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**CONCENTRATING SOLAR TROUGH COLLECTORS BASED ON AIR
AS HEAT TRANSFER FLUID FOR ELECTRICITY GENERATION
AND HIGH-TEMPERATURE PROCESS HEAT**

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Abstract

The use of air as heat transfer fluid (HTF) in solar parabolic trough collectors at operating temperatures up to 600 °C is investigated for electricity generation and high-temperature process heat. A novel solar receiver design that addresses the low density and thermal conductivity of gaseous HTFs is developed. It consists of an array of helically coiled absorber tubes contained side-by-side within an insulated groove having a rectangular windowed opening. Secondary concentrating optics are incorporated to boost the geometric concentration ratio to 97×. The multiple absorber tubes are connected via two axial pipes serving as feeding and collecting manifolds. The proof of concept is delivered by on-sun experiments with a 1 m-long solar receiver prototype composed of 7 absorber tubes, mounted on a 4.85 m-aperture solar trough concentrator. Feeding rates in the range of 5 – 20 l_n/min to each absorber tube led to air outlet temperatures of 621 – 449 °C and a peak receiver efficiency of 64%. The steady-state energy conservation equation coupling radiation, convection, and conduction is formulated and solved numerically using the finite volume technique. The solar flux distribution incident at each absorber tube is determined by Monte Carlo ray-tracing using spectrally and directionally dependent optical properties. Model validation is accomplished by comparison to experimental results.

The validated heat transfer model is further applied to a parameter study of the array of coiled absorber tubes. Numerical simulations are performed for variable solar direct normal irradiance (DNI) and incidence angle, and the annual mean solar collector efficiencies are calculated. For a typical yearly DNI of 2400 kWh/m² and air outlet temperatures in the range 420 – 660 °C, the maximum attainable annual mean DNI-to-thermal efficiency varies from 0.49 to 0.37.

Industrial scale demonstration is realized with a 212 m-long parabolic trough concentrator and associated solar receiver designed for air operating

temperatures of 500 °C. The fully pneumatic solar concentrator is based on a stack of reflective polymeric films mounted on a rigid concrete support structure and protected by a transparent polymeric envelope. The 1.2 MW_{th} solar collector unit, comprising the solar concentrator and solar receiver, is modeled by formulating the energy conservation equation coupling radiation, convection, and conduction and solving it numerically using the finite volume technique. Model validation is accomplished by comparison to on-sun experimental data obtained from tests of the first solar collector unit after its commissioning in Ait Baha, Morocco. The mean absolute and RMS deviations between measured and computed air outlet temperatures of the solar receiver are 7.3 °C and 9.5 °C, respectively. The validated model is further applied to identify heat losses and predict year-round operation. For a DNI of 2400 kWh/m²/yr, the solar plant can deliver 1810 MWh of thermal energy at 500 °C.

On the route from a conventional straight tubular absorber to the array of coiled absorber tubes, alternative solar receivers are explored with a potential for low-cost construction. The receiver designs are based on multiple parallel straight absorber tubes arranged side-by-side on a circular arc to delimit a groove having a rectangular windowed aperture. The modeling techniques of the validated heat transfer models are applied to the numerical simulation of the most promising straight tubular receivers, and their efficiencies are compared to the performance of the array of coiled tubes. At solar noon on summer solstice in Sevilla, Spain, DNI-to-thermal efficiencies of 0.51 – 0.54 are simulated for straight tubular receivers and 0.56 for the array of coiled tubes at an operating temperature of 500 °C. Material cost and practical considerations are also discussed.

The optical and mechanical design of the pneumatic solar trough concentrator is outlined. The theory of a new mirror regulation mechanism based on inflatable sleeves is introduced, including the effects of weight. Predictions by theory are compared with experiments, and a model-based control system for the automatic regulation of inflation pressures is proposed. In addition, further experimental results and details of the operation of the industrial scale solar concentrator in Ait Baha, Morocco are presented.

The narrow-angle spectral specular reflectance and angular scattering of conventional and novel reflective materials for solar concentrators are measured over the wavelength range 300 – 2500 nm at incidence angles ranging from 15° to 60° using a spectroscopic goniometry system. The solar-weighted specular reflectance at near normal incidence and an acceptance half-angle of 17.5 mrad is 0.941 for back-silvered glass, 0.908 – 0.926 for silvered polymer films, 0.895 for aluminized polyester film, 0.939 – 0.954 for silvered aluminum sheets, and 0.860 for aluminized aluminum sheet. The angular scattering, quantified in terms of the standard deviation of a Gaussian distribution, is found to be negligible for aluminized polyester (< 0.05 mrad) and back-silvered glass (< 0.07 mrad), and noticeable for silvered polymer films (0.27 – 1.12 mrad) and silvered aluminum sheets (0.12 – 1.66 mrad). In addition, the spectral narrow-angle transmittance of semi-transparent materials suitable for protective covers is measured, yielding solar-weighted normal transmittance values of 0.913 and 0.946 for 100 μm thin films of ETFE (ethylene-tetrafluoroethylene) and FEP (fluorinated ethylene propylene), respectively. The measured optical properties are incorporated in a Monte Carlo ray-tracing program and applied to analyze the optical performance of solar concentrators.

Zusammenfassung

Die Nutzung von Luft als Wärmeübertragungsfluid (HTF) in Parabolrinnen-Sonnenkollektoren bei Betriebstemperaturen bis zu 600 °C zur Erzeugung von Elektrizität und industrieller Prozesswärme wird untersucht. Dazu wird ein neuartiges Design eines Strahlungsempfängers entwickelt, das die tiefe Dichte und Wärmeleitfähigkeit von gasförmigen Fluiden miteinbezieht. Es besteht aus einer Reihe von schraubenförmigen Absorberröhren, die nebeneinander in einer isolierten Nut mit einer rechteckigen Fensteröffnung eingebettet sind. Durch die Verwendung eines Sekundärkonzentrators wird eine geometrische Solarkonzentration von $97\times$ erreicht. Die schraubenförmigen Absorberröhren sind über zwei axiale Rohre verbunden, welche als Verteiler und Sammler der kalten respektive warmen Luft dienen. Der Machbarkeitsbeweis wird mit einem 1 m-langen Strahlungsempfängerprototypen bestehend aus 7 Absorberröhren erbracht, der von einem Solarrinnen-Konzentrator mit einer Aperturweite von 4.85 m bestrahlt wird. Mit Luftspeisungen von 5 – 20 l_n/min zu jeder Absorberröhre wurden Luftaustrittstemperaturen von 621 – 449 °C und eine maximale Effizienz des Strahlungsempfängers von 64% erreicht. Die stationäre Energieerhaltungsgleichung für gekoppelte Strahlung, Wärmeleitung und Konvektion wird aufgestellt und numerisch gelöst mittels Finite-Volumen-Verfahren. Die Solarflussverteilung auf jeder Absorberröhre wird mittels Monte Carlo Raytracing für spektral- und richtungsabhängige optische Materialeigenschaften ermittelt. Die Modellvalidierung erfolgt durch Vergleichen mit experimentellen Daten.

Das validierte Model wird weiterverwendet für eine Parameterstudie des Arrays von schraubenförmigen Absorberröhren. Numerische Simulationen werden durchgeführt für variable direkt normale Sonneneinstrahlung (DNI) und Einstrahlwinkel und die Jahresdurchschnittseffizienz des Sonnenkollektors wird berechnet. Bei einer typischen jährlichen DNI von 2400 kWh/m² und

Luftauslasstemperaturen von 420 – 660 °C liegt die maximal erreichbare Jahresdurchschnittseffizienz von DNI zu Wärme im Bereich 0.49 – 0.37.

Die grosstechnische Demonstration wird mit einem 212 m-langen Parabolrinnen-Konzentrator und dazugehörendem Strahlungsempfänger für Betriebstemperaturen bis zu 500 °C realisiert. Der vollständig pneumatische Solarkonzentrator basiert auf einem Stapel von reflektierenden Polymerfilmen, die auf einer starren Betonstruktur montiert und durch eine transparente Polymerhülle geschützt sind. Für die 1.2 MW_{th} Sonnenkollektor-Einheit, bestehend aus Konzentrator und Strahlungsempfänger, wird die Energieerhaltungsgleichung für gekoppelte Strahlung, Wärmeleitung und Konvektion aufgestellt und numerisch gelöst mittels Finite-Volumen-Methode. Das Modell wird im Vergleich zu experimentellen Daten validiert, welche in Sonnenkollektoren-Tests mit der ersten Einheit nach ihrer Inbetriebnahme in Ait Baha, Marokko, gesammelt werden. Die mittlere absolute und quadratische Abweichung zwischen der gemessenen und berechneten Luftauslasstemperatur beträgt 7.3 °C respektive 9.5 °C. Das validierte Modell wird weiterverwendet zur Identifizierung von Energieverlusten und Vorhersage der Ganzjahresproduktion. Bei einer jährlichen DNI von 2400 kWh/m²/yr liefert die Solaranlage 1810 MWh an thermischer Energie bei 500 °C.

Auf dem Pfad von einem konventionellen geraden Absorberrohr zum Array schraubenförmiger Absorberröhren werden alternative Strahlungsempfänger-Designs mit Potenzial zur kostengünstigen Herstellung untersucht. Die Strahlungsempfänger bestehen aus mehreren geraden Absorberrohren, die nebeneinander auf einem Kreisbogen angeordnet sind und so eine Rinne bilden mit einer rechteckigen Fensteröffnung. Die Modellieretechniken der validierten numerischen Modelle werden auf die vielversprechendsten Designs mit geraden Absorberrohren angewendet und ihre Effizienz wird mit derjenigen des Arrays von schraubenförmigen Absorberröhren verglichen. Am Mittag der Sommersonnenwende in Sevilla, Spanien, beträgt die Effizienz von DNI zu Wärme bei 500 °C 0.51 – 0.54 für gerade Rohrabsober und 0.56 für das Array von schraubenförmigen Absorberröhren. Materialkosten und praktische Betrachtungen werden ebenfalls diskutiert.

Das optische und mechanische Design des pneumatischen Solarrinnen-Konzentrators wird dargelegt. Die Theorie eines neuen Spiegelregelungsmechanismus basierend auf aufblasbaren Schläuchen wird eingeführt unter Berücksichtigung von Gravitationseffekten. Die Theorie wird mit Experimenten verglichen, wovon ein modellbasiertes Steuersystem für die automatische Regulierung der Luftdrücke abgeleitet wird. Ausserdem werden weitere experimentelle Resultate und Details vom Betrieb des 212 m-langen Konzentrators der grosstechnischen Anlage präsentiert.

Der spektrale, gerichtete Reflexionsgrad und die Winkelstreuung konventioneller und neuartiger Spiegelmaterialien für Sonnenkonzentratoren werden mittels eines spektroskopischen Goniometers im Wellenlängenbereich 300 – 2500 nm unter Lichteinfallswinkel von 15° bis 60° gemessen. Der mit dem Sonnenspektrum gewichtete gerichtete Reflexionsgrad bei nahezu senkrechtem Lichteinfall und einem Öffnungshalbwinkel von 17.5 mrad beträgt 0.941 für Silberglas, 0.908 – 0.926 für versilberte Polymerfilme, 0.895 für aluminiumbeschichtetes Polyester, 0.939 – 0.954 für versilberte Aluminiumbleche und 0.860 für aluminiumbeschichtetes Aluminium. Die Winkelstreuung, angegeben als Standardabweichung einer Gaussverteilung von der gespiegelten Richtung, ist vernachlässigbar für aluminiumbeschichtetes Polyester (< 0.05 mrad) und Silberglas (< 0.07 mrad), und bemerkbar für versilberte Polymerfilme (0.27 – 1.12 mrad) und versilberte Aluminiumbleche (0.12 – 1.66 mrad). Zusätzlich wird der spektrale, gerichtete Transmissionsgrad innerhalb kleiner Öffnungswinkel von für Schutzhüllen geeigneten semi-transparenten Materialien gemessen. Der solargewichtete Transmissionsgrad bei senkrechtem Lichteinfall beträgt 0.913 respektive 0.946 für 100 μm ETFE (Ethylen-Tetrafluorethylen) und FEP (Fluor-Ethylen-Propylen) Dünnsfilme. Die gemessenen optischen Eigenschaften werden in ein Monte Carlo Raytracing Programm integriert und verwendet, um die optische Performance von Konzentratoren zu untersuchen.

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Nomenclature

Latin characters

A_j	fixing point of j^{th} pneumatic sleeve at central beam	-
A	aperture area, surface area, cross-sectional area	m^2
AR	(height-to-diameter) aspect ratio	-
a	aperture width	m
	discretized heat conduction coefficients	W/K
	power law constant	mrاد
a_{ins}	thermally insulated area per unit receiver length	m
$B_{1,2}$	constants	-
BRDF	bi-directional reflectance distribution function	sr^{-1}
b	(power law) exponent	-
C_j	center point of the j^{th} circular arc	-
C	solar concentration	suns
	minor loss coefficient	-
C_g	geometric concentration ratio	$\times$
c	arbitrary radiative property	-
c_f	Fanning friction coefficient	-
c_p	specific heat capacity at constant pressure	J/(kg K)
D	(pipe) diameter	m
De	Dean number	-
d	distance, linear dimension	m
d_{pixel}	pixel size	μm
E	Young's modulus	N/m ²
E_{DNI}	direct normal irradiance (DNI)	W/m ²
E_{λ}	spectral irradiance	W/(m ² .nm)
$\hat{\mathbf{e}}_x, \hat{\mathbf{e}}_y, \hat{\mathbf{e}}_z$	unit vectors of Cartesian coordinates	-
e	equivalent surface roughness	m
$\mathbf{F}$	focal point	-
F	force	N
	fraction; factor	-

F_{k-j}	configuration factor from surface segment k to j	-
$F_{\Delta\lambda T}$	blackbody fractional function of interval $\Delta\lambda T$	-
f	focal length	m
	Darcy friction factor	-
	scattering function	mrad ⁻¹
g	gravitational acceleration	m/s ²
He	helical number	-
h	height	m
	heat transfer coefficient	W/(m ² K)
	scale factor	-
I_j	modified j^{th} order Bessel functions of the first kind	-
I'_λ	spectral intensity	W/(m ² nm sr)
IAM	incidence angle modifier	-
J	(Jacobian) matrix	-
K_j	modified j^{th} order Bessel functions of the second kind	-
k	thermal conductivity	W/(m K)
k_0	nominal transverse stiffness of mirror films	N/m
L	length	m
l	axial length (in trough direction)	m
M	half-width of reference beam profile	pixel
m	fin parameter	m ⁻¹
	(concentrator) slope	-
	discrete convolution coordinate	pixel
	number of internal reflections	-
$\dot{m}$	mass flow rate	kg/s
m'_{tube}	mass of absorber tubes per unit receiver length	kg/m
$\mathbf{N}_j$	j^{th} node of arcspline concentrator	-
N	number	-
	half-width of sample beam profile	pixel
Nu	Nusselt number	-
n	number (of surface segments; per receiver section)	-
	position on camera pixel matrix	pixel
	discrete coordinate (parallel to plane of incidence)	pixel
	refractive index	-
$\hat{\mathbf{n}}$	unit normal vector	-
P	point (on concentrator)	-
Pr	Prandtl number	-

p	pressure	Pa
Δp	pressure drop	Pa
p_{coil}	coil pitch	m
Δp_j	differential pressure on j^{th} arc	Pa
$p_{\text{sleeve},j}$	inflation pressure of j^{th} sleeve	mbar
Q_{yr}	yearly energy	kWh/yr
$\dot{Q}$	heat transfer rate	W
$\dot{q}$	local heat transfer rate (per absorber tube, receiver section)	W
$\dot{q}'$	heat transfer per unit receiver length	W/m
$\dot{q}''$	heat flux	W/m ²
R	(circular arc; coil) radius	m
	reflectance	-
$R_{\text{specular},\text{solar}}$	solar-weighted specular reflectance	-
$R_{\text{specular},\lambda}$	spectral specular reflectance	-
R_{air}	gas constant of air	J/(kg K)
$\Re$	random number	-
R	ratio	-
Ra	Rayleigh number	-
Re	Reynolds number	-
$\mathbf{r}$	position vector	-
r	radial coordinate	m
	reference beam profile	pixel ⁻¹
r_{tube}	tube radius	m
$\mathbf{S}$	limiting truncation point of trumpet without shading	-
S	stretched mirror film length	m
S_{sleeve}	inflated sleeve chord length	m
$\hat{\mathbf{s}}$	unit direction vector	-
s	sample beam profile (reflected/transmitted)	pixel ⁻¹
$\mathbf{T}$	truncation point of secondary reflectors	-
T	temperature	K
ΔT	temperature difference	K
T_{solar}	solar-weighted narrow-angle transmittance	-
T_{λ}	spectral narrow-angle transmittance	-
t	thickness	m
	time	s
	length parameter	m

U	bulk mean flow velocity	m/s
	voltage	V
u	focal function	m
	arbitrary coordinate	-
V	volume	m ³
$\dot{V}$	volume flow rate	m ³ /s
v	pneumatic control variables (input)	Pa; mbar
$\dot{W}_{\text{pump}}$	pumping power	W
w	width	m
	measured shape variables (output)	cm; mrad
x	x -coordinate (transversal; parallel to plane of incidence)	m
y	y -coordinate (in trough direction; perpendicular to plane of incidence)	m
z	z -coordinate (surface normal, optical axis)	m

Greek characters

α	surface absorptivity, window absorptance	-
	kinetic energy factor	-
α_{incl}	surface inclination angle from vertical	deg
β_{base}	base sheet opening angle	deg
β_{groove}	groove opening angle	deg
β_{track}	tracking angle	deg
γ	intercept factor	-
δ_{kj}	Kronecker delta	-
ε	surface emissivity, window emittance	-
	strain	-
$\varepsilon_{\text{blocking, bars}}$	fraction of solar radiation blocked by bars	-
$\varepsilon_{\text{pump}}$	specific pumping power demand	-
ζ	toroidal angle	rad
η	efficiency	-
η_{optical}	optical efficiency	-
η_{th}	thermal efficiency	-
θ	cone half-angle	mrad
	zenith angle	rad
	incidence angle	deg

θ	angular deviation of reflected ray from focus	mrad
θ_{skew}	skew angle	deg
θ_{ilt}	tilt angle of tilted and skewed coiled absorber tubes	deg
κ_{air}	specific heat ratio of air	-
λ	wavelength	μm
μ	dynamic viscosity	Pa s
ξ_{gate}	relative linear shift of knife gate valve [0 = fully closed, 1 = fully open]	-
ρ	surface reflectivity, window reflectance	-
	density	kg/m^3
ρ_{λ}	spectral surface reflectivity	-
σ	standard deviation of angular scattering	mrad
	Stefan-Boltzmann constant	$\text{W}/(\text{m}^2 \text{K}^4)$
	stress	N/m^2
τ	(window) transmittance	-
τ_{λ}	spectral internal transmissivity	-
ν	Poisson ratio	-
Φ	rim angle of solar concentrator	deg
	radiant flux	W
Φ_{ilt}	focal plane tilt angle	deg
φ	slope angle of solar concentrator	deg
ϕ	azimuth angle, poloidal angle	rad
	circular sector angle	deg
χ	convolution coordinate	m
ω	solid angle	sr

Subscripts and superscripts

I	first gray band (transparent window)
II	second gray band (opaque window)
400	(thermal conductivity) at 400 °C
acc	acceptance (angle)
amb	ambient
app	apparent
b	blackbody
c	characteristic

cond	conduction
conv	convection
eff	effective
el	electrical
exp	experiment
ext	external
h	directional-hemispherical (reflectance)
i	inner; incident
in	inlet
ins	insulation
<i>i</i>	finite volume index
<i>j</i>	circular arc index
	finite volume/surface segment index; fluid volume index
<i>k</i>	finite volume/surface segment index
<i>l</i>	gray band index
lm	log-mean (temperature difference)
m	(bulk) mean
n	normal conditions; surface slope error
o	outer
out	outlet
p	primary concentrator
r	reflected
rad	thermal radiation
s	surface; scattering (angle)
sim	simulation
src	source (divergence angle)
sup	support (mirror film)
t	transmitted
th	thermal
win	window
<i>x</i>	local coordinate (from leading edge); in plane of incidence
<i>y</i>	in trough direction; perpendicular to plane of incidence
yr	yearly (per year), annual mean
λ	spectral (per unit wavelength)
$\Delta\lambda$	gray band
$\parallel$	parallel polarization (p)
$\perp$	perpendicular polarization (s)

'	directional; per unit length
"	per surface area

Abbreviations

AM	air mass
AR	anti-reflection (coating)
ASC	arcspline concentrator
boPET	bi-axially oriented polyethylene terephthalate (film)
CCD	charge-coupled device
CSP	concentrated solar power
DGW	double-glazed window
DSG	direct steam generation
ETFE	ethylene tetrafluoroethylene
FS	full signal output
FWHM	full width at half maximum
HTF	heat transfer fluid
IR	infrared
LTP	line-to-point (focus)
MC	Monte Carlo
ORC	organic Rankine cycle
PID	proportional-integral-derivative (controller)
PTC	parabolic trough collector
PV	photovoltaics
PVC	polyvinyl chloride
PVC-PES	polyvinyl chloride coated polyester (film)
PVD	physical vapor deposition
PTFE	polytetrafluoroethylene
Rd	reading
RH	relative humidity
RMS	root-mean-square
SGW	single-glazed window
SPHER	small particle heat exchanger receivers
TES	thermal energy storage
TPES	total primary energy supply
UV	ultraviolet

1 Introduction

In 2013, the world's total primary energy supply (TPES) reached 565×10^{18} J or 13.5 billion tonnes of oil equivalent [1]. Besides residual energy sources from the formation of our planet (nuclear, geothermal) and tidal energy resulting from the gravitational field of the moon, all primary energy sources or 95% of the 2013 global TPES originate from the sun. In fact, the solar radiation arriving at the earth's surface in one year, estimated from the global mean surface flux [2], amounts to 3.0×10^{24} J or more than 5000 times the world's current TPES. Resource depletion and climate change will necessitate a shift from fossil fuels to renewable energy sources, which raises the question: "How can solar radiation arriving at the earth's surface be effectively converted into useful forms of energy?"

1.1 Motivation for concentrating solar thermal power

Special emphasis has received the electricity sector, accounting for roughly one third of global TPES today [1]. In contrast to conventional renewable electricity-generating technologies such as biomass and hydropower that are inherently dispatchable, newer technologies, besides geothermal energy, usually rely on an intermittent, deterministic (tidal) or stochastic (solar, wind, waves, etc.) source, posing challenges from a grid stability point of view. A technology that is able to store sunlight on a large scale and convert it into electricity is concentrating solar power (CSP) equipped with thermal energy storage (TES). Unlike photovoltaic (PV) cells that convert sunlight directly into an electric current, CSP plants focus solar radiation to heat up a fluid, which is then used to drive a power block or to charge the TES for temporally decoupled electricity generation. Furthermore, the thermal energy can be used to provide industrial process heat, accounting for roughly a sixth of global TPES in 2011 [3]. In order to afford the

future deployment of CSP, innovative technologies that overcome the technical and economical drawbacks of current plants are required [4-6].

1.2 Motivation for solar troughs

CSP technologies are classified following their solar concentrating optics into line- and point-focus systems. Line-focus devices are also referred to as 2D or trough concentrators because their geometry may be described in 2D and extruded in the third dimension along the focal line, while point-focus systems are often referred to as 3D concentrators because they require 3D shapes to focus radiation onto a point. The most common line-focus system is the parabolic trough collector (PTC), which has proven its ability for power production in over 30 years of commercial operation [7, 8]. Thanks to their extruded shape, troughs are easier to manufacture and more scalable than 3D structures, which offers the potential of lower costs per square meter of collecting mirror area. PTCs are traditionally manufactured from expensive back-silvered glass reflectors supported by steel structures [9], which suffer from lack of rigidity at elevated wind speeds. Lighter aluminum reflectors are used in an enclosed trough system to produce steam for solar thermal enhanced oil recovery [10, 11]. A greenhouse made of glass protects the solar collectors from wind and dust, which allows for significant weight and cost reductions. Lightweight enclosed trough systems based on inflated metallized polymer reflectors protected by a weather-resistant transparent film at the top have been investigated recently, offering further saving potential [12-14].

1.3 Motivation for air as heat transfer fluid

Conventional PTCs use thermal oils as heat transfer fluid (HTF), which have several limitations. Their operating temperature is restricted to below 400 °C, which in turn limits the attainable thermal-to-electric efficiency of the power block. Furthermore, thermal oils are flammable, hazardous in case of leakage, and require troublesome TES concepts involving molten salt tanks coupled via a heat exchanger, which involves cost and energy penalties [8].

Molten salts, on the other hand, are stable up to 500 – 600 °C depending on their chemical composition, and allow for direct coupling of the solar field and

TES [15]. However, molten salts have also high freezing points at 120 – 220 °C, which poses technical challenges and requires a backup system for heat tracing to prevent the HTF from freezing in the pipes under all circumstances.

An alternative to synthetic HTFs is direct steam generation (DSG), where water is fed to the solar field to produce saturated or super-heated steam that can be directly fed to a steam turbine, thus eliminating the cost and energy penalties associated with the steam generators. Challenges of DSG arise from the phase change of the HTF, which requires sophisticated control systems to deal with transients and stratification of the two-phase water/steam flow in the horizontal absorber tubes as well as complex TES concepts [16]. The feasibility of DSG has been proven in several demonstration projects [9].

Gaseous HTFs such as air, on the other hand, are rather used in point-focusing solar towers and less common in PTCs due to their low density and thermal conductivity, which requires solar receivers with larger flow cross-sections and heat transfer areas. Nevertheless, air shows several intriguing advantages over the aforementioned HTFs: 1) no costs; 2) no relevant operating temperature limits; 3) no HTF degradation nor phase change; 4) near-ambient operating pressure, allowing for thin-walled ducts and avoiding leakage concerns; 5) neither corrosiveness nor toxicity; 6) direct coupling to a low-cost TES based on a thermocline pebble stone packed bed, eliminating costs associated with heat exchangers; and 7) straightforward system integration into industrial processes where heating is achieved by combustion. This leads to the question that shall be addressed in this thesis: “How can air be used as heat transfer fluid in parabolic trough collectors?”

1.4 Thesis outline

The focus in this thesis is on the experimental and numerical investigation of solar air receivers for line-focusing solar concentrators. Sidetracks, primarily regarding the solar concentrator, are explored as far as they contribute to the superior objective of producing economically viable concentrating solar thermal collectors.

Chapter 2 introduces the conceptual designs of the novel solar air receiver and solar concentrating optics. A thermodynamic heat transfer model of the solar

receiver including radiation, conduction, and convection is formulated and coupled to an optical Monte Carlo ray-tracing program for the solar radiation. The proof of concept of the new design is delivered by on-sun tests of a 1 m-long receiver prototype performed at the Airlight Energy test facility in Biasca, Switzerland. Furthermore, the numerical model is validated by comparison to experiments.

In Chapter 3, the validated heat transfer model is applied to a parameter study of the new receiver design. Its characteristics and features are discussed, crucial design parameters are defined, and their effects assessed by yearly performance predictions.

Chapter 4 introduces the engineering design of the industrial scale solar collector comprising the pneumatic parabolic trough concentrator and new solar air receiver. The numerical heat transfer model of the solar receiver is extended to account for the relevant engineering details, and validated by comparison to on-sun tests performed with a $1.2 \text{ MW}_{\text{th}}$ solar collector unit at the Airlight Energy pilot plant in Ait Baha, Morocco. A detailed energy balance of experiments is presented, technical improvements are identified and their effectiveness is assessed by annual mean performance predictions using the validated model.

In Chapter 5, alternative solar air receivers with potentially lower construction costs are explored. The conceptual design is introduced and the modelling techniques of Chapters 2 and 4 are applied. Yearly performance predictions are performed and a comparison of different solar receiver concepts is included.

In Chapter 6, the optical and pneumatic design of the solar trough concentrator based on inflated polymeric films is outlined, including the film length regulation via inflatable sleeves. Further experimental results from operation are given, and a control system for automatic pressure regulation is introduced.

In Chapter 7, the materials, methods, and results of the optical characterization of materials for solar concentrators are presented. The narrow-angle spectral specular reflectance and angular scattering of solar reflector materials as well as narrow-angle transmittance of protective semi-transparent

materials are measured in the optical lab of the Professorship of Renewable Energy Carriers (PREC) at ETH Zurich.

Chapter 8 summarizes the key findings of this thesis and gives an outlook for recommended future research topics on solar thermal trough collectors with emphasis on their application to industrial process heat.

2 Conceptual design and proof of concept¹

In this chapter, the conceptual design of a novel solar air receiver for solar trough concentrators is introduced and investigated numerically and experimentally. As an introduction, a review of solar air receiver designs is included.

2.1 Review of solar air receivers

The literature on solar trough receivers based on air as heat transfer fluid (HTF) is limited. Accordingly, the review is extended to central receiver systems designed for point-focusing solar towers and summarizes line-focus receivers in the end.

2.1.1 Central receivers

Air is used in several high-temperature central receiver designs for point-focusing solar towers, which have been reviewed in [17] and classified into volumetric receivers, small particle receivers, and tubular receivers.

Volumetric receivers are made of porous metallic or ceramic structures, which absorb concentrated solar radiation within their volume and provide a large surface area for heat transfer to the gaseous HTF. A comprehensive review of volumetric receivers until 2009 is given in [18]. Volumetric receivers are classified following their application into open-loop systems at atmospheric pressure for Rankine cycles and closed-loop pressurized systems for direct Brayton or combined cycles. The first demonstration plant of a solar tower with an open volumetric air receiver coupled to a 1.5 MW_{el} Rankine cycle steam turbine and thermal energy storage has been built in Jülich, Germany [19]. The modular solar receiver design comprising a matrix of ceramic absorbers has been

¹ Material in this chapter has been published in P. Good, G. Ambrosetti, A. Pedretti, and A. Steinfeld, "An array of coiled absorber tubes for solar trough concentrators operating with air at 600 °C and above," *Solar Energy*, vol. 111, pp. 378-395, 2015.

developed in a series of research projects (HiTRec, SOLAIR) focusing on performance, reliability, and up-scaling [20, 21]. Pressurized volumetric receivers generally require an airtight window that withstands the operating pressures and temperatures, which increases the complexity and costs of the design [18]. A directly-irradiated annular pressurized receiver (DIAPR) with a frustum-shaped fused-silica window [22] and a pin fin ceramic absorber [23] was successfully tested for 250 hours at operating pressures and temperatures up to 20 bar and 1200 °C, respectively [24]. In the REFOS project, compressed air at 15 bar was preheated to 800 °C for a solar hybrid combined cycle power plant using a metallic absorber and a domed quartz window [25]. A pressurized receiver for solar-driven gas turbines, in which the window is replaced by a cylindrical silicon carbide cavity surrounded by ceramic foams for effective heat transfer to the annular air flow, was developed at ETH Zurich [26, 27]. A 50 kW_{th} solar receiver module was successfully tested for 93 hours at 2 – 6 bar absolute pressure and reached a peak air outlet temperature of 1200 °C [27].

In small particle heat exchanger receivers (SPHER), submicron particles are suspended in air and absorb concentrated solar radiation entering through the windowed aperture throughout the gas volume. Thanks to their high surface-to-volume ratio, the small particles transfer heat effectively to the surrounding HTF until they vaporize or combust, leaving a clean exit air stream that can be fed to a gas turbine [28]. HTF outlet temperatures of 2000 K have been reached with carbon particles suspended in air at atmospheric pressure, intended for process heat applications [29]. More recently, lab-scale experimentation with a pressurized SPHER pointed out clogging issues of the particle injector [30]. The pressurized window and the solid-gas suspension system that maintains the desired particle concentration within the receiver remain the technical challenges of small particle receivers [17].

A tubular receiver comprised of multiple parallel straight absorber tubes that form a frustum-shaped cavity for absorbing concentrated solar radiation and are thermally insulated on the backside was tested for use in a solar hybrid microturbine system [31]. The receiver reached the target operating temperature of 800 °C but at efficiencies below the design point due to flaws in the thermal insulation and sub-design mass flow rates.

2.1.2 Trough receivers

A few solar air receivers have been tested on line-focusing solar concentrators. Tests with pressurized CO_2 at 70 bar and 400 °C as HTF in conventional parabolic trough solar receivers comprising a selective-coated and vacuum-insulated stainless steel absorber tube confined by a transparent glass jacket have been performed at the Plataforma Solar de Almería [32]. The challenges remaining are the high pressure in the solar field and the stability of the selective coating when aiming at higher operating temperatures. The use of air at near-atmospheric pressure in conventional tubular trough receivers would cause either prohibitive pressure drops or heat losses if the absorber tube diameter were increased. This favors the utilization of cavity receivers, in which the flow cross-section and heat transfer area are decoupled from the aperture through which the major heat losses occur. Cylindrical cavity receivers for line-focus systems were investigated for liquid HTFs flowing through an annular cross-section [33] and multiple parallel absorber tubes [34]. A cylindrical cavity receiver containing a single absorber tube for atmospheric air has been investigated numerically and experimentally [35, 36], and a directly irradiated cylindrical cavity with a rectangular windowed aperture has been proposed [37]. Moreover, a small-scale volumetric receiver for process heat applications using air at atmospheric pressure and temperatures up to 250 °C was tested at DLR [38]. The receiver consisted of an outer rectangular cold air duct, a nested inner hot air duct, a porous absorber placed along the focal line, and a rectangular windowed aperture. Fresh air is distributed over the full receiver's length via the cold air duct, percolates through the porous structure where it is heated by absorbed solar radiation, and is collected by the hot duct.

The receiver design investigated in this thesis may be regarded as an implementation of the volumetric receiver concept using tubular absorbers. Its conceptual design is introduced in the next section after the solar concentrating optics.

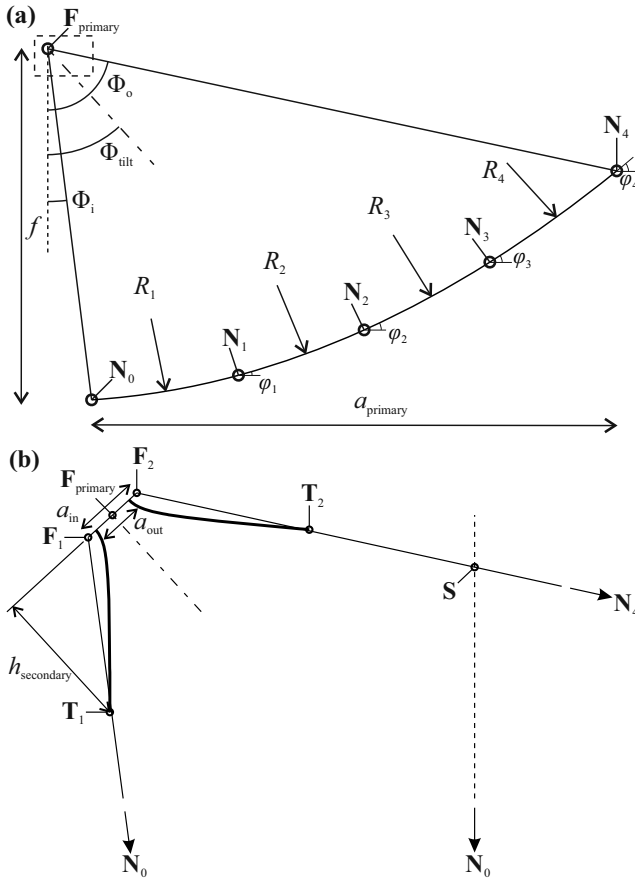


Fig. 2.1: Scheme of one wing of the solar concentrating optics, combining an imaging primary concentrator composed of a 4-arcs-approximated parabola (a) with a non-imaging secondary concentrator composed of a tilted, truncated, linear trumpet (b).

2.2 Design

2.2.1 Solar concentrating optics

An imaging primary concentrator is combined with a non-imaging secondary concentrator. The primary concentration is achieved by a one-axis tracking linear trough concentrator composed of two-wing reflectors, each consisting of four tangentially adjacent circular arc segments that closely approximate the parabolic

Table 2.1: Geometric parameters of the combined optical concentrator system.

baseline dimensions	Primary concentrator			
		index j	radius R_j	slope angle φ_j
focal length f	3.25 m	1	6.78 m	15.3°
aperture width a_{primary}	4.85 m	2	7.80 m	24.4°
inner rim angle Φ_i	7.1°	3	9.52 m	32.3°
outer rim angle Φ_o	77.9°	4	12.18 m	39.1°
Secondary concentrator				
inlet aperture a_{in}	73.5 mm	height $h_{\text{secondary}}$		160 mm
outlet aperture a_{out}	50.0 mm	tilt angle Φ_{tilt}		42.5°

shape. This shape can be constructed by inflated mirror membranes that are inexpensive and allow for wide apertures up to 10 m [13]. The secondary concentration is obtained by two non-imaging secondary elements, one for each wing of the primary concentrator. To achieve a low average number of reflections and thus high optical efficiency, 2D-trough trumpets are used, each formed by two hyperbolic mirror profiles. These symmetric trumpets are accommodated in tandem with the primary concentrator by separating the focal lines of the two primary wings and by titling their focal planes by the arithmetic mean $\Phi_{\text{tilt}} = (\Phi_i + \Phi_o) / 2$ of the inner and outer rim angles. One wing of the primary concentrator is shown in Fig. 2.1a. A zoom on the focus and secondary optics (cf. dashed rectangle in Fig. 2.1a) is shown in Fig. 2.1b. The distance between the focal points F_1 and F_2 corresponds to the inlet aperture a_{in} of the trumpet. To prevent shading of the primary concentrator and still capture all rays reflected from N_0N_4 onto F_1F_2 , the trumpet is truncated to the left of point S at points T_1 and T_2 located on the lines F_1N_0 and F_2N_4 , respectively [39]. The dimensions are summarized in Table 2.1. The theoretical geometric concentration ratios of the primary and secondary concentrators are $C_{g,\text{primary}} = a_{\text{primary}} / a_{\text{in}} = 65.85$ and $C_{g,\text{secondary}} = a_{\text{in}} / a_{\text{out}} = 1.473$, respectively, yielding a total concentration of $C_g = a_{\text{primary}} / a_{\text{out}} = 97.0$.

2.2.2 Solar receiver

A fundamental limitation of tubular absorbers is the trade-off between flow resistance and heat transfer. Accordingly, for a fixed receiver length L as shown in Fig. 2.2a and given mass flow and heat transfer rates, an increase in the pipe diameter D leads to a lower pressure drop (Δp) and a higher temperature difference (ΔT) between absorber wall and HTF. One way to mitigate this problem is to branch the flow up into N_{tube} parallel channels (cf. Fig. 2.2b), which allows to achieve the same heat transfer at the same Δp and a relatively lower ΔT . To fit a large number of absorber tubes into a solar receiver, the multiple absorber tubes are arranged in cross-flow and connected via two manifold axial pipes: one feeding the cold HTF and one collecting the hot HTF. This is shown schematically in Fig. 2.2c. Such a flow configuration decouples the length L_{tube} of the absorber tubes from the solar receiver length L and offers large design flexibility. Accordingly, the solar receiver may be designed to accommodate a large number of preferably long cross-flow absorber tubes. The diameters of the run-back pipes may be chosen sufficiently large to avoid significant pressure drops. In principle, any kind of cross-flow absorber may be utilized in this solar receiver concept. A helically coiled tube having a cavity-type geometry is selected, as it offers several intriguing advantages: 1) high apparent absorptivity due to the cavity effect [40]; 2) low apparent emissivity as the cold air inlet is located near the aperture; 3) enhanced forced convection heat transfer coefficient due to secondary flow induced by centrifugal forces acting in the plane perpendicular to the axial flow direction [41]; 4) compact packing of tubes; 5) potential for low-cost mass production. Helical pipes have been previously applied to absorb spilled radiation in central receiver concepts for point-focusing solar concentrating towers [42, 43]. Recently, a rectangular cavity-shaped coiled absorber tube has been investigated as a solar receiver for parabolic dish concentrators [44].

