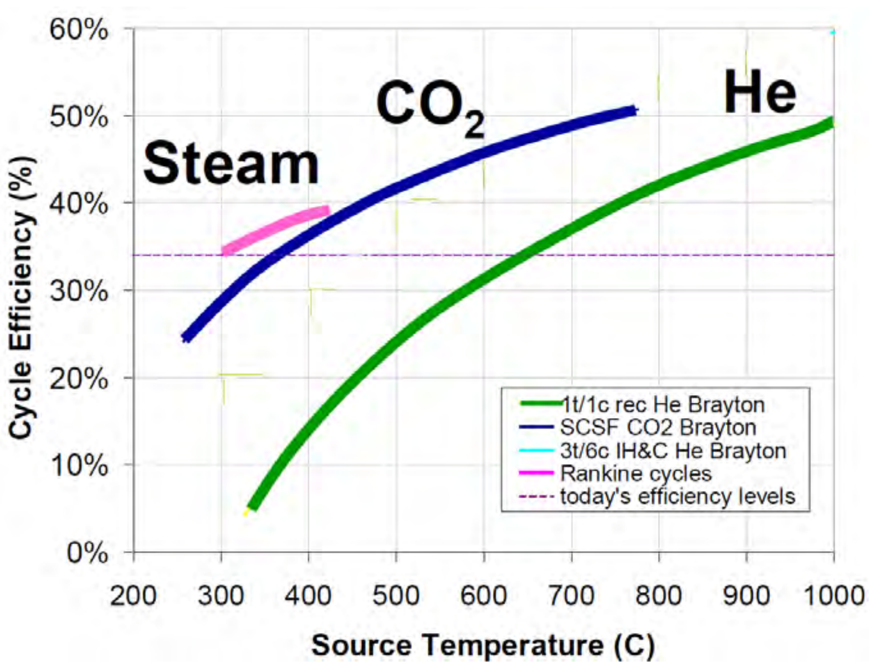
Future cost of generating electricity using solar tower technology can be reduced by:

1) Lowering the cost of the heliostats

2) Including thermal storages with the systems to increase the capacity factor

3) Increasing the operational temperature and employing more efficient power cycles

Supercritical CO_2 as the heat transfer and working fluids



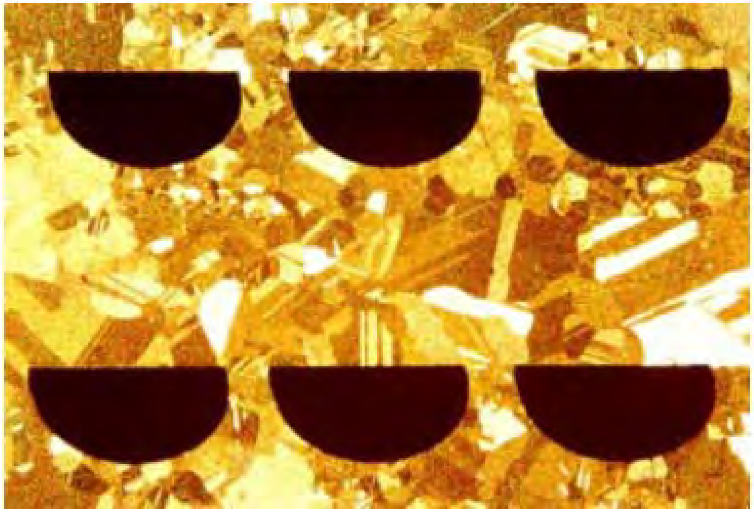
Ref (6)

- Carbon dioxide seems like a proper replacement for current heat transfer fluids (HTFs), i.e. oil, molten salt, and steam.
- Oil has low maximum operating temperature limit.
- Molten salt requires freeze protection units.
- Steam requires complex control systems.
- The main challenge about utilizing s-CO_2 as the HTF is the high operating pressure.

Compact heat exchanger technology

- Compact heat exchangers have already been extensively used for cooling the electronics. The high heat transfer rate in these heat exchangers is associated with the large area density (heat transfer area).

- Compact heat exchangers which are manufactured by diffusion bonding can tolerate very high pressures, i.e. more than 60 MPa.



Cross-section of a printed circuit heat exchanger
Ref (7)

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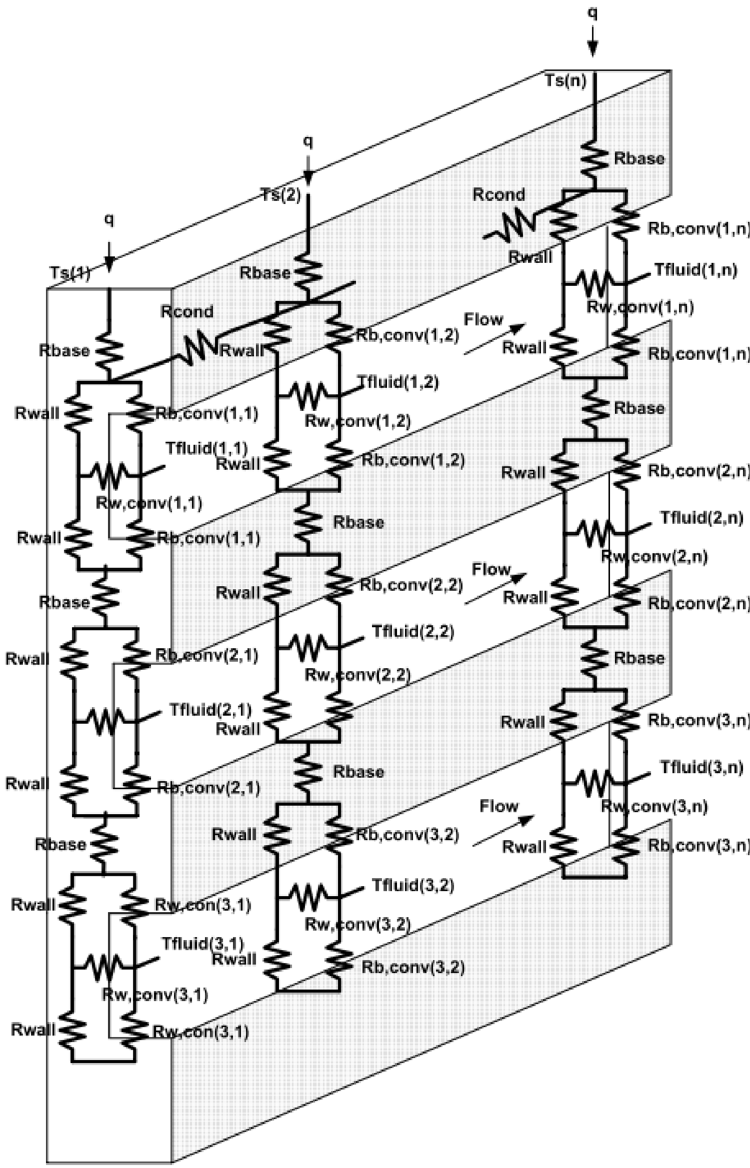
Thermal resistance network model



$$n = 200, q = \frac{Q}{n}, l = \frac{L}{n}$$

$$R_{base} = \frac{t}{k_s \times \frac{W}{2} \times l}$$

$$R_{wall} = \frac{\frac{b}{2}}{k_s \times l \times \frac{W - a}{2}}$$

$$R_{cond} = \frac{l}{k_s \times \frac{(W \times H - a \times b)}{2}}$$

$$R_{b,conv} = \frac{2}{h \times a \times l}$$

$$R_{w,conv} = \frac{1}{h \times b \times l}$$

Ref (8)

- Bulk fluid temperature can be found as:

$$T_{bulk,f}(l,k) = T_{f,in} + \frac{m}{\dot{m} \times C_p} \sum_{i=1}^{k} q(l,i)$$

where l represents the row number, and k is the grid number in the axial direction.