In the present study, an array of coiled absorber tubes is employed as cross-flow absorber for line-focusing solar trough concentrators. The absorber tubes are contained within an insulated groove having a rectangular windowed opening. The window is positioned at the focal plane of the solar trough concentrator, and the helically coiled tubes are placed close behind it. Secondary

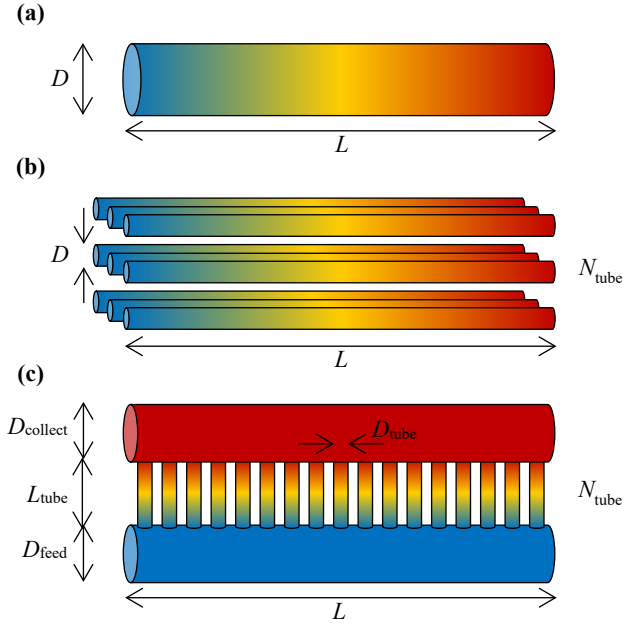


Fig. 2.2: a) Single tubular absorber; b) multiple parallel absorber tubes and c) array of parallel absorbers in cross-flow configuration, connected via feeding and collecting manifold pipes.

concentrating optics are incorporated to boost the solar concentration ratio (cf. Table 2.1). A cross-section of the novel solar receiver configuration is shown schematically in Fig. 2.3. Concentrated solar radiation from the secondary concentrator outlet enters through the windowed apertures and is absorbed by the array of helical tube absorbers. Cold air is supplied to each helical tube via the manifold feeding pipe, heated to the desired operating temperature by absorbed concentrated solar radiation, and released into the manifold collecting pipe leading to the power block or thermal storage. To compensate for pressure drops occurring in the feeding and collecting manifold pipes, adjustable knife gates are incorporated along the length of the solar receiver to establish uniform flow distribution through the absorber tubes. The diameters of the feeding and collecting pipes are chosen such that the pressure distribution is relatively flat and the overall pumping power remains below 1.5% of the expected electric

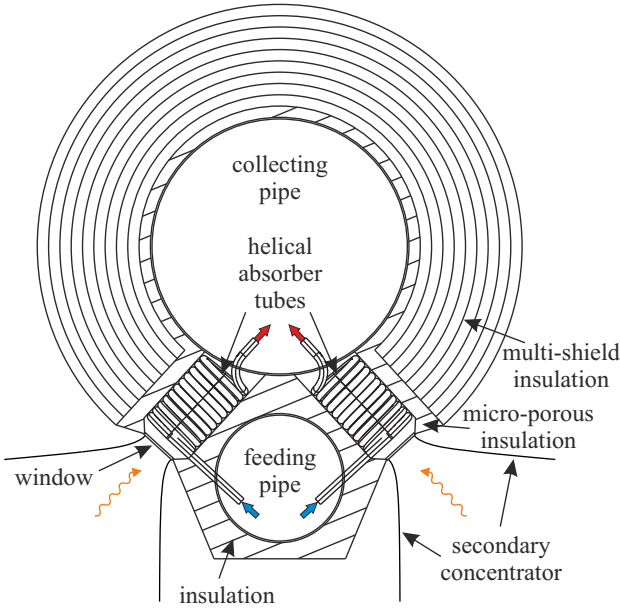


Fig. 2.3: Schematic cross-section of the solar receiver configuration comprising an array of helically coiled absorber tubes connected via two feeding/collecting manifold pipes. Window and secondary optics are incorporated at the focal plane of the solar trough concentrator.

power output. Further considerations include costs of the piping and thermal insulation material. For a 212 m-length solar collector, $T_{\text{in}} = 120\text{ }^{\circ}\text{C}$, and $T_{\text{out}} = 650\text{ }^{\circ}\text{C}$, the diameters of the manifold pipes are $D_{\text{feed}} = 0.4\text{ m}$ and $D_{\text{collect}} = 0.45\text{ m}$. Heat losses are minimized by: 1) a glass window placed at the aperture plane acting as a flow barrier to natural convection and radiation shield to IR thermal emission from the absorber tube, where optical reflection losses could be reduced by applying an anti-reflection (AR) coating; 2) high-temperature micro-porous insulation; and 3) multiple layers of low-emissivity foils surrounding the collecting pipe and providing low thermal inertia insulation. To avoid expensive refractory alloys, the receiver is designed for operating temperatures up to $650\text{ }^{\circ}\text{C}$ such that it can be constructed from conventional materials such as stainless steel for the helical tubes, rectangular groove wall, and piping, borosilicate or quartz glass for the window, and calcium silicate or glass wool for the insulation.

2.3 Heat transfer modeling

For design purposes and performance predictions, a heat transfer model of the array of coiled absorber tubes is developed. A cross-section of a single absorber tube with the relevant geometric parameters, surface temperatures, fluid states, and modes of heat transfer is shown in Fig. 2.4. Considering the system boundary spanned by the external window and groove wall surfaces #4 and #6 in the xz -plane, the steady-state energy conservation is given by

$$\dot{Q}_{\text{solar,incident}} - \dot{Q}_{\text{solar,reflected}} - \dot{Q}_{\text{reradiation}} - \dot{Q}_{\text{convection}} - \dot{Q}_{\text{conduction}} - \dot{Q}_{\text{gain,HTF}} = 0 \quad (2.1)$$

where $\dot{Q}_{\text{solar,incident}}$ is the solar radiation focused by the combined concentrator system onto the windowed aperture of the solar receiver, $\dot{Q}_{\text{solar,reflected}}$ is the solar radiation lost by reflection at the window, absorber tube, and groove wall, $\dot{Q}_{\text{solar,absorbed}} = \dot{Q}_{\text{solar,incident}} - \dot{Q}_{\text{solar,reflected}}$ is the solar radiation absorbed by the receiver (i.e. window, absorber tube, and groove wall), $\dot{Q}_{\text{reradiation}}$ denotes the thermal radiation heat losses including emission from the external glass surface and re-radiation from the absorber tube and rectangular groove leaving through the windowed aperture, $\dot{Q}_{\text{convection}}$ is the heat dissipated to ambient air by natural convection, $\dot{Q}_{\text{conduction}}$ represents the conduction heat losses through the insulation, and $\dot{Q}_{\text{gain,HTF}}$ is the useful heat extracted by the HTF. To account for the interaction with the neighbored absorber tubes, periodic boundary conditions are considered in the y -direction, which results in a net heat transfer of zero in this direction. The energy losses are: 1) optical losses due to reflection at the window, 2) heat losses by re-radiation from the external absorber tube and internal groove wall surfaces (#2 and #5) leaving through the semi-transparent window, and by thermal emission and free convection from the external window surface (#4), and 3) conduction heat losses from the external groove wall surface (#6).

Conduction

Conservation of energy in the solid domains is governed by the 3D steady-state heat conduction equation

$$\nabla \cdot (k \nabla T) = 0 \quad (2.2)$$

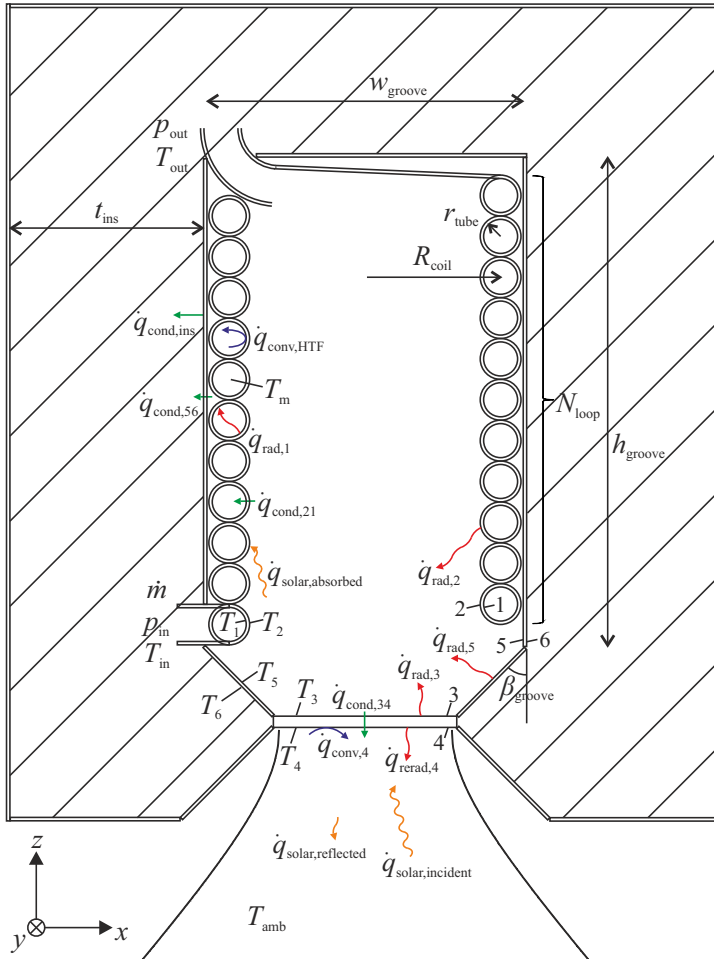


Fig. 2.4: Cross-section of a single coiled absorber tube with geometric parameters, surface numbering, temperatures, and modes of heat transfer. The y -coordinate indicates the axial direction of the solar trough concentrator.

with boundary conditions at surfaces #1–5

$$-k\nabla T|_s \cdot \hat{\mathbf{n}}_s = \dot{q}_s'' \quad (2.3)$$

where $-k\nabla T|_s$ is the conductive heat flux evaluated at the surface, $\hat{\mathbf{n}}_s$ is the surface normal vector, and $\dot{q}_s''$ is the net heat flux leaving the surface by radiation

and convection. Discretization of Eq. (2.2) into finite volumes and integration over the control volume yields a linear expression in terms of node temperatures for the net conductive heat transfer rate at each control volume (i, j, k) [45]

$$\begin{aligned} \dot{q}_{\text{cond},i,j,k} = & a_{i,j,k} T_{i,j,k} - (a_{i-1,j,k} T_{i-1,j,k} + a_{i+1,j,k} T_{i+1,j,k} \\ & + a_{i,j-1,k} T_{i,j-1,k} + a_{i,j+1,k} T_{i,j+1,k} + a_{i,j,k-1} T_{i,j,k-1} + a_{i,j,k+1} T_{i,j,k+1}) \end{aligned} \quad (2.4)$$

The calculation and the analytical expressions for the coefficients a are given in Appendix A. Conduction losses through the insulation are given by,

$$\dot{q}_{\text{cond,ins},j,k}'' = \frac{k_{\text{ins}}}{t_{\text{ins}}} (T_{6,j,k} - T_{\text{amb}}) \quad (2.5)$$

where t_{ins} and k_{ins} are the thickness and thermal conductivity of the insulation.

Radiation

The Monte Carlo (MC) ray-tracing technique is applied to the complete system including the primary imaging concentrator, secondary non-imaging concentrator, glass window, absorber tubes, and surrounding groove. Sunrays arrive at the primary concentrator aperture area A_{primary} under a specified incidence angle θ_{skew} and within a cone half-angle of $\theta_{\text{sun}} = 4.65$ mrad. The solar power is set such that a flat target with its surface normal vector parallel to the ray main direction receives the desired level of solar direct-normal irradiance (E_{DNI}),

$$\dot{Q}_{\text{solar,aperture}} = A_{\text{primary}} E_{\text{DNI}} \cos \theta_{\text{skew}} \quad (2.6)$$

The spectral distribution of solar radiation is sampled from the ASTM G173 – 03 reference spectrum for direct and circumsolar irradiance at an air mass of 1.5 (AM 1.5) [46]. The primary mirror is represented by circular trough segments with specular reflectivity ρ_{primary} and Gaussian angular scattering with standard deviation σ_s to account for surface imperfections and shape deviations. The zenithal component θ_s of the angular scattering is drawn from a Rayleigh distribution, whereas the azimuthal component ϕ_s is assumed to be uniformly distributed on the interval $0 - 2\pi$ [47]. Similar surface properties are applied to the hyperbolic trough surfaces of the secondary concentrator to account for

rejection and absorption of light by the trumpet. Rays missing the trumpet, commonly noted as spilled radiation, are considered to be lost. The spectrally and directionally selective optical properties of the window are given by the transmittance $\tau(\lambda, \theta)$ and reflectance $\rho(\lambda, \theta)$ data. For each ray intersecting the window, the probabilistic events of transmission, reflection, and absorption at the glass are given by [35],

$$\begin{aligned} 0 < \Re &\leq \tau'_{\lambda, \text{ray}} \quad (\text{transmission}) \\ \tau'_{\lambda, \text{ray}} < \Re &\leq \tau'_{\lambda, \text{ray}} + \rho'_{\lambda, \text{ray}} \quad (\text{reflection}) \\ \tau'_{\lambda, \text{ray}} + \rho'_{\lambda, \text{ray}} < \Re &< 1 \quad (\text{absorption}) \end{aligned} \quad (2.7)$$

where $\Re$ is a uniformly distributed random number taken from the interval (0,1). The helical tube geometry is approximated by a stack of N_{loop} tori with major and minor radii R_{coil} and r_{tube} , respectively. Solar-weighted radiative properties are used for the opaque absorber tubes and surrounding groove walls according to

$$c_{\text{solar}} = \frac{\int c_{\lambda} E_{\lambda, \text{solar}} d\lambda}{\int E_{\lambda, \text{solar}} d\lambda} \quad (2.8)$$

where c_{λ} is a spectral radiative property and $E_{\lambda, \text{solar}}$ is the spectral irradiance of the sun (ASTM G173 – 03 AM 1.5 reference spectrum for direct and circumsolar irradiance [46]). The values of the optical properties used in the MC simulations are listed in Table 2.2.

The thermal radiative exchange between the outer absorber tube surface (#2), the inner window surface (#3), and the inner groove surface (#5) is modeled by means of the gray-band approximated radiosity method for enclosures with diffuse surfaces and semi-transparent windows using two spectral bands [40]:

$$\begin{aligned} \sum_{j=1}^{n_2+n_3+n_5} \frac{\delta_{kj} - \rho_{\Delta\lambda_l, j} F_{k-j}}{1 - \rho_{\Delta\lambda_l, j} - \tau_{\Delta\lambda_l, j}} \dot{q}''_{\text{rad}, \Delta\lambda_l, j} = \\ \sum_{j=1}^{n_2+n_3+n_5} \frac{\delta_{kj} - (1 - \tau_{\Delta\lambda_l, j}) F_{k-j}}{1 - \rho_{\Delta\lambda_l, j} - \tau_{\Delta\lambda_l, j}} \varepsilon_{\Delta\lambda_l, j} \sigma F_{\Delta\lambda_l T_j} T_j^4 \end{aligned} \quad \delta_{kj} = \begin{cases} 1 & \text{if } k = j \\ 0 & \text{otherwise} \end{cases} \quad (2.9)$$

Table 2.2: Spectrally averaged hemispherical radiative properties of materials.

	Solar spectrum	1st gray band	2nd gray band
Quartz glass [48]			
spectral region	0.28 – 4.0 μm	0.16 – 3.6 μm	> 3.6 μm
transmittance	0.86	0.86	0.00
reflectance	0.14	0.14	0.21
emittance	0.00	0.00	0.79
Borosilicate glass [49, 50]			
spectral region	0.28 – 4.0 μm	0.3 – 2.7 μm	> 2.7 μm
transmittance	0.83	0.82	0.00
reflectance	0.15	0.14	0.12
emittance	0.02	0.04	0.88
AR-coated borosilicate glass [49, 51]			
spectral region	0.28 – 4.0 μm	0.3 – 2.7 μm	> 2.7 μm
transmittance	0.84	0.75	0.00
reflectance	0.12	0.19	0.12
emittance	0.04	0.06	0.88
Oxidized stainless steel [52]			
emissivity	0.88	0.85	0.67
reflectivity	0.12	0.15	0.33
Black paint coating [53]			
emissivity	0.92	0.92	0.92
reflectivity	0.08	0.08	0.08

In Eq. (2.9), T_j and $\dot{q}_{rad, \Delta\lambda_l, j}''$ are the temperature and the net leaving radiative flux of surface j , $\varepsilon_{\Delta\lambda_l, j}$, $\rho_{\Delta\lambda_l, j}$, and $\tau_{\Delta\lambda_l, j}$ denote the hemispherical emittance, reflectance, and transmittance of surface j in the spectral band $\Delta\lambda_l$, respectively ($\tau_{\Delta\lambda_l, j} = 0$ for opaque surface segments), $F_{\Delta\lambda_l T_j}$ is the fraction of total power emitted by a blackbody at temperature T_j in the band $\Delta\lambda_l$, F_{k-j} is the view factor from surface k to j , and δ_{kj} the Kronecker delta. The radiative fluxes originating from the absorber tube and surrounding groove surfaces (#2 and #5) that are incident on the inner window surface (#3) and leaving to the surroundings through the outer window surface (#4) are, respectively,

$$\begin{aligned}\dot{q}''_{\text{incident},\Delta\lambda_i,3} &= \frac{1}{1 - \rho_{\Delta\lambda_i,\text{win}} - \tau_{\Delta\lambda_i,\text{win}}} \left(\dot{q}''_{\text{emitted},\Delta\lambda_i,3} - \dot{q}''_{\text{rad},\Delta\lambda_i,3} \right) \\ \dot{q}''_{\text{leaving},\Delta\lambda_i,4} &= \frac{\tau_{\Delta\lambda_i,\text{win}}}{1 - \rho_{\Delta\lambda_i,\text{win}} - \tau_{\Delta\lambda_i,\text{win}}} \left(\dot{q}''_{\text{emitted},\Delta\lambda_i,3} - \dot{q}''_{\text{rad},\Delta\lambda_i,3} \right)\end{aligned}\quad (2.10)$$

Emission from the window to the surroundings is considered by

$$\dot{q}''_{\text{rerad},\Delta\lambda_i,4} = \dot{q}''_{\text{emitted},\Delta\lambda_i,4} - \varepsilon_{\Delta\lambda_i,\text{win}} \sigma F_{\Delta\lambda_i T_{\text{sky}}} T_{\text{sky}}^4 \quad (2.11)$$

where T_{sky} is the apparent sky temperature of the surroundings. The first gray band, denoted by the subscript I, covers the spectral region in which the window is semi-transparent. To account for the transparency of the window, i.e. both sides #3 and #4 see the absorber tube and the environment, the effective mean emissive flux $\dot{q}''_{\text{emitted},\Delta\lambda_i,3} = \dot{q}''_{\text{emitted},\Delta\lambda_i,4} = \varepsilon_{\Delta\lambda_i,\text{win}} \sigma (F_{\Delta\lambda_i T_3} T_3^4 + F_{\Delta\lambda_i T_4} T_4^4) / 2$ is used for the window emission terms in Eq. (2.9) to (2.11) in the first band I and $\dot{q}''_{\text{rad},\Delta\lambda_i,3}$ and $\dot{q}''_{\text{rerad},\Delta\lambda_i,4}$ are equally distributed between surfaces #3 and #4. The second spectral band, denoted by the subscript II, covers the infrared (IR) region beyond the cut-off wavelength of the glass, in which the window is completely opaque ($\tau_{\Delta\lambda_{\text{II}},\text{win}} = 0$). Eq. (2.9) then reduces to the standard enclosure theory's system of equations for opaque, gray-diffuse surfaces and the common expressions for the window emission terms $\dot{q}''_{\text{emitted},\Delta\lambda_{\text{II}},3} = \varepsilon_{\Delta\lambda_{\text{II}},\text{win}} \sigma F_{\Delta\lambda_{\text{II}} T_3} T_3^4$ and $\dot{q}''_{\text{emitted},\Delta\lambda_{\text{II}},4} = \varepsilon_{\Delta\lambda_{\text{II}},\text{win}} \sigma F_{\Delta\lambda_{\text{II}} T_4} T_4^4$ are used in Eq. (2.9) to (2.10) and (2.11), respectively. Thermal radiation exchanged among inner surface segments of the absorber tube (surface #1 in Fig. 2.4) is solved by the radiosity method for gray-diffuse enclosures

$$\begin{aligned}\sum_{j=1}^{n_i} \frac{1}{1 - \rho_{\Delta\lambda_i,j}} (\delta_{kj} - \rho_{\Delta\lambda_i,j} F_{k-j}) \dot{q}''_{\text{rad},\Delta\lambda_i,j} = \\ \sum_{j=1}^{n_i} \frac{\varepsilon_{\Delta\lambda_i,j}}{1 - \rho_{\Delta\lambda_i,j}} (\delta_{kj} - F_{k-j}) \sigma F_{\Delta\lambda_i T_j} T_j^4\end{aligned} \quad \delta_{kj} = \begin{cases} 1 & \text{if } k = j \\ 0 & \text{otherwise} \end{cases} \quad (2.12)$$

View factors are determined using MC ray-tracing and exploiting reciprocity and symmetry relations, where the helically coiled tube is represented by a stack of tori, as aforementioned. The thermal radiative properties of materials are averaged over the corresponding spectral bands, $c_{\Delta\lambda_i} = \int_{\Delta\lambda_i} c_{\lambda} d\lambda / \Delta\lambda_i$. The

hemispherical properties of the glass are obtained by integration of directional data over the hemisphere according to [54]

$$c_{\lambda} = \frac{\int_{2\pi} c'_{\lambda}(\hat{s}) \cdot I'_{\lambda}(\hat{s}) \cos \theta d\omega}{\int_{2\pi} I'_{\lambda}(\hat{s}) \cos \theta d\omega} = \frac{\int_0^{\pi/2} \int_0^{2\pi} c'_{\lambda}(\hat{s}) \cos \theta \sin \theta d\phi d\theta}{\int_0^{\pi/2} \int_0^{2\pi} \cos \theta \sin \theta d\phi d\theta} \quad (2.13)$$

where $\hat{s}(\theta, \phi)$ is the direction vector and I'_{λ} the spectral intensity. Temperature-dependent thermal conductivities and directional spectral radiative properties are taken for the materials borosilicate [50, 55], fused silica [48, 56], stainless steel AISI 316 [52, 57]). In addition, the transmittance and reflectance of a double-sided AR-coated borosilicate glass [58] have been measured experimentally as a function of wavelength (0.28 – 4.0 μm) and incidence angle (0 – 75°) using a spectroscopic goniometry system [51]. As a proxy of unavailable directional data for borosilicate glass in the IR beyond the cut-off wavelength at 2.7 μm , normal spectral emittance data of Pyrex 7740 borosilicate glass are used [49]. The total emissivity of black coating (Aremco HiE-Coat 840 M) was set to 0.92 [53]. The spectrally averaged hemispherical radiative properties of materials are listed in Table 2.2 for the two bands. For completeness, solar-weighted hemispherical properties of the glasses are also included, although transmission and reflection of sunrays at the window are modeled by Eq. (2.7) in the MC simulations.

Convection

Forced convection inside the absorber tube (#1) and natural convection occurring at the outside of the window (#4) are modeled by means of correlations from literature.

Due to a larger heat transfer enhancement by secondary flow and smaller pressure drop under laminar conditions compared to the turbulent regime [41], the helically coiled pipe is designed such that the flow stays laminar under all operation conditions. The critical Re in a curved pipe is given by [59]

$$\text{Re}_{\text{critical}} = 2100 \left[1 + 12 \left(\frac{r_{\text{tube}}}{R_{\text{coil}}} \right)^{1/2} \right] \quad (2.14)$$

which reduces to the critical Re of straight pipe flow in the limiting case $R_{\text{coil}}/r_{\text{tube}} \rightarrow \infty$. The friction coefficient for fully developed laminar flow in a helical pipe is calculated by [60],

$$\frac{c_{f, \text{curved}}}{c_{f, \text{straight}}} = \left[\left(1 - \frac{0.18}{[1 + (35/\text{He})^2]^{1/2}} \right)^b + \left(1 + \frac{r_{\text{tube}}/R_{\text{coil}}}{3} \right)^2 \left(\frac{\text{He}}{88.33} \right) \right]^{1/2} \quad (2.15)$$

where $b = 2$ for $\text{De} \leq 20$, $b = 1$ for $20 < \text{De} \leq 40$, and $b = 0$ for $\text{De} > 40$. $c_{f, \text{curved}}$ and $c_{f, \text{straight}} = 16/\text{Re}$ denote the laminar Fanning friction coefficients of the curved and straight pipes, respectively, $\text{De} = \text{Re}(r_{\text{tube}}/R_{\text{coil}})^{1/2}$ is the Dean number and $\text{He} = \text{De}(1 + [p_{\text{coil}}/(2\pi R_{\text{coil}})]^2)^{-1/2}$ is the helical number taking into account the effective radius of curvature due to the coil pitch, $p_{\text{coil}} = 2(r_{\text{tube}} + t_{\text{tube}})$. The peripherally averaged Nu for fully developed laminar flow in a helical pipe with axially uniform heat flux and peripherally constant wall temperature boundary conditions is calculated using the correlation [61]:

$$\text{Nu}_D = \left[\left(\frac{48}{11} + \frac{51/11}{[1 + 1342/(\text{Pr He}^2)]^2} \right)^3 + 1.816 \left(\frac{\text{He}}{1 + 1.15/\text{Pr}} \right)^{3/2} \right]^{1/3} \quad (2.16)$$

The convection heat transfer from absorber surface segment (j, k) to fluid volume j is evaluated as

$$\dot{q}_{\text{conv,HTF},j,k}'' = h_{\text{conv,HTF},j} (T_{1,j,k} - T_{m,j}) \quad (2.17)$$

where, $T_{m,j} = (T_{\text{in},j} + T_{\text{out},j})/2$ is the bulk mean temperature of the HTF in the j^{th} fluid volume. The heat gain of the HTF in volume j is calculated by

$$\dot{q}_{\text{gain,HTF},j} = \dot{m} c_{p,j} (T_{\text{out},j} - T_{\text{in},j}) = \sum_k A_{j,k} \dot{q}_{\text{conv,HTF},j,k}'' \quad (2.18)$$

where $A_{j,k}$ is the surface area of segment (j, k) and $c_{p,j}$ the specific heat of the fluid in volume j . For the thermophysical properties of air cubic polynomial least-square fits to data from [62] are evaluated at $T_{m,j}$.

The local heat transfer coefficient from the outer window surface to ambient air is calculated for laminar free convection at a vertical plate with uniform wall heat flux [63]

$$\text{Nu}_x = \frac{0.563}{\left[1 + (0.437/\text{Pr})^{9/16}\right]^{4/9}} \text{Ra}_x^{1/4} \quad (2.19)$$

which is valid for $0 < \text{Pr} < \infty$ and $\text{Ra}_x < 10^9$. The inclination of the hot window surface with respect to the gravitational axis during solar tracking, ranging from downward-facing horizontal ($\alpha_{\text{incl}} = -90^\circ$) to vertical ($\alpha_{\text{incl}} = 0^\circ$), is considered by replacing g by its component parallel to the surface, $g \cos \alpha_{\text{incl}}$ in the Ra definition of Eq. (2.19) [64]. Air properties are evaluated at the film temperature, $T_{\text{film}} = (T_4 + T_{\text{amb}})/2$. The convective heat flux from an outer window surface segment (j, k) to ambient is calculated by

$$\dot{q}_{\text{conv},4,j,k}'' = h_{\text{conv},4,j} (T_{4,j,k} - T_{\text{amb}}) \quad (2.20)$$

Numerical solution

Solid domains and the flow in the absorber tube are divided into finite control volumes. The computational grid is visualized in Fig. A.1 and selected parameters are summarized in Table A.1 in Appendix A. The energy balances are formulated by the net energy $\dot{q}_{\text{net}}$ leaving each control volume, $\dot{q}_{\text{net}} = \sum \dot{q}_{\text{out}} - \sum \dot{q}_{\text{in}} = 0$, which results in a simultaneous system of nonlinear algebraic equations in terms of the surface and fluid temperatures. The model is implemented in a Matlab program that incorporates the Monte Carlo ray-tracing results produced by the in-house code VeGaS [65]. The numerical solution process is accelerated by the analytical expression for the Jacobian, $\mathbf{J} = \partial \dot{q}_{\text{net}} / \partial T$.

Pumping power

Assuming fully developed flow [41], the pressure drop over the length of the absorber tube is

$$\Delta p = \int_0^{L_{\text{tube}}} -\frac{\partial p}{\partial x} dx = \frac{1}{r_{\text{tube}}} \int_0^{L_{\text{tube}}} c_{f,\text{curved}} \rho U^2 dx = \frac{\dot{m}^2}{\pi^2 r_{\text{tube}}^5} \int_0^{L_{\text{tube}}} \frac{c_{f,\text{curved}}}{\rho} dx \quad (2.21)$$

where $L_{\text{tube}} = N_{\text{loop}} [(2\pi R_{\text{coil}})^2 + p_{\text{coil}}^2]^{1/2}$ is the absorber tube length, the Fanning friction coefficient of the curved pipe is calculated by Eq. (2.15), and the bulk mean velocity is replaced by $U = \dot{m} / (\pi r_{\text{tube}}^2 \rho)$ in the last step. The required pumping power $\dot{W}_{\text{pump}}$ for compression of air at temperature T_{in} on the cold side from outlet pressure p_{out} to inlet pressure $p_{\text{in}} = p_{\text{out}} + \Delta p$ is calculated assuming an ideal gas and isentropic pumping efficiency $\eta_{\text{pump, isentropic}}$

$$\dot{W}_{\text{pump}} = \frac{\dot{m}}{\eta_{\text{pump, isentropic}}} \frac{\kappa_{\text{air}} R_{\text{air}}}{\kappa_{\text{air}} - 1} T_{\text{in}} \left[1 - \left(\frac{p_{\text{out}}}{p_{\text{out}} + \Delta p} \right)^{\frac{(\kappa_{\text{air}} - 1)}{\kappa_{\text{air}}}} \right] \quad (2.22)$$

where the specific heat ratio $\kappa_{\text{air}} = 1.4$ and gas constant $R_{\text{air}} = 287 \text{ J/(kg K)}$. When operating with ambient air, as it is the case in all numerical simulations and experiments throughout this article, p_{out} and T_{in} correspond to the ambient pressure p_{amb} and temperature T_{amb} , respectively.

Theoretical performance

The optical efficiencies of the primary concentrator, secondary concentrator, and solar receiver including the window are, respectively,

$$\eta_{\text{optical, primary}} = \frac{\dot{Q}_{\text{solar, intercept}}}{\dot{Q}_{\text{solar, aperture}}} = \rho_{\text{primary}} \gamma_{\text{primary}} \quad (2.23)$$

$$\eta_{\text{optical, secondary}} = \frac{\dot{Q}_{\text{solar, incident}}}{\dot{Q}_{\text{solar, intercept}}} \quad (2.24)$$

$$\eta_{\text{optical, receiver}} = \frac{\dot{Q}_{\text{solar, absorbed}}}{\dot{Q}_{\text{solar, incident}}} \quad (2.25)$$

where $\dot{Q}_{\text{solar, intercept}}$ is the concentrated solar radiation incident on the inlet aperture area A_{in} of the secondary concentrator, and γ_{primary} is the intercept factor, i.e. the fraction of sunrays reflected by the primary concentrator that intercept A_{in} . The thermal efficiency is defined by

$$\eta_{\text{th}} = \frac{\dot{Q}_{\text{gain, HTF}}}{\dot{Q}_{\text{solar, absorbed}}} = 1 - \frac{\dot{Q}_{\text{heat losses}}}{\dot{Q}_{\text{solar, absorbed}}} \quad (2.26)$$

where $\dot{Q}_{\text{heat losses}} = \dot{Q}_{\text{reradiation}} + \dot{Q}_{\text{convection}} + \dot{Q}_{\text{conduction}}$ are the total heat losses by thermal re-radiation, natural convection, and heat conduction. The solar-to-thermal receiver efficiency is given by the product of its optical and thermal efficiencies,

$$\eta_{\text{receiver}} = \frac{\dot{Q}_{\text{gain,HTF}}}{\dot{Q}_{\text{solar,incident}}} = \eta_{\text{optical,receiver}} \eta_{\text{th}} \quad (2.27)$$

Thus, the overall collector efficiency is defined by

$$\eta_{\text{collector}} = \frac{\dot{Q}_{\text{gain,HTF}}}{\dot{Q}_{\text{solar,DNI}}} = \cos \theta_{\text{skew}} \eta_{\text{optical,primary}} \eta_{\text{optical,secondary}} \eta_{\text{receiver}} \quad (2.28)$$

where $\dot{Q}_{\text{solar,DNI}} = A_{\text{primary}} E_{\text{DNI}}$ is the available DNI on the trough aperture area at normal incidence. The specific pumping power is normalized to the total expected electric power output,

$$\varepsilon_{\text{pump}} = \frac{\dot{W}_{\text{pump}}}{\eta_{\text{th-el}} \dot{Q}_{\text{gain,HTF}}} \quad (2.29)$$

where $\eta_{\text{th-el}}$ is the thermal-to-electric conversion efficiency of the heat engine.

2.4 Experimental

The solar receiver prototype fabricated comprises an array of 7 solar absorber tubes. Its lateral cross-section is shown in Fig. 2.5a and the axial cross-section of a single absorber tube is shown in Fig. 2.5b. The helical absorber tubes are coiled AISI 316 stainless steel pipes, sandblasted and coated with a double layer of black paint (Aremco HiE-Coat 840-M, maximum operating temperature: 1093 °C) for high solar absorptivity. The linear trumpet secondary concentrator is constructed from a hyperbolic, extruded aluminum profile to which silvered aluminum sheets (Almeco vega SP298, rated specular reflectivity: 94% [66]) are attached. To prevent thermal degradation, the reflective surfaces are thermally connected to the aluminum profiles by use of thermal paste and active water-cooling is applied on the backsides. The rectangular window is made of borosilicate glass coated with a double-sided broadband AR coating [58]. The walls containing the absorber tubes and holding the window are made of stainless steel AISI 304 and are fixed to the trumpet support structure. For an enhanced

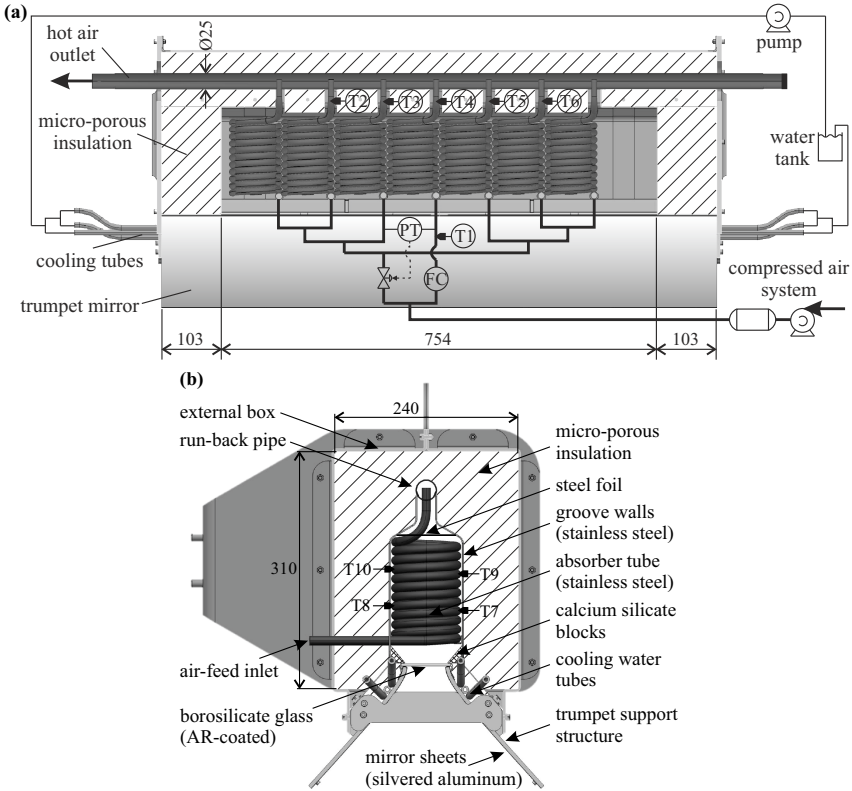


Fig. 2.5: a) Lateral cross-section of the solar receiver prototype, comprising 7 solar absorber tubes, and experimental setup. b) Axial cross-section of a single solar absorber tube.

absorptivity, the steel foil on the top is also painted black. The geometric dimensions of the solar receiver prototype are listed in Table 2.3. The entire solar receiver is insulated by a 70 mm-thick layer of Microtherm micro-porous insulation and a 20 mm-thick block of calcium silicate towards the aperture. The assembly is contained in an outer stainless steel box that is also fixed to the trumpet support structure and sustains the weight of all components. Because the inflated primary trough can only be realized with large lengths, one wing of the solar trough concentrator of 1.2 m length and 4.85 m aperture is fabricated out of four PVD coated aluminized mirror sheets (Almeco vega WR293, rated

Table 2.3: Geometric parameters of solar receiver prototype.

Absorber tube			
R_{coil}	40 mm	N_{loop}	11
r_{tube}	5 mm	t_{tube}	1 mm
Window			
a_{win}	54 mm	t_{win}	3.3 mm
Surrounding groove			
w_{groove}	94 mm	h_{groove}	144 mm
β_{groove}	45°	t_{groove}	0.5, 2, 0.1 mm

specular reflectivity: $\geq 88\%$ [66]) pressed on a steel frame having the shape of the inflated arcspline concentrator with a profile geometry given in Table 2.1. The entire system was mounted on a 2-axis tracker for the purpose of testing at normal solar incidence angle regardless of the actual position of the sun.

Ambient air is pressurized to 4 bar by a compressor unit and provided to the mass flow control system. The mass flow rate through the central absorber tube is regulated by an electronic mass flow controller (Bronkhorst F201-AV, flow range: 1.4 – 70 l_n/min, accuracy incl. linearity: $\pm 0.5\%$ Rd plus $\pm 0.1\%$ FS, control stability: $\leq \pm 0.1\%$ FS). The mass flow rates to the other six absorber tubes are controlled by a 1/4-inch proportional valve (Regtronic, linearity: $\leq \pm 0.5\%$ FS, hysteresis: $\leq \pm 0.5\%$ FS), which is regulated by a PID controller (Sensortechnics BTEL5P10D4A differential pressure transmitter, range: -10 to 10 mbar, linearity and hysteresis: $\pm 0.1\%$ FS) such that the pressure difference between the inlets to the 7 absorber tubes is zero (cf. Fig. 2.5a). Air inlet and outlet temperatures and absorber tube wall temperatures at strategic locations (cf. Fig. 2.5) are measured by K-type thermocouples. The trumpet secondary mirrors are actively cooled by a closed-loop water circuit driven by a pump (flow rate: 3 l/min). The DNI is measured by a pyrhelimeter (Kipp & Zonen CHP-1, acceptance angle: 5°, tracking error tolerance: 0.75°, linearity: $\pm 0.2\%$). The temperature distribution of the receiver window is recorded using an IR thermal image camera (NEC TH78-390, accuracy: $\pm \max(2^\circ\text{C}, 2\% \text{ Rd})$, temperature range: $-20 - 1000^\circ\text{C}$, spectral range: 8 – 14 μm).

2.5 Results

2.5.1 Numerical results

The window is a key component in the solar receiver design. Thus, two standard glass materials of industrial grade that withstand the temperatures are considered: borosilicate and quartz. Their optical properties are listed in Table 2.2. A windowless (open) solar receiver is also modeled by $\tau_{\text{win}} = \rho_{\text{win}} = 0$, $\varepsilon_{\text{win}} = 1$, and $T_3 = T_4 = T_{\text{sky}}$. The stainless steel absorber tube and groove walls are considered as black painted and oxidized respectively (cf. Table 2.2). Natural convection is neglected ($h_{\text{conv},4} = 0 \text{ W}/(\text{m}^2 \text{ K})$), as justified for a downward-facing absorber tube ($\alpha_{\text{incl}} = -90^\circ$), i.e. when the normal vector of the receiver aperture is pointing in the direction of the gravitational force such that stratification occurs and heat losses by natural convection are negligible [67]. The rectangular groove is assumed to be perfectly insulated ($k_{\text{ins}} = 0 \text{ W}/(\text{m K})$). Accordingly, only radiative losses to the surroundings occur, whose apparent sky temperature is set to ambient temperature ($T_{\text{sky}} = T_{\text{amb}}$). Simulations are performed for ambient air inlet temperature, $T_{\text{in}} = T_{\text{amb}} = 20^\circ$, and zero skew angle ($\theta_{\text{skew}} = 0^\circ$). The solar power input is adjusted such that at a certain flow rate the desired air outlet temperature is reached, and restricted to below the limit imposed by perfectly reflective primary and secondary concentrators ($\rho_{\text{primary}} = \rho_{\text{secondary}} = 1$, $\gamma_{\text{primary}} = 1$) and maximum DNI of $1050 \text{ W}/\text{m}^2$.