- The equivalent thermal resistance is given as:

$$R_{eq}(l,k) = \frac{T_j(k) - T_{bulk,f}(l,k)}{q(l,k)}$$

- Writing the energy balance around grid k leads to:

$$-T_j(k-1) + \left[2 + R_{cond} \sum_{l=1}^{m} \frac{1}{R_{eq}(l,k)}\right] T_j(k) - T_j(k+1) = R_{cond}\left[q + \sum_{l=1}^{m} \frac{T_{bulk,f}(l,k)}{R_{eq}(l,k)}\right]$$

Therefore, a system of linear equations is obtained for n unknown junction temperatures.

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- Laminar flow (Re<=2300)

$$Nu = 8.235\,(1 - 2.0421\,\alpha + 3.0853\,\alpha^2 - 2.4765\,\alpha^3 + 1.0578\,\alpha^4 - 0.1861\,\alpha^5)$$

α is the aspect ratio <=1

- Turbulent flow (Re>=5000)

$$Nu = \frac{\frac{f_c}{2}\,(Re - 1000)\,Pr}{1 + 12.7\,(\mathrm{Pr}^{\frac{2}{3}} - 1)\sqrt{\frac{f_c}{2}}}, f_c = 0.25\left(\frac{1}{1.8\log Re\ - 1.5}\right)^2$$

- Transition region (2300<Re<5000)

$$Nu = Nu_{2300} + \frac{Nu_{5000} - Nu_{2300}}{5000 - 2300}\,(Re - 2300)$$

- Pressure drop at the entrance and exit of the channels:

$$\Delta P = \frac{C\rho V^2}{2}$$

where *C* is taken as 0.5 for the entrance, and 1 for the exit.

- The friction loss inside the channel:

$$\Delta P = \frac{f\,\frac{L}{D}\,\rho V^2}{2}$$

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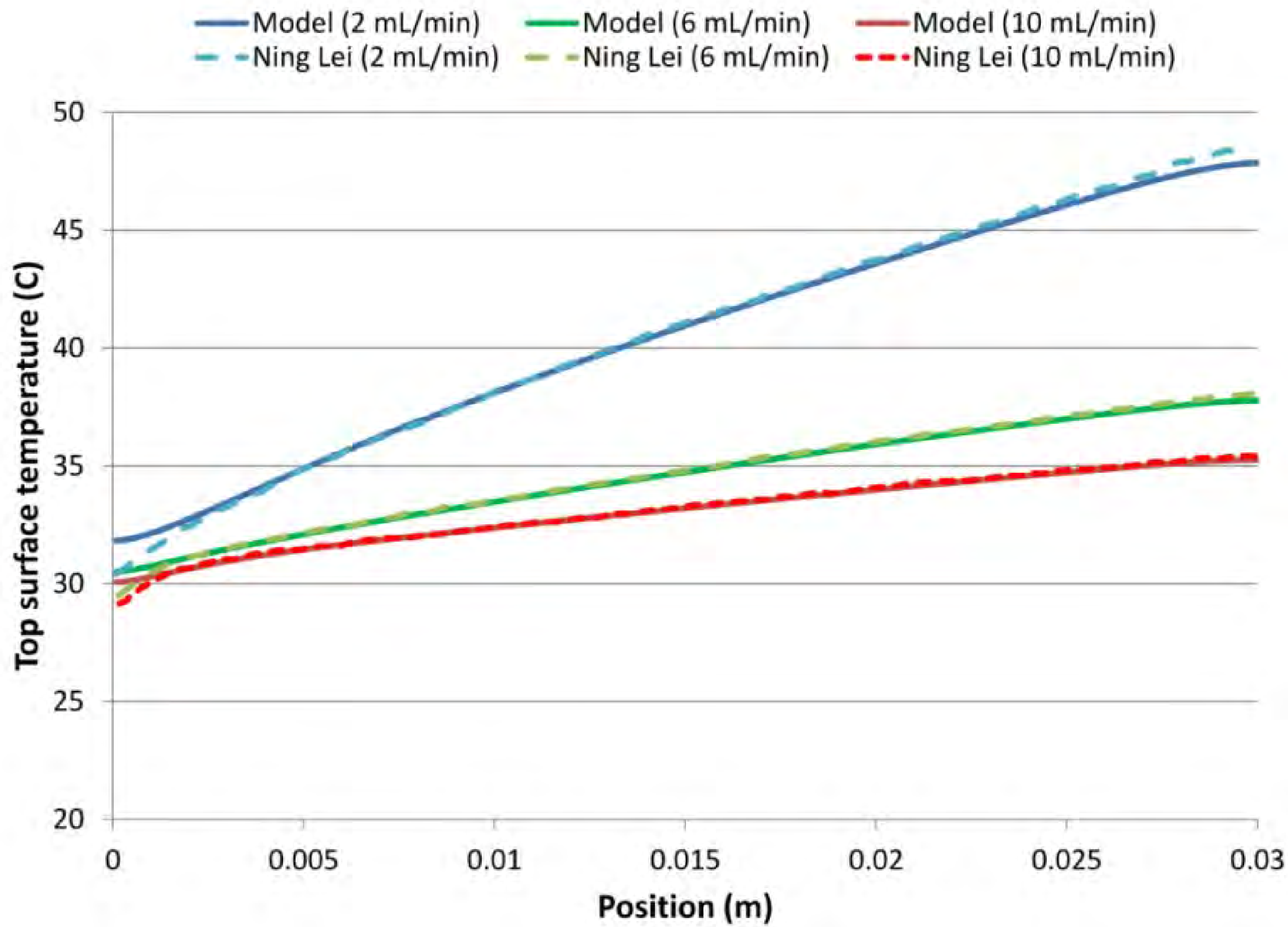
Model validation



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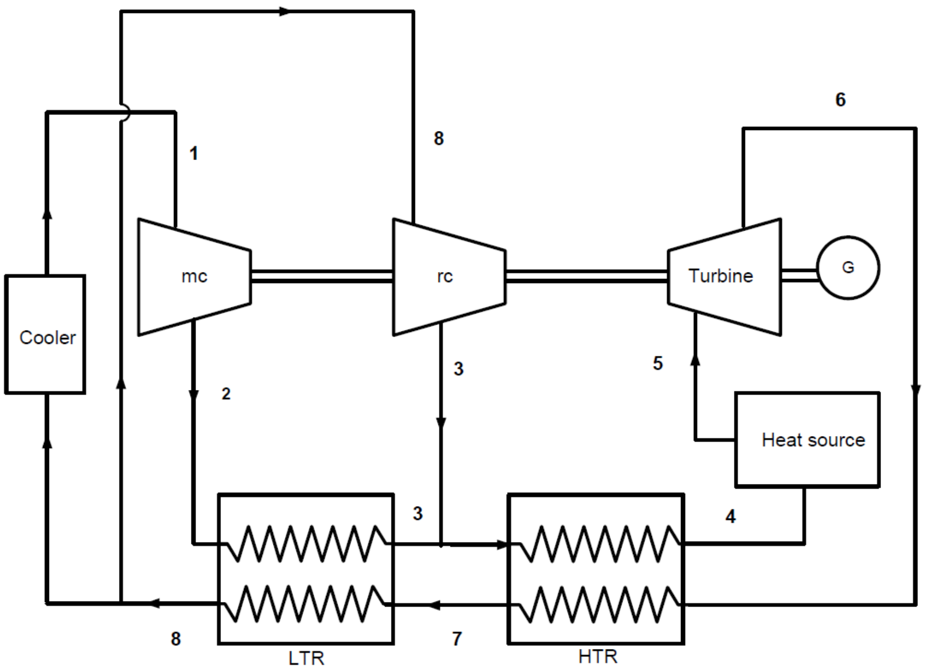
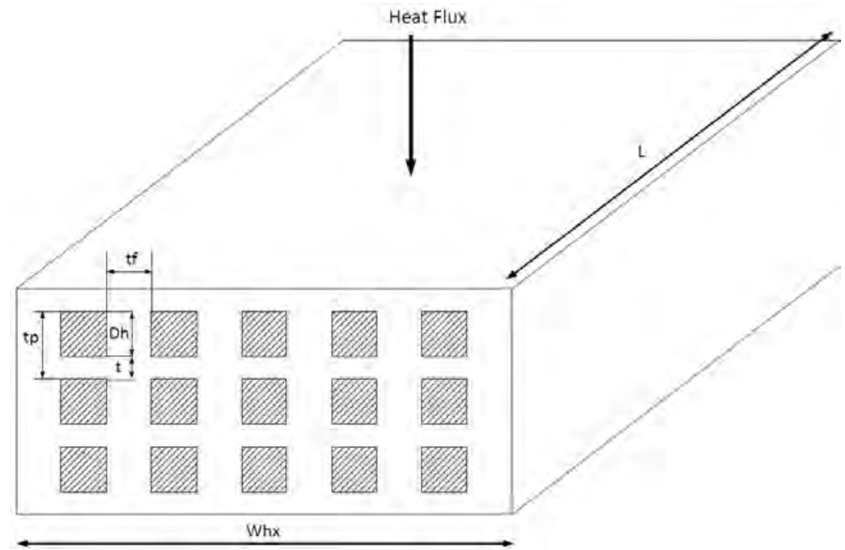
Direct s-CO_2 receiver in a recompression Brayton cycle





$P_5 = 20$ Mpa, $T_5 = 700°C$, $T_1 = 35°C$

Under Optimized Condition

$\eta = 48.6\ \%$

$T_4 = 530$

$h_5 - h_4 = 213\,\frac{kJ}{kg}$

Mass flow rate: 1 kg/s

Heat flux: 500 kW/m^2

W_{hx}=0.6 m

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Material selection

- Inconel 625, which is a nickel based alloy, is selected as the heat exchanger material because of its good corrosion resistance in high temperature s-CO_2 environment.

Nickel	58.0 min.
Chromium	20.0-23.0
Iron	5.0 max.
Molybdenum	8.0-10.0
Niobium (plus Tantalum)	3.15-4.15
Carbon	0.10 max.
Manganese	0.50 max.
Silicon	0.50 max.
Phosphorus	0.015 max.
Sulfur	0.015 max.
Aluminum	0.40 max.
Titanium	0.40 max.
Cobalt[a]	1.0 max.

Melting range is 1290-1350 C, and maximum operating temperature is 982 C.

Temp °C	Mean Linear Expansion[a] µm/ µm•°C	Thermal Conductivity [b,c] W/m•°C	Electrical Resistivity[c] µΩ-cm
-157	-	7.2	-
-129	-	7.5	-
-73	-	8.4	-
-18	-	9.2	-
21	-	9.8	129
38	-	10.1	130
93	12.8	10.8	132
204	13.1	12.5	134
316	13.3	14.1	135
427	13.7	15.7	136
538	14.0	17.5	138
649	14.8	19.0	138
760	15.3	20.8	137
871	15.8	22.8	136
927	16.2	-	-
982	-	25.2	135
1093	-	-	134

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Geometric optimization

- Design variables

1) Number of rows $3 \leq m \leq 10$
2) Hydraulic diameter $0.5\, mm \leq D_h < 3mm$
3) Horizontal distance between the channels $1mm \leq t_f < 5mm$

- Objective functions

1) Pressure drop
2) $\mathrm{RR} = \left(\frac{\mathrm{T_{surface,mean}} - \mathrm{T_{inlet}}}{\mathrm{Q''}}\right) \times 10000\ \frac{\mathrm{K}}{\mathrm{W\ cm^2}}$

- Constraints

1) Mechanical strength of the system (Checked by ASME pressure vessel code)

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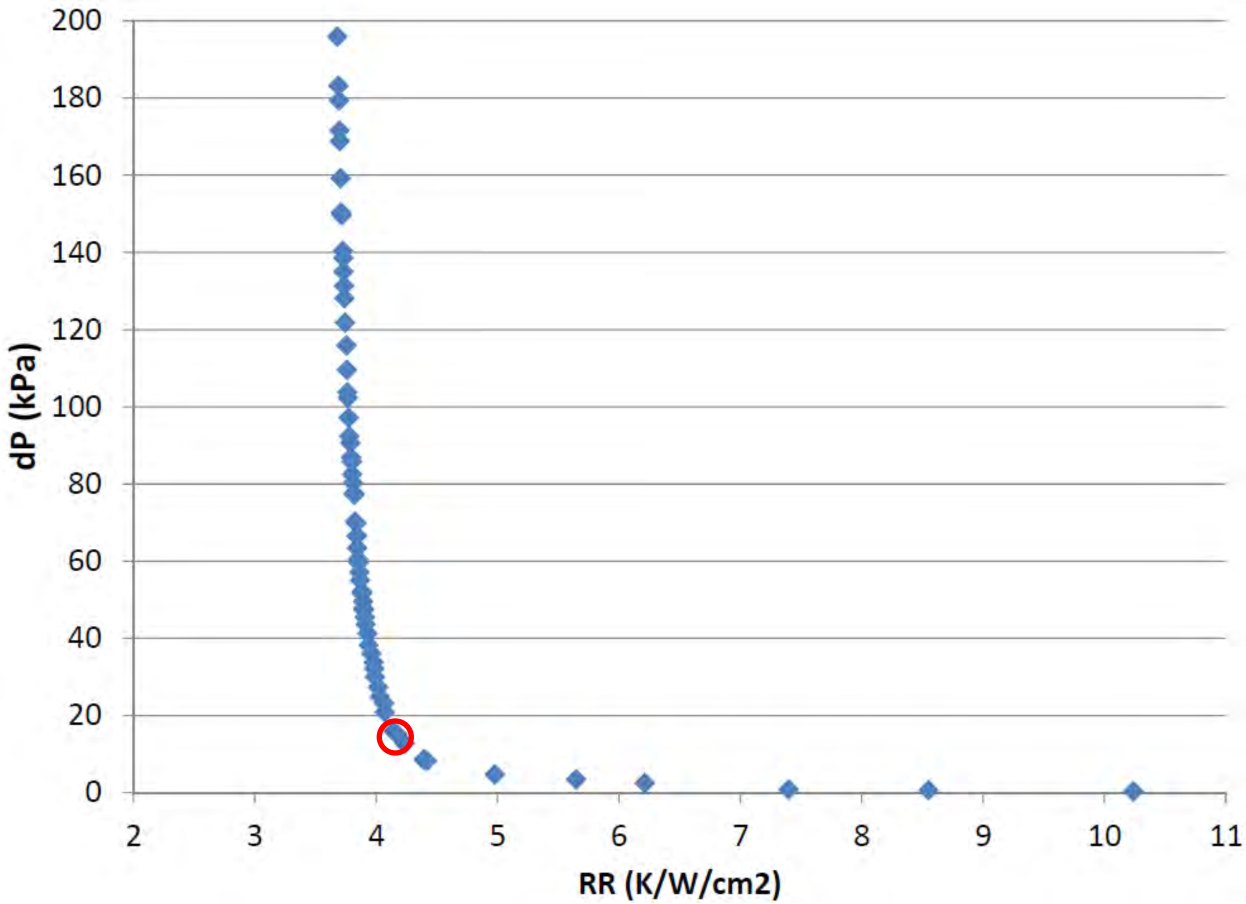
Pareto front of two objective optimization