The thermal and solar receiver efficiencies as a function of mass flow rate are shown in Fig. 2.6 at three different operating temperatures. Mass flow is indicated in l_n/min , i.e. in liters of air under normal conditions ($p_n = 101325 \text{ Pa}$, $T_n = 0^\circ \text{ C}$, $\rho_n = 1.293 \text{ kg}/\text{m}^3$) per minute. $\eta_{\text{optical,receiver}}$ is 90.9%, 91.3%, and 98.2% for borosilicate, quartz glass, and a windowless receiver, respectively, the latter of which representing the apparent solar absorptance of the cavity receiver. The glass windows effectively reduce re-radiation losses significantly, thus yielding 4 – 17% higher η_{th} compared to that of the windowless case. The windowed receivers also outperform the windowless receiver in terms of η_{receiver} , except for low operating temperatures and high flow rates. Due to the shorter cut-off wavelength of borosilicate ($2.7 \mu\text{m}$) compared to quartz ($3.6 \mu\text{m}$), the former is a better thermal radiative shield and yields up to 2% higher η_{receiver} than that of

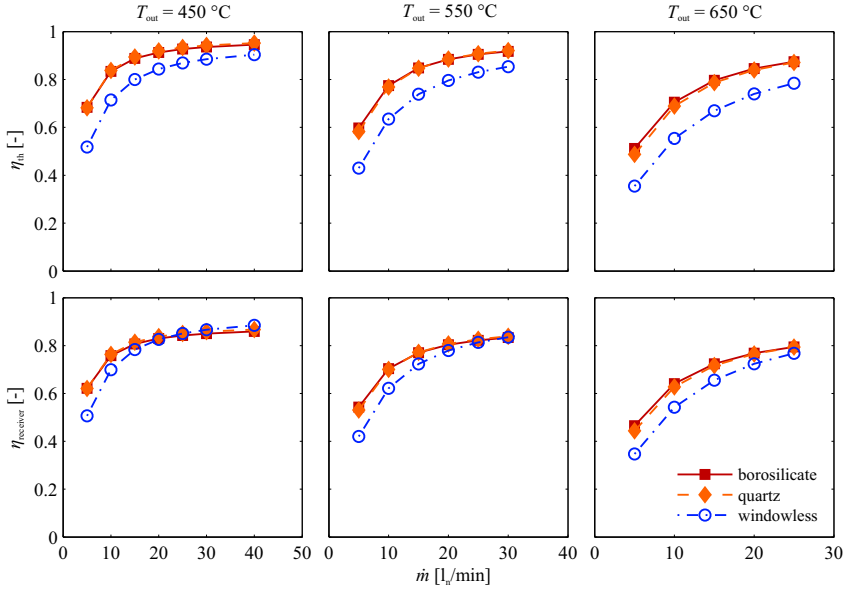


Fig. 2.6: Thermal and solar receiver efficiencies as a function of flow rate and operating temperature for borosilicate (solid, squares) and quartz (dashed, diamonds) windows, and for a windowless receiver (dash-dotted, circles).

the latter at high operating temperatures and low flow rates, in spite of the lower solar transmittance and higher absorptance of borosilicate with respect to quartz (cf. Table 2.2). Note also that borosilicate is usually less expensive than pure fused silica.

Fig. 2.7 a and b show, respectively, the thermal radiative flux incident on the rectangular aperture and the temperature of the external window surface T_4 (or the sky temperature for the windowless case) as a function of the air mass flow rate, assuming the same operating conditions as in Fig. 2.6. For comparison, the blackbody hemispherical total emissive power, $\dot{q}''_{\text{emitted,b}} = \sigma T_b^4$, at various temperatures T_b is also indicated in Fig. 2.7a. The radiative flux emitted by the absorber tube through the rectangular aperture is similar among the windowed and windowless receivers, and thus the absorber tube temperature is mainly determined by the operating temperature and mass flow rate. Further, $\dot{q}''_{\text{incident,3}}$ is

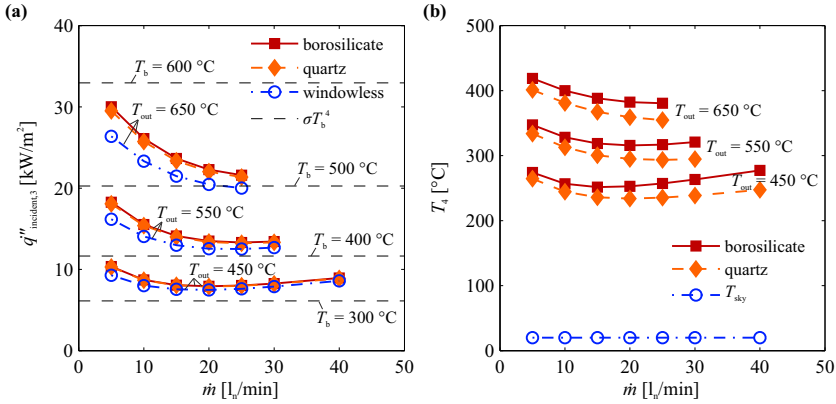


Fig. 2.7: a) Thermal radiative flux emitted by the absorber tube through the rectangular aperture; and b) temperature of the external window surface as a function of mass flow rate at different operating temperatures.

substantially lower than the blackbody hemispherical total emissive flux at temperature T_{out} , which results from feeding the cold air to the helical tube near the rectangular aperture such that a positive temperature gradient develops axially along the absorber tube. This effect increases with mass flow rate due to improved cooling of the absorber tube. The temperature of the external window surface is mainly a function of the air outlet temperature. Because of absorption in the near IR, the borosilicate window reaches a higher temperature than the quartz window, but the difference is less than 31°C . The low thermal conductivity of glass (e.g. $k_{400} = 1.84 \text{ W/(m K)}$ for fused silica, and $k_{400} = 1.66 \text{ W/(m K)}$ for borosilicate) causes temperature differences from 5 to 20°C between the inner and outer glass surfaces of the 3.3 mm thick window.

The pressure drop and the corresponding specific pumping power demand as a function of mass flow rate at three operating temperatures ($T_{\text{out}} = 450, 550, 650^\circ\text{C}$) are shown in Fig. 2.8 a and b, respectively. The pressure drop increases with operating temperature due to the decreasing density and increasing viscosity of the gas. However, since hotter air also has a higher enthalpy, the specific pumping power demand in percentage of the expected electrical output —

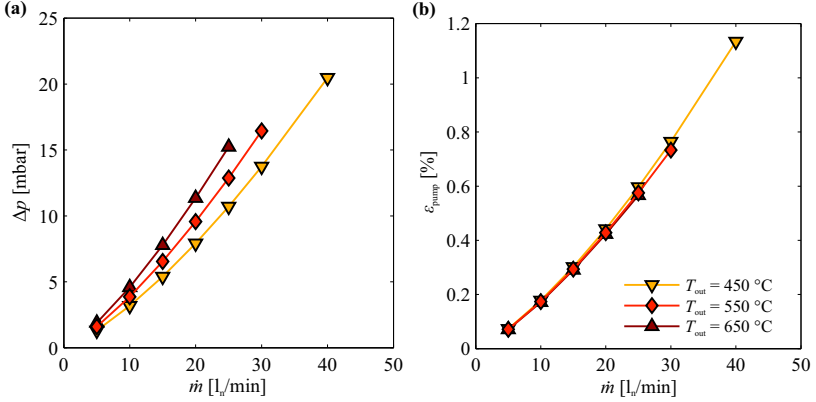


Fig. 2.8: a) Pressure drop and b) specific pumping power demand as a function of mass flow rate at three operating temperatures.

assuming $\eta_{pump, isentropic} = 95\%$ and $\eta_{th-el} = 35\%$ — actually decreases with increasing operating temperature. The pressure drop remains below 20 mbar and, consequently, the specific pumping power demand does not exceed 1% of the expected electric power output. This is the case under all operating conditions, except for $T_{out} = 450$ °C and $\dot{m} = 40$ L_n/min for which $\Delta p = 20.5$ mbar, $\varepsilon_{pump} = 1.1\%$.

In practice, the solar receiver is not perfectly insulated and the effect of the thermal conductivity of the insulation needs to be considered. Fig. 2.9 shows η_{th} for a solar receiver with a borosilicate window and a certain layer of insulation around the rectangular groove wall (surface #6) with the conduction heat transfer coefficient $h_{cond, ins} = k_{ins} / t_{ins}$ varying from 0 to 8 W/(m² K). The results are shown for different flow rates (10, 15, 25 L_n/min) and operating temperatures (450, 550, 650 °C). For comparison, $h_{cond, ins}$ obtained with a 50 mm thick layer of micro-porous insulation (Microtherm, $k_{400} = 0.026$ W/(m K) [68]), earth-alkali silicate wool (Promaglaf, $k_{400} = 0.1$ W/(m K) [69]), and calcium silicate blocks (Monalite, $k_{400} = 0.25$ W/(m K) [69]) are also indicated.

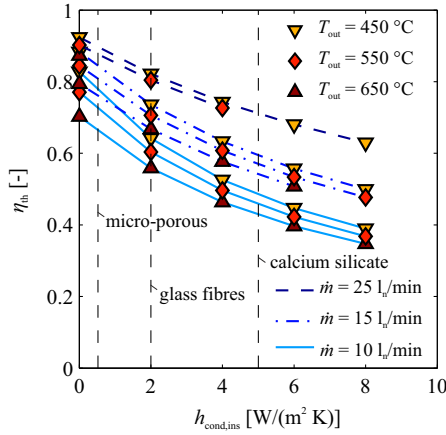


Fig. 2.9: Thermal efficiency as function of the conduction heat transfer coefficient of insulation at different operating conditions.

2.5.2 Model validation

A typical experimental run over time is shown in Fig. 2.10. The measured DNI is above 900 W/m^2 over a large part of the experiment and peaks at 945 W/m^2 . With a constant air inlet temperature of $20 \text{ }^\circ\text{C}$, the steady-state air outlet temperature from the central coiled absorber tube (T4) reached 542 and $610 \text{ }^\circ\text{C}$ at mass flow rates of 15 and $10 \text{ l}_w/\text{min}$, respectively. The air outlet temperatures from the neighboring absorber tubes (signals T3 and T5) followed closely the outlet temperature from the central absorber tube ($\pm 20 \text{ }^\circ\text{C}$ under steady-state conditions), thus indicating that the central absorber tube is well insulated from end effects. Since the model introduces periodic boundary conditions in the y -direction, the central absorber tube was used as the representative one in the analysis that follows. The relevant operating conditions of the central coiled absorber tube under approximate steady-state conditions, which were used for the model validation, are summarized in Table 2.4.

The primary trough concentrator was characterized by solar flux measurements using Gardon type flux gauges in combination with a CCD camera to record the flux map on a Lambertian target. The solar power input was obtained by integrating the solar flux distribution incident at the secondary

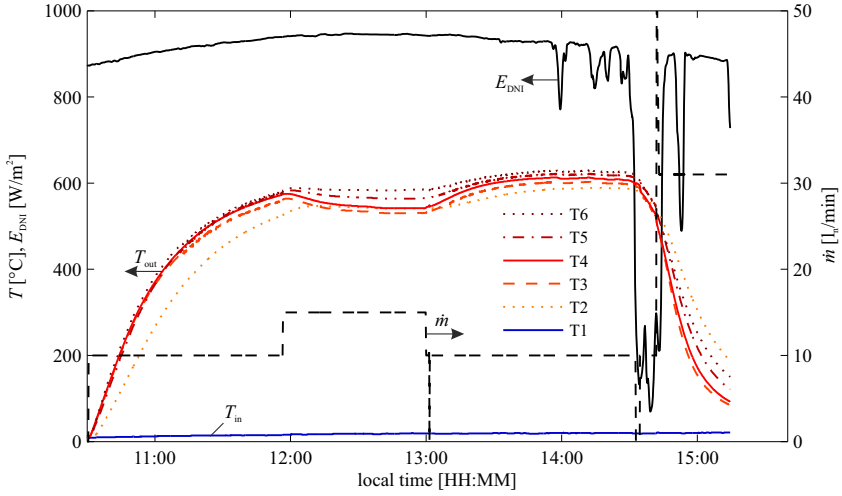


Fig. 2.10: Typical experimental run showing the measured air inlet (T1) and outlet temperatures (T2–T6) and DNI on the left ordinate, and the applied mass flow rate through the central absorber tube on the right ordinate as a function of time.

Table 2.4: DNI, flow rate, and operating temperatures of absorber tube under approximate steady-state conditions in ascending order of air outlet temperature.

Exp. run	E_{DNI} [W/m^2]	$\dot{m}$ [l_n/min]	T_{in} [$^{\circ}\text{C}$]	T_{out} [$^{\circ}\text{C}$]	Comments
1	620	25	11	264	fluctuating DNI
2	949	25	11	367	wind ~ 10 m/s
3	977	20	10	401	wind ~ 10 m/s
4	921	20	9	449	
5	832	15	10	512	
6	945	15	19	542	
7	892	10	12	581	
8	890	10	20	609	
9	843	5	15	621	

concentrator's inlet plane. The model parameters of the primary concentrator (ρ_{primary} , σ_s) were fitted to match the calculated and experimentally measured solar power input to the receiver. The procedure is described in Appendix Appendix B. The best agreement was obtained with $\rho_{\text{primary}} = 0.83$, $\sigma_s = 3.5$ mrad,

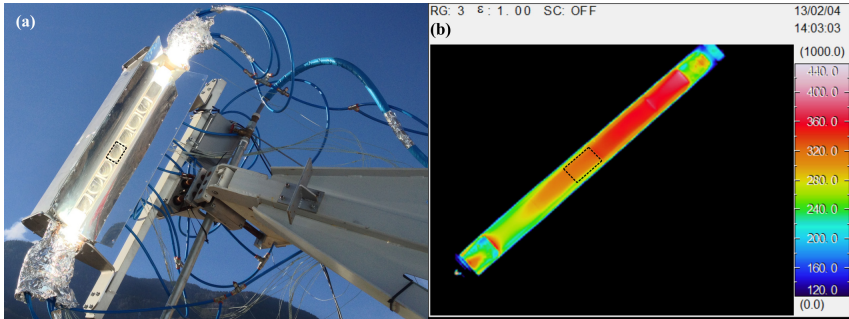


Fig. 2.11: Photograph (a) and thermal image (b) of window measured by IR thermography during experiment 7. The temperature scale is in °C. The window aperture in front of the central absorber tube is indicated by the black dashed rectangle.

which yields $\gamma_{\text{primary}} = 92\%$ and $\eta_{\text{optical,primary}} = 77\%$. The experimentally determined reflectivity value of 83% as compared to the rated value (88%) is attributed to outdoor exposure and soiling of the primary concentrator. For the secondary concentrator, the rated specular reflectivity $\rho_{\text{secondary}} = 0.94$ and no surface scattering was used. The measured optical properties of the AR-coated window are summarized in Table 2.2. These data were utilized in the MC ray-tracing simulation, which yielded $\eta_{\text{optical,secondary}} = 97\%$ and $\eta_{\text{optical,receiver}} = 92\%$. In combination with the DNI measurements during testing, the solar power input to the receiver was determined in each steady-state experiment.

Since the model predictions indicate that the absorber tube temperature, and thus the thermal radiative flux incident on the inner window surface, window temperature, and heat losses through the windowed aperture are mainly a function of operating temperature and mass flow rate, this correlation is analyzed in the experiments. The temperature distribution on the absorber tube was measured at four locations by thermocouples T7 to T10 (cf. Fig. 2.5b). In addition, the outer window temperature was captured using IR thermography. A photograph of the solar receiver during tracking and a thermal image of the same scene are shown in Fig. 2.11 a and b, respectively. The region of interest containing the central absorber tube is indicated by the black dashed rectangle. All thermal images were corrected for the actual emissivity of the glass [49]. The average emittance of the glass over the spectral range of the camera (8 – 14 μm)

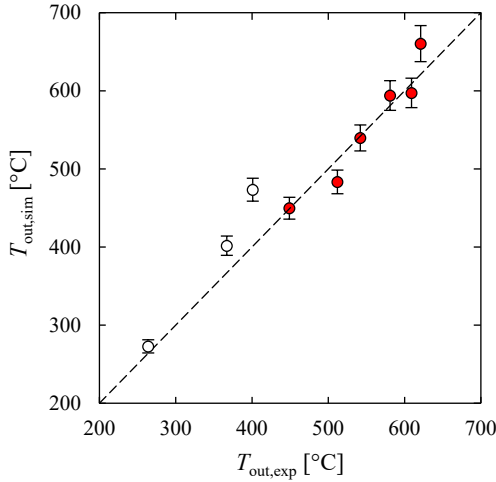


Fig. 2.12: Parity plot of the air outlet temperatures: numerically simulated vs. experimentally measured.

was calculated as 0.88. In the following graphs, the data points obtained from the experiments with transient DNI and wind (runs 1–3 in Table 2.4) are represented by empty markers.

Fig. 2.12 shows a parity plot of the numerically simulated vs. the experimentally measured air outlet temperatures. The largest deviations are obtained for the windy experiments (+72 °C), while all other data points agree within ± 40 °C. An effective conduction heat transfer coefficient of the insulation, $h_{cond,ins,eff} = 3.7 \text{ W}/(\text{m}^2 \text{ K})$, is applied in the numerical simulation to reproduce the air outlet temperature experimentally measured, which, for an insulation thickness $t_{ins} = 70 \text{ mm}$, yielded an average effective thermal conductivity of $0.26 \text{ W}/(\text{m K})$. This value is roughly an order of magnitude greater than the rated one by the manufacturer (Microtherm, $k_{400} = 0.026 \text{ W}/(\text{m K})$). This discrepancy is explained by additional heat sinks that were present but not captured by the model, namely: 1) radiation spillage caused by misalignment and tracking errors of the primary concentrator; 2) heat losses by thermal bridges from the hot groove walls to the water-cooled secondary optics; 3) release of hot air at the top due to chimney effect; 4) forced convection by wind. While these additional losses affect the energy balance, their significance is secondary and may either

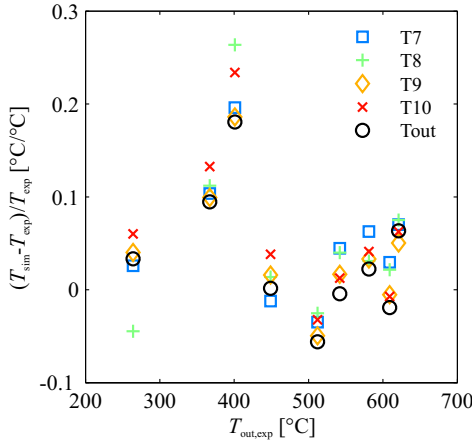


Fig. 2.13: Relative difference between numerically simulated and experimentally measured absorber tube temperatures (T7–T10) and air outlet temperature (Tout) as a function of measured air outlet temperature.

be reduced or eliminated. The main heat losses are dominated by re-radiation and natural convection through the windowed aperture.

The comparison of the numerically simulated temperature distribution on the helical absorber tube and the experimentally measured temperatures by thermocouples T7 to T10 is shown in Fig. 2.13 by plotting the relative difference between simulated and measured values, $(T_{sim} - T_{exp})/T_{exp}$, as a function of measured air outlet temperature $T_{out,exp}$. The relative deviation of the simulated air outlet temperature $T_{out,sim}$ from $T_{out,exp}$ is also plotted. The percentage deviation from all measured temperatures lies within $\pm 7.5\%$ except for the windy experiments (runs 2–3 in Table 2.4), for which the maximum percentage error amounts to 26.4%. The correlation between the average temperature of the window's external glass surface, which essentially determines the heat losses through the windowed aperture, and the air outlet temperature, $T_{win,ext}$ vs. T_{out} , predicted by the model and measured in the experiments, is shown in Fig. 2.14. An agreement within 20 °C is obtained for outlet temperatures > 440 °C, whereas the discrepancy is significantly larger for those experiments affected by wind or transient DNI (runs 1–3 in Table 2.4).

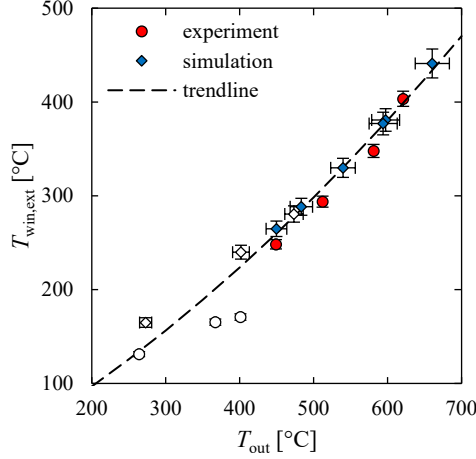


Fig. 2.14: Experimentally measured temperature of the window's external glass surface as a function measured air outlet temperature (circles), numerically simulated external window temperature as a function of simulated outlet temperature (diamonds), and polynomial trendline (dashed).

2.5.3 Experimental receiver performance

The experimentally measured η_{receiver} and $\eta_{\text{collector}}$ are shown in Fig. 2.15 a and b, respectively. All error bars are calculated from the accuracy of the measurement equipment using a first order Taylor series expansion. Considering the filled markers only, a maximum $\eta_{\text{receiver}} = 64\%$ at $T_{\text{out,exp}} = 449\text{ }^{\circ}\text{C}$ is achieved with $\dot{m} = 20\text{ l}_n/\text{min}$. A maximum $T_{\text{out,exp}} = 621\text{ }^{\circ}\text{C}$ with $\eta_{\text{receiver}} = 25\%$ is obtained with $\dot{m} = 5\text{ l}_n/\text{min}$. At $T_{\text{out,exp}} = 542\text{ }^{\circ}\text{C}$ and $\dot{m} = 15\text{ l}_n/\text{min}$, the measured efficiency still amounts to $\eta_{\text{receiver}} = 56\%$. The thermal power $\dot{Q}_{\text{gain,HTF}}$ delivered by the central absorber tube varies from 68.7 W at a mass flow rate of 5 l_n/min to 196 W at 20 l_n/min. The maximum pressure drop predicted by the model for the steady-state experimental conditions amounts to 10.1 mbar, and the specific pumping power assuming efficiencies of 95% and 35% for isentropic pumping and thermal-to-electric conversion, respectively, remains below 1% of the expected electricity generation in all cases.

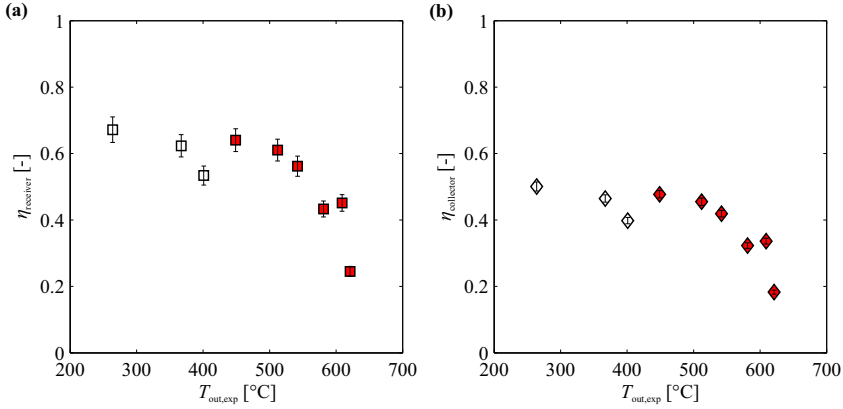


Fig. 2.15: Experimentally demonstrated a) receiver efficiencies and b) overall collector efficiencies as a function of measured outlet temperature. Empty markers represent experiments with fluctuating DNI and wind.

The different loss mechanisms quantified by the heat transfer model in percentage of the available DNI are plotted in Fig. 2.16 as a function of the numerically simulated air outlet temperature for the conditions $T_{in} = T_{amb} = T_{sky} = 20\text{ }^{\circ}\text{C}$, $E_{DNI} = 1000\text{ W/m}^2$, and $\theta_{skew} = 0^{\circ}$. Optical losses by the primary concentrator, secondary concentrator, and window amount to 23.3%, 2.2%, and 5.8%, respectively. The heat losses through the windowed aperture due to natural convection (2.1 – 4.2%) and re-radiation (3.3 – 14.3%) vary from 5.3% at 400 $^{\circ}\text{C}$ to 18.5% at 650 $^{\circ}\text{C}$. Conduction losses through the insulation vary between 11.6% and 20.1%, and $\eta_{collector}$ ranges from 51.9% at an operating temperature of 400 $^{\circ}\text{C}$ to 30.2% at 650 $^{\circ}\text{C}$.

2.6 Conclusion

An entirely novel solar receiver for parabolic trough concentrators using air as HTF at operating temperatures exceeding 600 $^{\circ}\text{C}$ was designed, fabricated, tested, and numerically simulated. Model validation is accomplished by comparison to experimental results in terms of the temperature distributions on the absorber tube and the external surface of the window. On-sun experiments were performed with a 1 m-long solar receiver prototype comprising 7 coiled

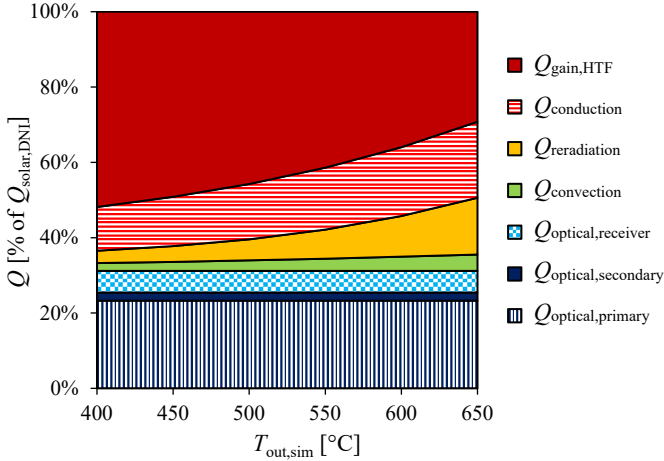


Fig. 2.16: Area plot of the losses in percentage of available DNI, plotted as a function of $T_{\text{out,sim}}$ for $T_{\text{in}} = T_{\text{amb}} = T_{\text{sky}} = 20^{\circ}\text{C}$, $E_{\text{DNI}} = 1000 \text{ W/m}^2$, and $\theta_{\text{skew}} = 0^{\circ}$.

absorber tubes. With feeding rates in the range of $5 - 20 \text{ l}_n/\text{min}$ to each absorber tube, air outlet temperatures of $621 - 449^{\circ}\text{C}$ and receiver efficiencies of $25 - 64\%$ were achieved, while the pressure drop and pumping power demand remained below 10 mbar and 1% of the expected electric power output, respectively. Major heat losses occur at the primary mirror by absorption and spillage of sunlight, at the window by solar reflection, thermal re-radiation, and natural convection, and by other loss mechanisms including conduction through the insulation and chimney effect.

3 Parametric study

In this chapter, the validated heat transfer model of Chapter 2 is applied to a parameter study of the new receiver design based on an array of coiled absorber tubes. From the geometric parameters that define the geometry of the helically coiled tubes, meaningful dimensionless numbers are defined. Numerical simulations are performed for various values of DNI and skew angle. The aim is to identify trends and peculiarities of the new solar trough receiver design and to find good parameters in terms of annual mean efficiency, material use and costs.

3.1 Design parameters

The geometry of the helically coiled absorber tube is defined by four parameters, the tube's outer diameter $D_{\text{tube,o}}$ and wall thickness t_{tube} , the coil's centerline diameter D_{coil} , and the number of loops N_{loop} . With these parameters two meaningful dimensionless parameters are defined: the coil-to-tube radius ratio

$$R_{\text{coil-tube,i/o}} = \frac{D_{\text{coil}}}{D_{\text{tube,i/o}}} \quad (3.1)$$

and the height-to-diameter aspect ratio (AR) of the coiled absorber tube,

$$AR = \frac{h_{\text{coil}}}{D_{\text{coil,i}}} = \frac{D_{\text{tube,o}} N_{\text{loop}}}{D_{\text{coil}} - D_{\text{tube,o}}} = \frac{N_{\text{loop}}}{R_{\text{coil-tube,o}} - 1} \quad (3.2)$$

$R_{\text{coil-tube}}$ governs the friction factor and Nu of flow inside the coiled tube due to its appearance in the definition of De. In addition, manufacturability constraints impose a lower limit on $R_{\text{coil-tube}}$ depending on the tube material and wall thickness, and the pipe bending method. AR governs the thermal re-radiation from the coiled absorber tube due its appearance in the view factor expression from the hot top surface to the colder window, and determines the penetration depth of solar radiation into the absorber cavity. Leaving spaces between

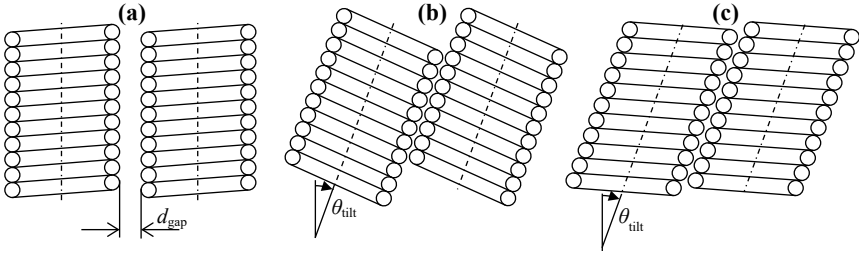


Fig. 3.1: Longitudinal section of array of absorber tubes with gaps (a), tilted absorber tubes (b), and skewed absorber tubes (c).

neighbored absorber tubes as shown in Fig. 3.1a introduces an additional parameter, the gap distance d_{gap} , from which the gap-to-coil ratio is defined as

$$R_{\text{gap-coil}} = \frac{d_{\text{gap}}}{D_{\text{coil}} + D_{\text{tube,o}}} = \frac{d_{\text{gap}} / D_{\text{tube,o}}}{R_{\text{coil-tube,o}} + 1} \quad (3.3)$$

Introducing gaps decreases the number of coils per unit receiver length

$$n'_{\text{coil}} = \frac{1}{D_{\text{coil}} + D_{\text{tube,o}} + d_{\text{gap}}} = \frac{1 / D_{\text{tube,o}}}{(1 + R_{\text{coil-tube,o}})(1 + R_{\text{gap-coil}})} \quad (3.4)$$

which reduces fabrication costs and accelerates assembly. A further design consideration is shown in Fig. 3.1b, where the coiled absorber tubes are tilted by an angle θ_{tilt} to better match the asymmetric distribution of skew angles in N-S oriented PTCs. Tilted absorber tubes require more space in the longitudinal and vertical directions, which reduces the number of coils by a factor $n'_{\text{coil,tilted}} = n'_{\text{coil}} \cos \theta_{\text{tilt}}$. A design that is as compact as the non-tilted absorber tubes and allows for skew angle matching is obtained with skewed absorber tubes, shown in Fig. 3.1c. The centerline length of the coiled absorber tube is given by

$$L_{\text{tube}} = N_{\text{loop}} \left[(\pi D_{\text{coil}})^2 + p_{\text{coil}}^2 \right]^{1/2} = N_{\text{loop}} \pi D_{\text{coil}} \left[\frac{(\pi D_{\text{coil}})^2}{(\pi D_{\text{coil}})^2 - D_{\text{tube,o}}^2} \right]^{1/2} \quad (3.5)$$

where $p_{\text{coil}} = D_{\text{tube,o}} / [1 - (D_{\text{tube,o}} / (\pi D_{\text{coil}}))^2]^{1/2}$ is the coil's pitch. The total material use for the array of coiled absorber tubes per unit axial length is

$$m'_{\text{coil}} = 2\rho_{\text{steel}} n'_{\text{coil}} \pi D_{\text{tube}} L_{\text{tube}} t_{\text{tube}} \quad (3.6)$$

Table 3.1: Baseline geometric design parameters.

Geometric dimensions		Dimensionless numbers	
t_{tube}	0.5 mm	$R_{\text{coil-tube,i}}$	7.27
$D_{\text{tube,o}}$	12 mm	$R_{\text{coil-tube,o}}$	6.67
D_{coil}	80 mm	AR	1.94
M_{loop}	11	$R_{\text{gap-coil}}$	0
d_{gap}	0 mm	n'_{coil}	10.9 m^{-1}
L_{tube}	2.77 m		
m'_{coil}	9.07 kg/m		

where the factor 2 is due to the two focal lines of the solar collector system. The baseline geometric dimensions and dimensionless numbers are given in Table 3.1 and correspond to the parameters used for the pilot plant in Morocco described in the next chapter. The only difference to the receiver prototype geometry tested in Chapter 2 is the reduction of the tube wall thickness from 1.0 to 0.5 mm. Variation of geometric parameters generally alters the pressure drop and pumping power required to circulate the air through the array of coiled absorbers tubes. This is considered by the overall efficiency defined as

$$\eta_{\text{overall}} = \frac{\dot{Q}_{\text{gain,HTF}} - \dot{W}_{\text{pump}} / \eta_{\text{th-el}}}{\dot{Q}_{\text{solar,DNI}}} = \eta_{\text{collector}} (1 - \varepsilon_{\text{pump}}) \quad (3.7)$$

which is used for comparison of receiver designs with different pressure drops.

3.2 Methods

Solar resource

The available solar irradiance at a solar collector depends on numerous factors such as the geographical location and orientation of the collector, daytime, season, and meteorological conditions. For one-axis tracking parabolic trough collectors the solar irradiance on the concentrator's aperture is described by two variables: the instantaneous DNI and skew angle. θ_{skew} is deterministic and is calculated from the zenith and azimuth angles (θ_z , ϕ_s) of the sun, and the azimuth

and tilt angles (ϕ , β) of the solar collector. Generally, the solar incidence angle on a solar collector tilted by an angle β is given by [70, 71]

$$\cos \theta = \cos \theta_z \cos \beta + \sin \theta_z \sin \beta \cos(\phi_s - \phi) \quad (3.8)$$

In one axis tracking solar trough collectors, the angle β is continuously adjusted to maximize Eq. (3.8) such that the vectors pointing to the sun and in the direction of the tracking axis, and the normal vector of the concentrator's aperture lie in the same plane, which yields for the tracking angle

$$\tan \beta_{\text{track}} = \tan \theta_z \cos(\phi_s - \phi) \quad (3.9)$$

Subbing Eq. (3.9) into (3.8) and rearranging yields for the skew angle

$$\sin \theta_{\text{skew}} = \sin \theta_z \sin(\phi_s - \phi) \quad (3.10)$$

It is noted that for ϕ_s measured from the N-S axis, $\phi = 0^\circ$ for E-W and $\phi = -90^\circ$ for N-S oriented tracking axes. The solar angles at a certain location at any time of the year may be calculated from the latitude, solar declination, and hour angle [70, 71]. DNI, on the other hand, is a stochastic variable that depends on the instantaneous optical thickness of the earth's atmosphere. Hottel [72] derived an empirical correlation for the atmosphere's transmittance of DNI under clear sky conditions based on the solar zenith angle and the location's altitude. Real DNI data has been recorded by several national and international weather station networks [73, 74]. In addition, commercial software are available that create stochastic sub-hourly time series from measured time-averaged data and perform spatial interpolation based on satellite images [75, 76]. In this thesis, measured DNI data from the Baseline Surface Radiation Network (BSRN) [74] are used for the calculation of the annual power output of the solar collector. BSRN stations record the DNI in time intervals of 1 minute and thus allow for capturing the temporal fluctuations of the power output due to nonlinear system response, thermal inertia, etc. The distribution of yearly solar irradiance incident on the aperture of a PTC with the tracking axis oriented N-S (a) and E-W (b) over skew angle and DNI is shown in Fig. 3.2. Commercial PTCs are usually oriented N-S because of lower cosine losses, although E-W oriented collectors show other advantages such as less seasonal variations, repetitive skew angles every day, and smaller tracking angles and shading losses, offering the potential for more compact solar fields.

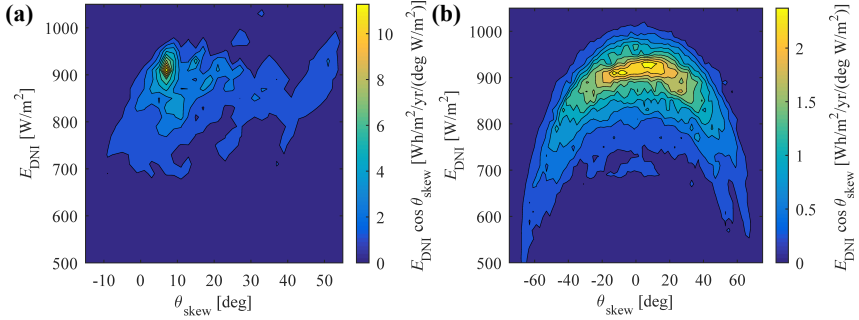


Fig. 3.2: Contour plots of yearly DNI per unit DNI per unit skew angle incident on a solar parabolic trough's aperture with the tracking axis oriented N-S (a) and E-W (b) for Sede Boqer, 2011.

Simulation parameters

Simulations are performed for Sede Boqer, Israel (30.860° N, 34.779° E, altitude: 500 m), representing a typical arid place in the desert with a yearly available DNI of $E_{\text{DNI,yr}} = \int_{\text{yr}} E_{\text{DNI}}(t) dt = 2400 \text{ kWh/m}^2/\text{yr}$. The solar trough collector is oriented N-S and tracks from -70° to 70° . The solar concentration system described in Chapter 2 is simulated using slightly different optical material properties. Most importantly, a semi-transparent topsheet with a solar-weighted normal transmittance $T_{\text{normal,solar}} = 0.92$ is introduced, which protects the primary concentrator and solar receiver from environmental impacts (e.g. wind, dust). The solar-weighted specular reflectance $R_{\text{specular,solar}}$ of the primary and secondary concentrators is set to 0.92 and 0.94, respectively, and $T_{\text{normal,solar}}$ of the AR-coated receiver window is 0.95. A comprehensive assessment of optical properties of materials for solar concentrators is included in Chapter 7. The coiled absorber tubes are assumed as perfectly insulated ($k_{\text{ins}} = 0 \text{ W/(m K)}$) because the study aims at assessing the effects of the absorber tube's geometry on the inevitable heat losses through the windowed aperture while heat losses through the thermal insulation mainly depend on its technical implementation. T_{in} and T_{out} of the air at the coiled absorber tubes are set to 240°C and 580°C ,

respectively, envisioning solar-thermal power generation via a Rankine cycle steam turbine. The air inlet and outlet temperatures at the heat recovery steam generators identified by a feasibility study¹ are 570 °C and 250 °C, respectively, thus considering a temperature drop of 10 °C due to transmission losses in the feeding and collecting pipes. The ambient temperature in the enclosed solar collector is set to $T_{\text{amb}} = T_{\text{sky}} = 50$ °C. Numerical simulations are performed for E_{DNI} and θ_{skew} in the range 100 – 1100 W/m² and 0 – 60° in intervals of 100 W/m² and 10°, respectively, spanning a 2D grid over the occurring solar irradiance conditions. The instantaneous power rates at the actual solar conditions are calculated by bilinear interpolation of the simulation results over E_{DNI} and θ_{skew} . Annual mean efficiencies are calculated according to the definitions of their instantaneous equivalents with the instantaneous powers replaced by their integrals over the year

$$Q_{\text{yr}} = \int_{\text{yr}} \dot{Q}(t) dt \quad (3.11)$$

3.3 Results

Optical efficiency

The optical efficiency of the complete solar collector system comprising the transparent topsheet, primary concentrator, secondary concentrator, receiver window, and coiled absorber tube is defined as

$$\eta_{\text{optical}} = \frac{\dot{Q}_{\text{solar,absorbed}}}{\dot{Q}_{\text{solar,DNI}}} = \cos \theta_{\text{skew}} \eta_{\text{primary}} \eta_{\text{secondary}} \eta_{\text{receiver}} \quad (3.12)$$

At normal solar incidence, $\eta_{\text{optical}}(\theta_{\text{skew}} = 0) = \eta_{\text{optical},0} = 0.773$ for the baseline geometry. The angular dependence is shown in Fig. 3.3 by the incidence angle modifier (IAM) defined as

$$\text{IAM}(\theta_{\text{skew}}) = \frac{\eta_{\text{optical}}(\theta_{\text{skew}})}{\eta_{\text{optical},0}} \quad (3.13)$$

¹ Feasibility Study Report, “50 MWel CSP Project,” technical report on behalf of Airlight Energy SA, 13.12.2013.

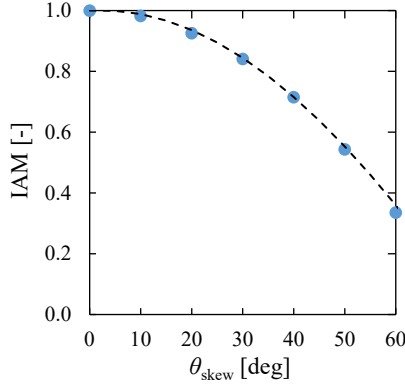


Fig. 3.3: Incidence angle modifiers simulated by MC ray-tracing (markers) and computed from empirical correlation [77] (dashed line) as a function of skew angle.

For comparison, the empirical IAM derived from test results of the SEGS LS-2 solar PTC is plotted [77]

$$\text{IAM}(\theta_{\text{skew}}) = \cos \theta_{\text{skew}} + 8.84 \cdot 10^{-4} \theta_{\text{skew}} - 5.369 \cdot 10^{-5} \theta_{\text{skew}}^2 \quad (3.14)$$

showing a close resemblance. Both $\eta_{\text{optical},0}$ and IAM are nearly independent of the geometry of the coiled absorber tubes due to the cavity effect [40].

Thermal efficiency

The heat losses of the baseline coiled absorber tube through the windowed aperture by thermal re-radiation and natural convection are shown in Fig. 3.4 as a function of DNI and skew angle. The increasing heat losses towards low DNI values are due to less effective cooling of the absorber tube's inlet region close to the window at lower mass flow rates (cf. Fig. 2.7). The same effect occurs for decreasing solar power with increasing skew angles. More importantly, the penetration depth of concentrated solar radiation into the coiled absorber tubes decreases with increasing skew angle such that the hottest region of the absorber tube moves closer to the window. It is noted that the mismatch between the line-focus solar radiation and 3D coiled absorber tubes originates from their invention in the context of line-to-point focusing secondary concentrators [14, 78], which have later been replaced by line-focus secondary concentrators for practical reasons.

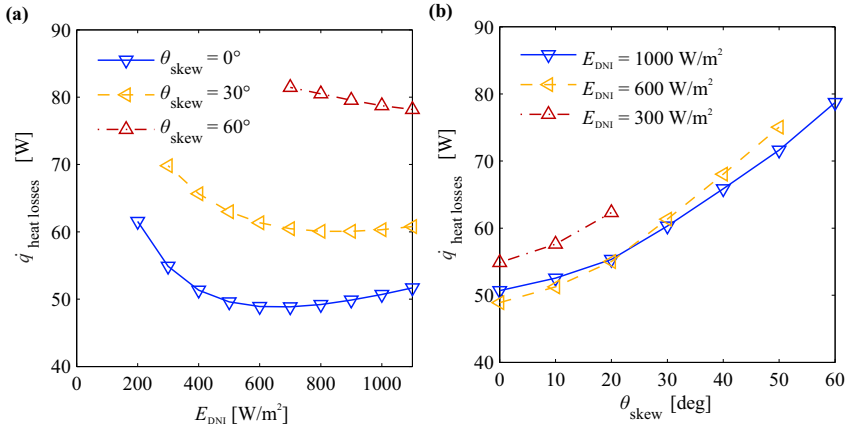


Fig. 3.4: Heat losses of coiled absorber tube (baseline geometry) as a function of DNI (a) and skew angle (b).

The thermal efficiency defined in Eq. (2.26) is shown as a function of skew angle in Fig. 3.5 for various absorber tube geometries. The blue solid lines with downward-facing triangular markers represent the baseline geometry and the varied parameters are indicated in the graphs. Designs with smaller AR show a better performance at oblique angles but suffer from higher losses at near-normal incidence (Fig. 3.5a). A similar effect may be observed if gaps are introduced between the absorber tubes (Fig. 3.5b). Tilting effectively shifts the thermal performance maximum from 0° towards θ_{tilt} , where the skewed absorber tubes generally outperform the tilted coils (Fig. 3.5 c and d).

Annual mean overall efficiency

The annual mean overall efficiencies are plotted in Fig. 3.6 as a function of AR (a), $R_{\text{coil-tube}}$ (b), $R_{\text{gap-coil}}$ (c), and θ_{tilt} of the tilted and skewed absorber tubes (d). The constant geometric parameters are indicated in the graphs. The aspect ratio of the baseline geometry ($\text{AR} = 1.94$) shows a good compromise between effective heat transfer at normal incidence and inactive absorber tube surface at large θ_{skew} . The performance optimum for the coil-to-tube radius ratio ($R_{\text{coil-tube}} = 6.67$) is explained by two competing effects with decreasing $R_{\text{coil-tube}}$: 1) enhanced convection heat transfer coefficient due to enforced secondary flow in coiled

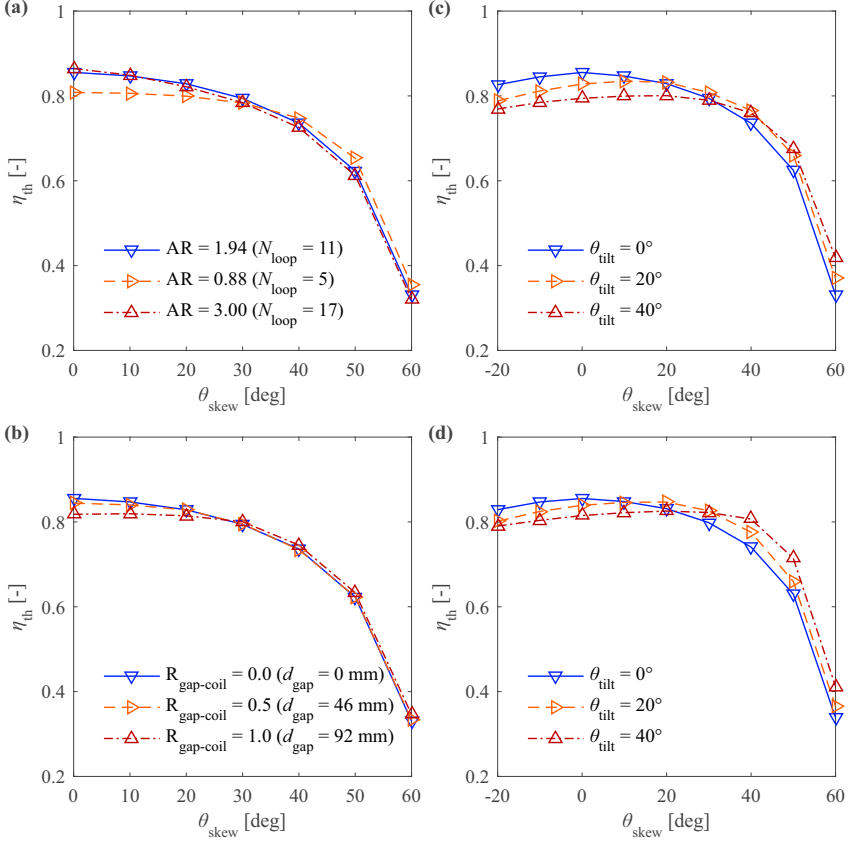


Fig. 3.5: Thermal efficiency as a function of θ_{skew} at $E_{DNI} = 1000 \text{ W/m}^2$ for different aspect ratios (a), gap-to-coil ratios (b), and tilt angles of the tilted (c) and skewed (d) absorber tubes.

absorber tubes; and 2) increasing absorption of solar radiation at the lowermost loop of the absorber tube close to the windowed aperture. $\eta_{overall, yr}$ is stable for small gap-to-coil ratios, thus offering the potential for reducing the number of absorber tubes by at least 10%. Furthermore, the baseline $\eta_{overall, yr}$ can be increased by 0.5% (absolute) by skewing the absorber tubes by 20° .

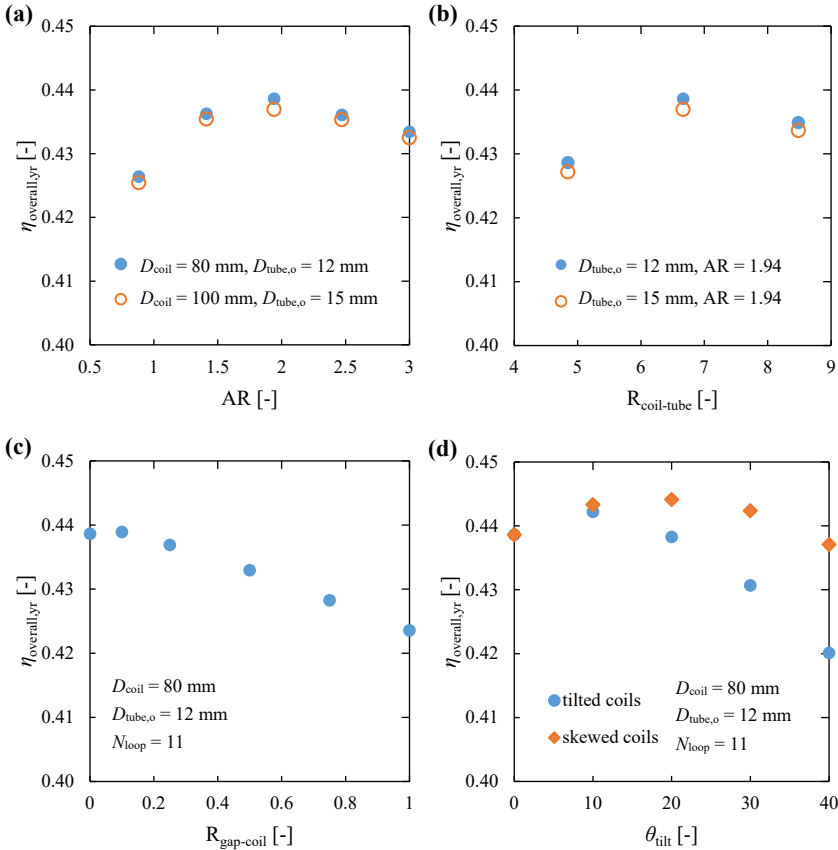


Fig. 3.6: Annual mean overall efficiency for Sede Boqer (2011) as a function of AR (a), $R_{\text{coil-tube}}$ (b), $R_{\text{gap-coil}}$ (c), and θ_{tilt} (d).

Pressure drop and operating temperature

Pressure drop and pumping power of the array of coiled absorber tubes may be controlled by scaling their coil and tube diameters while keeping the dimensionless numbers constant. The resulting $\eta_{\text{overall,yr}}$ as a function of the annual specific pumping power is shown in Fig. 3.7, indicating a flat optimum. The effect of operating temperature on $\eta_{\text{overall,yr}}$ is shown in Fig. 3.8. For air outlet temperatures in the range 420 – 660 °C, the annual mean overall efficiency varies from 0.49 to 0.37.

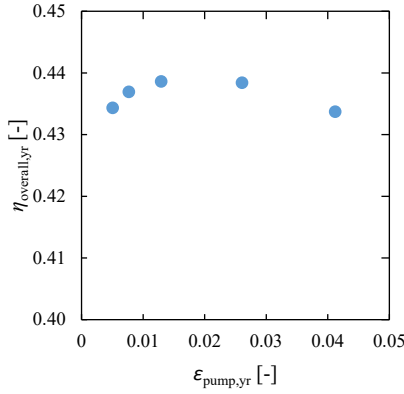


Fig. 3.7: Annual mean overall efficiency as a function of specific pumping power for $N_{\text{loop}} = 11$, $\text{AR} = 1.94$, $R_{\text{coil-tube,o}} = 6.67$, $R_{\text{gap-coil}} = 0$, and $D_{\text{tube,o}} = 7.5, 9.0, 12, 15, 18$ mm.

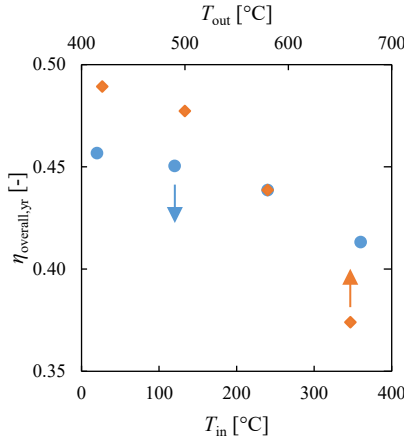


Fig. 3.8: Annual mean overall efficiency for baseline geometry of coiled tubes with variable inlet temperature at $T_{\text{out}} = 580$ °C (circles), and variable outlet temperature at $T_{\text{in}} = 240$ °C (diamonds).

Material use

Fig. 3.9 shows $\eta_{\text{overall,yr}}$ of various absorber tube geometries as a function of the material used for their fabrication from 0.5 mm-thick pipes (stainless steel, $\rho_{\text{steel}} =$

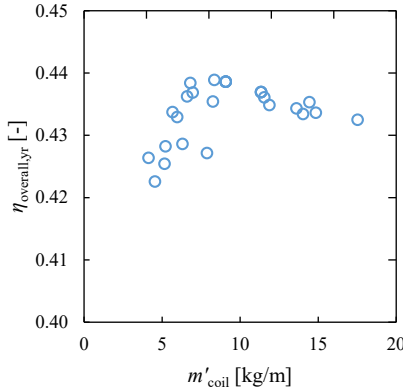


Fig. 3.9: Annual mean overall efficiency vs. material use for absorber tubes ($t_{\text{tube}} = 0.5 \text{ mm}$) per meter length of receiver with two focal lines.

8000 kg/m^3). While a lack of material seems prohibitive, excess material seems to be counterproductive, yielding a sweet spot near the baseline geometry.

3.4 Conclusion

A parametric study of the array of coiled absorber tubes was performed and characteristic dimensionless numbers were identified. Heat losses through the windowed aperture show a significant skew angle dependence due to the 3D nature of the coiled absorber tubes. This effect can be partially counteracted by tilting (skewing) the absorber tubes towards the skew rays of solar radiation. The robustness of the annual mean overall efficiency to the absorber tube geometry allows tuning for other design criteria such as material use and cost, e.g. by introducing gaps between the coils.

4 Industrial scale demonstration¹

After the successful proof of concept with air temperatures above 600 °C measured for the first time on a parabolic trough [79], the coiled tube cross-flow receiver concept was integrated with a pneumatic solar trough concentrator based on a stack of polymeric films supported by a concrete structure [13]. To experimentally demonstrate the complete solar concentrating system on an industrial scale, a 3.6 MW_{th} pilot plant comprised of three 212 m-long solar trough collectors and integrated with a thermocline-based pebble stone thermal energy storage (TES) [80, 81] has been constructed in Ait Baha, Morocco [82]. In this chapter, the engineering details of the design are presented. A heat transfer model is formulated for the solar concentrator and solar receiver, which is validated by comparison to experimental results obtained after the commissioning of the first solar collector unit. A detailed energy balance is presented, based on which improvement potentials are identified. The model is further applied for yearly performance predictions of the improved design during regular operation.

4.1 Design and modelling

The heat transfer analysis of an array of helically coiled absorber tubes is presented in detail in Chapter 2. In this chapter, the 3D model is extended to simulate the concentrating optics and energy transfer of the complete solar collector unit comprising the solar concentrator and the solar receiver, including the axial integration over the solar receiver's length along with calculations of pressure drop and mass flow distribution.

¹ Material in this chapter has been published in P. Good, G. Ambrosetti, A. Pedretti, and A. Steinfeld, "A 1.2 MW_{th} solar parabolic trough system based on air as heat transfer fluid at 500 °C — Engineering design, modelling, construction, and testing," *Solar Energy*, vol. 139, pp. 398-411, 2016.

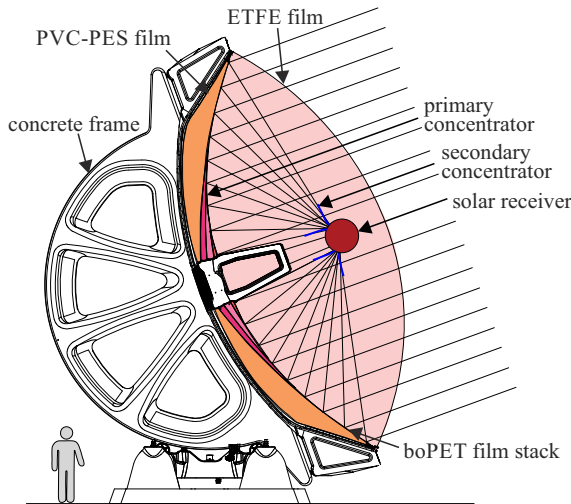


Fig. 4.1: Schematic cross-section of solar parabolic trough collector comprising a pneumatic trough concentrator based on a stack of polyester films whose topmost is aluminized, a semi-transparent ETFE at the top, a PVC-coated polyester foil at the bottom, and solar receiver, all mounted on a concrete support structure.

4.1.1 Solar concentrating optics

A schematic of the solar parabolic trough concentrator is shown in Fig. 4.1. Each of the two mirror halves is constructed from a multi-layer stack of bi-axially oriented polyester films (boPET) whose topmost film is aluminized in order to obtain a reflective surface. The mirror foils and solar receiver are mounted on a rigid concrete support structure and are protected from external impacts (e.g. wind, dust) by an inflated membrane envelope, consisting of a semi-transparent ethylene tetrafluoroethylene (ETFE) foil at the top and a polyvinyl chloride coated polyester (PVC-PES) film at the bottom [13, 82].

The solar concentrating optics encompasses an imaging primary concentrator in tandem with a non-imaging secondary concentrator [79]. The primary concentration is achieved by a one-axis tracking arcspline concentrator (ASC) [13, 14, 83] composed of two wing reflectors, each consisting of four tangentially adjacent circular arc segments. The secondary concentration is obtained by two 2D truncated trumpet secondary concentrators, one for each wing of the primary concentrator. The symmetric trumpets are accommodated in

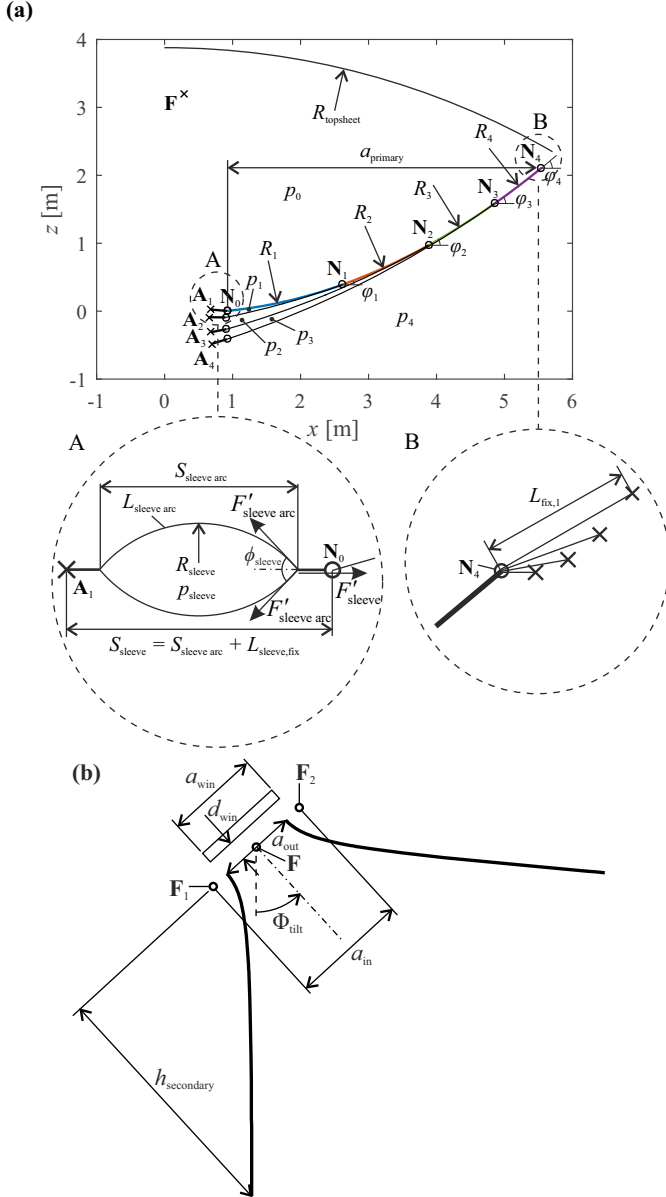


Fig. 4.2: One wing of the arcspine concentrator constructed from four films (a) and hyperbolic secondary concentrator with windowed aperture of solar receiver (b). Details of film fixation at the inner rim point with inflatable sleeve (A) and at the outer rim point with variable fixing lengths (B) are shown.

Table 4.1: Geometric parameters of optical components: semi-transparent topsheet, nominal arcspline concentrator, trumpet secondary concentrator, and solar receiver window.

Semi-transparent topsheet				
radius R_{topsheet}	10 m	thickness t_{topsheet}	100 μm	
Primary concentrator				
baseline dimensions		index j	radius R_j	slope angle φ_j
focus F	(0.292, 3.195) m	1	7.001 m	19.97°
outer rim N ₄	(5.545, 2.109) m	2	8.649 m	29.20°
aperture width a_{primary}	4.625 m	3	10.763 m	35.32°
outer rim angle Φ_{o}	78.32°	4	12.907 m	39.18°
Secondary concentrator				
inlet aperture a_{in}	73.7 mm	height $h_{\text{secondary}}$	160 mm	
outlet aperture a_{out}	50.0 mm	tilt angle Φ_{tilt}	42.5°	
Solar receiver window				
aperture a_{win}	55.0 mm	thickness t_{win}	3.3 mm	
focus distance d_{win}	3.0 mm			

tandem with the primary concentrator by separating the focal lines of the two primary wings and titling their focal planes outwards by an angle Φ_{tilt} [79]. The theoretical geometric concentration ratios of the primary and secondary concentrators are $C_{\text{g,primary}} = a_{\text{primary}} / a_{\text{in}} = 62.8$ and $C_{\text{g,secondary}} = a_{\text{in}} / a_{\text{out}} = 1.473$, respectively, yielding a total concentration ratio of $C_{\text{g}} = a_{\text{primary}} / a_{\text{out}} = 92.5$. A schematic of one wing of the solar concentrator is shown in Fig. 4.2. The geometric parameters of the ASC, in our case four radii R_j and nodal slopes φ_j , are determined using a direct optical approach that guarantees maximum theoretical concentration with a minimum number of arcs [14, 83]. Geometric parameters of all optical components are summarized in Table 4.1. The geometry of the ASC is mechanically determined by applying gradually decreasing pressures p_1 to p_4 from the top to the bottom chambers, where p_0 is fixed and corresponds to the inflation pressure of the collector [13, 14]. The manufacturing tolerances can be partially compensated by adjusting the four pressures p_j [13]. To render the arcspline shape completely controllable, pneumatic sleeves are introduced at the inner ends of the mirror films that allow for individual foil

length regulation by increasing and releasing the pressure in the respective sleeve.

4.1.2 Solar air receiver

The conceptual solar receiver design was introduced in Chapter 2. Air at ambient pressure is used as the HTF, which has roughly 1000 times lower density and 10 times lower thermal conductivity compared to commonly used HTFs such as thermal oil, molten salt, or water/steam. This is accounted for by the cross-flow design of the solar receiver based on an array of helically coiled absorber tubes. A lateral view, cross-section, and 3D exploded view of the engineered solar receiver are shown in Fig. 4.3. Concentrated solar radiation delivered by the secondary concentrators enters through the windowed apertures and is absorbed by the array of helical tube absorbers. Cold air is supplied to each helical tube by the manifold feeding pipe via T-branches and flexible hoses, heated to the desired operating temperature, and released into the manifold collecting pipe leading to the TES or user. The window, which has an anti-reflection (AR) coating to minimize optical losses, reduces effectively heat losses from the absorber tube by natural convection and IR thermal emission [79]. The array of coiled absorber tubes and the hot air collecting pipe are lined by a first layer of thermal insulation. On the collecting pipe, a second layer of lightweight, low- c_p insulation is provided by multiple radiation shields [84]. The trumpet secondary concentrators are actively refrigerated and thermally insulated from the hot diverging groove walls by solid insulation blocks, which support also the window. The air inlet and outlet to the feeding and collecting axial manifold pipes are located at the same end of the collector. To compensate for pressure drops, adjustable knife gate valves are incorporated along the length of the pipes to establish uniform flow distribution through the absorber tubes. A cross-section of one coiled absorber tube is shown schematically in Fig. 4.4. Indicated are the geometric parameters and a selection of temperatures and heat fluxes. The geometric parameters are listed in Table 4.2. Further constructive details are given in Section 4.2.2.

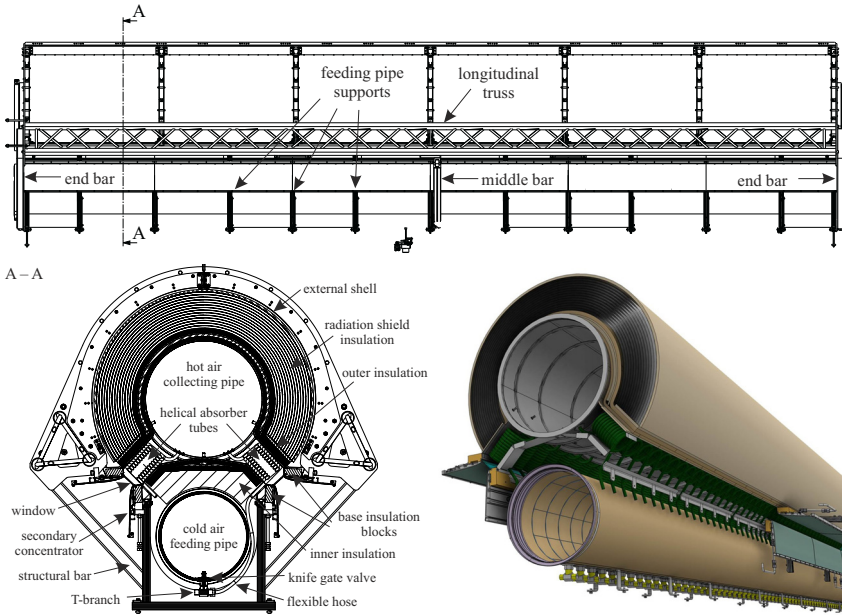


Fig. 4.3: Solar air receiver based on an array of coiled absorber tubes in cross-flow configuration. Lateral view (top), cross-section (left), and 3D CAD model (right) of 5.88 m-long receiver section.

4.1.3 Array of helical absorber tubes

Radiation

The solar flux distribution on each absorber tube is determined by Monte Carlo (MC) ray-tracing using experimentally measured spectrally and directionally dependent optical properties for the semi-transparent topsheet, primary concentrator, secondary concentrator, and receiver window [85, 86]. Further details about the material properties are given in Section 4.2. The spectral and directional characteristics of the sun are modelled by the ASTM G173 – 03 (AM 1.5) reference spectrum for direct and circumsolar irradiance [46] and the Buie sunshape model with a circumsolar ratio of 5% [87]. The optical efficiency of the primary solar concentrator is defined as the solar power input to the receiver

$$\frac{\dot{Q}_{\text{solar, input}}}{\dot{Q}_{\text{solar, aperture}}} = A_{\text{primary}} E_{\text{DNI}} \cos \theta_{\text{skew}},$$

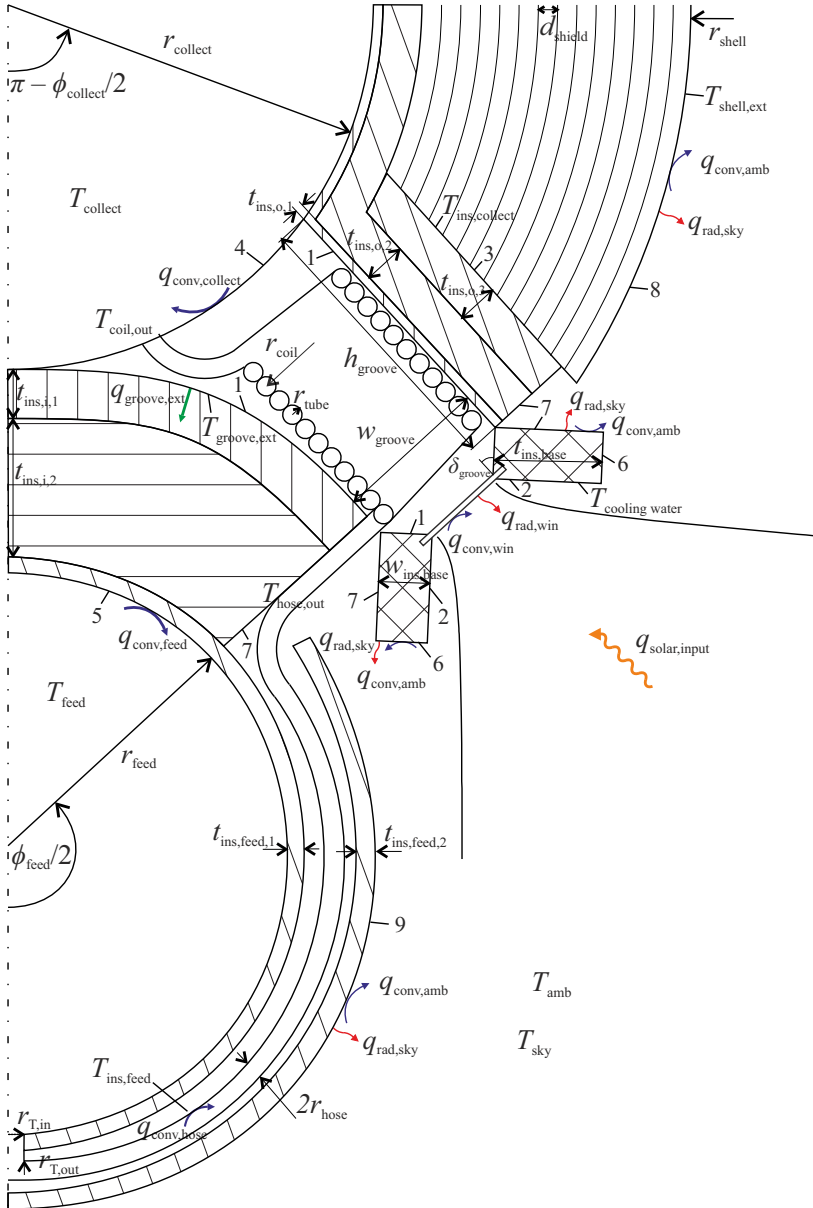


Fig. 4.4: Schematic cross-section of one coiled absorber tube with geometric parameters, surface numbering, and selected temperatures and heat fluxes.

Table 4.2: Geometric parameters of solar receiver. Spatial dimensions are given in mm, angles (greek letters) in degrees, and numbers are dimensionless (N) or per receiver section (n).

Array of absorber tubes				Manifold pipes			
absorber tube		outer insulation		feeding pipe		collecting pipe	
L_{tube}	2962	$t_{\text{ins,o},1}$	6	ϕ_{feed}	259	ϕ_{collect}	221
$r_{\text{i,tube}}$	5.5	$t_{\text{ins,o},2}$	25	$r_{\text{i,feed}}$	166.5	$r_{\text{i,collect}}$	226.5
t_{tube}	0.5	$t_{\text{ins,o},3}$	25	$r_{\text{o,feed}}$	175	$r_{\text{o,collect}}$	227
N_{loop}	11	inner insulation		$t_{\text{ins,feed},1}$	10	$t_{\text{ins,collect},1}$	6
r_{coil}	40	$t_{\text{ins,i},1}$	39	$t_{\text{ins,feed},2}$	10	$t_{\text{ins,collect},2}$	25
n_{coil}	64	$t_{\text{ins,i},2}$	89	metallic support		radiation shields	
surrounding groove		base insulation		$r_{\text{i,fin}}$	215	N_{shield}	14
w_{groove}	94	$w_{\text{ins,base}}$	31.8	$r_{\text{o,fin}}$	275	d_{shield}	11.8
h_{groove}	180	$t_{\text{ins,base}}$	67.1	t_{fin}	4	t_{shield}	0.05
t_{groove}	0.5	structural bar		n_{fin}	13		
β_{groove}	45	l_{receiver}	5880	T-branch		external shell	
flexible hose		h_{bar}	50	$r_{\text{T,in}}$	7.75	$r_{\text{o,shell}}$	435
$r_{\text{i,hose}}$	6.75	h_{bar}	50	$r_{\text{T,out}}$	6.1	t_{shell}	0.5
L_{hose}	410	n_{bar}	3				

$$\eta_{\text{optical,concentrator}} = \frac{\dot{Q}_{\text{solar,input}}}{\dot{Q}_{\text{solar,aperture}}} = \tau_{\text{topsheet}} \rho_{\text{primary}} \gamma_{\text{primary}} \quad (4.1)$$

τ_{topsheet} is the solar-weighted transmittance of the semi-transparent topsheet, ρ_{primary} the specular reflectance of the primary concentrator, and γ_{primary} the intercept factor of the primary concentrator, i.e. the fraction of reflected solar radiation that intercepts the inlet aperture of the secondary concentrator. The optical efficiency of the solar receiver is defined as the ratio of absorbed solar power by the receiver to its solar power input

$$\eta_{\text{optical,receiver}} = \frac{\dot{Q}_{\text{solar,absorbed}}}{\dot{Q}_{\text{solar,input}}} = (1 - \varepsilon_{\text{blocking,bars}}) \eta_{\text{optical,secondary}} \gamma_{\text{secondary}} \alpha_{\text{receiver}} \quad (4.2)$$

$\varepsilon_{\text{blocking,bars}}$ is the fraction of concentrated solar radiation intercepting the inlet aperture of the secondary concentrator that is blocked by the water-cooled bars, $\eta_{\text{optical,secondary}}$ the optical efficiency of the non-imaging secondary concentrator, $\gamma_{\text{secondary}}$ the fraction of solar radiation leaving through the secondary concentrator's outlet that is incident on the receiver's windowed aperture, and α_{receiver} the absorptance of the receiver. Blocking by the water-cooled bar in the middle of each receiver module and by the two adjacent bars at the modules' ends is determined as

$$\begin{aligned} l_{\text{blocking,middle}} &= l_{\text{bar}} + h_{\text{bar}} \tan \theta_{\text{bar}} \\ l_{\text{blocking,end}} &= 2l_{\text{bar}} + h_{\text{bar}} \tan \theta_{\text{bar}} \\ \varepsilon_{\text{blocking,bars}} &= \frac{l_{\text{blocking,middle}} + l_{\text{blocking,end}}}{l_{\text{receiver}}} \end{aligned} \quad (4.3)$$

Here, l_{receiver} and l_{bar} are the axial lengths of the receiver module and bar in trough direction, respectively, h_{bar} is the bar's profile height, and θ_{bar} the projected incidence angle of concentrated solar radiation on the bars in the plane perpendicular to the tilted focal plane. θ_{bar} is related to the incidence or skew angle θ_{skew} of solar radiation on the primary concentrator's aperture by $\tan \theta_{\text{bar}} = \tan \theta_{\text{skew}} / \cos(2\varphi - \Phi_{\text{tilt}})$.

Conduction

Energy conservation in the solid domains of thermal insulation surrounding the helical absorber tube is governed by the 2D steady-state heat conduction equation:

$$\nabla \cdot (k \nabla T) = 0 \quad (4.4)$$

with three different boundary conditions: fixed wall temperature (surfaces 1–3), forced convection (surfaces 4–5), and combined natural convection and thermal emission (surfaces 6–7).

$$\begin{aligned} T_{s,i} &= T_i \text{ for } i = 1, 2, 3 \\ -k \nabla T|_{s,i} \cdot \hat{\mathbf{n}}_{s,i} &= h_i (T_{s,i} - T_i) \text{ for } i = 4, 5 \\ -k \nabla T|_{s,i} \cdot \hat{\mathbf{n}}_{s,i} &= h_{\text{amb},i} (T_{s,i} - T_{\text{amb}}) + \varepsilon_i \sigma (T_{s,i}^4 - T_{\text{sky}}^4) \text{ for } i = 6, 7 \end{aligned} \quad (4.5)$$

In Eq. (4.5), the subscript s,i denotes evaluation at surface i with normal vector $\hat{\mathbf{n}}_{s,i}$, h is the convection heat transfer coefficient, ε the surface emissivity and σ the Stefan-Boltzmann constant. The fixed wall temperatures are $T_1 = T_{\text{groove,ext}}$, $T_2 = T_{\text{cooling water}}$, $T_3 = T_{\text{ins,collect}}$, and forced convection occurs to the HTF in the feeding and collecting manifold pipes at temperatures $T_4 = T_{\text{collect}}$ and $T_5 = T_{\text{feed}}$, respectively.

Convection

Flow in the manifold pipes is mostly turbulent ($\text{Re} \gg 2300$) and becomes laminar towards the opposite ends to the inlet/outlet of the solar receiver. Forced convection is calculated for fully developed turbulent flow [88],

$$\text{Nu}_D = \frac{(f/8)(\text{Re}_D - 1000)\text{Pr}}{1 + 12.7(f/8)^{1/2}(\text{Pr}^{2/3} - 1)} \quad (2300 < \text{Re}_D < 5 \times 10^6) \quad (4.6)$$

where the Darcy friction factor f is supplied via Eq. (4.30). In the laminar region, the analytical solution for fully developed laminar flow and axially uniform wall heat flux is used [41],

$$\text{Nu}_D = 4.364 \quad (\text{Re}_D < 2300) \quad (4.7)$$

The transitional flow regime ($2300 < \text{Re}_D < 3100$) occurs only in a very short region of the manifold pipes, where the Gnielinski correlation yields conservative estimates of heat losses [89]. Natural convection from the external insulation surfaces (6–7) is usually laminar ($\text{Ra} \ll 10^9$) and is calculated for each surface based on its length L , inclination angle from the vertical α_{incl} , and average surface temperature. For downward facing horizontal surfaces ($\alpha_{\text{incl}} = -90^\circ$), $\text{Nu}_L = 0.27\text{Ra}_L^{1/4}$ ($10^5 \leq \text{Ra}_L \leq 10^{10}$) [41]. For $-90^\circ < \alpha_{\text{incl}} \leq 60^\circ$, the average Nu is determined by a correlation for laminar convection from a vertical plate with uniform wall heat flux [90],

$$\text{Nu}_L = 0.68 + \frac{0.670\text{Ra}_L^{1/4}}{\left[1 + (0.437/\text{Pr})^{9/16}\right]^{4/9}} \quad (\text{Ra}_L \leq 10^9) \quad (4.8)$$

where the gravitational acceleration g in the definition of Ra is replaced by its component parallel to the surface $g \cos \alpha_{\text{incl}}$. For strongly upward facing surfaces

($\alpha_{\text{incl}} > 60^\circ$), $\text{Nu}_L = 0.54 \text{Ra}_L^{1/4}$ ($10^4 \leq \text{Ra}_L \leq 10^7$) [91]. Heat fluxes in the thermal insulation and coiled absorber tube are coupled via the surface wall temperature $T_1 = T_{\text{groove,ext}}$ and heat flux $\dot{q}''_{\text{groove,ext}} = k \nabla T|_{\text{s},1} \cdot \hat{\mathbf{n}}_{\text{s},1}$ of the external groove surface. Temperature-dependent thermal conductivity data are used for the insulating materials.

Numerical solution

The governing equations are discretized according to finite control volumes and reformulated as a linear system of equations in terms of the grid point temperatures [45]. Rectangular grids are used for the base and outer groove insulation, whereas a cylindrical grid is applied to the inner insulation. Interface conductivities are calculated by the harmonic mean. The numerical solution is found by solving iteratively until the root-mean-square (RMS) error of the initial and final temperature distribution is below 0.01 K.

4.1.4 Manifold pipes

Heat losses of the manifold pipes are analyzed by 1D axisymmetric heat transfer models of the feeding and collecting pipes coupling radiation, conduction and convection. Eq. (4.4) written in cylindrical coordinates and considering radial gradients only yields,

$$\frac{d}{dr} \left(rk \frac{dT}{dr} \right) = 0 \quad (4.9)$$

with boundary condition

$$-\phi rk \frac{dT}{dr} \Big|_{\text{s}} = \dot{q}'_{\text{rad}} + \dot{q}'_{\text{conv}} \quad (4.10)$$

where ϕ is the sector angle of the manifold pipe sector. Integrating Eq. (4.9) twice over r yields for the conduction heat transfer through an annular sector with inner and outer radii r_i and r_o , respectively,

$$\dot{q}'_{\text{cond}} = \frac{\phi k (T_i - T_o)}{\ln(r_o / r_i)} \quad (4.11)$$

Thermal radiative exchange between the shields of the collecting pipe insulation is calculated by the radiosity method for diffuse concentric cylinders [54]

$$\dot{q}'_{\text{rad,shields}} = \frac{\phi_{\text{collect}} r_i \sigma (T_i^4 - T_o^4)}{\frac{1}{\varepsilon_i} + \left(\frac{1}{\varepsilon_o} - 1 \right) \frac{r_i}{r_o}} \quad (4.12)$$

Radiation shields are assumed diffuse because of their corrugation to increase stiffness and facilitate construction. Their apparent emissivity is given by:

$$\varepsilon_{\text{app}} = \frac{\varepsilon}{\varepsilon \left(1 - \frac{A_{\text{smooth}}}{A_{\text{corrugated}}} \right) + \frac{A_{\text{smooth}}}{A_{\text{corrugated}}}} \quad (4.13)$$

where A_{smooth} and $A_{\text{corrugated}}$ are the surface areas of the smooth and corrugated shields, respectively. Heat transfer by natural convection between shields is calculated using an effective thermal conductivity for the annulus [92]

$$\frac{k_{\text{eff}}}{k} = 0.386 \left(\frac{\text{Pr}}{0.861 + \text{Pr}} \right)^{1/4} \text{Ra}_{L_c}^{1/4} \quad (4.14)$$

where the modified length scale L_c in the definition of Ra is [41]

$$L_c = 2 \left(\frac{[\ln(r_o / r_i)]^4}{(r_i^{-3/5} + r_o^{-3/5})^5} \right)^{1/3} \quad (4.15)$$

For $k_{\text{eff}} / k \leq 1$, heat transfer occurs by pure conduction ($k_{\text{eff}} = k$), which is desirable for thermal insulation applications and was considered in the design of the multi-shield insulation. Natural convection from the external surfaces (8–9) to ambient air is calculated by the Nu correlation for a horizontal cylinder [93],

$$\text{Nu}_D = \left[0.6 + 0.387 \left(\frac{\text{Ra}_D}{\left[1 + (0.599 / \text{Pr})^{9/16} \right]^{16/9}} \right)^{1/6} \right]^2 \quad (4.16)$$

where Ra and Nu are based on the external diameter of the insulation. Thermal emission from the external surface to the surroundings is calculated as

$$\dot{q}'_{\text{rad,sky}} = \phi_{\text{collect}} r_{o,\text{shell}} \varepsilon \sigma (T_{\text{shell,ext}}^4 - T_{\text{sky}}^4) \quad (4.17)$$

and solar radiation transmitted by the topsheet that is incident on the external surface of the collecting pipe insulation is considered as

$$\dot{q}'_{\text{solar,shell}} = 2r_{\text{o,shell}} \varepsilon \tau_{\text{topsheet}} E_{\text{DNI}} \cos \theta_{\text{skew}} \quad (4.18)$$

At the feeding pipe, additional heat losses occur through the metallic plates that support the feeding pipe and act as annular fins. Considering the problem as steady-state 1D axisymmetric, and assuming that the thickness t_{fin} and thermal conductivity k of the fin are constant, the convective heat transfer coefficient h is uniform over the surface, radiation from the surface is negligible, and the outer edge of the fin (tip) is adiabatic, the closed form solution for the heat transfer rate from an annular fin with $m^2 \equiv 2h / (kt_{\text{fin}})$ is [41]

$$\dot{q}_{\text{fin}} = \phi_{\text{feed}} r_{\text{i,fin}} t_{\text{fin}} k m (T_{\text{i}} - T_{\text{amb}}) \frac{K_1(mr_{\text{i,fin}}) I_1(mr_{\text{o,fin}}) - I_1(mr_{\text{i,fin}}) K_1(mr_{\text{o,fin}})}{K_0(mr_{\text{i,fin}}) I_1(mr_{\text{o,fin}}) - I_0(mr_{\text{i,fin}}) K_1(mr_{\text{o,fin}})} \quad (4.19)$$

Here, $r_{\text{i,fin}}$ and $r_{\text{o,fin}}$ are the inner and outer radii and T_{i} the base temperature of the fin, and I_j and K_j are modified j^{th} order Bessel functions of the first and second kinds, respectively. $\dot{q}_{\text{fin}}$ are integrated into the heat transfer model of the feeding pipe insulation as heat sink between the internal and external insulation layers, T_{i} is set to the temperature $T_{\text{feed,ins}}$ between the two layers, and h is calculated by Eq. (4.8) for a vertical plate with length scale $L = r_{\text{o,fin}} - r_{\text{i,fin}}$. Heat transfer by forced convection from the flexible hoses to the adjacent thermal insulation is considered by

$$\dot{q}_{\text{hose}} = h A_{\text{hose}} \Delta T_{\text{lm}} = \dot{m}_{\text{coil}} c_p (T_{\text{hose,in}} - T_{\text{hose,out}}) \quad (4.20)$$

where the convection heat transfer coefficient is calculated by Eq. (4.7), $A_{\text{hose}} = 2\pi r_{\text{i,hose}} L_{\text{hose}}$ is the heat transfer area, $\Delta T_{\text{lm}} = (\Delta T_{\text{in}} - \Delta T_{\text{out}}) / \ln(\Delta T_{\text{in}} / \Delta T_{\text{out}})$ the log-mean temperature difference between the tube wall and HTF, and $\dot{m}_{\text{coil}}$ the mass flow rate through the hose and coiled absorber tube downstream. Assuming that the tube wall temperature is constant and equal to $T_{\text{ins,feed}}$ of the surrounding insulation, Eq. (4.20) can be solved for the HTF outlet temperature from the hose,

$$T_{\text{hose,out}} = T_{\text{ins,feed}} + (T_{\text{hose,in}} - T_{\text{ins,feed}}) \exp\left(-\frac{h A_{\text{hose}}}{\dot{m}_{\text{coil}} c_p}\right) \quad (4.21)$$

$\dot{q}_{\text{hose}}$ are integrated in the feeding pipe model as heat source between the internal and external thermal insulation layers.