$$\mathrm{D_h} = 2.8\ \mathrm{mm}, \mathrm{t_f} = 5\mathrm{mm}, \text{number of rows} = 3$$

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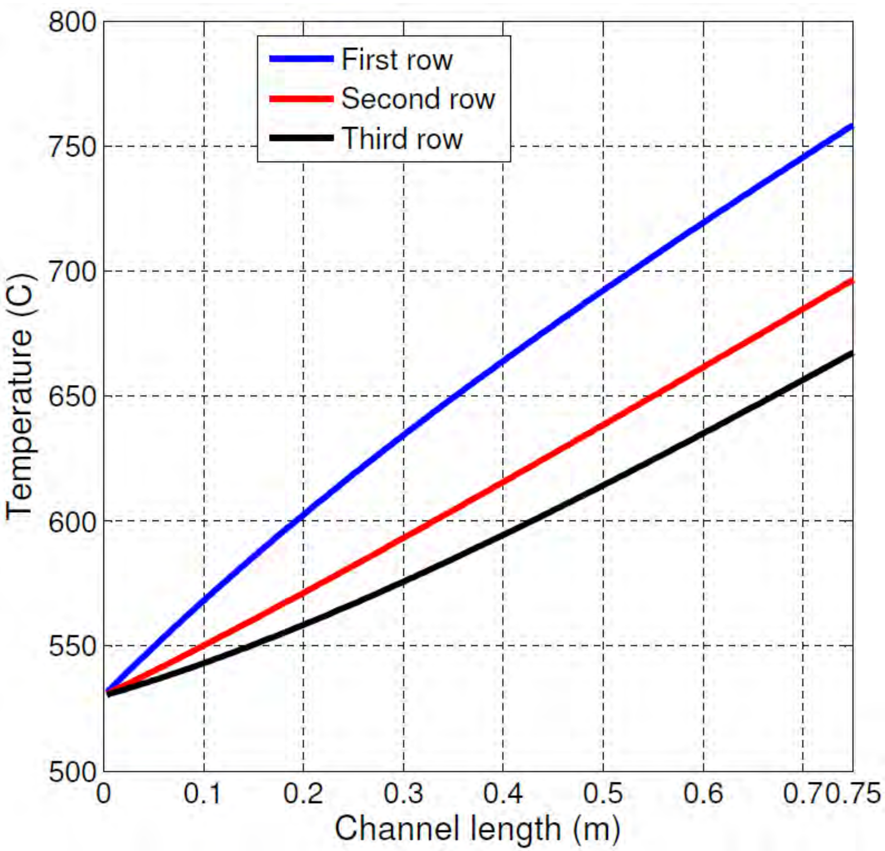
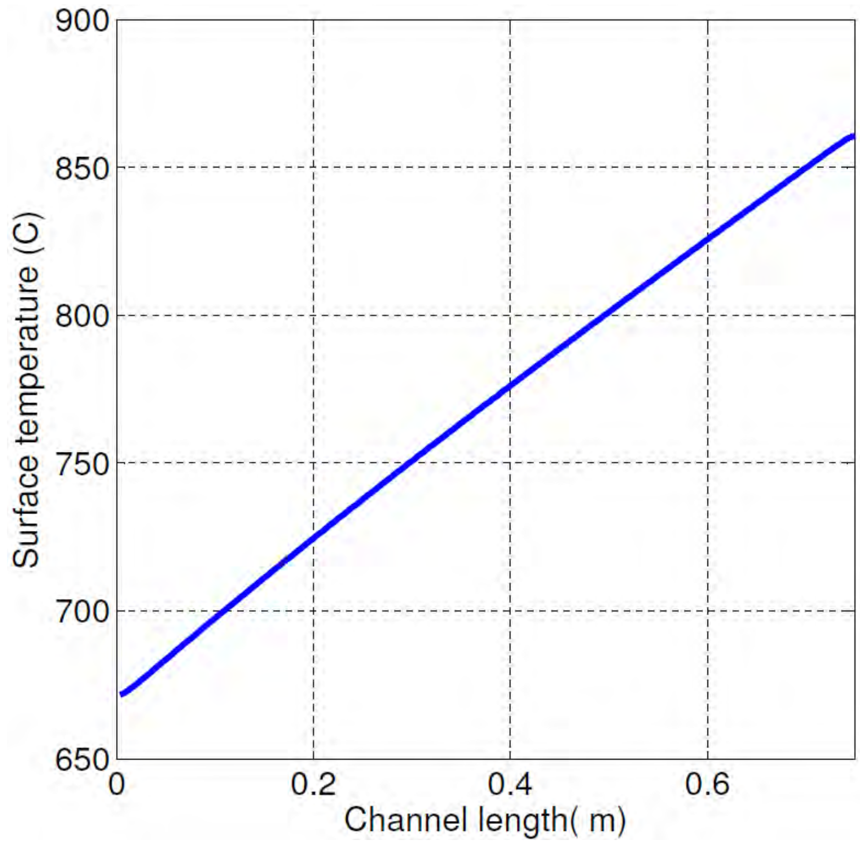
Optimization results