The governing mass and energy conservation equations for the HTF flow in the manifold pipes derived from a differential axial section are

$$\frac{d\dot{m}_{\text{manifold}}}{dy} = -\dot{m}'_{\text{crossflow}} \quad (4.22)$$

$$c_p \dot{m}_{\text{manifold}} \frac{dT_{\text{manifold}}}{dy} = -\dot{m}'_{\text{crossflow}} c_p (T_{\text{crossflow}} - T_{\text{manifold}}) - / + \dot{q}'_{\text{net,feed/collect}} \quad (4.23)$$

where y is the axial coordinate measured from the receiver's inlet/outlet, $\dot{m}'_{\text{crossflow}} = \dot{m}'_{\text{coil,left}} + \dot{m}'_{\text{coil,right}}$ the mass flow through the cross-flow absorbers per unit axial length, and $T_{\text{crossflow}} = (\dot{m}'_{\text{coil,left}} T_{\text{coil,left}} + \dot{m}'_{\text{coil,right}} T_{\text{coil,right}}) / \dot{m}'_{\text{crossflow}}$ the average temperature of the cross-flow. $\dot{q}'_{\text{net,manifold}} = \dot{q}'_{\text{manifold}} - \dot{q}'_{\text{crossflow}}$ is the net heat transfer leaving the feeding/collecting pipes. The sign change in the last term originates from reverse flow directions in the feeding and collecting pipes. Given the mass flow distribution through the array of absorber tubes, integration of Eq. (4.22) is straightforward, whereas Eq. (4.23) needs to be solved numerically because of temperature-dependent heat flux $\dot{q}'_{\text{net,manifold}}(T_{\text{manifold}})$. The numerical solution is obtained by discretization of the manifold pipes into axial segments and integration over the control volumes. Axial integration is performed in flow direction of the respective manifold pipe, i.e. in positive and negative y -direction for the feeding and collecting pipes, respectively. The integration step derived from an energy balance on a control volume of the collecting pipe is

$$T_{\text{collect},i} = \frac{\dot{m}_{\text{manifold},i+1} c_p T_{\text{collect},i+1} + n_{\text{coil}} \dot{m}_{\text{crossflow},i} c_p T_{\text{crossflow},i} - \dot{q}_{\text{net,collect},i}}{\dot{m}_{\text{manifold},i} c_p} \quad (4.24)$$

where n_{coil} is the number of coiled absorber tubes on each wing. For the feeding pipe, the air leaving through the cross-flow branches is at the same temperature as the flow in the manifold pipe ($T_{\text{crossflow}} = T_{\text{manifold}}$) and Eq. (4.24) simplifies to

$$T_{\text{feed},i+1} = T_{\text{feed},i} - \frac{\dot{q}_{\text{net,feed},i}}{\dot{m}_{\text{manifold},i+1} c_p} \quad (4.25)$$

The heat gain by the HTF is calculated as

$$\dot{Q}_{\text{gain,HTF}} = \dot{m}_{\text{manifold},0} c_p (T_{\text{collect},0} - T_{\text{feed},0}) \quad (4.26)$$

where subscript “0” denotes quantities at the solar receiver’s inlet/outlet. The solar-to-thermal receiver efficiency is defined as

$$\eta_{\text{receiver}} = \frac{\dot{Q}_{\text{gain,HTF}}}{\dot{Q}_{\text{solar,input}}} \quad (4.27)$$

4.1.5 Pressure drop

The pressure distribution of the HTF flow through the solar receiver is calculated using the generalized Bernoulli equation [94]

$$\left(p + \rho \alpha \frac{U^2}{2} \right)_1 = \left(p + \rho \alpha \frac{U^2}{2} \right)_2 + \Delta p_{\text{total},1-2} \quad (4.28)$$

where potential energy is neglected and p is the thermodynamic pressure, U the mean velocity averaged over the cross-section, α the kinetic energy factor (fully developed laminar flow in circular pipe: $\alpha = 2$, turbulent flow: $\alpha \approx 1$), and $\Delta p_{\text{total},1-2}$ the total pressure loss due to friction and dynamic losses between sections 1 and 2. Pressure drop by friction is evaluated using the Darcy-Weissbach equation [94]

$$\Delta p_{\text{friction}} = f \frac{L}{D} \rho \frac{U^2}{2} \quad (4.29)$$

where f is the Darcy friction factor, and L and D the length and diameter of the pipe section, respectively. Friction factors in the feeding and collecting manifold pipes and in the flexible hoses are calculated using the relationship by Churchill [95] that is valid for the laminar and turbulent regime [94],

$$f = 8 \left[\left(\frac{8}{\text{Re}_D} \right)^{12} + \frac{1}{(B_1 + B_2)^{1.5}} \right]^{1/12} \quad (4.30)$$

$$B_1 = \left[2.457 \ln \left(\frac{1}{(7 / \text{Re}_D)^{0.9} + (0.27e / D)} \right) \right]^{16}, \quad B_2 = \left(\frac{37530}{\text{Re}_D} \right)^{16}$$

where e is the equivalent roughness of the pipe surface. Friction factors in the helically coiled absorber tubes are calculated using the relation by Manlapaz and Churchill [60] for fully-developed laminar flow

$$\frac{f_{\text{curved}}}{f_{\text{straight}}} = \left[\left(1 - \frac{0.18}{\left[1 + (35 / \text{De})^2 \right]^{1/2}} \right)^b + \left(1 + \frac{r_{\text{tube}} / r_{\text{coil}}}{3} \right)^2 \left(\frac{\text{De}}{88.33} \right) \right]^{1/2} \quad (4.31)$$

where $b = 2$ for $\text{De} \leq 20$, $b = 1$ for $20 < \text{De} \leq 40$, and $b = 0$ for $\text{De} > 40$. Here, $\text{De} = \text{Re}(r_{\text{tube}} / r_{\text{coil}})^{1/2}$ is the Dean number and f_{curved} and $f_{\text{straight}} = 64 / \text{Re}$ denote the Darcy friction factors of laminar flow in curved and straight pipes, respectively. Dynamic losses resulting from flow disturbances by pipe fittings, valves, and flow junctions are calculated by

$$\Delta p_{\text{dynamic}} = C_0 \rho_0 \frac{U_0^2}{2} \quad (4.32)$$

where C_0 is the dimensionless local loss coefficient referring to the dynamic pressure $p_{\text{dynamic},0} = \rho_0 U_0^2 / 2$ at section 0. The loss coefficients of sudden pipe diameter contractions and expansions are approximated by:

$$C_{\text{contraction},1} = 0.5 \left(1 - \frac{A_1}{A_2} \right) \quad (4.33)$$

$$C_{\text{expansion},1} = \left(1 - \frac{A_1}{A_2} \right)^2 \quad (4.34)$$

where C_1 refers to section 1 with the smaller area $A_1 < A_2$. Pressure drop from the feeding manifold pipe to the T-branches across the knife gate valves is modelled as discharge from a large tank, for which follows from Eq. (4.33) for $A_2 \rightarrow \infty$ $C_{\text{gate,entry}} = 0.5$, followed by a sudden cross-section expansion from the gate area to the inlet area of the T-branch given by Eq. (4.34). The geometry of the knife gate is shown schematically in Fig. 4.5. The gate area is the overlapping region of two circles, i.e. the sum of two segments of a circle, which is a function of the relative linear shift ξ_{gate} that is defined as 0 for completely closed and 1 for completely open valves, respectively.

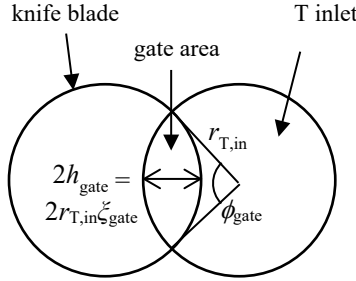


Fig. 4.5: Geometry of knife gate valve.

$$A_{\text{gate}} = r_{\text{T,in}}^2 (\phi_{\text{gate}} - \sin \phi_{\text{gate}})$$

$$\phi_{\text{gate}} = 2 \arccos \left(1 - \frac{h_{\text{gate}}}{r_{\text{T,in}}} \right) = 2 \arccos (1 - \zeta_{\text{gate}}) \quad (4.35)$$

Imperfect sealing of the knife gate valves is considered by a parallel flow that bypasses the gate through a leak with area A_{leak} and loss coefficient calculated as for the gate, $C_{\text{leak}} = 0.5 + (1 - A_{\text{leak}} / A_{\text{T,in}})^2$. For the T-branches that divide the flow into two streams feeding the coils on the left and right wings, $C_{\text{T},0} = 1.75$, where subscript “0” denotes the common section [94]. Exit flow from the coils into the collecting manifold pipe is considered as discharge into a large tank, for which follows from Eq. (4.34) for $A_2 \rightarrow \infty$ $C_{\text{coil,exit}} = 1$.

Using the axial discretization, mass conservation Eq. (4.22) for the mass flow in the manifold pipes after i receiver sections counted from the solar receiver’s inlet/outlet yields,

$$\dot{m}_{\text{manifold},i} = \dot{m}_{\text{manifold},i-1} - n_{\text{coil}} (\dot{m}_{\text{coil,left},i} + \dot{m}_{\text{coil,right},i}) \quad (4.36)$$

and for the total mass flow through the solar collector

$$\dot{m}_{\text{manifold},0} = \sum_{i=1}^{N_{\text{receiver}}} n_{\text{coil}} (\dot{m}_{\text{coil,left},i} + \dot{m}_{\text{coil,right},i}) \quad (4.37)$$

where N_{receiver} is the number of receiver sections in one collector. The Bernoulli Eq. (4.28) delivers an expression for the pressure drop through the cross-flow branches on the left and right wings,

$$p_{\text{feed},i} - p_{\text{collect},i} - \Delta p_{\text{gate},i} - \Delta p_{\text{T},i} = \begin{cases} \Delta p_{\text{hose,left},i} + \Delta p_{\text{coil,left},i} \\ \Delta p_{\text{hose,right},i} + \Delta p_{\text{coil,right},i} \end{cases} \quad (4.38)$$

with the recursion formulae for the pressures in the feeding and collecting manifold pipes

$$\begin{aligned} p_{\text{feed},i} &= p_{\text{feed},i-1} - \Delta p_{\text{feed},i} \\ p_{\text{collect},i} &= p_{\text{collect},i-1} + \Delta p_{\text{collect},i} \end{aligned} \quad (4.39)$$

Eqs. (4.37) and (4.38) deliver $2N + 1$ equations for the $2N + 3$ unknowns $\dot{m}_{\text{coil}}$, $\dot{m}_{\text{manifold},0}$, $p_{\text{feed},0}$, $p_{\text{collect},0}$. The numerical solution for $\dot{m}_{\text{coil}}$ and $p_{\text{collect},0}$ is found by using the initialization at the receiver's inlet,

$$\begin{aligned} \dot{m}_{\text{manifold},0} &= \dot{m}_{\text{inlet}} \\ p_{\text{feed},0} &= p_{\text{inlet}} \end{aligned} \quad (4.40)$$

The pressure drop of the solar receiver is evaluated as

$$\Delta p_{\text{receiver}} = p_{\text{feed},0} - p_{\text{collect},0} \quad (4.41)$$

The electric pumping power required to circulate the process air through the solar receiver is calculated by

$$\dot{W}_{\text{pump}} = \frac{1}{\eta_{\text{pump}}} \left(\frac{\dot{m}_{\text{inlet}} R_{\text{air}} T_{\text{inlet}}}{p_{\text{outlet}}} \right) \Delta p_{\text{receiver}} \quad (4.42)$$

where $R_{\text{air}} = 287 \text{ J/(kg K)}$ and $\eta_{\text{pump}} = 0.8$ is the pumping efficiency of the fan.

4.2 Solar pilot plant

The layout of the solar plant constructed in Ait Baha, Morocco (30.217° N, 9.149° W, altitude: 256 m) is shown schematically in Fig. 4.6, along with photographs of the plant and a solar collector in tracking mode. It is composed of three 212 m-long solar collector units, integrated to a thermal energy storage (TES) based on a thermocline packed bed of rocks that is immersed in the ground. The solar field has a total active aperture area of 5880 m². It provides thermal power to the TES and to an air-to-oil heat exchanger coupled to an organic Rankine cycle (ORC) for the purpose of boosting electricity production from waste heat of a cement factory [82]. The concrete frame of each solar collector unit is built as an assembly of 12 elementary modules, each 17.64 m-long, which

constitute the basic constructive element of a pre-cast, pre-stressed fiber-reinforced high performance concrete, manufactured on site. Each module comprises one central and two external longitudinal beams that are supported at either ends by concrete wheels based on a foundation group for decoupling the permanent load (~ 140 tons/module) from transient phenomena such as wind, thermal expansion, or seismic action. Each wheel is driven by a dedicated electric engine via a gearbox and precision-encoders. In addition, the torsions in all driving shafts are monitored by torsion elements. The tracking mechanism guarantees a tracking accuracy of $\pm 0.05^\circ$.

4.2.1 Solar trough concentrator

The semi-transparent topsheet is composed of 12 m-long and 1.8 m-wide ETFE films (Toray Toyoflon [96], solar weighted normal transmittance: 0.913 [85], $\pm_{0.05}^{0.00}$ due to soiling) that are oriented transversally and welded together piece-by-piece to provide the required length of 216 m. Each mirror foil is composed of three 1.85 m-wide boPET films (Toray Lumirror U50 [97], thickness: 23 μm) that are welded together longitudinally to provide the required width of 5.4 m. The topmost foil is aluminized to achieve high reflectivity at low cost. Its solar-weighted specular reflectance at near normal incidence is 0.895 [85], $\pm_{0.02}^{0.00}$ due to dust. The welded mirror films are brought on rolls and then rolled out over the collector's length. The films are clamped at both ends by plastic sliders that are pulled in axial direction through extruded aluminum channels by electric motors. This provides an air-tight partitioning of the pressure chambers between the mirror films over the collector's length. The lowermost pressure chamber (p_4) is closed by the PVC-PES film at the bottom and extends over the left and right mirror wings. At both ends of the solar collector, the pressure chambers are sealed by a vertical mirror shape adjusted metallic plate for each mirror foil. The pneumatic sleeves are fabricated from two 350 mm-wide and 0.5 mm-thick PVC films that are welded together on both sides to form an inflatable sleeve (non-inflated (flat) width: 190 mm). Each sleeve has a length of 8.82 m, i.e. there are two sleeves along the module's length.

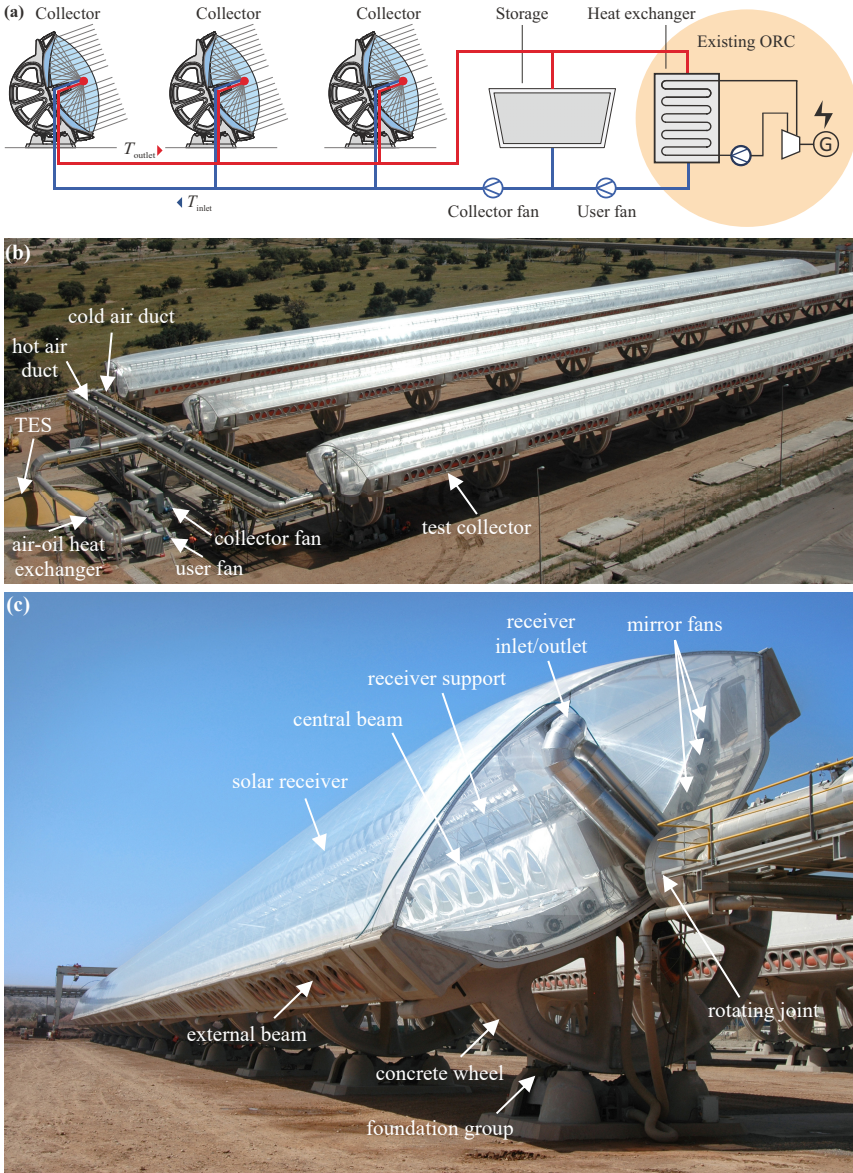


Fig. 4.6: Pilot plant in Ait Baha, Morocco: a) scheme; b) photograph of the three solar trough concentrators, piping, and TES; c) photograph of one solar collector unit in tracking mode.

4.2.2 Solar air receiver

A solar collector unit comprises 12 modules. Each module contains three 5.88 m-long sections. A photograph of an assembled solar receiver section is shown in Fig. 4.7. The trumpet secondary concentrators are fabricated from 1.96 m-long extruded aluminum profiles featuring an internal channel for the cooling water flow. The receiver windows are made of borosilicate glass and AR-coated [98], with a solar-weighted normal transmittance of 0.95 [85], $\pm_{0.01}^{0.00}$ due to soiling. They are sustained by the base insulation blocks made of calcium silicate-based ceramic (Promat Monalite M1, $k_{400} = 0.23 \text{ W/(m K)}$) [69]). The feeding pipe is constructed from a steel grid that spans an airtight, temperature resistant tubular PTFE/glass fiber fabric. At the bottom, it features an axial, perforated steel plate with threads for the T-branches, denoted as T inlet plate. The knife gate valves are constructed by a second axial, perforated steel plate, referred to as knife blade, which is pushed onto the T inlet plate by springs while allowing for sliding. The position of the knife blade is controlled by a step motor for each receiver section. Thermal expansion of the feeding pipe is accounted for by expansion bellows placed between the receiver sections. The absorber tubes are fabricated from stainless steel (grade AISI 316L) and coated by black paint for enhanced solar absorptance. The collecting pipe is assembled from 2.94 m-long stainless steel (grade AISI 321) pipe segments, i.e. two segments per receiver section, each equipped with 13 expansion bellows. The thermal insulation layers on the feeding pipe, inner and outer surrounding groove walls, and on the collecting pipe are made of glass wool (Promat Promaglaf G, $k_{400} = 0.087 \text{ W/(m K)}$, inclined hatching in Fig. 4.4; and Promaglaf HT, $k_{400} = 0.12 \text{ W/(m K)}$, vertical hatching in Fig. 4.4 [69]). The hollow volume between the manifold pipes and the absorber tubes is filled by flocked glass wool insulation (Promat Promaglaf G, horizontal hatching in Fig. 4.4). The radiation shield insulation is constructed from corrugated aluminum foils ($A_{\text{smooth}} / A_{\text{corrugated}} = 0.83$) that are kept at the design distance by circumferential glass wool spacers placed in axial intervals of 0.5 m (cf. Fig. 4.7).

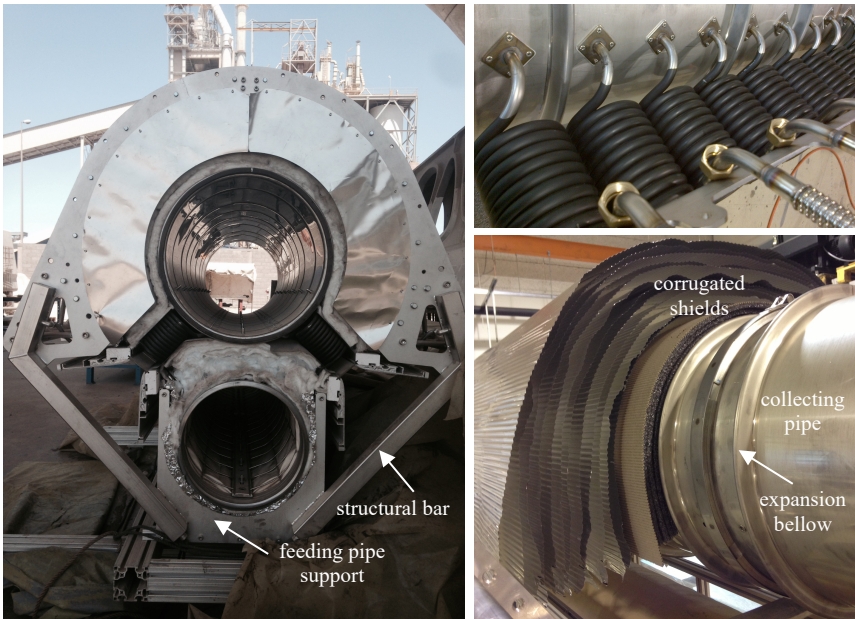


Fig. 4.7: Photographs of a solar receiver section assembled on the ground (left), with an array of coiled absorber tubes (top) and corrugated radiation shields (bottom) attached to the collecting pipe.

4.2.3 Measurement instrumentation

A weather station measures the atmospheric pressure, temperature, relative humidity (RH) and wind speed. DNI is measured by a pyrheliometer (Kipp & Zonen CHP 1). A second weather station measures T and RH inside the solar collector. The overpressure inside the solar collector that creates the convex shapes of the ETFE and PVC-PES foils is measured by a differential pressure sensor and sustained round-the-clock by a two-stage air inflation system. The principal fan is placed between the primary and secondary air filters and sucks in air from atmosphere. An auxiliary fan delivers the filtered air to the solar collector. A PID controller stabilizes the collector's inflation pressure at 130 Pa. Pressure chambers 1–3 of either wing of the ASC are driven each by two frequency-synchronized fans located at both ends of the solar collector that evacuate air from the chambers. The common chamber 4 is driven by four fans,

two located at each end of the solar collector. The chamber pressures p_1 to p_4 are measured at both ends by differential pressure transmitters with respect to the ambient pressure inside the collector. An automatic control system based on a dedicated PID controller for each chamber holds the pressures at the desired set points. Inflation and deflation of the 192 pneumatic sleeves is driven by a single blower for the complete collector via axial pumping and suction lines and a series of valves. A pressure sensor reads sequentially the pressures of eight sleeves by opening the corresponding valves. After every pressure measurement, the control system connects the sleeve for a certain amount of time to the pumping or suction line, depending if the sleeve needs to be inflated or deflated, respectively, such that the pressure approaches and remains at the desired set point.

Mirror shape measurements are performed on each wing of the primary concentrator by a vertically downward facing digital camera (Basler acA2500-14gm, pixel size: $2.2 \mu\text{m} \times 2.2 \mu\text{m}$, focal length: 25 mm) mounted on a linear drive that scans the aperture in transversal x -direction. The camera is aligned with the optical z -axis of the trough concentrator by the reflection from a horizontal flat glass mirror aligned by a precision level at zero tracking angle. The resulting alignment error of the camera is $\pm 0.05^\circ$. During an aperture scan, a software identifies the position on the camera pixel matrix of the solar receiver — appearing in the camera image as a vertical black strip — for each transversal position. The recorded position n (in pixel) is a measure for the angular deviation θ_r of the specularly reflected on-axis ray from the focus, related by the imaging equation

$$\theta_r = \arctan \left(n \frac{d_{\text{pixel}}}{f_{\text{camera}}} \right) \quad (4.43)$$

where d_{pixel} is the pixel size and f_{camera} the focal length of the camera. The concentrator shape scan $\theta_r(x)$ corresponding to the nominal arcspline geometry is plotted in Fig. 4.8, showing the four spherical aberrations of the ASC. The concentrator shape scan of a perfectly parabolic solar concentrator is $\theta_r(x) = 0$. The maximum tolerable angular deviation varies over x with the focal distance r and the angle under which the tilted focal plane is seen from the reflection point on the mirror

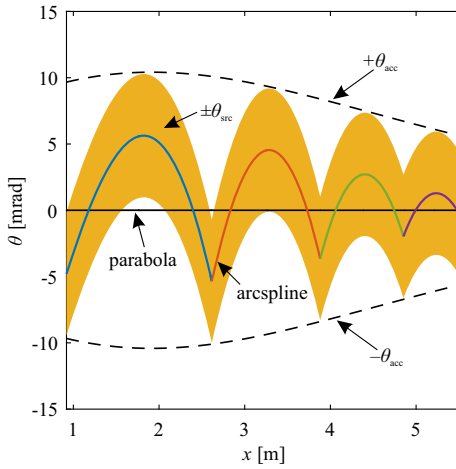


Fig. 4.8: Ideal concentrator shape scans for perfect parabola (black solid) and nominal arcspline geometry (colored arcs). Inlet aperture of secondary concentrator (black dashed) and solar band (orange area) are also shown.

$$\theta_{\text{acc}} = \arctan\left(\frac{a_{\text{in}} \cos(\Phi - \Phi_{\text{tilt}})}{2r}\right) \quad (4.44)$$

The acceptance angle $\theta_{\text{acc}}(x)$ is indicated by black-dashed lines in Fig. 4.8. The incident solar radiation arrives within a cone half-angle of $\theta_{\text{src}} = \theta_{\text{sun}} / \cos \theta_{\text{skew}}$ which is shown in Fig. 4.8 by the orange area for $\theta_{\text{sun}} = 4.65$ mrad and $\theta_{\text{skew}} = 0^\circ$ and denoted as the solar band. The intercept factor of the primary concentrator can be inferred from the mirror shape scan as the area fraction of the solar band that falls within $\pm\theta_{\text{acc}}$

$$\hat{\gamma}_{\text{primary}} = \frac{1}{2\theta_{\text{src}} a_{\text{primary}}} \int_0^{a_{\text{primary}}} \max\{\min[\theta_r(x) + \theta_{\text{src}}, \theta_{\text{acc}}] - \max[\theta_r(x) - \theta_{\text{src}}, -\theta_{\text{acc}}], 0\} dx \quad (4.45)$$

The HTF fan is driven by an asynchronous motor whose rotational speed is controlled via the frequency at the inverter. Mass flow rate is measured at the solar receiver's inlet and outlet by averaging Pitot tubes and differential pressure transmitters (Endress + Hauser Deltatop DP62D and Deltabar M PMD55, 0 – 3.1 kg/s, ± 0.02 to 0.035 kg/s under all experimental conditions). For pressure and

temperature compensation, absolute pressure transmitters and temperature probes (Endress + Hauser Cerabar M PMP51, 0 – 1100 mbar, ± 1.65 mbar; and Omnigrad S TC61, 0 – 500/700 °C on cold/hot side, ± 1.5 °C) are installed upstream and downstream both Pitot tubes, respectively. Due to an insufficient straight pipe section downstream the solar receiver's outlet, the mass flow measurement on the hot side is located downstream the rotating joint outside the solar collector. The air outlet temperature of the solar receiver is measured by J-type thermocouple (Endress + Hauser, Omnigrad S TC61, 0 – 700 °C, ± 1.5 °C). Pressure drop of the solar receiver is measured by differential pressure transmitters (Sensortech BTE5000, 0 – 70 mbar, ± 0.5 mbar) installed at the solar receiver's inlet and outlet. The air outlet temperatures of one coiled absorber tube per receiver section on the left and right wings are measured using K-type thermocouples.

4.3 Experimental

4.3.1 Off-sun measurements

Pressure regulation of the pneumatic ASC is performed off-sun with the collector in the horizontal position ($\beta_{\text{track}} = 0^\circ$). Shape measurements are shown in Fig. 4.9a. Distortions that manifest themselves as discontinuities in the measured mirror shape occur at a transversal position of ± 4.4 m (cf. Fig. 4.9a left), which coincides with the position of the longitudinal weld seams of the mirror films. Shape distortion is caused by longitudinal tension in the films and can be reduced or eliminated by adjusting the film mounting process. The final intercept factors for each solar collector module, calculated by Eq. (4.45) from concentrator shape scans, are shown in Fig. 4.9b. The maximum intercept factor amounts to 0.94. The low values at the first and last positions are due to end effects that cause wrinkling and shape distortions of the mirror films, which could be reduced by matching the shapes of the vertical end plates precisely to the pneumatic mirror shape. Steady wind has little effect on concentrator shape because it is absorbed by the control system of differential pressures on the mirror films, but sudden wind loads lead to shape fluctuation, with the intercept factor remaining within 1% for moderate wind of e.g. 4 m/s.

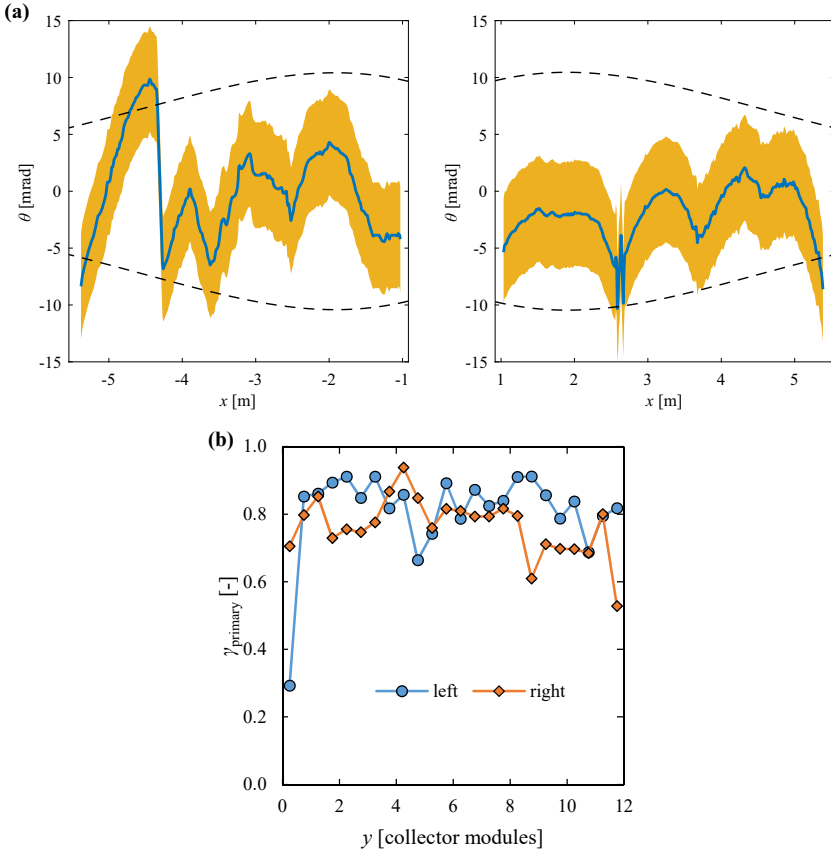


Fig. 4.9: a) Experimentally measured concentrator shape scans on left and right wings. b) Intercept factors calculated from concentrator shape scans after pressure regulation at 24 axial positions on both wings.

Hot air at 119 °C is fed to the solar receiver. The experimentally measured coil outlet temperatures $T_{\text{coil,out,exp}}$ in steady-state and the numerically simulated temperatures $T_{\text{coil,out,sim}}$ under the same experimental conditions are shown in Fig. 4.10. Additionally shown are the numerically simulated temperature distributions of the airflow in the feeding ($T_{\text{feed,sim}}$) and collecting manifold pipes ($T_{\text{collect,sim}}$), and mass flow distribution $\dot{m}_{\text{manifold,sim}}$. The deviation between the experimentally measured and numerically simulated receiver outlet temperatures

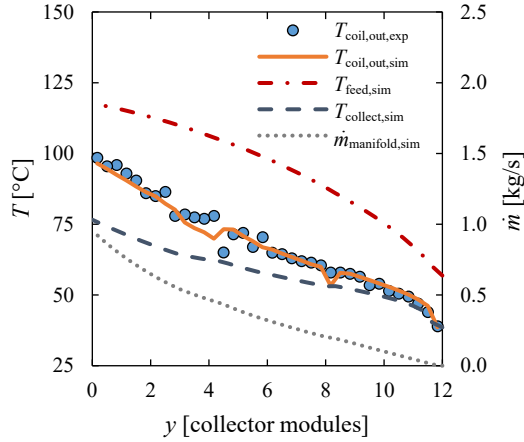


Fig. 4.10: Off-sun heat loss test with $\dot{m}_{\text{inlet}} = 0.954 \text{ kg/s}$ and $T_{\text{inlet}} = 119 \text{ }^{\circ}\text{C}$. Experimentally measured and numerically simulated coil outlet temperatures, and simulated air temperatures and mass flow rate in feeding and collecting manifold pipes as a function of axial position.

is $1.5 \text{ }^{\circ}\text{C}$. Of the total heat loss of the HTF flow of 40 kW , 42% is lost in the feeding pipe, 31% in the flexible hoses, 25% in the array of coiled absorber tubes, and 2% in the collecting pipe.

4.3.2 On-sun measurements

A representative on-sun test is shown in Fig. 4.11a, performed on November 4, 2015. Plotted are the temporal variation of DNI, air mass flow rate, pressure drop, inlet and outlet temperatures of the solar receiver unit, and outlet temperatures from left and right coils of a solar receiver section. The solar receiver is heated up for about one hour until steady-state operation is reached. The air outlet temperature from the solar receiver is stabilized by decreasing stepwise the air mass flow rate. At the end of test, the pneumatic ASC is defocalized first before the solar collector is stopped and returned into the horizontal position. The measured air outlet temperatures from the left and right arrays of coiled absorber tubes in steady-state and the relative linear positions of the knife blades are shown in Fig. 4.11b. The gray bars indicate shaded regions caused by: I) end

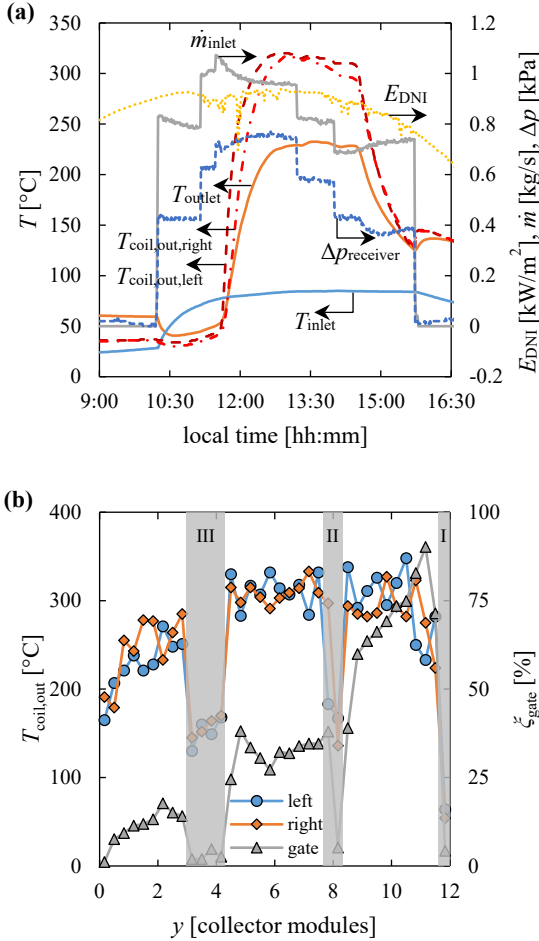


Fig. 4.11: a) Temporal variation of DNI, air mass flow rate, pressure drop, inlet and outlet temperatures of the solar receiver unit, and outlet temperatures from left and right coils of a solar receiver section; b) axial distribution of coil outlet temperatures measured on the left and right wings in steady-state and linear position of knife gate valves.

effect due to skew angle, II) covered primary mirror, and III) covered protective topsheet. The latter two shaded regions were due to technical problems with the corresponding components during commissioning and served as a safe spot for installing and parking measurement equipment during solar tracking. The

average intercept factor, calculated from the mirror shape scans, was 0.56. The significant reduction of concentrator performance during solar tracking is due to temperature effects (elongation of mirror films because of thermal expansion) and gravity effects (deflection of concrete beams and receiver displacement because of weight).

The experimentally measured and numerically simulated axial distributions of air outlet temperatures from the left and right arrays of coiled absorber tubes are shown in Fig. 4.12 a and b, respectively. The set mass flow rates are plotted on the right vertical axis. The mean absolute and root-mean square (RMS) deviations between measured and computed temperatures are 24 °C and 34 °C, respectively, primarily attributed to spatial and temporal fluctuations of the primary mirror shape over the solar collector's length. A parity plot of the experimentally measured and numerically simulated air outlet temperatures of the solar receiver and pressure drops across the solar receiver at steady-state conditions is shown in Fig. 4.12 c and d, respectively. The mean absolute and RMS deviations between measured and computed temperatures are 7.3 °C and 9.5 °C, respectively, and between measured and computed pressures are 0.25 mbar and 0.41 mbar, respectively. The maximum and RMS percentage deviations between the measured and computed thermal power output $\dot{Q}_{\text{gain,HTF}}$ of the solar receiver are 13% and 6%, respectively, and between the measured and computed pumping power $\dot{W}_{\text{pump}}$ are 7% and 3%, respectively.

Energy balance

Optical losses in percentage of the solar radiation arriving at the primary's aperture, excluding the shaded regions, occur by: 1) reflection and absorption at the semi-transparent topsheet: 15 – 19% for skew angles 34 – 53°, 2) absorption at the primary reflector: 10%; and 3) intercept losses (average): 32 – 37%, yielding average optical efficiencies of 0.42 – 0.36 for the primary solar concentrator. Losses of the solar receiver, quantified in percentage of the solar radiative power input incident on the secondary concentrator's inlet, are classified into optical (18 – 24%) and thermal losses (21 – 55%), yielding receiver efficiencies in the range 0.24 – 0.57 for all steady-state experimental conditions. Optical losses of the solar receiver are further classified into blocking

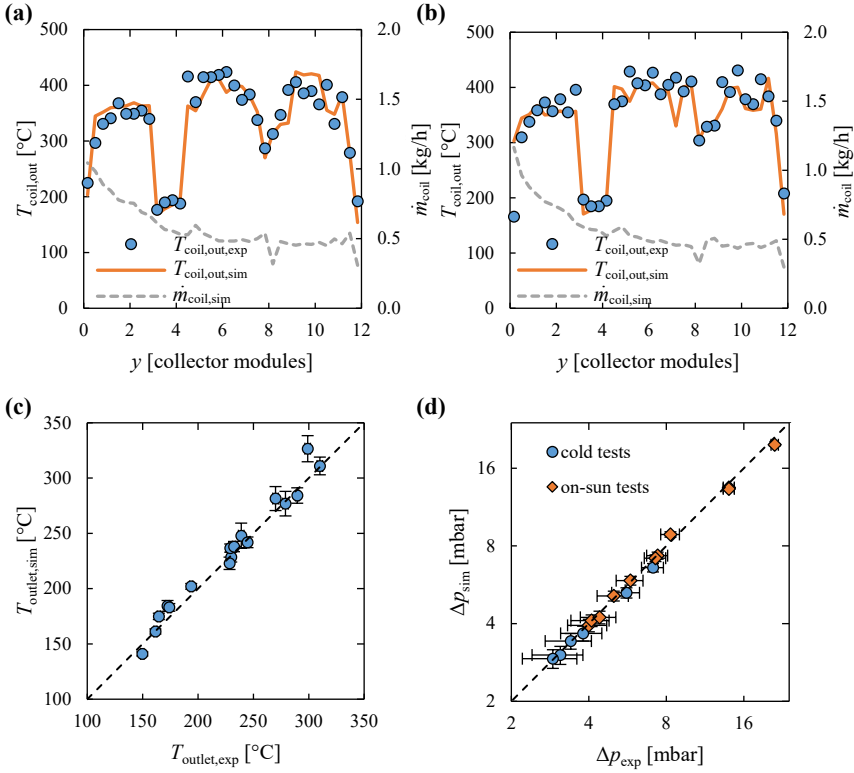


Fig. 4.12: Experimentally measured and numerically simulated coil outlet temperatures, and set mass flow rates for left (a) and right (b) wings. c) Parity plot of experimentally measured vs. numerically simulated air outlet temperatures of the solar receiver at steady-state conditions. d) Parity plot of numerically simulated vs. experimentally measured pressure drop.

by the structural bars (3.8 – 4.9%), absorption by the secondary concentrator (1.5%), radiation spillage between the secondary concentrator’s outlet and the receiver’s aperture (5.9%), and reflection from the solar receiver including the window (7.4 – 11.5%). Heat losses occur in the feeding pipe (0 – 8% for T_{inlet} in the range 30 – 140 $^{\circ}\text{C}$), in the coiled absorber tubes by thermal re-radiation (4 – 13%) and natural convection (3 – 7%) through the window and heat conduction through the thermal insulation (12 – 27%), and in the collecting pipe (2.1 – 3.3%) for T_{outlet} in the range 150 – 310 $^{\circ}\text{C}$. The air pumping power consumption for the

solar collector unit varies from 0.17 kW at $\dot{m}_{\text{inlet}} = 0.51 \text{ kg/s}$ to 5.9 kW at $\dot{m}_{\text{inlet}} = 2.0 \text{ kg/s}$, accounting for 0.5% to 7.8% of the solar collector unit's thermal power output multiplied by a factor 0.35 for heat-to-electricity conversion.

Technical improvement and performance predictions

The degradation of optical properties of materials can be diminished by applying appropriate cleaning of the semi-transparent topsheet and filtering of the air that inflates the solar collector. Shape distortions can be diminished by reducing longitudinal tension in the mirror films, avoiding receiver displacement, and adjusting pressures. Minor modifications are required in the solar receiver. Blocking of concentrated solar radiation by the structural bars can be halved by reducing their profile width and height from 50 mm to 25 mm. Such downsizing of the structural bars increases the nominal and annual mean optical efficiencies of the solar receiver by 1.2% and 1.5%, respectively. Solar radiation spillage between the secondary concentrator's outlet and the solar receiver's aperture can be reduced by decreasing their distance from 3 mm to 1.5 mm. The knife gate valves need to be redesigned for airtightness such that equal mass flow rates can be established throughout the solar receiver's length. The thermal insulation can be improved by replacing the calcium silicate based insulation blocks and glass wool insulation by micro-porous insulation (Microtherm Super G, $k_{400} = 0.0225 \text{ W/(m K)}$ [68]).

Simulations are performed for Tamanrasset, Algeria (22.790° N, 5.529° E, altitude: 1385 m) using measured DNI data from the baseline surface radiation network for the year 2013 with a temporal resolution of one minute. The solar collector tracks $\pm 70^\circ$ because of limitations of the two-wing design and is oriented such that the tracking axis coincides with the N-S direction. The solar receiver is operated with ambient air at $T_{\text{inlet}} = 25^\circ \text{C}$ and delivers hot air at $T_{\text{outlet}} = 500^\circ \text{C}$ with $\dot{m}_{\text{inlet}}$ adjusted accordingly. Temperatures in- and outside the solar collector are assumed as $T_{\text{amb}} = 50^\circ \text{C}$ and $T_{\text{sky}} = 25^\circ \text{C}$, respectively. Numerical simulations are performed for E_{DNI} and θ_{skew} in the range 100 – 1200 W/m^2 and 0 – 50° in intervals of 100 W/m^2 and 10°, respectively, and the instantaneous power at the actual solar conditions is calculated by bilinear

interpolation. The yearly energy delivered by the solar collector and lost by the different mechanisms are calculated by

$$Q_{\text{yr}} = \int_{\text{yr}} \dot{Q}(t) dt \quad (4.46)$$

Of the total available DNI of 2400 kWh/m²/yr, 80% arrive at the primary solar concentrator's aperture, whereas cosine losses and non-collected energy due to the tracking limits amount to 9% and 11%, respectively. The annual mean optical efficiency of the primary concentrator is 0.75. Losses occurring at the semi-transparent topsheet, primary reflector, and intercept losses account for 9.6%, 9.4%, and 5.7% of the solar radiation arriving at the primary's aperture, respectively. The annual mean optical efficiency of the solar receiver is 0.90, with optical losses in percentage of the solar power input occurring by blocking at the bars (1.6%), absorption in the secondary concentrator (0.8%), radiation spillage (1.3%), and reflection from the receiver (6.6%). Heat losses from the coiled absorber tubes occur by re-radiation (11%) and natural convection (4.4%) from the window, and heat conduction through the thermal insulation (3.3%), where heat leaving through the inner insulation (1.3%) is actually not lost but preheats the cold air flow in the feeding pipe. Heat losses from the collecting pipe amount to 2.6% of the solar power input, requiring that the average outlet temperature from the coils is roughly 15 °C higher than the desired T_{outlet} . The pumping power remains below 1% of the thermal power output (multiplied by a factor 0.35). The annual mean DNI-to-heat efficiency and peak efficiency at $E_{\text{DNI}} = 1000 \text{ W/m}^2$, $\theta_{\text{skew}} = 0^\circ$ are 0.39 and 0.61, respectively.

4.4 Comparison to state-of-the-art

These efficiency figures are roughly 10% (absolute) lower than the values reported for commercial state-of-the-art parabolic trough collectors such as the third-generation EuroTrough (Skal-ET) installed in the Andasol power plants [99, 100]. This discrepancy is attributable to major differences in the design of the solar concentrator and receiver. The semi-transparent topsheet reduces the optical efficiency due to Fresnel losses occurring particularly at large skew angles [85]. At normal solar incidence, the optical efficiency of the inflated collector system amounts to 0.69, compared to 0.76 and 0.79 for the EuroTrough

and next generation Ultimate Trough, respectively [101, 102]. On the other hand, the convex topsheet is easier to clean and allows for operation under harsh conditions such as in the proximity of factories for delivering industrial process heat. The voluminous solar air receiver suffers from higher heat losses than conventional selectively coated and vacuum insulated absorbers for liquid HTFs. The thermal efficiency of the air-based receiver, defined as the heat transferred to the HTF divided by the solar power absorbed by the receiver, at $E_{\text{DNI}} = 1000 \text{ W/m}^2$, $\theta_{\text{skew}} = 0^\circ$ amounts to 0.88, compared to 0.95 – 0.96 for state-of-the-art parabolic trough receivers such as Solel's UVAC 3 [103] or Schott's PTR 70 [104]. On the other hand, the cavity-type solar air receiver uses no selective coatings and vacuum insulation, which are subject to temperature limitations and degradation. Future R&D must focus on improving the annual mean optical efficiency of the system. Major potential is identified in the intercept factor and the use of alternative materials with higher solar-weighted reflectance for the primary concentrator and higher transmittance for the protective topsheet. On the receiver side, heat losses have to be reduced and a piping system for air at near-ambient pressure with low cost and thermal inertia needs to be developed to centralize the heat of multiple solar collector units [105]. The direct capital costs for the solar field, HTF piping system, and TES are estimated at 500 EUR/m² of collecting aperture area², which are in the cost range stated for commercial state-of-the-art parabolic trough collectors [106, 107]. The main advantages of the air-based system analyzed in this work are its higher operating temperature, ease of integration with industrial process heat and thermal storage, and potential for efficiency improvements and cost reductions.

4.5 Conclusion

The design, modelling, fabrication, and testing of a pilot-scale solar trough concentrator system, based on a stack of reflective polymeric films mounted on a rigid concrete support and combined with a cross-flow solar receiver using air as heat transfer fluid, was described as an alternative to conventional solar parabolic trough collectors. A numerical optical and heat transfer analysis of the

² Internal cost estimates by Airlight Energy SA

complete solar collector unit was formulated and experimentally validated by comparison to measured data collected during the commissioning phase of the first-of-its-kind solar pilot plant in Ait Baha, Morocco. The maximum and RMS percentage deviations between the measured and computed thermal power output of the solar concentrating system are 13% and 6%, respectively. For a yearly DNI of 2400 kWh/m^2 , model simulations predict an annual thermal energy output at 500°C of 926 kWh/m^2 of primary solar concentrator aperture. The validated optical and heat transfer models can be further applied for the design and optimization of the solar concentrating optics and solar receiver technology.

5 Straight tubular solar receivers¹

5.1 Introduction

The receiver design based on an array of coiled absorber tubes is one particular technical implementation of the cross-flow receiver concept, where the cross-flow solar absorbers are connected via axial feeding and collecting manifolds. The design evolved from a series of attempts to increase the surface area for heat transfer starting from the most basic straight tubular solar absorber. The initial receiver design comprised a black absorber tube enclosed by an insulated cylindrical cavity with a rectangular aperture and was investigated numerically and experimentally [35, 36]. Based on numerical analyses, an improved receiver design was presented recently, where the absorber has been eliminated and the HTF flows in the directly irradiated cylindrical groove with a windowed aperture [37]. Furthermore, the effect of enhanced surface area for heat transfer by V-corrugations was quantified [37]. However, realizing the theoretical benefit of surface corrugation in practice seems challenging for several reasons such as manufacturability (airtightness, thermal expansion bellows), increased pumping power, and cost.

In this chapter, alternative practical receiver designs are explored, evolving on the route from the single straight tube receiver to the array of coiled absorber tubes. Numerical methods are introduced to simulate the thermal performance of the most promising receiver designs, and the calculated efficiencies are compared to the performance of the array of coiled tubes studied in the previous chapters. Material use and cost considerations are also included.

¹ Material in this chapter has been extracted from M. Querci, “Solar air receivers for parabolic trough concentrators based on multiple parallel straight absorber tubes,” M.Sc. thesis, ETH Zurich, Zurich, Switzerland, 2014, conducted under the direct supervision of P. Good.

5.2 Design and modeling

5.2.1 Solar receiver design

The solar receivers presented in this section are specifically designed for the solar concentrating optics with separated focal lines introduced in Chapter 2 (focal half-distance $F_x = 0.280$ m). However, the designs can be adjusted to any line-focus concentrator. The most basic solar receiver is comprised of a single straight absorber tube and is shown in Fig. 5.1. It may be regarded as a hybrid design between the absorber tube contained in a cylindrical cavity [36] and the directly irradiated circular groove [37], where the absorber tube is kept and the surrounding cylindrical cavity is removed. Metallic base sheets leading from the windowed apertures to the absorber tube close the illuminated chamber. The tube diameter is chosen such that excessive pumping power requirements are avoided. From Eq. (2.29), the specific pumping power is reformulated as

$$\varepsilon_{\text{pump}} = \frac{\dot{V} \Delta p}{\eta_{\text{th-el}} \eta_{\text{pump}} \dot{Q}_{\text{gain,HTF}}} = \frac{\Delta p}{\eta_{\text{th-el}} \eta_{\text{pump}} \rho c_p (T_{\text{outlet}} - T_{\text{inlet}})} \quad (5.1)$$

For meaningful comparisons among different receiver designs, the tube diameter is designed for Δp -equality with the array of coiled absorber tubes including the collecting manifold pipe under the same peak power conditions of $E_{\text{DNI}} = 1000 \text{ W/m}^2$ and $\theta_{\text{skew}} = 0^\circ$. Using an empirical power law expression for the Fanning friction coefficient for fully developed turbulent flow in a straight pipe [108]

$$c_f = 0.046 \text{Re}_D^{-1/5} \quad (3 \times 10^4 < \text{Re} < 10^6) \quad (5.2)$$

the tube diameter may be expressed analytically by

$$D_{\text{tube}} = 0.666 \dot{m}^{3/8} \left(\frac{\mu^{1/5} L}{\rho \Delta p} \right)^{5/24} \quad (5.3)$$

An alternative to surface corrugation for enhanced heat transfer is the flow division into multiple parallel channels. Eq. (5.3) yields for the pipe diameters as a function of the number of tubes N_{tube} for equal pressure drop

$$D_{\text{tube}} \propto \left(\frac{\dot{m}}{N_{\text{tube}}} \right)^{3/8} \quad (5.4)$$

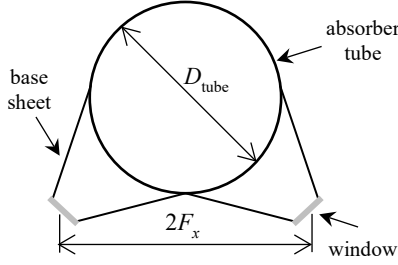


Fig. 5.1: Schematic of single straight tube solar receiver.

The benefit of flow division may be expressed in terms of the required temperature difference ΔT between tube wall and HTF to drive the convective heat transfer to the fluid $\dot{Q}_{\text{gain,HTF}} = L\pi\text{Nu}_d k \Delta T$. Using another power law expression for uniform heating in fully developed turbulent flow that is valid for gases [108],

$$\text{Nu}_d = 0.022 \text{Pr}^{1/2} \text{Re}_d^{4/5} \quad (0.5 < \text{Pr} < 1.0) \quad (5.5)$$

yields for the temperature difference

$$\Delta T = \frac{8.615 \mu^{5/6} \dot{m}^{1/2} c_p (T_{\text{outlet}} - T_{\text{inlet}})}{k (\text{Pr} N_{\text{tube}})^{1/2} L^{5/6} (\rho \Delta p)^{1/6}} \quad (5.6)$$

Accordingly, for gaseous HTFs in turbulent flow ΔT decreases with the square root of the number of tubes, $\Delta T \propto (N_{\text{tube}})^{-1/2}$. D_{tube} decreases with less than the square-root of N_{tube} at constant Δp such that only a small number of tubes fit into a solar receiver, which limits the flexibility in tube arrangement. Two feasible configurations with the defining geometric parameters are shown in Fig. 5.2. The upper tubes are positioned on a circular arc with central angle ϕ_{arc} and radius $R_{\text{arc}} = D_{\text{tube,o}} / [2 \sin(\phi_{\text{arc}} / (2N_{\text{tube}}))]$. This yields equal illuminated surface segments for all tubes with central angle $\phi_{\text{illuminated}} = \pi - \phi_{\text{arc}} / N_{\text{tube}}$. ϕ_{arc} is tuned such that all tubes receive similar amounts of concentrated solar radiation. $\beta_{\text{base,o}}$ and $w_{\text{base,o}}$ are chosen to minimize direct solar irradiation on the outer base sheets while keeping $\phi_{\text{illuminated}}$ for the outermost tubes. The angle of the inner base sheets $\beta_{\text{base,i}}$ is calculated such that the cusp is at the same z-coordinate as the maximum

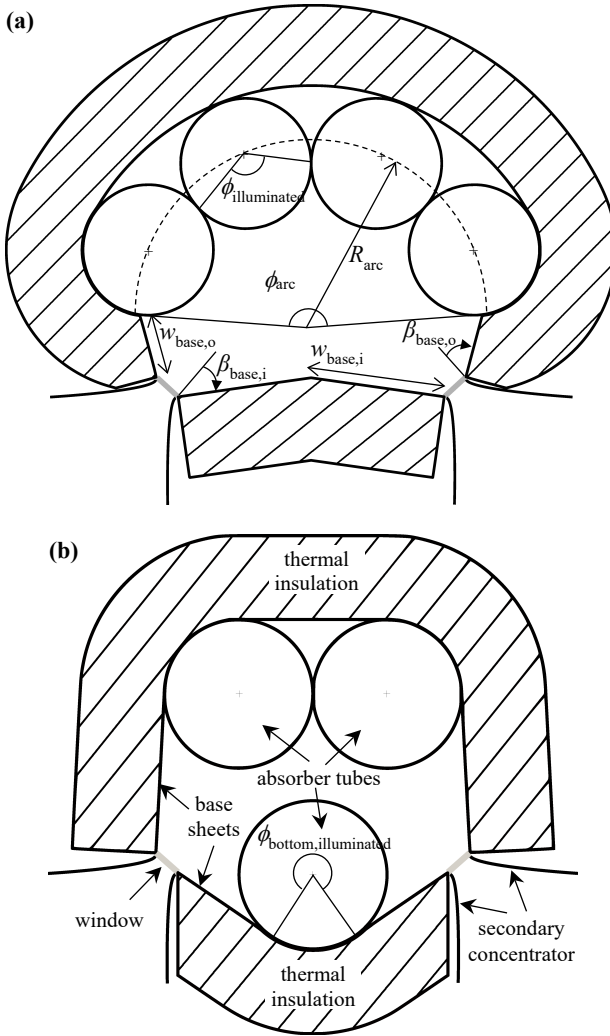


Fig. 5.2: Multiple parallel straight tubes solar receivers and geometric parameters without (a) and with bottom tube (b).

window height to avoid direct radiation spillage through the windows in the absence of a bottom tube. In the case of a bottom tube, the inner base sheets are aligned tangentially to the tube. The bottom tube's position is defined by its

illuminated segment angle $\phi_{\text{bottom,illuminated}}$ such that it receives the same amount of solar radiation as the upper tubes and direct solar irradiation on the outer base sheets is eliminated. For the single straight tube design, the angles β_{base} are chosen as a tradeoff between maximizing the illuminated surface area of the absorber tube and minimizing direct solar irradiation on the base sheets.

5.2.2 Heat transfer modeling

The relevant heat transfers are indicated in the schematic cross-section shown in Fig. 5.3. The steady-state energy conservation is again given by Eq. (2.1), and the application of the modelling techniques introduced in Section 2.3 to straight tubular receivers is straightforward.

Radiation

The solar radiation incident on the receiver window ($\dot{Q}_{\text{solar,incident}}$), reflected ($\dot{Q}_{\text{solar,reflected}}$) and absorbed ($\dot{Q}_{\text{solar,absorbed}}$) by the solar receiver are simulated by MC ray-tracing as outlined in Section 2.3. The thermal radiative exchange between the illuminated absorber tube segments, base sheets and semi-transparent window is calculated using the gray-band approximated radiosity method Eq. (2.9). Thermal radiation inside the absorber tubes is considered by Eq. (2.12). All view factors are calculated by MC ray-tracing exploiting symmetry and reciprocity relations. Thermal emission from the external window surface to the surroundings is given by Eq. (2.11).

Conduction

Heat conduction in the receiver window, base sheets and absorber tubes is governed by the 2D steady-state heat conduction equation (2.2) with boundary conditions at surfaces #1–5 given by Eq. (2.3). Heat losses from surface #6 through the insulation are calculated as 1D heat conduction using Eq. (2.5) for straight surfaces and Eq. (4.11) for curved surfaces. Heat transfer in trough direction is neglected because the temperature gradients in axial direction are usually below 3 °C/m.

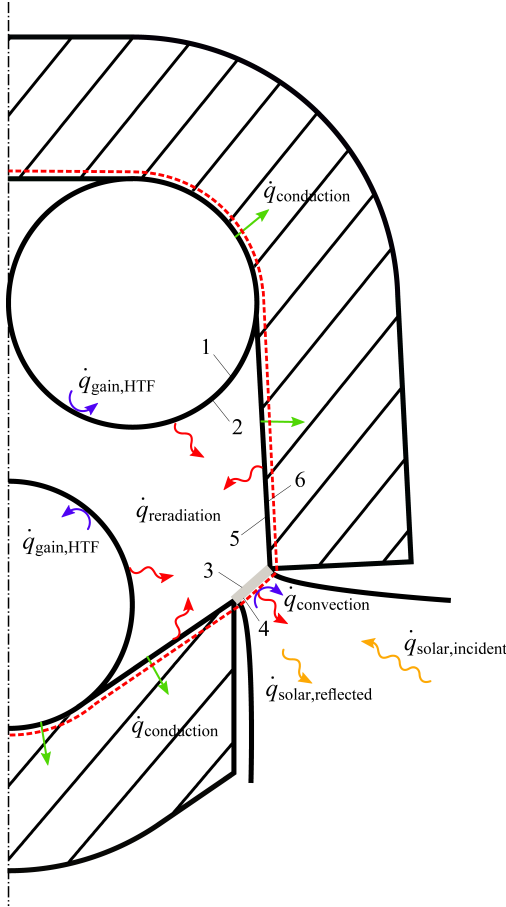


Fig. 5.3: Schematic cross-section of straight tubular solar receiver with surface numbering, system boundary (red dashed line), and relevant heat transfers.

Convection

Natural convection from the external window surface to the environment is calculated by Eq. (2.19). Forced convection from the internal surface of the absorber tubes to the HTF is calculated by the Gnielinski equation (4.6) for fully developed turbulent flow with the Darcy friction factor supplied via the correlation by Petukhov for smooth pipes [109]

$$f = (0.790 \ln \text{Re}_D - 1.64)^{-2} \quad (10^4 < \text{Re} < 5 \times 10^6) \quad (5.7)$$

Discretization and numerical solution

The window, base sheets and absorber tubes are discretized into finite half-volumes with the centroids located at the surface. The symmetry of the solar receiver is exploited to reduce computational cost. Furthermore, the receiver is divided into N axial sections, and the 2D steady-state energy conservation equation for all control volumes is solved successively for every axial section using the recursion formula

$$T_i = T_{i-1} + \frac{\dot{q}_{\text{gain,HTF},i}}{\dot{m}c_p} \quad (5.8)$$

where T_i is the HTF outlet temperature of the i^{th} axial section and $T_0 = T_{\text{inlet}}$. Average heat transfers per unit length are computed by integration of the corresponding local rates over the receiver's length

$$\dot{Q}' = \frac{1}{L} \int_0^L \dot{q}' dy \quad (5.9)$$

5.3 Results

Straight tubular receivers are dimensioned and numerically simulated for two different technical applications: electricity generation and industrial process heat. The optical properties of the semi-transparent topsheet, primary and secondary concentrators, and receiver window are chosen as in Section 3.2. The external receiver surfaces #6 are insulated by a 0.2 m-thick layer of thermal insulation with three different thermal conductivities: $k_{\text{ins}} = 0$ W/(m K), yielding the maximum theoretical performance if the receiver were perfectly insulated; $k_{\text{ins}} = 0.05$ W/(m K), representing a realistic case; and $k_{\text{ins}} = 0.10$ W/(m K), a conservative case.

5.3.1 Electricity generation

For electricity generation via a Rankine cycle steam turbine, the operating temperatures of the solar air receiver are fixed at $T_{\text{inlet}} = 250$ °C, $T_{\text{outlet}} = 570$ °C (cf. Section 3.2). Fig. 5.4 shows the average heat losses per unit length through the windowed aperture $\dot{Q}'_{\text{heat losses}} = \dot{Q}'_{\text{reradiation}} + \dot{Q}'_{\text{convection}}$ as a function of E_{DNI} (a) and θ_{skew} (b) for straight tubular receivers with different numbers of tubes. Their

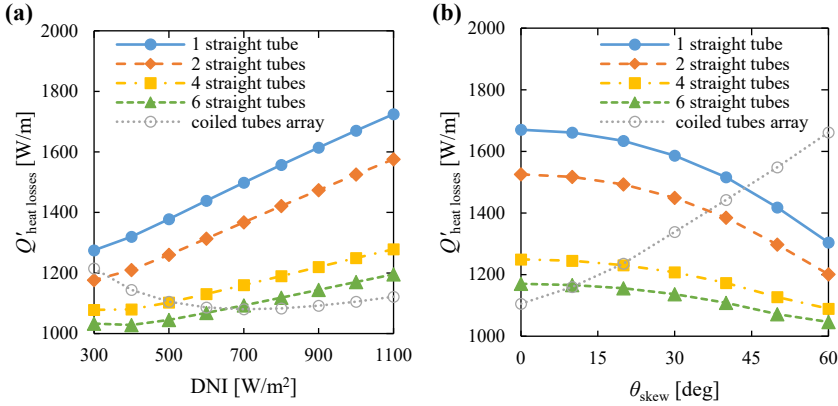


Fig. 5.4: Average heat losses through the windowed aperture ($k_{\text{ins}} = 0$ W/(m K)) per unit length for various numbers of straight tubes and the array of coiled tubes as a function of a) E_{DNI} at $\theta_{\text{skew}} = 0^\circ$, and b) θ_{skew} at $E_{\text{DNI}} = 1000$ W/m².

Table 5.1: Geometric parameters of four straight tubular receivers with different numbers of absorber tubes and an array of coiled tubes with collecting manifold pipe designed for operating temperatures of 250/570 °C at the receiver's inlet/outlet.

Parameter	1 tube	2 tubes	4 tubes	6 tubes	Array of coiled tubes	
$w_{\text{base,i}}$ [m]	0.263	0.260	0.260	0.260	N_{loop}	11
$w_{\text{base,o}}$ [m]	0.263	0.166	0.089	0.034	$D_{\text{tube,o}}$ [m]	0.015
$\beta_{\text{base,i}}$ [deg]	35	39.4	39.4	39.4	D_{coil} [m]	0.100
$\beta_{\text{base,o}}$ [deg]	35	42.5	66.5	42.5	L_{tube} [m]	3.46
ϕ_{arc} [deg]	0	121	199	248	n'_{coil} [1/m]	8.70
R_{arc} [m]	inf	0.395	0.372	0.384	m'_{coil} [kg/m]	11.0
N_{tube}	1	2	4	6	collecting pipe	
$D_{\text{tube,o}}$ [m]	0.510	0.398	0.313	0.271	$D_{\text{collect,0}}$ [m]	0.55
$\phi_{\text{illuminated}}$ [deg]	180	120	130	139	m'_{collect} [kg/m]	6.91
m'_{tube} [kg/m]	6.4	10.0	15.7	20.4	$m'_{\text{coil+collect}}$ [kg/m]	17.9
a_{ins} [m]	1.85	2.31	2.70	2.98	a_{ins} [m]	2.39

geometric parameters are summarized in Table 5.1. In contrast to the array of coiled tubes, the straight tubular absorbers show decreasing $\dot{Q}'_{\text{heat losses}}$ also

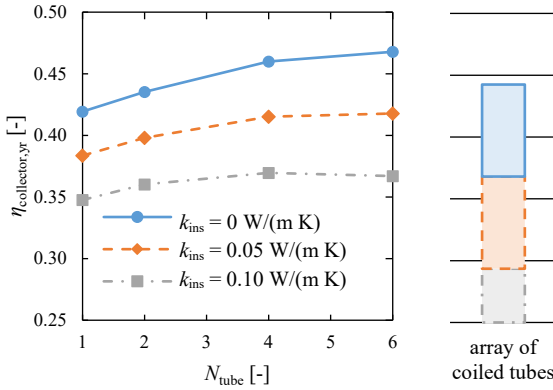


Fig. 5.5: Annual mean solar collector efficiencies with straight tubular receivers as a function of number of tubes for different thermal conductivities of insulation in Sede Boqer, 2011.

towards lower DNI values (cf. Fig. 5.4a) because they achieve the temperature gain of the HTF over the full receiver length and do not rely on the formation of a temperature gradient in cross-flow direction (cf. Fig. 2.7). Furthermore, the negative effect of θ_{skew} is eliminated due to the transition from a 3D array to a 2D groove solar receiver geometry (cf. Fig. 3.4).

Fig. 5.5 shows the annual mean solar collector efficiencies as a function of the number of absorber tubes for the three values of k_{ins} . The efficiencies are calculated for Sede Boqer, Israel, using the same methodology as in Chapter 3. Dividing the flow into multiple parallel channels is beneficial up to $N_{\text{tube}} \approx 4$, above which the increasing conduction heat losses through the insulation start to offset the enhanced convection heat transfer. Conduction heat losses are proportional to the receiver's external surface area that needs to be covered by thermal insulation ($\dot{Q}'_{\text{conduction}} \propto a_{\text{ins}}$), which is given in Table 5.1 in units $[\text{m}^2/\text{m}] = [\text{m}]$. For comparison, the 1D heat conduction model of the thermal insulation of the straight tubular receivers is applied one-to-one to the surrounding groove walls and collecting manifold pipe of the coiled absorber tubes. The annual mean collector efficiencies obtained for the array of coiled tubes are indicated in Fig. 5.5 by the stacked column. The significantly higher sensitivity of the cross-flow

Table 5.2: Geometric parameters of straight tubular receivers without and with an additional tube at the bottom (+1) and an array of coiled tubes with collecting manifold pipe designed for operating temperatures of 25/600 °C at the receiver's inlet/outlet.

Parameter	2 tubes	2+1 tubes	4 tubes	4+1 tubes	Array of coiled tubes	
$w_{\text{base,i}} [\text{m}]$	0.259	0.214	0.260	0.229	N_{loop}	11
$w_{\text{base,o}} [\text{m}]$	0.250	0.300	0.126	0.117	$D_{\text{tube,o}} [\text{m}]$	0.012
$\beta_{\text{base,i}} [\text{deg}]$	39.4	81.8	39.4	78.2	$D_{\text{coil}} [\text{m}]$	0.080
$\beta_{\text{base,o}} [\text{deg}]$	47	39.5	57.5	48.5	$L_{\text{tube}} [\text{m}]$	2.77
$\phi_{\text{arc}} [\text{deg}]$	33	0	166	164	$n'_{\text{coil}} [1/\text{m}]$	10.9
$R_{\text{arc}} [\text{m}]$	1.125	inf	0.356	0.413	$m'_{\text{coil}} [\text{kg/m}]$	8.70
N_{tube}	2	3	4	5	collecting pipe	
$D_{\text{tube,o}} [\text{m}]$	0.323	0.280	0.252	0.233	$D_{\text{collect,0}} [\text{m}]$	0.454
$\phi_{\text{illuminated}} [\text{deg}]$	163.5	180	139	139	$m'_{\text{collect}} [\text{kg/m}]$	5.71
$\phi_{\text{bottom,illuminated}} [\text{deg}]$	-	291	-	298	coiled tubes + pipe	
$m'_{\text{tube}} [\text{kg/m}]$	8.1	10.6	12.7	14.6	$m'_{\text{coil+collect}} [\text{kg/m}]$	14.4
$a_{\text{ins}} [\text{m}]$	1.94	1.92	2.31	2.18	$a_{\text{ins}} [\text{m}]$	1.92

receiver design to k_{ins} originates from the heat losses and temperature drop in the collecting pipe, which requires the coiled absorber tubes to operate at temperatures $T_{\text{coil,out}} > T_{\text{outlet}}$. For $k_{\text{ins}} = 0.05 \text{ W/(m K)}$, the annual pumping power amounts to 1.5% of yearly electric power output for all solar receiver designs (assuming $\eta_{\text{th-el}} = 0.35$ and $\eta_{\text{pump}} = 0.8$).

5.3.2 Industrial process heat

For process heat applications, air is assumed to be fed in from ambient ($T_{\text{inlet}} = 25 \text{ °C}$), which eliminates the costs for the return pipe, and heated to $T_{\text{outlet}} = 600 \text{ °C}$. Four different solar receivers have been designed for these operating conditions, two of them featuring an additional absorber tube at the bottom (+1). The geometric parameters are summarized in Table 5.2. Fig. 5.6 shows the local heat losses through the windowed aperture (a) and thermal efficiencies $\eta'_{\text{th}} = 1 - \dot{q}'_{\text{heat losses}} / \dot{q}'_{\text{solar,absorbed}}$ (b) as a function of the HTF temperature in the tubes for the perfectly insulated case. Despite the smaller heat transfer surface area the 2+1 tubes design loses less heat by re-radiation and

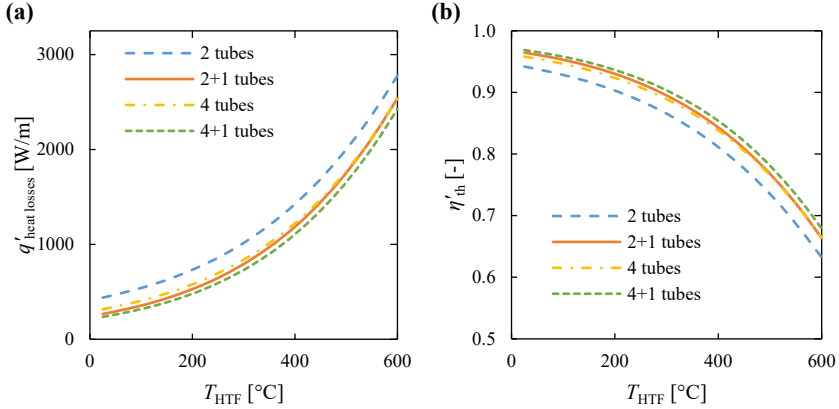


Fig. 5.6: Local heat losses (a) and thermal efficiency (b) as a function of air temperature for different straight tubular receivers with $k_{\text{ins}} = 0 \text{ W/(m K)}$ at $E_{\text{DNI}} = 1000 \text{ W/m}^2$ and $\theta_{\text{skew}} = 0^{\circ}$.

convection than the 4 tubes design. This is due to several advantages of the bottom tube: 1) it captures concentrated solar radiation that would otherwise hit the outer base sheets; 2) it captures thermal radiation from the upper tubes that would otherwise hit the inner base sheets; and 3) it allows for more compact solar receiver designs, which also decreases conduction heat losses through the insulation (cf. a_{ins} in Table 5.2).

The annual mean solar collector efficiencies for the three different values of k_{ins} are given in Table 5.3 for the four straight tubular receivers and the array of coiled tubes with the baseline geometry. The specific pumping power $\varepsilon_{\text{pump, yr}}$ remains below 0.5% in all cases. Under these operating conditions, the straight tubular receivers seem to outperform the cross-flow receiver, particularly for less performing thermal insulation. It is noted, however, that the thermally insulated surface of the straight tubular receivers is mostly convex while the cross-flow receiver includes concave corners between the coiled tubes and the collecting pipe, which is not captured by the surface area parameter a_{ins} . Accordingly, the 1D heat conduction model of the thermal insulation for the straight tubular receivers is extended to 2D, as outlined in Section 4.1.3 for the array of coiled tubes. The results for the 2+1 straight tubes are also included in Table 5.3.

Table 5.3: Annual mean collector efficiencies $\eta_{\text{collector,yr}}$ in Sede Boqer (2011) of straight tubular receivers and the array of coiled absorber tubes with 1D thermal insulation layers of various conductivities k_{ins} , and a 2D thermal insulation for the 2+1 tubes.

k_{ins} [W/(m K)]	0.00	0.05	0.10
2 straight tubes	0.452	0.424	0.395
2+1 straight tubes	0.471	0.444	0.416
4 straight tubes	0.471	0.441	0.412
4+1 straight tubes	0.479	0.446	0.413
array of coiled tubes	0.437	0.342	0.255
2+1 straight tubes (2D thermal insulation)	0.471	0.415	0.358

5.3.3 Comparison with other receivers

Finally, the straight tubular receiver concept is compared with the directly irradiated cylindrical groove proposed in [37], and the real cross-flow receiver of Chapter 4 based on the array of coiled tubes. Two different absorber tube configurations, the most straightforward 1 tube and the most promising 2+1 tubes designs, with a 0.2 m thick layer of thermal insulation ($k_{\text{ins}} = 0.05$ W/(m K)) are selected for this comparison. The cross-flow receiver with coiled absorber tubes is simulated using the validated heat transfer model of Section 4.3 including the technical improvements. The efficiency of the directly irradiated cylindrical groove is calculated using numerically simulated heat loss data from [37]. For comparability, the smooth cylindrical groove with a single-glazed (SGW) or double-glazed window (DGW) is selected. Numerical simulations are performed for $E_{\text{DNI}} = 847$ W/m², $\theta_{\text{skew}} = 13.9^\circ$ corresponding to solar noon on summer solstice in Sevilla, Spain [37]. Given the close resemblance of the optical designs of all solar receivers in terms of geometric concentration ratio and optical components, their optical efficiencies are adjusted to the case of a cylindrical groove with a SGW aperture ($\eta_{\text{optical},0} = 0.697$, $\text{IAM}(13.9^\circ) = 0.965$ [37]) such that the receivers absorb the same solar flux per m² aperture. The solar receivers are operated with ambient air at $T_{\text{inlet}} = 25$ °C and deliver process heat at $T_{\text{outlet}} = 500$ °C with $\dot{m}$ (L for the cylindrical grooves, respectively) adjusted

Table 5.4: Design and performance parameters of different air receivers for solar trough concentrators.

	Array of coiled tubes	Straight tubes		Cylindrical groove	
		1 tube	2+1 tubes	SGW	DGW
collector length L , m	212	212	212	247	250
aperture width a_{primary} , m	9.7	9.7	9.7	9.0	9.0
geometric concentration C_g	97.0	97.0	97.0	97.5	97.5
inner tube diameter D_i , m	0.453 ^a	0.419	0.279	0.45	0.45
mass flow $\dot{m}$, kg/s	1.99	1.81	1.91	2	2
pressure drop Δp , mbar	11.5 ^b	10.8	11.6	10.7 ^c	10.8 ^c
thermal efficiency, η_{th}	0.84	0.76	0.80	0.78	0.77 ^d
specific pumping power $\varepsilon_{\text{pump}}$	0.70%	0.66%	0.71%	0.65%	0.66%

^a Inner diameter of collecting manifold pipe.

^b Sum of pressure drops occurring in coiled tube and collecting pipe.

^c Calculated from pressure drop of single straight tube using Eq. (5.3).

^d Effective η_{th} calculated as the product of actual η_{th} (0.83) and η_{optical} of the additional window (0.93).

accordingly. The ambient and apparent sky temperatures are $T_{\text{amb}} = 45^\circ\text{C}$ and $T_{\text{sky}} = 1.85^\circ\text{C}$, respectively. The key parameters and results for the different receiver designs are given in Table 5.4.

5.4 Conclusion

Multiple parallel straight tubes were applied as solar absorbers in a receiver for line-focus solar radiation using air as heat transfer fluid. The thermodynamics of such receivers were investigated numerically coupling radiation, conduction, convection, and friction in the tubes. Flow division into multiple tubes reduces effectively heat losses through the windowed aperture while heat losses through the thermal insulation are increased, yielding an optimum around three tubes for the investigated conditions. In contrast to the array of 3D coiled tubes, straight tubular receivers are insensitive to the solar incidence angle due to their 2D groove-like geometry, which can allow for higher annual mean thermal efficiencies. At solar noon on summer solstice in Sevilla, Spain, the straight tubular receivers and the directly irradiated cylindrical grooves show similar thermal efficiencies, while the cross-flow array of coiled absorber tubes

outperforms the 2D groove-like receivers by 5 – 10% under these high-DNI low-skew conditions.

Besides efficiency, other factors such as manufacturability, operability, and costs require consideration. The directly irradiated cylindrical groove is the most compact receiver design requiring the least amount of thermal insulation. However, the heat transfer fluid is in direct contact with the window, which poses a series of challenges: 1) The window fixture needs to be airtight and accommodate thermal expansion. 2) Dust particles in the HTF might deposit on the window and degrade its solar transmittance and lifetime. 3) Expensive quartz glass is required in the high-temperature region of the solar receiver. These problems are mitigated by the tubular receivers, in which the window acts merely as a radiation shield and convection barrier. Besides the single straight tube design, cross-flow receivers are generally more compact and require less thermal insulation. Another cost indicator is the mass of high-temperature stainless steel m'_{tube} used per meter tube length. m'_{tube} in kg/m of receiver is given in Table 5.1 and Table 5.2 for the straight absorber tubes and coiled tubes with collecting pipe, assuming a constant wall thickness of 0.5 mm for all the tubes. While the tubing of the cross-flow receiver seems rather heavy, its weight can potentially be reduced by further decreasing the wall thickness of the small-diameter coiled tubes. From an assembly point of view, straight tubular receivers appear favorable because of a much lower number of parts. Finally, it is noted that neither the straight tubular receivers nor the directly irradiated cylindrical groove have been fabricated such that further design issues, cost and performance penalties might arise during their construction.