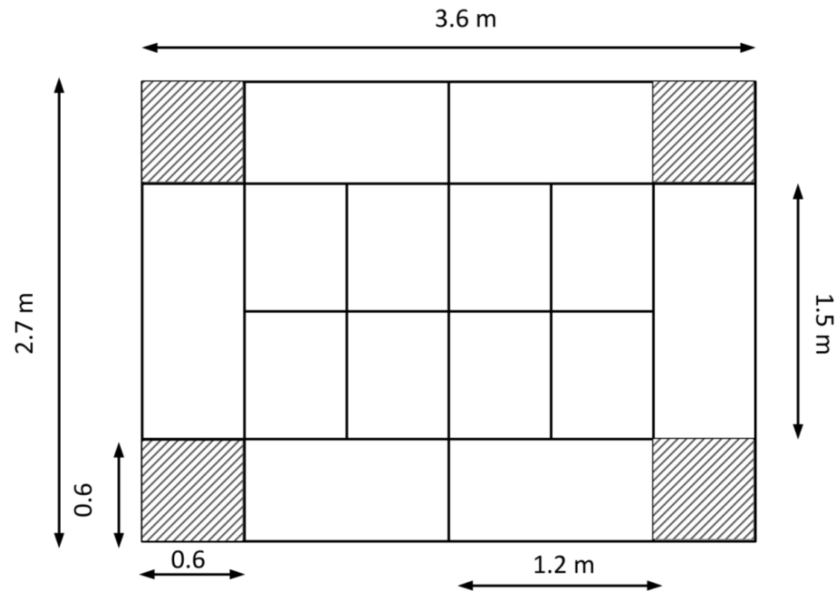


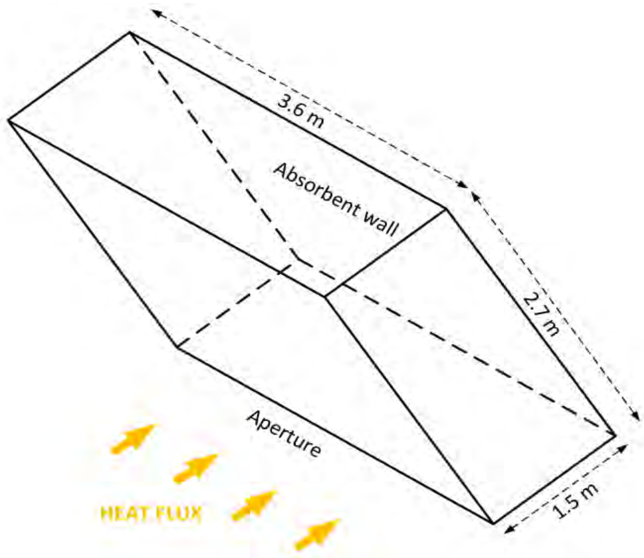
S-CO_2 temperature inside the channels



Surface temperature

Design of a 3MW$_{th}$ receiver





Tilt angle of the receiver is 35 °

Total number of panels that is required for a 3 MW$_{th}$ receiver to heat s-CO_2 from 530℃ to 700℃ at 20 Mpa is:

$$n_{panels} = \frac{3000}{213} \cong 14$$

Convective heat loss

- Natural convection (Clausing method)

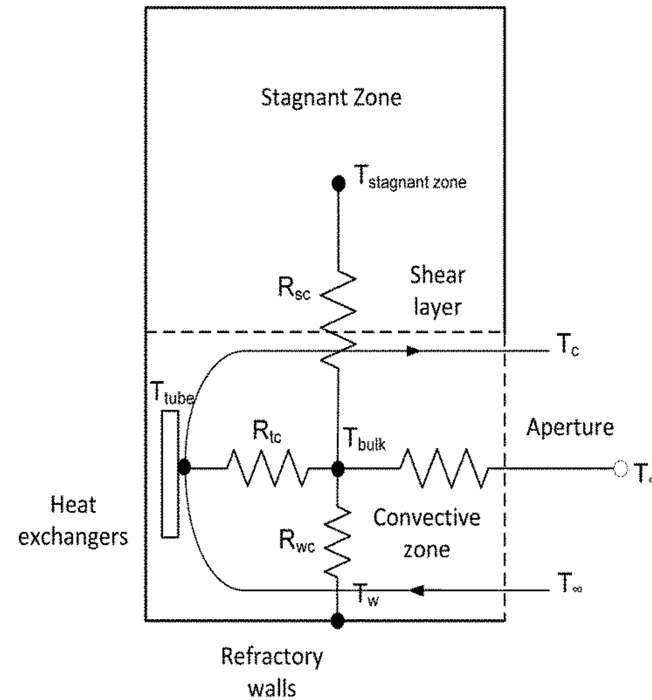
$$Q_a = \rho_\infty u_\infty \, c_p A_a (T_c - T_\infty)$$

$$V_b = \sqrt{g\,\beta\,(T_c - T_\infty) L_a}\,,\; V_a = 0.5\,[(C_3 V_b)^2 + (C_4 V_{wind})^2]^{0.5}$$

$$Q_c = h_{conv}\, A_{cav}\, (T_{surface} - T_{ref})$$

$$Nu = 0.082\,(Gr.\mathrm{Pr})^{\frac{1}{3}}\left[-0.9 + 2.4\left(\frac{T_{surface}}{T_\infty}\right) - 0.5\left(\frac{T_{surface}}{T_\infty}\right)^2\right]$$

$$Q_c = Q_a$$



- Forced convection (Siebers, D. and Kraabel, J., 1984)

$$Nu_{forced} = 0.287\, Re_w^{0.8}\, Pr^{1/3}$$

- Combined convective heat transfer coefficient is given by:

$$h_{mix} = h_{Natural} + h_{forced}$$

- For each panel the convective heat loss is calculated by:

$$Q_{conv,loss,i} = h_{mix,i}\, A_i\, (T_i - T_{bulk})$$

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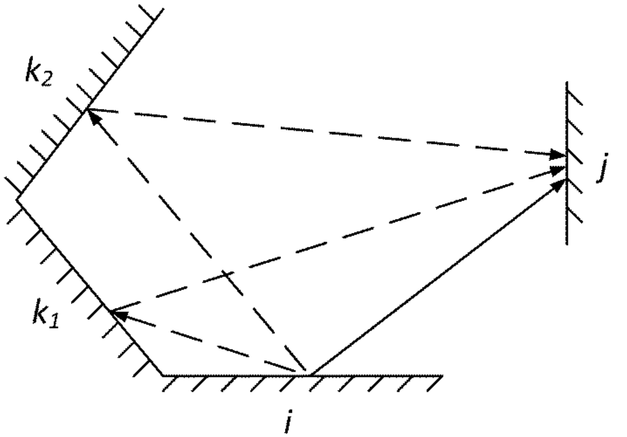
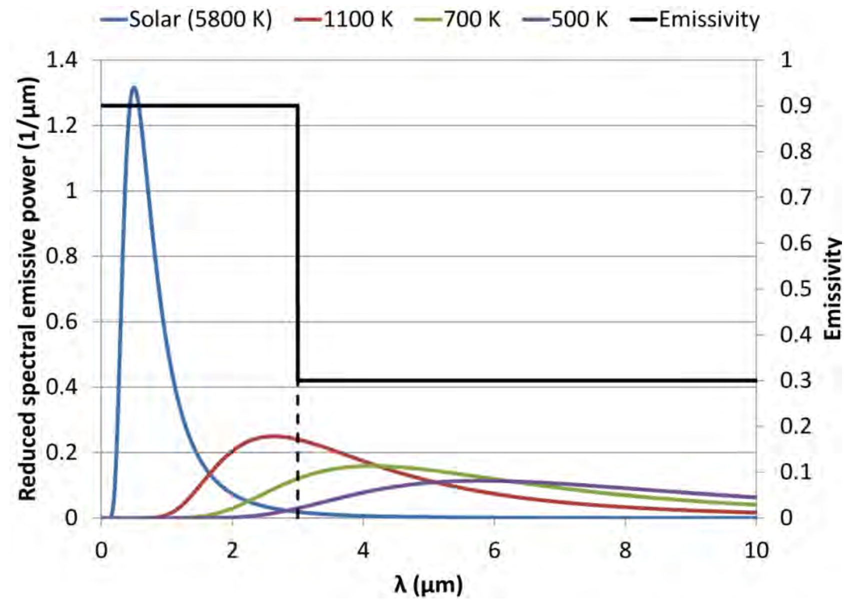
Radiative heat loss (model developed by Teichel)





$$\dot{Q}_{rad,thermal,i,j} = \varepsilon_{j,therm}\, \varepsilon_{i,therm}\, \sigma\, A_i\, \hat{F}_{i,j,therm} \left(\left(1 - f_{0-\lambda_{step},T_i}\right) T_i^4 - \left(1 - f_{0-\lambda_{step},T_j}\right) T_j^4 \right) + \varepsilon_{j,solar}\, \varepsilon_{i,solar}\, A_i\, \hat{F}_{i,j,solar} \left(f_{0-\lambda_{step},T_i} T_i^4 - f_{0-\lambda_{step},T_j} T_j^4 \right)$$

$$\dot{Q}_{rad,solar,i,j} = f_{0-\lambda_{step},T_{sun}} \hat{F}_{i,j,solar}\, A_i\, \varepsilon_{solar,i}\, \varepsilon_{solar,j} \left(Flux_{solar,i} - Flux_{solar,j} \right) + \left(1 - f_{0-\lambda_{step},T_{sun}}\right) A_i\, \hat{F}_{i,j,therm}\, \varepsilon_{i,therm}\, \varepsilon_{j,therm} \left(Flux_{solar,j} - Flux_{solar,i} \right)$$

$$\dot{Q}_{rad,i,j} = \dot{Q}_{rad,thermal,i,j} + \dot{Q}_{rad,solar,i,j}$$

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Energy balance on cavity surfaces

1) Active surfaces:

$$Flux_i\ A_i = Q_{gain,i} + Q_{conv,loss,i} + Q_{rad,loss,i}$$

2) Passive surfaces:

$$0 = 0 + Q_{conv,loss,i} + Q_{rad,loss,i}$$

$$if\ Q_{rad,loss,i} = A_i \sum_{j=1}^{N} \left[h_{rad,therm,i,j}\ (T_i - T_j)\right] + \dot{Q}_{rad,solar,i}$$

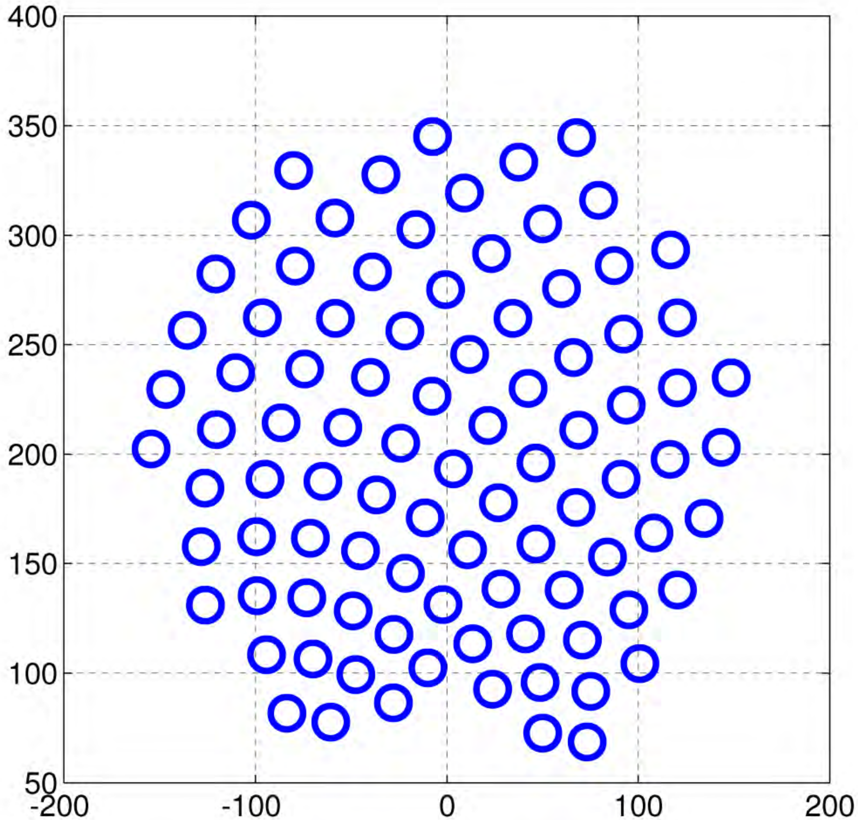
$$T_i = \frac{A_i \sum_{j=1}^{N}\left[h_{rad,therm,i,j} T_j\right] - \dot{Q}_{rad,solar,i} + A_i h_{mix,i} T_{bulk}}{A_i \sum_{j=1}^{N}(h_{rad,therm,i,j}) + A_i h_{mix,i}}$$

3) Corners:

$$Flux_i\ A_i = 0 + Q_{conv,loss,i} + Q_{rad,loss,i}$$

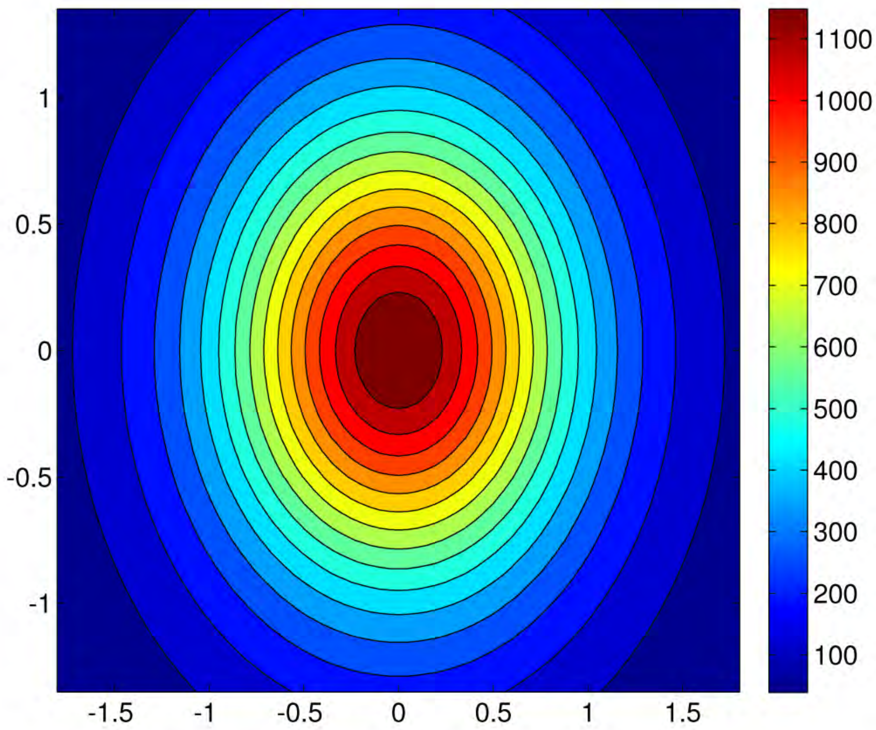
$$T_i = \frac{Flux_i\ A_i + A_i \sum_{j=1}^{N}\left[h_{rad,therm,i,j} T_j\right] - \dot{Q}_{rad,solar,i} + A_i h_{mix,i} T_{bulk}}{A_i \sum_{j=1}^{N}(h_{rad,therm,i,j}) + A_i h_{mix,i}}$$