6 Arcspline solar concentrator

In this chapter, the pneumatic arcspline concentrator (ASC) described in Sections 4.1.1 and 4.2.1 is designed for maximizing the solar power input to the secondary optics and solar receiver. The theory of a new mirror regulation mechanism based on inflatable sleeves is developed and extended to include weight effects. Experimental results from the operation of the industrial scale solar concentrator are presented, from which conclusions are drawn for further improvement. In addition, a model-based control system for the automatic regulation of inflation pressures is described.

6.1 Theory

6.1.1 Optical design

A schematic cross-section of one wing of the ASC is shown in Fig. 4.2a. The nominal geometry of the ASC is determined using a direct optical approach [14]. The design procedure of an ASC for a flat horizontal receiver that achieves the theoretical geometric concentration ratio of a parabola with a minimum number of arcs is outlined in [83]. Here, the ASC is designed for a tilted receiver given the mechanical constraints of 1) $N = 4$ circular arcs; 2) fixed coordinates of the focus $\mathbf{F}(F_x, F_z)$ and outer rim point $\mathbf{N}_4(N_{x,4}, N_{z,4})$; and 3) given inlet apertures a_{primary} and a_{in} of the primary and secondary concentrators, respectively, and focal plane tilt angle Φ_{tilt} . A point on the j^{th} arc is given by

$$\mathbf{P}[x(\varphi), z(\varphi)] = \mathbf{C}_j + R_j(\sin \varphi, -\cos \varphi) \quad (6.1)$$

where $\mathbf{C}_j$ is the center point of the j^{th} circular arc and $\varphi = \arctan(dz / dx)$ the surface slope angle. A ray coming at an angle θ_i measured counterclockwise from the optical z -axis and undergoing specular reflection at point $\mathbf{P}(\varphi)$ is parametrized by

$$\mathbf{r}_{\text{ray}}(\varphi, t; \theta_i) = \mathbf{P}(\varphi) + t[-\sin(2\varphi + \theta_i), \cos(2\varphi + \theta_i)] \quad (6.2)$$

where the parameter t is the distance travelled by the ray after reflection. The tilted focal plane is parametrized by the distance u from the focus,

$$\mathbf{r}_{\text{receiver}}(u; \Phi_{\text{tilt}}) = \mathbf{F} + u(\cos \Phi_{\text{tilt}}, \sin \Phi_{\text{tilt}}) \quad (6.3)$$

Intersecting Eq. (6.2) with Eq. (6.3) yields for the distance of the intersection point from the focus

$$u(\varphi; \theta_i, \Phi_{\text{tilt}}) = \frac{(x(\varphi) - F_x) \cos(2\varphi - \theta_i) + (z(\varphi) - F_z) \sin(2\varphi - \theta_i)}{\cos(\Phi_{\text{tilt}} - 2\varphi + \theta_i)} \quad (6.4)$$

Eq. (6.4) is called the focal function, which needs to fulfill the following condition for full interception [83]

$$\left. \begin{aligned} u(x; \theta_i = +\theta_{\text{src}}, \Phi_{\text{tilt}}) &\leq a_{\text{in}} / 2 \\ u(x; \theta_i = -\theta_{\text{src}}, \Phi_{\text{tilt}}) &\geq -a_{\text{in}} / 2 \end{aligned} \right\} \forall x \in [N_{x,0}, N_{x,N}] \quad (6.5)$$

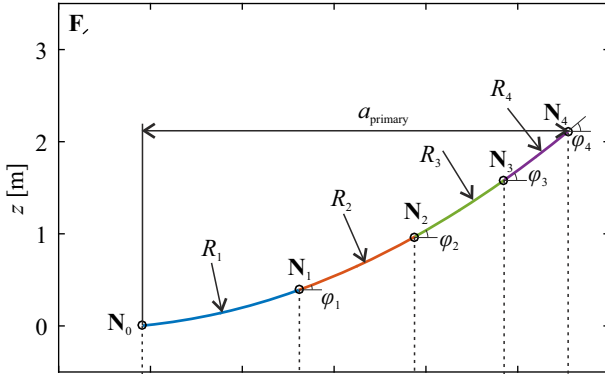
where θ_{src} is the maximum angle of incidence at the primary aperture and $N_{x,0} = N_{x,N} - a_{\text{primary}}$ the inner rim point of the concentrator. The ASC is designed starting from $N_{x,N}$ and ending at $N_{x,0}$. The final result for $\theta_{\text{src}} = 4.75$ mrad is shown in Fig. 6.1. The slope angle φ_N at $\mathbf{N}_N$ is chosen such that $u(N_{x,N}; \theta = -\theta_{\text{src}}, \Phi_{\text{tilt}}) = -a_{\text{in}} / 2$. The radius R_N of the outermost arc is then chosen such that $\max[u(x; \theta = +\theta_{\text{src}}, \Phi_{\text{tilt}})] = +a_{\text{in}} / 2$, which defines the outermost arc. For the most effective use of arcs the N^{th} arc is extended inwards until $u(N_{x,N-1}; \theta = -\theta_{\text{src}}, \Phi_{\text{tilt}}) = -a_{\text{in}} / 2$, where the node $\mathbf{N}_{N-1}(N_{x,N-1}, N_{z,N-1})$ is the outer endpoint of arc $N-1$. Using the tangency condition between arcs and following this procedure until $N_{x,0}$ yields an ASC with full intercept, i.e. all incident solar radiation is focused onto the inlet aperture of the secondary concentrator. The geometric parameters of the nominal ASC are summarized in Table 6.1.

6.1.2 Mechanical design

The ASC is constructed using a stack of N inflatable films as shown for the right wing in Fig. 4.2a. From the nominal arcspline geometry and the given clamping points $\mathbf{A}$ of the supporting films the support arc radii $R_{\text{sup},j}$ are calculated as

$$R_{\text{sup},j} = \frac{(N_{x,j-1} - A_{x,j})^2 + (N_{z,j-1} - A_{z,j})^2}{2[(N_{x,j-1} - A_{x,j}) \sin \varphi_j - (N_{z,j-1} - A_{z,j}) \cos \varphi_j]} \quad (6.6)$$

(a)



(b)

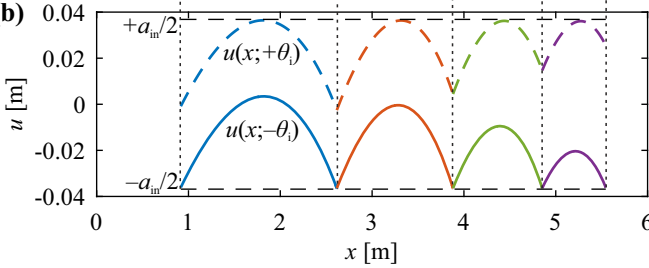


Fig. 6.1: Nominal arcspline designed for full interception of a tilted flat receiver ($\Phi_{\text{tilt}} = 42.5^\circ$, $\theta_{\text{src}} = 4.75$ mrad, $C_{\text{g,primary}} = 62.8\times$) with a minimum number of arcs ($N = 4$): (a) mirror profile $z(x)$; and (b) focal function from Eq. (6.4) for positive (dashed) and negative (solid) edge-rays.

It is noted that for mechanical constructability, the clamping point $\mathbf{A}_1$ of the reflective top film needs to lie on the first arc of the ASC, whose radius R_1 is already determined by the optical design.

Film pressures and tensions

A force balance for each arc of the reflective top film yields N equations,

$$\sum_{k=1}^j F'_k = (p_0 - p_j) R_j \quad (6.7)$$

where F'_j is the tension of the j^{th} film. The same force balance for each support arc yields additional $N - 1$ equations,

$$F'_j = (p_{j-1} - p_j) R_{\text{sup},j} \quad (6.8)$$

p_0 is the inflation pressure of the transparent topsheet and is fixed at $p_0 \approx 130$ Pa. Accordingly, there are $2N - 1$ equations for the $2N$ unknowns (N pressures and N tensions), leaving one degree of freedom for which the differential pressure across the mirror film stack $\Delta p_N = p_0 - p_N$ is chosen. This yields a linear system of equations for the N tensions and $N - 1$ pressures. The analytical solution for the differential pressures is

$$\Delta p_j = p_0 - p_j = \Delta p_N \prod_{k=j+1}^N \frac{(R_{\text{sup},k} - R_k)}{(R_{\text{sup},k} - R_{k-1})} \quad (6.9)$$

Film lengths

The tension relates the film's stretched length S to its unstretched length L_0 according to [14]

$$L_0 = S \cdot e^{-F'/k_0} \quad (6.10)$$

where $k_0 = t_0 E / (1 - \nu^2)$ is the nominal transverse stiffness of the film expressed by the Young's modulus E , Poisson ratio ν and thickness of the unstretched film t_0 for a linearly elastic isotropic material in plane strain stress state [14]. The stretched lengths are calculated from the arcspline and support arcs geometry as

$$S_1 = \sum_{k=1}^N (\varphi_k - \varphi_{k-1}) R_k + L_{\text{fix},1} \quad (6.11)$$

$$S_j = (\varphi_j - \varphi_{\text{sup},0,j}) R_{\text{sup},j} + \sum_{k=j}^N (\varphi_k - \varphi_{k-1}) R_k + L_{\text{fix},j}, \quad j = 2, \dots, N$$

where $L_{\text{fix},j}$ are the fixation lengths of the films at the outer rim point as shown in Fig. 4.2a. The sensitivity of the ASC geometry to deviations in the films' lengths was analyzed quantitatively [13] and qualitatively [14] in previous works. Errors in film length are caused by manufacturing tolerances, thermal and hygroscopic expansion, shrinkage (residual strain-relief of the polymeric films), creep [14], and the Poisson effect by longitudinal tension originating from the film mounting procedure. For an isotropic material, the initial length contraction of the transversally unstretched film ($\sigma_{xx} = F' / t_0 = 0$) by a longitudinally pulling force $F_y = \sigma_{yy} t_0 L_0$ is given by Hooke's law under plane stress, where ε and σ denote strain and stress, respectively.

Table 6.1: Complete optical and mechanical design of the nominal arcspline.

a_{primary} [m]	f [m]	F_x [m]	F_z [m]	Φ_i [°]	Φ_o [°]
4.625	3.246	0.292	3.230	15.017	77.953
E [Pa]	ν [-]	L_0 [m]	t_0 [m]	$L_{\text{sleeve arc}}$ [m]	$L_{\text{sleeve,fix}}$ [m]
5.10E+09	0.38	5.332	2.30E-05	0.190	0.060
index j					
Parameter	0	1	2	3	4
R_j [m]		7.012	8.640	10.743	12.886
φ_j [°]	5.632	19.626	28.918	35.094	38.999
$N_{x,j}$ [m]	0.919	2.586	3.862	4.844	5.545
$N_{z,j}$ [m]	0.018	0.391	0.967	1.580	2.109
$C_{x,j}$ [m]		0.231	-0.316	-1.332	-2.564
$C_{z,j}$ [m]		6.996	8.529	10.370	12.123
$A_{x,j}$ [m]		0.681	0.660	0.683	0.707
$A_{z,j}$ [m]		-0.003	-0.094	-0.292	-0.459
$R_{\text{sup},j}$ [m]			10.376	13.401	14.974
$\varphi_{\text{sup},j}$ [°]			9.932	15.242	18.259
$C_{\text{sup},x,j}$ [m]			-0.899	-2.618	-3.765
$C_{\text{sup},z,j}$ [m]			10.164	12.696	13.832
$L_{\text{fix},j}$ [m]		0.184	0.142	0.103	0.069
S_j [m]		5.334	5.335	5.338	5.347
Δp_j [Pa]		7.1	13.8	24.7	50.0
F'_j [N/m]		49.8	69.2	146.1	379.2
$S_{\text{sleeve},j}$ [m]		0.240	0.235	0.230	0.231
$f_{\text{sleeve},j}$ [°]		65.700	80.838	91.518	90.039
$p_{\text{sleeve},j}$ [mbar]		1.8	3.4	8.8	22.2
$F'_{\text{sleeve arc}}$ [N/m]		29.7	45.4	104.7	268.3

$$\Delta L = \varepsilon_{xx} L_0 = -\frac{\nu F_y}{E t_0} \quad (6.12)$$

Pneumatic sleeves

The diversity and temporal variation of the aforementioned effects call for a compensation mechanism. Too short film lengths can partially be corrected by increasing the pressures p_j , which yields N control variables [13, 14]. The ASC geometry has $2N$ degrees of freedom (N slope angles and N radii), which requires N additional control variables to render the ASC shape fully controllable. For this purpose, inflatable sleeves are introduced between the inner clamping points and mirror films, which allow for length regulation of each film via the sleeve inflation pressure. A cross-section of an inflated sleeve is shown in Fig. 4.2a. 2D force balances on a sleeve arc and on the junction of the two arcs yields for the tension

$$F'_{\text{sleeve arc}} = p_{\text{sleeve}} R_{\text{sleeve}} = \frac{F'_{\text{sleeve}}}{2 \cos(\phi_{\text{sleeve}} / 2)} \quad (6.13)$$

The chord length of the sleeve arc $S_{\text{sleeve,arc}}$ is a function of the circular segment angle ϕ_{sleeve} and the constant sleeve arc length $L_{\text{sleeve,arc}}$

$$S_{\text{sleeve arc}} = L_{\text{sleeve arc}} \frac{2 \sin(\phi_{\text{sleeve}} / 2)}{\phi_{\text{sleeve}}} \quad (6.14)$$

where strain of the sleeve arcs is negligible. For a desired $S_{\text{sleeve,arc}}$, Eq. (6.14) needs to be solved numerically for the exact solution of ϕ_{sleeve} . To facilitate design, the sine function in Eq. (6.14) can be accurately approximated over the relevant range $[0, \pi/2]$ by a 5th order Taylor series at 0, which yields the analytical expression

$$\phi_{\text{sleeve}} = 2 \sqrt{10 - 2 \sqrt{30 \frac{S_{\text{sleeve arc}}}{L_{\text{sleeve arc}}} - 5}} \quad (6.15)$$

The required sleeve pressure is given by

$$p_{\text{sleeve}} = \frac{F'_{\text{sleeve}}}{S_{\text{sleeve arc}}} \tan(\phi_{\text{sleeve}} / 2) \quad (6.16)$$

For $p_{\text{sleeve}} = 0$ $\phi_{\text{sleeve}} = 0$ (flat sleeve), and for $p_{\text{sleeve}} \rightarrow 0$ $\phi_{\text{sleeve}} = \pi$ (circular sleeve) such that sleeve length regulation is theoretically possible on the interval $S_{\text{sleeve arc}} = (2L_{\text{sleeve arc}} / \pi, L_{\text{sleeve arc}}]$. In practice, the maximum inflation pressure of

the sleeves imposes a lower limit on $S_{\text{sleeve arc}}$. The pressures and tensions for the nominal geometry of the ASC are given in Table 6.1.

Effect of weight

So far, the films, sleeves, and film-sleeve-fixtures have been considered as weightless. The area-specific weight of the 23 μm thick boPET films is 0.31 Pa, which is at least one order of magnitude smaller than the differential pressures on the films and is thus negligible. The shape of a single film under combined weight and uniform pressure is derived in [14]. The area-specific weight of the 0.5 mm thick PVC sleeve arcs is 6.9 Pa, which is at least one order of magnitude smaller than the sleeve inflation pressures and thus negligible for the calculation of the sleeve shape. Conversely, the length-specific weight of the film-sleeve-fixtures in the real system is 24 N/m, which is comparable to the film tensions and requires further analysis. Considering a point mass at the film-sleeve junction with weight F'_{weight} , the 2D force balance at the junction reads

$$\begin{aligned} F' \cos \varphi_0 - F'_{\text{sleeve}} \cos \varphi_{\text{sleeve}} - F'_{\text{weight}} \sin \beta_{\text{track}} &= 0 \\ F' \sin \varphi_0 - F'_{\text{sleeve}} \sin \varphi_{\text{sleeve}} - F'_{\text{weight}} \cos \beta_{\text{track}} &= 0 \end{aligned} \quad (6.17)$$

where φ_0 and φ_{sleeve} are the slope angles of the film and sleeve at node $\mathbf{N}_0$, and β_{track} the tracking angle of the solar concentrator. Solving Eq. (6.17) for the sleeve tension and slope angle yields

$$\begin{aligned} F'_{\text{sleeve}} &= \left(F'^2 + F_{\text{weight}}'^2 - 2F'_{\text{weight}} F' \sin(\varphi_0 + \beta_{\text{track}}) \right)^{1/2} \\ \varphi_{\text{sleeve}} &= \arcsin \left(\frac{F' \sin \varphi_0 - F'_{\text{weight}} \cos \beta_{\text{track}}}{F'_{\text{sleeve}}} \right) \end{aligned} \quad (6.18)$$

In this case, the film and sleeve pressures required to establish the desired ASC geometry have to be calculated numerically by simultaneously solving Eqs. (6.7) to (6.8), (6.13) and (6.18). For $F'_{\text{weight}} = 24 \text{ N/m}$, the theoretically calculated pressures are far too high to be reached in practice and would cause prohibitive tensions in the films. The reason is that the innermost node $\mathbf{N}_0(N_{x,0}, N_{z,0})$ of the reflective top film is pulled down by the fixture weight and could only be maintained on the first arc of the nominal ASC geometry using high film pressures and tensions that compensate F'_{weight} . Accordingly, the nominal ASC

shape of Table 6.1 cannot be established in the real system. With N_0 too low, the average mirror slope to the outer rim point N_4 is generally too steep, which can be compensated in the real system by lowering the solar receiver, i.e. the focus $F(F_x, F_z)$. Mathematically, the problem can be formulated as finding the focal length f and focus height F_z of a parabola $z(x) = F_z - f + (x - F_x)^2 / (4f)$ that passes through the points $N_0(N_{x,0}, N_{z,0})$ and $N_4(N_{x,4}, N_{z,4})$. The solution is:

$$f = \frac{(N_{x,4} + N_{x,0} - 2F_x)(N_{x,4} - N_{x,0})}{4(N_{z,4} - N_{z,0})} \quad (6.19)$$

$$F_z = N_{z,0} + f - (N_{x,0} - F_x)^2 / (4f)$$

Based on these calculations and initial concentrator shape measurements performed on the real system, the solar receiver has been lowered by 35 mm and a new nominal ASC for the new focus position has been designed according to Section 6.1.1 (cf. Table 4.1).

6.2 Experimental

Off-sun concentrator shape measurements on the first 212 m-long solar collector unit have been performed every 8.82 m using the camera system described in Section 4.2.3, and the inflation pressures were adjusted manually based on the theory in Section 6.1.2. The concentrator shape scans for the left and right wings at one axial position and calculated intercept factors $\hat{\gamma}_{\text{primary}}$ at all 24 locations after pressure regulation are shown in Fig. 4.9 a and b, respectively. The average intercept factor was 0.78.

Beam deflection

The arcspline geometry variation within a collector module is investigated by shape measurements performed in axial intervals of 1 m. The intercept factors for the right wing of collector module #4 are shown in Fig. 6.2 as a function of the axial position measured from the head of the concrete beam. Intercept variations are in the range 0.80 ± 0.14 . The deflection of the central beam is calculated from the mirror shape scans via the concentrator's slope deviation from a reference parabola,

$$\Delta z = \int_{a_{\text{primary}}} -\Delta m \, dx = \int_{a_{\text{primary}}} \left(\tan(\varphi_{\text{parabola}} + \theta_r / 2) - \tan \varphi_{\text{parabola}} \right) dx \quad (6.20)$$

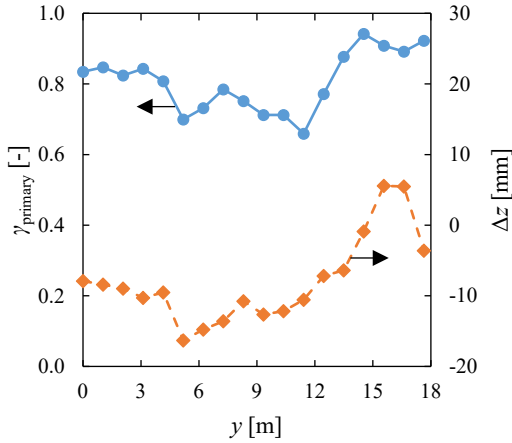


Fig. 6.2: Collector module #4, right wing. Intercept factors (circular, solid) and deflection (diamonds, dashed) calculated from mirror shape scans performed in spatial intervals of 1 m.

The result is also shown in Fig. 6.2, indicating some deflection towards the middle of the beam. Distortions of the deflection curve are explained by longitudinal height variations of the fixing points of the solar receiver, the pneumatic sleeves at the central beam, and the mirror films at the external beam. For restricting the average errors of the concentrator's surface slope and reflected ray direction to 0.5 and 1 mrad, respectively, Δz should be limited to ± 3 mm.

Solar tracking

The effects of operating conditions in solar tracking on concentrator performance are assessed by aperture scans acquired during solar tracking under the shaded region III. The tracking angle, ambient temperature inside the collector, and intercept factors calculated from the mirror shape scans over time are shown in Fig. 6.3. The intercept factors during solar tracking decrease roughly by factors of 0.75 and 0.8 for the left and right wings. The reasons for this significant reduction are classified into temperature effects (elongation of mirror films due to thermal expansion) and gravity effects (weight influence of pneumatic sleeves, variable deflection of concrete beams, and receiver displacement due to weight).

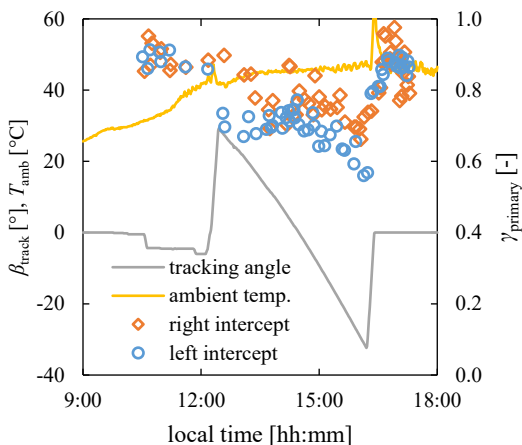


Fig. 6.3: Collector module #4 left and right wings. Tracking angle and ambient temperature of the collector (left vertical axis) and intercept factors calculated from concentrator shape scans taken during solar tracking (right vertical axis).

Model-based control system

The substantial variations of the solar concentrator's performance during solar tracking call for a control system that automatically adjusts the inflation pressures of the mirror films and sleeves to the actual conditions. The simplest case is a feedforward controller that adjusts the pressures in the arcspline chambers to the actual temperature and tracking angle of the solar collector based on empirical data. A more refined method uses a feedback control system that regulates the local sleeve pressures based on concentrator shape measurements. This approach requires a measuring system and an optical-mechanical model of the ASC to compute the pressure adjustments. The concentrator shape scans provide complete information about the ASC geometry as long as all circular arcs are distinguishable. Accordingly, this optical measuring technique could readily be applied in such a control system. A model that calculates the resulting ASC geometry for a given set of pressures is obtained by solving the force balances in Eqs. (6.7), (6.8), (6.13) and (6.18). The large number of geometric parameters (focus and clamping point positions, film lengths, sleeve weight, camera alignment) and associated tolerances generally require a parameter

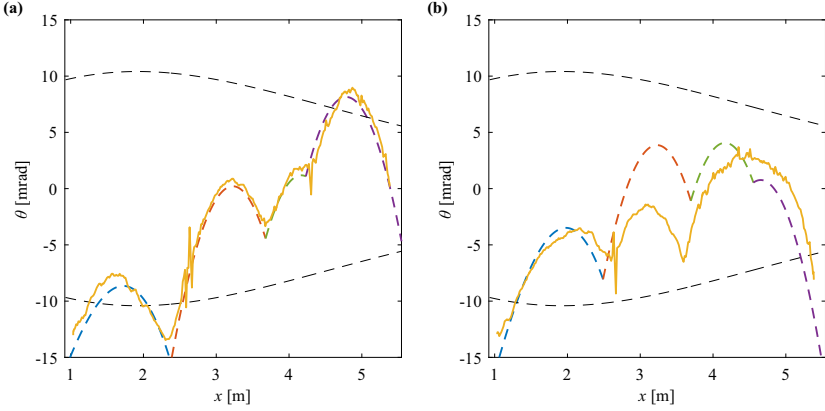


Fig. 6.4: Collector module #5 right wing. Experimentally measured (solid) and computed (dashed arcs) mirror shape scans: a) model calibration run; b) model prediction for a decrease of all sleeve pressures.

identification for model calibration. The highest uncertainties and temporal fluctuations are attributed to the unstretched mirror film lengths L_0 . Accordingly, the model is calibrated by identifying L_0 for the $N = 4$ films by least-square minimization. The agreement between the calibrated model and the experimental shape measurement used for parameter identification is shown in Fig. 6.4a. The concentrator shapes measured experimentally after an adjustment of all sleeve pressures by 1 – 6 mbar and predicted by the model are shown in Fig. 6.4b.

The calibrated model is used to quantify the effects of the film and sleeve pressures $v = [\Delta p_f, p_{\text{sleeve}_f}]$ (control variables) on the concentrator shape $w = [N_{x,1}, \dots, N_{x,N-1}, \theta_{r,1-}, \theta_{r,1}, \dots, \theta_{r,N-1}, \theta_{r,(N-1)+}]$ (measured output variables), where $\theta_{r,1-}$ and $\theta_{r,(N-1)+}$ are the measured angular deviations from the focus at the inner and outer x -limits of a mirror shape scan, respectively. The identified matrix $\mathbf{J} = \partial w / \partial v \approx \Delta w / \Delta v$ of the pneumatic ASC system, linearized about its nominal operating point given in Table 6.1, is provided in Table 6.2. The required pressure corrections are calculated by matrix inversion, $\Delta v = \mathbf{J}^{-1} \Delta w$, where Δw are the deviations of the actual from the nominal ASC shape. For large deviations from the nominal operating point, the strong non-linearity of the pneumatic ASC system needs to be considered. It is further noted that the recognition of all

Table 6.2: Matrix of sensitivities of nodal x-coordinates N_x and angular deviations θ_r of reflected rays from focus to changes in differential film pressures Δp and sleeve pressures p_{sleeve} .

		[Pa]				[mbar]			
	$\partial w / \partial v$	Δp_1	Δp_2	Δp_3	Δp_4	$p_{\text{sleeve},1}$	$p_{\text{sleeve},2}$	$p_{\text{sleeve},3}$	$p_{\text{sleeve},4}$
[cm]	$N_{x,1}$	40.4	-24.0	-0.3	0.0	-14.5	11.0	0.1	0.0
	$N_{x,2}$	-0.5	23.3	-14.4	-0.1	-5.9	-5.3	3.4	0.0
	$N_{x,3}$	0.0	-0.9	15.5	-9.0	-2.6	-2.7	-2.0	1.4
[mrad]	$\theta_{r,1-}$	16.1	-0.9	-1.5	-0.7	-12.7	1.7	0.6	0.1
	$\theta_{r,1}$	-10.5	4.3	1.8	0.3	5.3	-3.8	-0.5	0.0
	$\theta_{r,2}$	0.3	-4.4	0.1	1.0	2.7	2.2	-0.5	-0.2
	$\theta_{r,3}$	0.0	0.4	-1.9	-0.4	0.9	1.0	0.7	0.0
	$\theta_{r,3+}$	0.0	0.0	0.9	-2.6	0.5	0.6	0.5	0.4

circular arcs in a shape scan can be difficult in practice, particularly when operating far away from the nominal ASC geometry.

6.3 Conclusion

The arcspline solar concentrator was designed for maximum intercept factor with the solar receiver's secondary concentrator in a tilted focal plane. The theory of the film length regulation mechanism based on inflatable sleeves was developed, including the effect of gravity. The weight of the film-sleeve-fixtures in the real system was found to distort the shape of the ASC significantly such that the optical design could not be exactly established in practice. In addition, concrete beam deflection, receiver displacement and temperature variations during solar tracking were found to decrease the concentrator's performance. To overcome these challenges, a model-based control system is proposed that automatically adjusts the inflation pressures according to recurring mirror shape measurements.

7 Optical material characterization^{1,2}

7.1 Introduction

Knowledge of the optical properties of reflective materials is of paramount importance for the design of solar concentrating systems. Of interest is the reflectance weighted by the solar irradiance spectrum, which strongly affects the attainable solar concentration ratio. The spectral directional-hemispherical reflectance $R_{h,\lambda}$ at near normal incidence, with ASTM G173 – 03 for direct normal irradiance (DNI) at AM 1.5 [46], is employed as standard for concentrating solar collectors [110]. Absolute reflectometers such as the one by NIST was developed to measure bi-directional reflectance in the range 200 – 2500 nm [111]. Measurements of specular reflectance were reported for several reflective materials, including back-silvered glass, metallized polymer films, and polished aluminum [112, 113]. However, measured spectral data over the solar spectrum and at various incidence angles are generally not available for defined acceptance angles suited for solar applications. In addition to the reflectance, measurement of the narrow-angle surface scattering is needed for characterizing the imperfections of the solar concentrator, as these strongly influence the solar flux distributions. Surface scattering is generally understood to be a wave phenomenon of phase differences caused by surface roughness, and it is often treated as diffraction for optically smooth surfaces [114-116]. For solar concentrators, the aim is to find a convenient description of the specular reflected beam shape that can then be further applied for design purposes and performance

¹ Material in this chapter has been published in P. Good, T. Cooper, M. Querci, N. Wiik, G. Ambrosetti, and A. Steinfeld, "Spectral reflectance, transmittance, and angular scattering of materials for solar concentrators," *Solar Energy Materials and Solar Cells*, vol. 144, pp. 509-522, 2016.

² Data in this chapter has been published in P. Good, T. Cooper, M. Querci, N. Wiik, G. Ambrosetti, and A. Steinfeld, "Spectral data of specular reflectance, narrow-angle transmittance and angle-resolved surface scattering of materials for solar concentrators," *Data in Brief*, vol. 6, pp. 184-188, 2016.

predictions, e.g. by Monte Carlo (MC) ray-tracing. The angular scattering distribution may be accurately described by Gaussian probability density functions [112], which are straightforward to integrate into MC simulations. In addition, the standard deviation σ is a quantitative indicator of the severity of scattering and can be compared to that derived from other sources of beam spreading such as surface slope and tracking errors [112]. Reflected beam profiles of silvered polymer films applied to different substrates were measured at various incidence angles using adjustable source and detector slits combined with Fourier transform analysis [113]. More recently, angle-resolved reflectance was measured using quasi-monochromatic LEDs (455, 533, 631 nm) as light source and a CCD detector [117] and the bi-directional reflectance was measured using the principle of a Coblentz sphere [118]. In this work, a spectroscopic goniometry system is employed that enables the spectral and directional measurement of reflectance, transmittance, and scattering with high accuracy over the wide range of wavelengths and incidence angles relevant for solar concentrating applications.

In this chapter, the spectral specular reflectance of conventional and novel solar reflective materials is reported, measured over the solar spectrum (300 – 2500 nm) at incidence angles ranging from 15° to 60°. The angles of source divergence (6 mrad) and detector acceptance (17.5 mrad = 1°) in the plane of incidence are chosen to closely resemble the sun angle (4.65 mrad) and a typical solar receiver's acceptance angle for full interception (1°). In addition, the spectral narrow-angle transmittance of suitable semi-transparent materials for protective covers of solar concentrators is measured with the same experimental setup and characteristic angles. Furthermore, the narrow-angle surface scattering is measured over the wavelength range 350 – 1050 nm, which covers 75% of the DNI (ASTM G173 – 03 AM 1.5 [46]). Finally, the measured optical properties are applied in MC ray-tracing simulations of two solar concentrators, namely a parabolic trough and a parabolic dish, to elucidate their effect on the solar flux distribution and solar concentration ratio at the focal plane, and draw conclusions for further development.

7.2 Materials

7.2.1 Reflective materials

Three types of solar reflective materials are investigated: 1) back-silvered glasses; 2) metallized polymer films; and 3) metallized aluminum sheets. An overview of the characterized materials is given in Table 7.1.

Back-silvered glass

This is a widely-used reflector material, fabricated by applying a reflective silver layer to the backside of the glass substrate by a wet chemical process and covered with a protective paint [119]. State-of-the-art parabolic trough concentrators are often constructed from 4 mm-thick back-silvered glasses. Attempts to decrease material use and cost while increasing the reflectance have triggered the development of thinner mirrors at the expense of reduced rigidity. In this study, three samples are tested with varying thickness (1 – 4 mm).

Metallized polymeric film

This lightweight alternative offers reduced material cost and weight. In this study, two samples for outdoor use are investigated: a silvered acrylic film featuring a protective copper layer at the backside, and a silvered film protected by multiple layers of semi-transparent polymers on both sides. The polymeric top layer should be highly transparent to solar radiation and resistant to abrasion and UV radiation [120]. A cost-effective alternative to silvered films are aluminized films. In contrast to silver, aluminum is highly reflective in the UV spectral range, enabling the use of inexpensive, non-UV-resistant substrate such as polyethylene terephthalate (boPET) [12, 13, 82]. However, aluminized boPET requires a resistant top layer for protection from weather and abrasion.

Metallized aluminum sheet

Front-surface metallized aluminum offers a compromise between glass and polymer mirrors in terms of rigidity and weight/material content. Silvered aluminum sheets, originally developed for lighting applications, are interesting materials for solar concentrators, provided they are protected from weather. They

Table 7.1: Reflective and semi-transparent solar materials considered in this study.

Sample name	Product description	Thickness	Intended environment	Performance reported by manufacturer	
Back-silvered glass				$R_{\text{direct,solar}}$ [121]	
AgGlass4mm	flat glass mirror (2014)	4 mm	outdoor	≥ 0.945 [122]	
AgGlass2mm	flat glass mirror (2013)	2 mm	outdoor	≥ 0.945 [122]	
AgGlass1mm	flat glass mirror (2008)	1 mm	outdoor	≥ 0.945 [122]	
Metallized polymer films				$R_{\text{h,solar}}$	$R_{\text{specular,660nm,15}^\circ,25\text{mrad}}$
AgFilm#1	silvered acrylic film	117 μm	outdoor	0.94	> 0.95 [123]
AgFilm#2	silvered polymer film	100 μm	outdoor	0.93	0.94 [124]
AlFilm	aluminized boPET	23 μm	indoor	[97]	
Metallized aluminum (Al) sheets				$R_{\text{h,solar/light}}$	$R_{\text{specular, 60}^\circ \text{ ISO 7668}}$
AgSheet#1	silvered Al sheet	0.5 mm	indoor	≥ 0.95	≥ 0.92 [125]
AgSheet#2	silvered Al sheet	0.4 mm	indoor (light)	≥ 0.98	≥ 0.93 [126]
AgSheet#3	silvered Al sheet	0.3 mm	indoor (light)	≥ 0.98	≥ 0.93 [127]
AlSheet	aluminized Al sheet	0.4 mm	outdoor	≥ 0.89	≥ 0.88 [128]
Transparent polymer films					
ETFE100 μm	ETFE film		100 μm	outdoor	[96]
FEP100 μm	FEP film		100 μm	outdoor	[129]
Transparent glass				$T_{\text{normal,solar}}$	
Borosilicate3.3mm	borosilicate substrate	3.3 mm	outdoor	0.92 [50]	
BorosilicateAR3.3mm	AR-coated borosilicate	3.3 mm	outdoor	0.97 [98]	

comprise a pure silver layer applied onto an anodized aluminum substrate by physical vapor deposition (PVD) and a reflectance enhancing dielectric layer. In this study, three samples for indoor, solar, and lighting applications are tested. For completeness, an aluminized aluminum reflector for outdoor use is also characterized. The aluminized reflector has a similar layer system as the silvered sheets, except that silver is replaced by a pure aluminum layer and a UV- and weather-resistant coating is applied on top.

7.2.2 Semi-transparent materials

Transparent materials are used to protect reflectors [10, 11]. A lightweight alternative to glass are semi-transparent polymeric films [12, 13, 82], such as ethylene-tetrafluoroethylene (ETFE) and fluorinated ethylene propylene (FEP) because of their favorable optical, chemical, and mechanical properties [130]. The spectral narrow-angle transmittance of 100 μm thin films of ETFE and FEP is measured. For comparison, the narrow-angle transmittance spectra of a borosilicate substrate and an anti-reflection (AR) coated borosilicate are also measured.

7.3 Reflectance and transmittance

The spectral specular reflectance $R_{\text{specular},\lambda}$ at a certain incidence angle θ and azimuth angle ϕ is defined as the fraction of spectral radiant flux incident on a surface element dA from a solid angle ω_{src} that is reflected in the specular direction within a solid angle ω_{acc} . Assuming that the incident radiance on the sample is uniform and isotropic within ω_{src} , this may be expressed as a bi-conical reflectance using the bi-directional reflectance distribution function (BRDF) of the surface [131]:

$$R_{\text{specular},\lambda}(\lambda, \theta, \phi, \omega_{\text{src}}, \omega_{\text{acc}}) \equiv \frac{d\Phi_{\text{r},\lambda}(\lambda, \theta, \phi, \omega_{\text{acc}})}{d\Phi_{\text{i},\lambda}(\lambda, \theta, \phi, \omega_{\text{src}})} = \frac{\int_{\omega_{\text{acc}}} \int_{\omega_{\text{src}}} \text{BRDF}(\lambda, \theta_{\text{i}}, \phi_{\text{i}}, \theta_{\text{r}}, \phi_{\text{r}}) \cos \theta_{\text{i}} d\omega_{\text{i}} \cos \theta_{\text{r}} d\omega_{\text{r}}}{\int_{\omega_{\text{src}}} \cos \theta_{\text{i}} d\omega_{\text{i}}} \quad (7.1)$$

Here the reflectance is denoted by R to indicate that solar reflective materials usually consist of several layers and thus the measured reflectance is an overall extensive property resulting from multiple interactions of light with optical interfaces rather than an intensive property such as the reflectivity ρ of a single surface. The solar-weighted specular reflectance is given by:

$$R_{\text{specular},\text{solar}} = \frac{\int R_{\text{specular},\lambda} E_{\lambda,\text{solar}} d\lambda}{E_{\text{DNI}}} \quad (7.2)$$

where $E_{\lambda, \text{solar}}$ and E_{DNI} are the spectral and total direct normal solar irradiance, respectively (ASTM G173 – 03 AM 1.5 [46]). Numerical integration is performed over the spectral range 300 – 2500 nm using the weighed ordinate method [132]. The spectral narrow-angle transmittance T_λ is defined as the fraction of spectral radiant flux incident from a solid angle ω_{src} on a surface element dA of a semi-transparent medium layer that is transmitted and leaves the medium along the incident direction within a solid angle ω_{acc} ,

$$T_\lambda(\lambda, \theta, \phi, \omega_{\text{src}}, \omega_{\text{acc}}) \equiv \frac{d\Phi_{t, \lambda}(\lambda, \theta, \phi, \omega_{\text{acc}})}{d\Phi_{i, \lambda}(\lambda, \theta, \phi, \omega_{\text{src}})} \quad (7.3)$$

For a smooth single-layered non-scattering semi-transparent medium, the spectral transmittance T_λ is given in terms of its surface's spectral reflectivity ρ_λ and medium's spectral transmissivity τ_λ [40]:

$$T_\lambda = \frac{\tau_\lambda (1 - \rho_\lambda)^2}{1 - \rho_\lambda^2 \tau_\lambda^2} \quad (7.4)$$

The solar-weighted narrow-angle transmittance is given by:

$$T_{\text{solar}} = \frac{\int T_\lambda E_{\lambda, \text{solar}} d\lambda}{E_{\text{DNI}}} \quad (7.5)$$

7.3.1 Experimental setup

Measurements are performed using a spectroscopic goniometry system [51], shown schematically in Fig. 7.1. A xenon-arc lamp (rated input power: 150 W) is used as light source (1). An aspherical Czerny-Turner type double monochromator (2) enables spectral measurements in the range 300 – 2500 nm with a full bandwidth at half maximum (FWHM) 15 – 30 nm. A mechanical beam chopper (3) modulates the nearly monochromatic light at a frequency of 417 Hz. The diverging light beam is collimated by a MgF_2 plano-convex lens (4). A calcite Glan-Thompson polarizer (5) is inserted after the collimating lens for materials showing polarization-dependent optical properties. The last component on the source side is an iris (6) with adjustable size acting as the aperture stop. On the detector side, the light beam is focused by a second MgF_2 plano-convex lens (4) to an adjustable mechanical slit (8). With this arrangement,

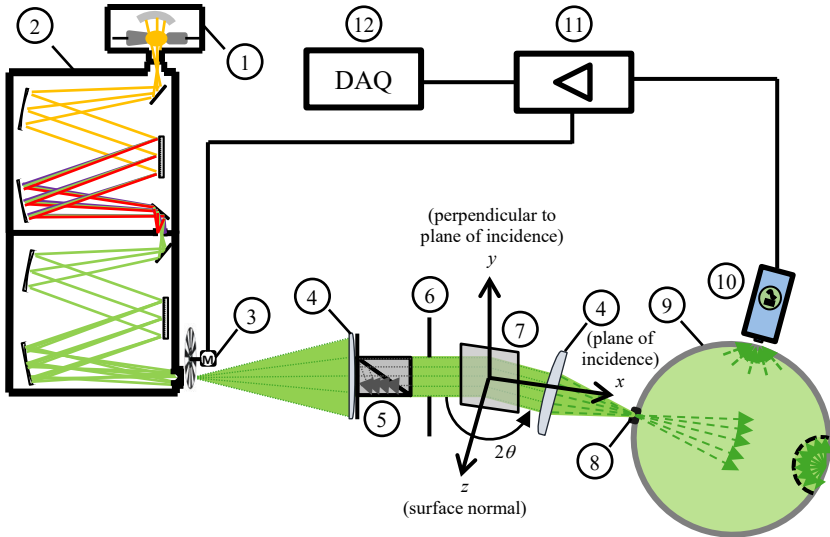


Fig. 7.1: The spectroscopic goniometry system comprised of: (1) Xe-arc lamp, (2) double monochromator, (3) chopper, (4) imaging lens pair, (5) polarizer, (6) iris, (7) sample, (8) mechanical slit, (9) integrating sphere, (10) photodetector, (11) lock-in amplifier, and (12) data acquisition system. The x - y - z coordinate system is centered at the pivot point and x - z defines the plane of incidence.

the monochromator exit slit is imaged onto the mechanical slit which acts as the field stop and allows to precisely adjust the desired acceptance angle of the detector. The light passing through is then collected in an integrating sphere (diameter: 50 mm) (9). The diffused light is measured at the detector port by a thermoelectrically cooled Si or PbS photodiode detector (10), depending on the wavelength (Si: 300 – 1000 nm, PbS: 1000 – 2800 nm). The detected photocurrent is further converted to a proportional voltage by a transimpedance amplifier. To compensate for the change in lens focal length at different wavelengths, the position of each lens in the imaging lens pair is automatically adjusted by a linear translation stage. In the spectral region 500 – 1000 nm where the Xe-arc lamp reaches its maximum emissive power, neutral density filters are placed in front of the second lens to prevent detector overloading. All components of the detector side are mounted on a rotatable arm that can be moved to angular positions ranging from 30° to 180° as measured from the

source beam direction (cf. Fig. 7.1). This imposes a lower limit on the minimum incidence angle for measuring reflectance of 15° . The sample (7) is mounted on a rotatable base plate that allows adjusting the desired incidence angle. Different sample holders are used for rigid and flexible materials. For rigid materials such as back-silvered glasses and aluminum sheets, the samples are softly clamped at their bottom edge. In this case, the maximum measurable incidence angle is only constrained by the samples size and is usually greater than 75° . Samples of flexible polymer films are fixed in a custom-made sample holder that allows for slight tensioning to create a flat surface. The maximum measurable incidence angle is then limited by the frame size to below 65° . The whole sample assembly is mounted on a three-axis linear translation table that allows for automatic adjustment of the sample position. For the reflectance measurement, the sample is always aligned such that the reflective surface passes through the pivot point of the rotatable detector arm. The lock-in amplifier (11) is frequency-synchronized with the chopper to measure the pulsed signal, thereby minimizing noise from other sources.