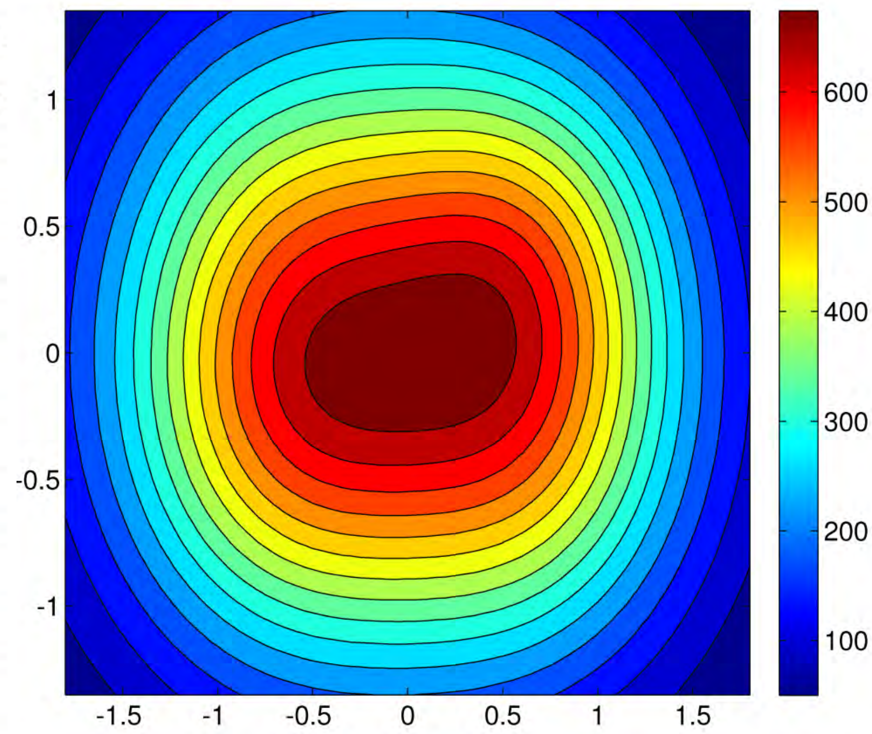
Solar field



Field parameters	
Location	
Dagget, CA	
Heliostats	
Number of heliostats	92
Width	8.84 m
Height	7.34 m
Reflectivity	0.88
Receiver	
Tower height	115 m
Tilt angle of the aperture	35°
Aperture width	3.6 m
Aperture height	2.7 m

Flux density distribution ($\mathbf{kW/m^2}$)



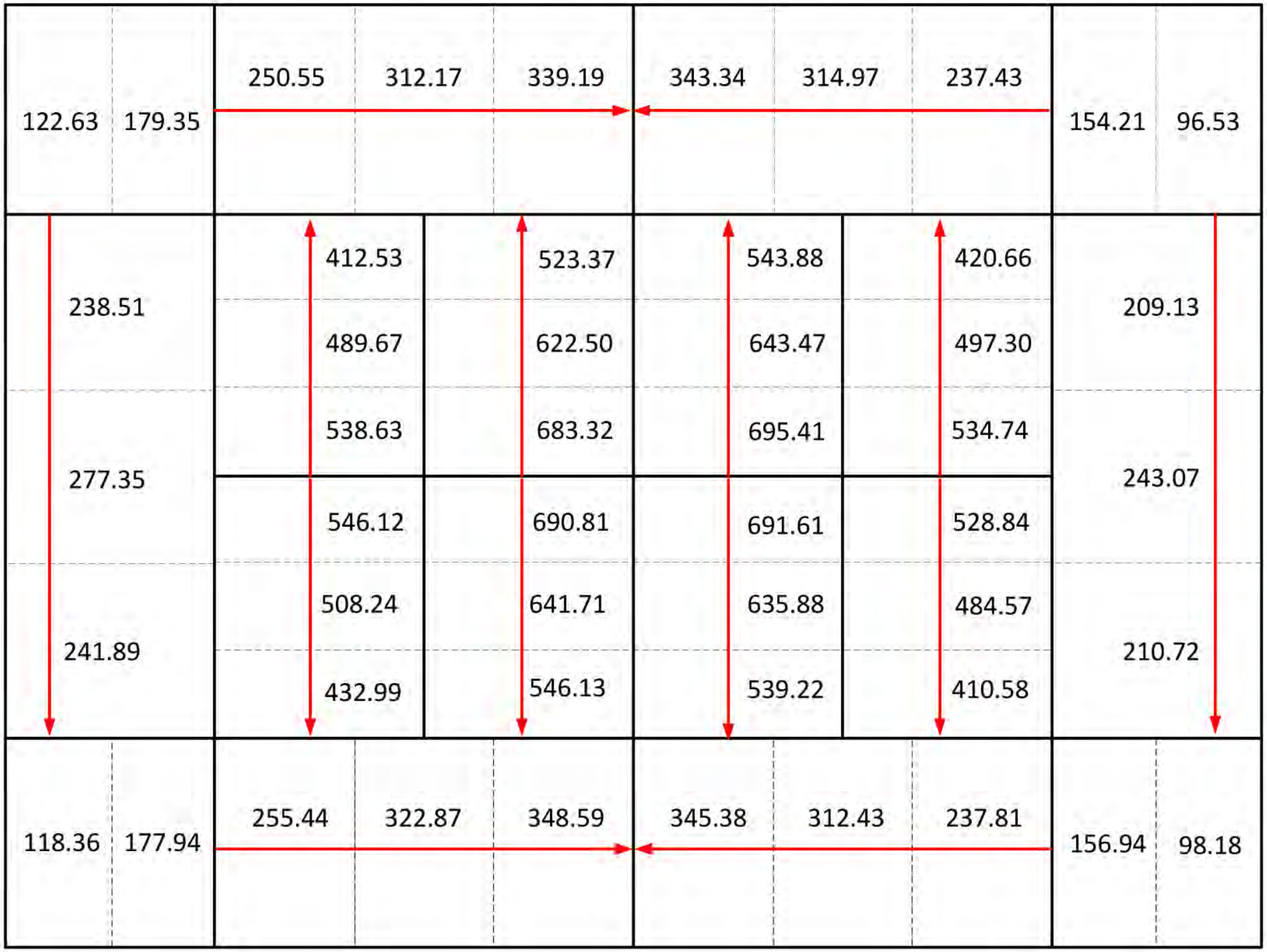


without aiming strategy

with aiming strategy

March 21st , noon

Flux density distribution $(\frac{kW}{m^2})$



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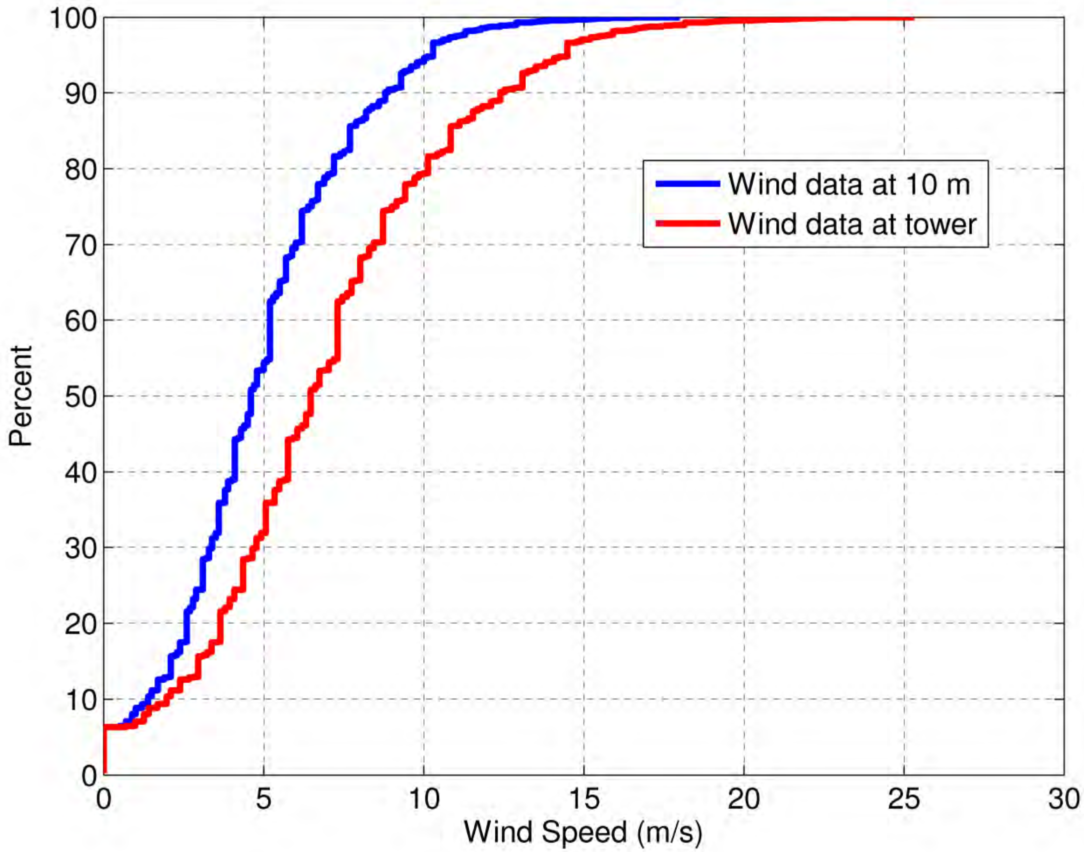
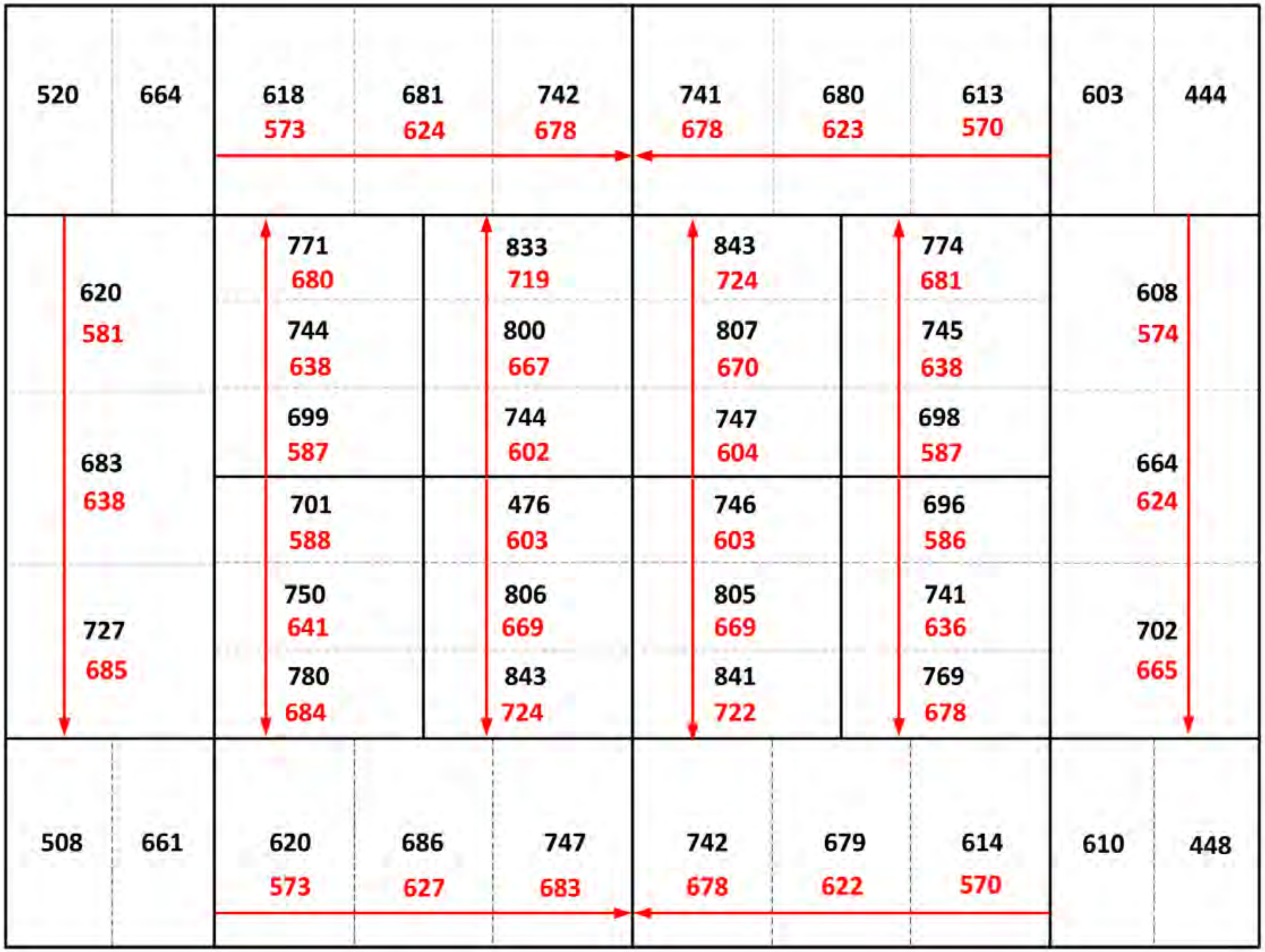
Wind speed profile for Dagget (TMY3 data)



$$\frac{V_1}{V_2} = \left(\frac{z_1}{z_2}\right)^{0.14}$$ (Duffie, J. A., Beckman, W.A., 2006)

Temperature distribution inside the cavity (°C)



Black→Surface Temperature   Red→Fluid temperature

Other important information

- Mean temperature of s-CO2 leaving the receiver: 691°C

- Maximum surface temperature: 843°C (Maximum allowable temperature for Inconel 625 is 982°C.

- Receiver efficiency:

$$\eta_{rec} = \frac{Q_{transfered\ to\ the\ fluid}}{Q_{received\ by\ the\ receiver}} \times 100 = 81.22\ \%$$

- Temperatures of the passive surfaces:

Top surface	169°C
Bottom surface	170°C
Left surface	173°C
Right surface	167°C

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Summary and conclusion

- A direct s-CO_2 receiver was designed based on the principles of compact heat exchangers.
- The receiver is expected to heat s-CO_2 from 530°C to 700°C. The geometry of the receiver was determined using multi-objective the Pareto based optimization approach by the simultaneous minimization of the unit thermal resistance and the pressure drop.
- A 3MW_{th} cavity receiver was designed using 14 individual panels.
- The heliostat field was designed, and the corresponding flux distribution on the receiver surface was obtained for March 21st.
- The radiative and convective heat transfer models were developed, and the bulk fluid and surface temperatures were obtained.
- The results showed that the s-CO_2 reached the design temperature while the surface temperatures remained below the maximum temperature limit of Inconel 625. The receiver efficiency was obtained as 81.22%, which is highly promising.
- The efficiency can be further improved by optimizing the geometry of the cavity receiver. Considering the appropriate thermal and mechanical performance of the CHEs, they can be seriously considered for the next generation of high temperature pressurized solar receivers.

Thank you for your attention,
Questions?

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