When compared to conventional circular-aperture measurements, slit-aperture measurements features two-fold advantages: it enables measurement of the specular reflectance and angular scattering in different directions and thus detection of anisotropic behavior [112]; and it uses only two lenses which reduces misalignment. Standard values for the source beam divergence (θ_{src}) and detector acceptance (θ_{acc}) angles for specular reflectance measurements of anodized aluminum surfaces are given in [133]. In the plane of incidence, the standard source half-angle is $0.375^\circ = 6.5 \text{ mrad}$, whereas the standard acceptance half-angle varies from $0.9^\circ = 15.7 \text{ mrad}$ at $\theta = 20^\circ$ to $2.2^\circ = 38.4 \text{ mrad}$ at $\theta = 60^\circ$ [133]. In the present study, the source divergence half-angle in the plane of incidence is set to $\theta_{\text{src},x} = 6 \text{ mrad}$, which includes roughly 97% of the direct normal and circumsolar irradiance assuming the standard solar scan sunshape [134]. The acceptance half-angle is set to $\theta_{\text{acc},x} = 17.5 \text{ mrad} = 1^\circ$ at all incidence angles. This choice of angles closely matches the operating conditions of reflective materials for solar concentrators. In the plane perpendicular to the plane of incidence, the acceptance angle ($\theta_{\text{acc},y} = 50 \text{ mrad}$) is chosen roughly 6

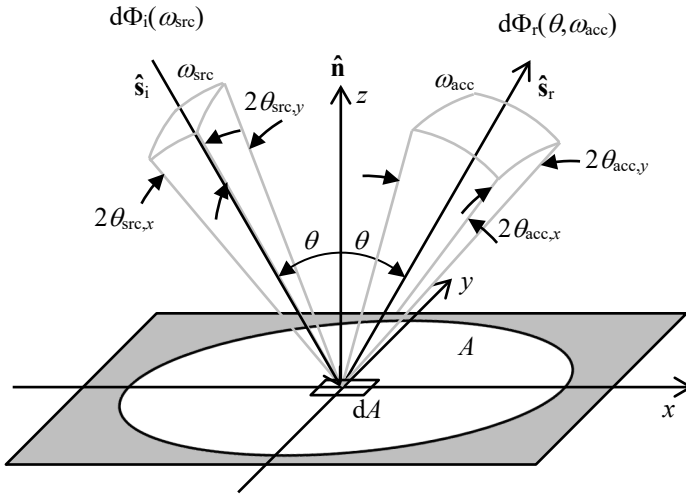


Fig. 7.2: Geometry of incident and specular reflected beams on a flat surface.

times larger than the source divergence angle ($\theta_{\text{src},y} = 8 \text{ mrad}$), similar to [112]. To compensate for changes in focal length due to spectral dispersion, the axial locations of the lenses are automatically adjusted by linear translation stages. The resulting maximum deviations of the source beam and detector acceptance angles occur at the minimum (300 nm) and maximum (2500 nm) measured wavelengths and are within the standard tolerances [133]. The geometry of the incident and specular reflected beams is shown in Fig. 7.2.

7.3.2 Methods

For each data point of $R_{\text{specular},\lambda}$, three sequential measurements are performed at the same wavelength λ : 1) a reference measurement of incident radiative flux with the detector arm located at 180° and the sample moved out of the beam; 2) a sample measurement of reflected radiative flux with the reflective sample placed in the source beam at an incidence angle θ and the detector arm rotated to 2θ ; and 3) a second reference measurement of incident flux to account for temporal fluctuations in Xe-arc lamp power, since a single-beam setup is used. The $R_{\text{specular},\lambda}$ is then calculated as the ratio of the voltage from the sample

measurement to the average voltage from the reference measurements, assuming that the measured voltage U is proportional to the radiant flux:

$$R_{\text{specular},\lambda}(\lambda, \theta) \equiv \frac{\Phi_{r,\lambda}(\lambda, \theta)}{\Phi_{i,\lambda}(\lambda)} = \frac{U_{\text{sample}}(\lambda, \theta)}{U_{\text{reference}}(\lambda)} \quad (7.6)$$

It is noted that the measurement is absolute in nature as the final result does not depend on any calibrated reference material. If the optical property of interest depends on the state of polarization of the incident light, the above measurement is performed twice, with the electric field of incident light oscillating once in the plane parallel to the plane of incidence (parallel ($\parallel$) or p-polarized), and once in the plane perpendicular to the plane of incidence (perpendicularly ($\perp$) or s-polarized). The optical property for unpolarized sunlight is then calculated as the average of p- and s-polarized reflectance [54]

$$R_{\text{specular},\lambda} = \frac{1}{2} (R_{\text{specular},\lambda,\parallel} + R_{\text{specular},\lambda,\perp}) \quad (7.7)$$

For measurement of T_λ , the exact same procedure is applied with the only difference to the reflectance measurement being that the detector arm remains at 180° also during the sample measurement.

7.3.3 Results and discussion

Specular reflectance

The measured specular reflectance spectra at near normal incidence (15°) of five samples of the solar reflective materials (listed in Table 7.1) are shown in Fig. 7.3. The spectral data are given in Table C.1. $R_{\text{specular},\text{solar}}$ for all characterized materials at all incidence angles are listed in Table 7.2. The spectral data measured at all incidence angles are available in [86]. While the spectral differences among the reflective materials are substantial, the dependence of specular reflectance on the incidence angle is generally weak. The maximum change in $R_{\text{specular},\text{solar}}$ between 15° and 60° is within 1.5% for all tested materials.

Back-silvered glass — The measured $R_{\text{specular},\text{solar}}$ of AgGlass4mm at $\theta = 15^\circ$ (0.941) is close to the solar direct reflectance [121] at near normal incidence stated by the manufacturer (≥ 0.945) [122]. Theoretically, the reflectance should increase with decreasing glass thickness due to reduced absorption in the glass

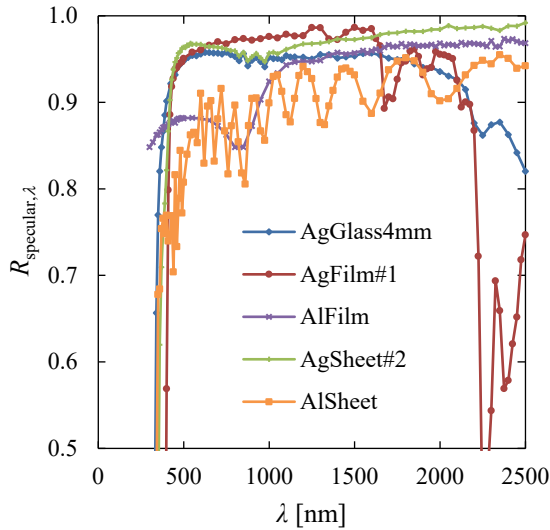


Fig. 7.3: Spectral specular reflectance at $\theta = 15^\circ$ and $\theta_{acc,x} = 17.5$ mrad of back-silvered glass, silvered and aluminized polymer films, and silvered and aluminized aluminum sheets.

layer, as observed for AgGlass2mm. The somewhat lower performance of AgGlass1mm is attributed to the age of that sample and improvements in the fabrication process over time.

Metallized polymer films — For the AgFilms#1 and #2, the solar-weighted directional-hemispherical reflectance $R_{h,solar}$ stated by the manufacturer is 1.4% and 2.2% higher than the measured $R_{specular,solar}$ at 15° , respectively (cf. Table 7.1 and Table 7.2). The spectra of $R_{h,\lambda}$ provided by the manufacturer [124] and $R_{specular,\lambda}$ measured at 15° incidence angle are shown in Fig. 7.4 for AgFilm#2. A significant difference, e.g. 5.5% at 500 nm, is observed especially in the visible range where the solar irradiance peaks. The difference decreases towards longer wavelengths until specular and hemispherical reflectance become identical beyond 2000 nm. A higher specular than hemispherical reflectance above 2250 nm is attributed to increased measurement uncertainty at longer wavelengths. Interestingly, AgFilm#2 and AlFilm feature comparable

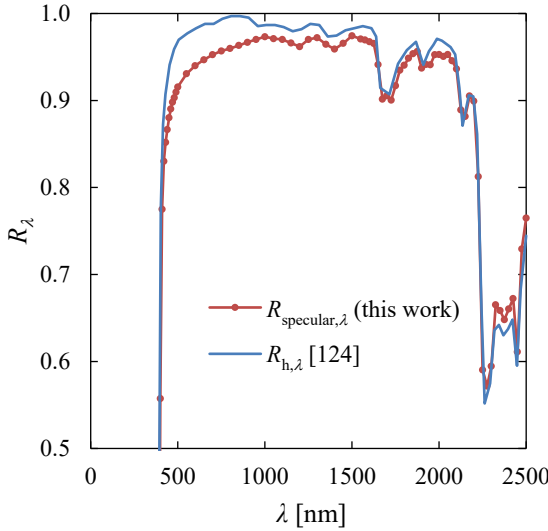


Fig. 7.4: AgFilm#2: spectral specular reflectance measured at $\theta = 15^\circ$ and $\theta_{\text{acc},x} = 17.5$ mrad and spectral directional-hemispherical reflectance given by the manufacturer.

$R_{\text{specular},\text{solar}}$, although the latter is cheaper. Note that aluminized boPET has no protective top layer but it is restricted to applications in a protected environment.

Metallized aluminum sheets — The measured $R_{\text{specular},\text{solar}}$ of silvered aluminum sheets lies between the specular and directional-hemispherical reflectance stated by the manufacturers [125-127] for solar radiation (AgSheet#1) or visible light (AgSheet#2 and #3). Reflectors for lighting (#2 and #3) generally show a higher $R_{\text{specular},\text{solar}}$ than that for solar applications (#1), but other factors such as durability and weather resistance should be taken into account for the selection of appropriate materials. The aluminized reflector for outdoor use shows interference patterns (cf. Fig. 7.3) caused by the protective, weather resistant, transparent top coating. In addition, $R_{\text{specular},\text{solar}}$ determined in this study for AlSheet is clearly lower than $R_{\text{h},\text{solar}}$ (≥ 0.89) and $R_{\text{specular},\text{ISO 7668 60}^\circ}$ (≥ 0.88) stated by the manufacturer [128]. A reason for this is the difference in acceptance angles used in this work (1°) and defined in ISO 7668 (2.2°), combined with significant angular scattering by the reflective surface. At 45°

Table 7.2: Solar-weighted narrow-angle transmittance of semi-transparent materials and solar-weighted specular reflectance of reflective materials as a function of incidence angle.

T_{solar}	0°	15°	45°	60°
ETFE100 μm	0.913	0.910	0.889	0.831
FEP100 μm	0.946	0.944	0.932	0.877
Borosilicate3.3mm	0.921			
BorosilicateAR3.3mm	0.950	0.949	0.940	0.880
$R_{\text{specular,solar}}$		15°	45°	60°
AgGlass4mm (2014)		0.942	0.935	0.934
AgGlass2mm (2013)			0.941	
AgGlass1mm (2008)			0.934	
AgFilm#1		0.926	0.922	0.913
AgFilm#2		0.908	0.901	0.893
AlFilm		0.895	0.898	0.894
AgSheet#1		0.939	0.941	0.944
AgSheet#2		0.948	0.949	0.949
AgSheet#3		0.954	0.951	0.952
AlSheet		0.860	0.865	

incidence angle and 550 nm, the relative difference between $R_{\text{specular},\lambda}$ measured with acceptance angles of 2.2° and 1° in the plane of incidence is 1.0% and decreases with increasing wavelength. Furthermore, $R_{\text{specular},\lambda}$ of AgSheet#2 and AlSheet was measured with the rolling marks of the aluminum substrate perpendicular to the plane of incidence, which yields $R_{\text{specular,solar}}$ that is roughly 0.5% lower than when measured with the rolling marks parallel to the plane of incidence.

Narrow-angle transmittance

The measured spectral normal transmittance $T_{\text{normal},\lambda}$ of the 100 μm ETFE and FEP films and the 3.3 mm borosilicate glasses (substrate and AR-coated) is shown in Fig. 7.5. The data are given in Table C.2. The spectral data at all incidence angles are available in [86]. Fluctuations at longer wavelength is attributed to thin-film interference. T_{solar} at different incidence angles are also

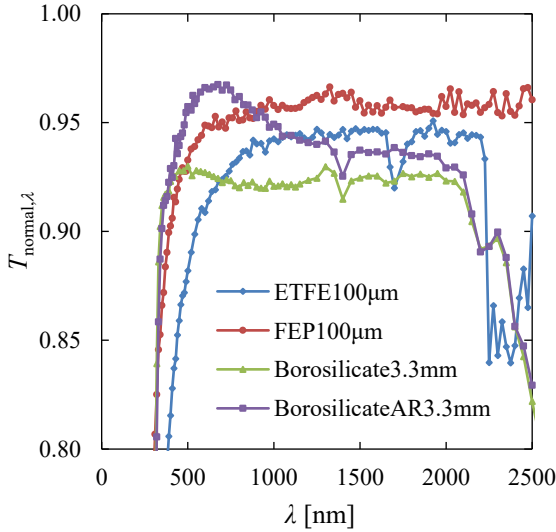


Fig. 7.5: Spectral normal transmittance at $\theta_{acc,x} = 17.5$ mrad of semi-transparent polymeric films and borosilicate glasses.

given in Table 7.2. While the 100 μm ETFE film shows a comparable value of T_{solar} as that for a 3.3 mm borosilicate glass, the 100 μm FEP film is almost as transparent as the AR-coated borosilicate glass.

7.4 Angle-resolved surface scattering

The narrow-angle scattering of reflective materials within the acceptance solid angle for specular reflectance is determined using a combined experimental-numerical approach. First, the shapes of the reference and specular reflected beams are measured. Then the scattering function is determined by a deconvolution of the incident beam from the reflected beam. Finally, the angular scattering is quantified by parameter estimation of statistical scattering models using the least-square method.

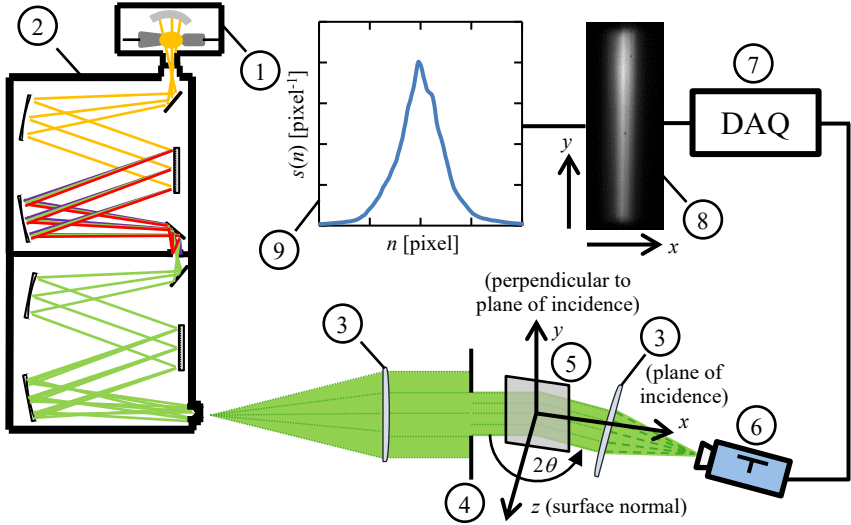


Fig. 7.6: Experimental setup for angular scattering measurements comprised of: (1) Xe-arc lamp, (2) double monochromator, (3) imaging lens pair, (4) iris, (5) sample, (6) CCD detector, (7) data acquisition system, (8) 2D grayscale image, and (9) normalized, averaged 1D beam profile.

7.4.1 Experimental setup

The modified experimental setup for the beam shape measurements is shown in Fig. 7.6. On the source side, the monochromator exit slit width is reduced to achieve a highly collimated beam in the plane of incidence ($\theta_{\text{src},x} = 0.31$ mrad), while the chopper is removed. On the detector side, the $f_{\text{acc}} = 75$ mm lens is replaced by a $f_{\text{acc}} = 250$ mm lens to refine the angular resolution of the setup. Most importantly, the detector assembly (adjustable slit + integrating sphere + photodetector) is replaced by an area scan CCD camera (Basler scA780-54gm, resolution: 782 x 582, pixel size: $8.3 \mu\text{m} \times 8.3 \mu\text{m}$, frame rate: 55 fps) with the sensor area placed at the rear focal point of the second lens. The CCD detector enables measurements in the spectral range 350 – 1050 nm, which includes 75% of ASTM G173 – 03 AM 1.5 reference spectrum for DNI [46].

The measurement procedure is similar to that described in the previous section. For every angular scattering experiment, three images are recorded at the same wavelength: 1) reference beam measurement with the sample moved

out of the beam and the detector arm positioned at 180° such that the image of the monochromator exit slit is projected at the CCD detector; 2) reflected/transmitted beam shape measurement with the sample moved into the path of light at an angle θ and the detector arm rotated to the corresponding angular position (2θ for reflection; 180° for transmission); and 3) a second reference beam measurement. The exposure time is adjusted for every experiment such that no saturation is detected in the recorded images. With the slit aperture setup, the scattering in the plane of incidence (x -direction) may be observed in each slit cross-section along the y -direction. This allows for averaging over y and extracting a 1D beam profile as a function of x . For this purpose, the 1D beam profiles at each slit cross-section in y are normalized for a unity area below the curve. The normalized 1D beam profiles are then averaged over a certain y -range to obtain a representative average beam profile. This measurement procedure defines a controlled experiment with high repeatability. For anisotropic materials that show visible marks originating from the fabrication of the substrate (e.g. sheet rolling, film extrusion), the measurement is performed once with the manufacturing marks parallel and once perpendicular to the plane of incidence.

7.4.2 Methods

Deconvolution

The sample beam profile $s(x)$ is a convolution of the reference beam profile $r(x)$ with the scattering function $f(x)$ of the material [112],

$$s(x) = (r * f)(x) = \int_{-\infty}^{\infty} r(\chi) f(x - \chi) d\chi \quad (7.8)$$

Due to the finite resolution of the CCD sensor, the measured beam profiles are discrete functions $r(n)$ and $s(n)$ related by a discrete convolution. Since the beam profiles are finite, i.e. $r(n) = 0$ for $|n| > M$ and $s(n) = 0$ for $|n| > N$, and thus $f(n) = 0$ for $|n| > N - M$, the equations for the $2N + 1$ non-zero elements of $s(n)$ are given by,

$$s(n) = \sum_{m=n-(N-M)}^{n+N-M} r(m) f(n-m), \quad \text{for } -N \leq n \leq N \quad (7.9)$$

This yields $2N + 1$ linear equations for the $2(N - M) + 1$ unknown non-zero elements of $f(n)$. The discrete scattering function can be determined by solving a linear least-square optimization problem. If needed, filtering can be applied *a posteriori* to obtain a smooth curve. For smoothing a scattering function, a non-causal weighted moving average filter is recommended since it does not introduce any phase delay. Finally, the imaging equation is applied to transform from pixel coordinates to scattering angle $\theta_{s,x}$:

$$\theta_{s,x} = \arctan\left(\frac{n \cdot d_{\text{pixel}}}{2f_{\text{acc}}}\right) \approx \frac{n \cdot d_{\text{pixel}}}{2f_{\text{acc}}} \quad (7.10)$$

where $d_{\text{pixel}} = 8.3 \mu\text{m}$ is the pixel size of the CCD detector and $f_{\text{acc}} = 250 \text{ mm}$ the focal length of the lens on the detector side. The profiles of the reference and sample beams are normalized for a unity area below the curve. Thus, the integral of the scattering function is 1 and $f(\theta_{s,x})$ becomes the probability density function of the scattering angle. The discrete deconvolution method provides an independent result for the scattering function without imposing any assumptions about its shape. However, for comparison among materials, a description based on a few characteristic parameters rather than a complete curve would be more convenient. Such a description is obtained by assuming a certain statistical scattering distribution model and identifying the best fitting parameters.

Statistical surface scattering models

It is generally assumed that surface scattering of solar reflective materials may be described by a scattering angle following a Gaussian distribution with zero mean and standard deviation σ [70, 112],

$$f(\theta_{s,x}) = \frac{1}{\sqrt{2\pi\sigma^2}} \exp\left(-\frac{\theta_{s,x}^2}{2\sigma^2}\right) \quad (7.11)$$

or by a superposition of two Gaussian distributions for materials that show different light scattering mechanisms, e.g. due to microscopic and macroscopic surface roughness of the reflective layer and the substrate, respectively [112]

$$f(\theta_{s,x}) = \frac{F_1}{\sqrt{2\pi\sigma_1^2}} \exp\left(-\frac{\theta_{s,x}^2}{2\sigma_1^2}\right) + \frac{1-F_1}{\sqrt{2\pi\sigma_2^2}} \exp\left(-\frac{\theta_{s,x}^2}{2\sigma_2^2}\right) \quad (7.12)$$

Using the experimentally measured reference beam profile and a Gaussian model for the scattering function, the scattered beam profile predicted by the model s_{model} is calculated according to Eq. (7.9). The standard deviations are determined by minimizing the deviations from the experimentally measured sample beam profile s_{exp} ,

$$\min_{\sigma} \sum_{n=-N}^N \left(s_{\text{model}}(n) - s_{\text{exp}}(n) \right)^2 \quad (7.13)$$

For anisotropic materials, the 1D scattering functions identified from the beam shape measurements in the two main directions of the material are finally combined to a 2D scattering function. Assuming orthotropic and independent scattering in the two main directions, the 2D scattering distribution function is the product of the 1D scattering functions in each direction. For single Gaussian distributions in both directions, this yields a bivariate Gaussian distribution:

$$f(\theta_{s,x}, \theta_{s,y}) = f(\theta_{s,x}) f(\theta_{s,y}) = \frac{1}{2\pi\sigma_x\sigma_y} \exp\left(-\frac{\theta_{s,x}^2}{2\sigma_x^2} - \frac{\theta_{s,y}^2}{2\sigma_y^2}\right) \quad (7.14)$$

7.4.3 Results

Angular scattering experiments are performed for the reflective materials AgGlass4mm, AgFilm#1 and #2, AlFilm, and AgSheet#1 and #2, and for the semi-transparent films ETFE100 μm and FEP100 μm . The estimated standard deviations of the best fit Gaussian scattering distribution as a function of wavelength and incidence angle for all tested materials are summarized in Table 7.3.

Highly specular materials

Very small angular scattering is measured for reflection from back-silvered glass and aluminized boPET, and for transmission through semi-transparent polymer films. The angular scattering of these materials is accurately described by a single Gaussian distribution according to Eq. (7.11), with standard deviations < 0.03 mrad for ETFE100 μm , < 0.05 mrad for FEP100 μm , < 0.05 mrad for AlFilm, and < 0.07 mrad for AgGlass4mm at all measured wavelengths and incidence angles. It is noted that these values are in the range of the angular resolution of the

Table 7.3: Best fit standard deviations of single Gaussian angular scattering distribution as a function of wavelength and incidence angle for all tested materials and manufacturing direction orientations.

Material	λ [nm]	350	400	500	555	600	700	800	950	1050
	θ [deg]	σ [mrad]								
ETFE100 μ m	0 – 60	< 0.033								
FEP100 μ m	0 – 45	< 0.033								
	60	0.049	0.049	0.044	0.041	0.038	0.037	0.039	0.033	0.033
AlFilm	15 – 45	< 0.033								
	60	0.042	0.048	0.042	0.038	0.042	0.040	0.038	0.036	0.039
AgGlass4mm	15	< 0.033								
	45	0.040	0.043	0.041	0.038	< 0.033		0.036	0.035	< 0.033
	60	0.060	0.064	0.057	0.055	0.053	0.053	0.058	0.053	0.049
AgFilm#1	15		0.34	0.33	0.33	0.32	0.31	0.31	0.30	0.29
marks parallel to	45		0.39	0.38	0.37	0.37	0.36	0.35	0.35	0.34
plane of incidence	60		0.49	0.50	0.50	0.50	0.49	0.48	0.46	0.47
AgFilm#1	15		0.76	0.70	0.70	0.68	0.62	0.60	0.55	0.58
marks perp. to	45		0.86	0.82	0.78	0.76	0.73	0.67	0.63	0.56
plane of incidence	60		1.01	0.99	0.93	0.93	0.90	0.80	0.71	0.76
AgFilm#2	15		0.65	0.59	0.57	0.55	0.50	0.46	0.40	0.35
marks parallel to	45		0.69	0.62	0.58	0.55	0.47	0.43	0.34	0.30
plane of incidence	60		0.71	0.62	0.56	0.51	0.42	0.37	0.29	0.27
AgFilm#2	15		0.72	0.66	0.64	0.62	0.56	0.52	0.47	0.45
marks perp. to	45		0.86	0.77	0.71	0.68	0.61	0.60	0.48	0.46
plane of incidence	60		1.12	0.98	0.93	0.88	0.77	0.85	0.71	0.55
AgSheet#1	15	0.58	0.56	0.55	0.55	0.54	0.53	0.52	0.51	0.51
marks parallel to	45	0.67	0.66	0.65	0.65	0.65	0.63	0.63	0.62	0.61
plane of incidence	60	0.83	0.82	0.82	0.82	0.81	0.80	0.80	0.80	0.79
AgSheet#1	15	0.40	0.37	0.31	0.29	0.27	0.23	0.21	0.18	0.16
marks perp. to	45	0.33	0.29	0.24	0.22	0.21	0.19	0.17	0.15	0.13
plane of incidence	60	0.27	0.24	0.21	0.20	0.19	0.16	0.15	0.14	0.12
AgSheet#2	15	1.04	1.03	1.01	1.00	1.00	0.99	0.99	0.98	0.97
marks parallel to	45	1.28	1.27	1.25	1.24	1.25	1.24	1.25	1.24	1.23
plane of incidence	60	1.66	1.66	1.64	1.64	1.65	1.64	1.66	1.66	1.65
AgSheet#2	15	0.38	0.34	0.28	0.26	0.24	0.22	0.20	0.18	0.17
marks perp. to	45	0.33	0.31	0.27	0.26	0.25	0.24	0.23	0.21	0.21
plane of incidence	60	0.32	0.31	0.29	0.28	0.28	0.27	0.27	0.26	0.26

experimental setup (0.033 mrad) and are thus subject to inaccuracies evolving from discrete sampling of the beam profiles. Standard deviations below the angular resolution of the setup are reported as < 0.033 mrad in Table 7.3.

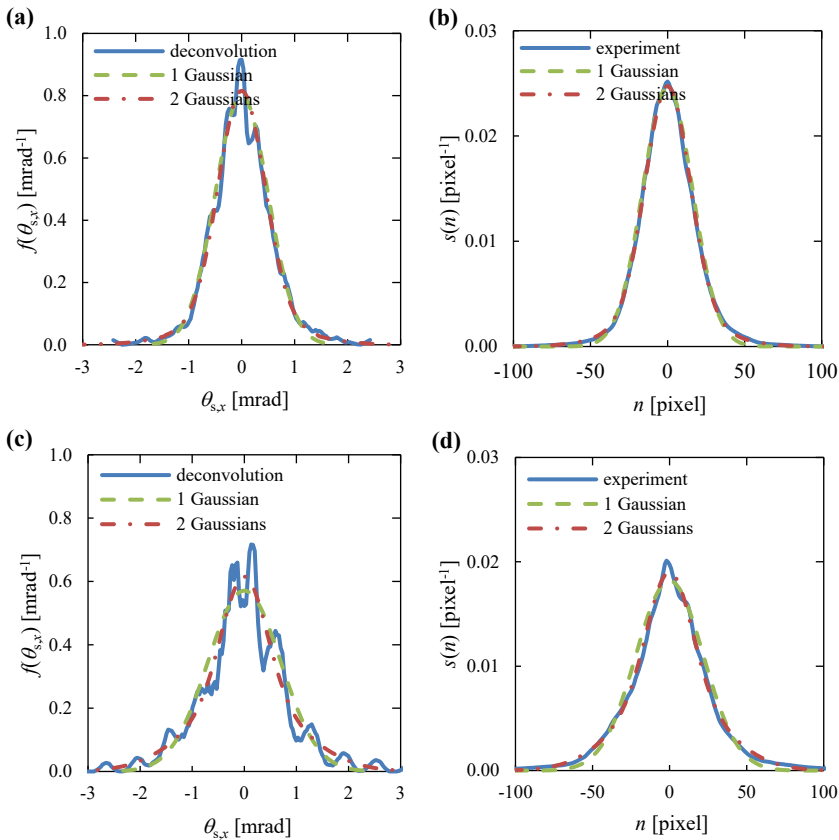


Fig. 7.7: AgFilm#1 with manufacturing marks parallel to plane of incidence at $\theta = 60^\circ$ (a,b); and perpendicular to plane of incidence at $\theta = 15^\circ$ (c,d). a)/c) Deconvolved, and best fit single and double Gaussian scattering functions; and b)/d) reflected beam profiles measured experimentally, and modelled with the single and double Gaussian scattering functions at 555 nm.

Silvered polymeric films

Significant light scattering is observed for reflection from silvered polymers. Fig. 7.7a shows the scattering functions obtained by deconvolution and by least-square fitting of a single Gaussian and a superposition of two Gaussian distributions for the AgFilm#1 with the manufacturing marks parallel to the plane of incidence. The best fits to the experimentally measured reflected beam profile

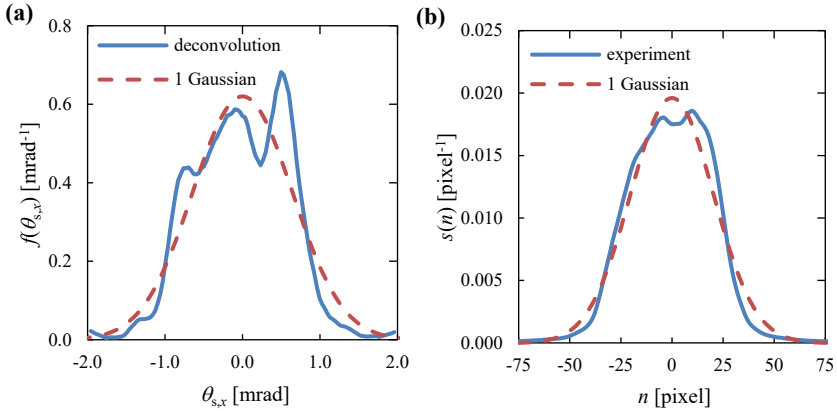


Fig. 7.8: AgSheet#1 with manufacturing marks oriented parallel to the plane of incidence at $\theta = 45^\circ$ and 555 nm. a) Deconvolved and best fit single Gaussian scattering functions; and b) reflected beam profiles measured experimentally, and modelled with the single Gaussian scattering function.

are shown in Fig. 7.7b. The single Gaussian distribution shows a good agreement in the central region of the reflected beam profile only, whereas the superposition of two Gaussian distributions according to Eq. (7.12) resolves also the tail regions. The best fit parameters of the double Gaussian distribution for all measured wavelengths and incidence angles are summarized in Table D.1 [86]. The scattering functions and reflected beam profiles for AgFilm#1 with the manufacturing direction perpendicular to the plane of incidence are shown in Fig. 7.7 c and d, respectively. In this direction the scatter is generally larger than when aligned with the marks parallel to the plane of incidence. Furthermore, the deconvolved scattering function indicates a presence of diffraction peaks, which cannot be captured by a Gaussian distribution. Nevertheless, the reflected beam profile is still reproduced with reasonable accuracy by a superposition of two Gaussian distributions.

Silvered aluminum sheets

Similar to silvered polymer films (cf. Fig. 7.7), the scattering of silvered aluminum sheets with the rolling marks of the aluminum substrate perpendicular to the plane of incidence is accurately described by a superposition of two

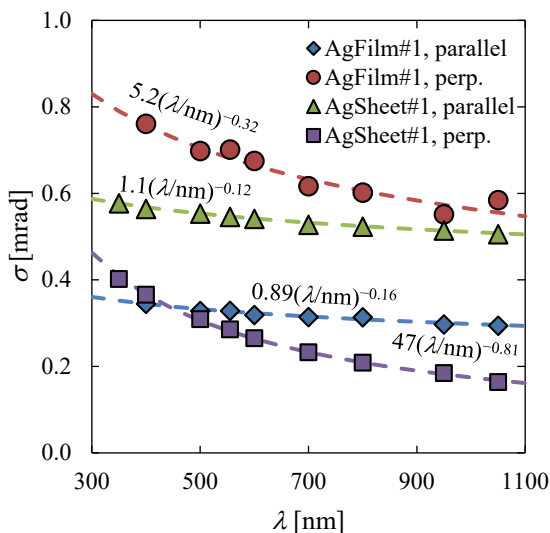


Fig. 7.9: AgFilm#1 and AgSheet#1 in the two main directions at $\theta = 15^\circ$. Standard deviation of angular scattering as a function of wavelength (markers) and power laws for the different series (dashed lines).

Gaussian distributions. However, the silvered sheets are highly anisotropic. The deconvolved and best fit Gaussian scattering functions and the experimentally measured and statistically modelled reflected beam profiles of the AgSheet#1 with the rolling marks parallel to the plane of incidence are shown in Fig. 7.8 a and b, respectively. For modeling such a beam shape, the superposition of two Gaussian distributions shows no benefit and is thus not shown in the figure. Due to the noticeable deviation between the experimentally measured and statistically modelled beam profiles it might be worth the effort to use the deconvolved scattering function rather than a Gaussian distribution for accurate simulations.

7.4.4 Discussion

Wavelength dependence

The standard deviations from Table 7.3 for AgFilm#1 and AgSheet#1 with the marks parallel and perpendicular to the plane of incidence at 15° incidence angle

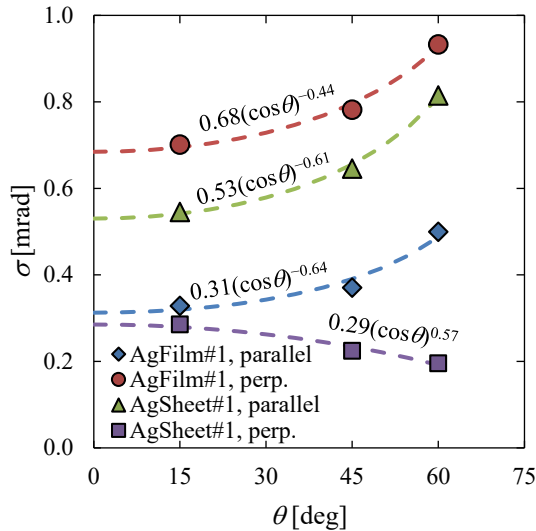


Fig. 7.10: AgFilm#1 and AgSheet#1 in the two main directions at 555 nm. Standard deviation of angular scattering as a function of incidence angle (markers) and power laws in $\cos \theta$ (dashed lines).

are plotted as a function of wavelength in Fig. 7.9. The standard deviation of the scattering distribution is generally decreasing at longer wavelength for all tested materials at all incidence angles. This is consistent with the theory of light scattering for smooth surfaces that predicts an inversely proportional relationship between scatter distribution width and wavelength [115, 135]. An empirical fit to the measured data points is obtained by a power law of the form $\sigma = a (\lambda / \text{nm})^b$. The trend lines and best fit equations are also shown in Fig. 7.9. The good agreement between the measured data and the power laws allows for interpolation and — to some extent — for certain extrapolation outside the measured range.

Incidence angle dependence

The standard deviation of the scattering distribution as a function of incidence angle at a wavelength of 555 nm is shown in Fig. 7.10 for the same materials and same manufacturing directions as in Fig. 7.9. The scatter generally increases with

increasing incidence angle for all tested materials except for the silvered aluminum sheets #1 and #2 with the rolling marks perpendicular to the plane of incidence. The scattering theory for smooth surfaces predicts a shift invariant scattering function in direction cosine space [115, 135, 136], which, for small scattering angles, stretches the scattering distribution in the plane of incidence by a factor of roughly $1/\cos\theta$. Based on purely geometric considerations for specular reflection from a surface with macroscopic slope errors, the scattering angle in the plane of incidence is twice the surface slope error independent of the incidence angle [70]. A simple description of incidence angle dependence that is straightforward to integrate in a MC program is obtained by a power law of $\cos\theta$. The trend lines and equations for the different materials and manufacturing directions are also indicated in Fig. 7.10.

Non-planarity of silvered glass

The scattering of AgGlass4mm in Table 7.3 is reported for the direction of perfect planarity approximately parallel to the plane of incidence, i.e. a projected angle in the plane of incidence of zero between the surface normal vectors of the glass front and silvered back surfaces. The reflected beam profiles at 555 nm and incidence angles of 15° and 60° measured for the perpendicular material orientation are shown in Fig. 7.11. In this direction, the main slit image is accompanied on either side by further images of the slit with lower intensity. This originates from a variation in glass thickness in x -direction, which causes a planarity error between the reflective silvered back surface and the front surface of the glass, denoted by $\theta_{n,x}$. Tracing a ray through refraction and reflection at a second surface reflector according to Snell's law [54] yields for the scattering angle $\theta_{s,x}$

$$\theta_{s,x} = 2\theta_{n,x} \left(1 - m \frac{\sqrt{(n_1/n_0)^2 - \sin^2 \theta}}{\cos \theta} \right) + O(\theta_{n,x}^2) \quad (7.15)$$

where n_0 and n_1 denote the indices of refraction of the surroundings and the glass, respectively, and m is the number of reflections at the back surface. For $m = 0$, the ray is directly reflected from the front surface and the reflection error is twice

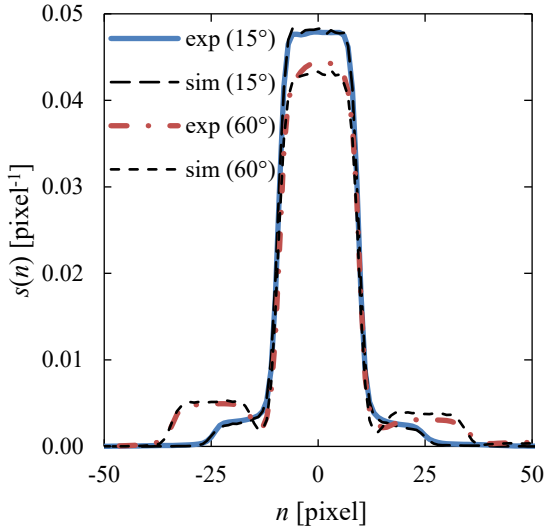


Fig. 7.11: AgGlass4mm with the direction of perfect planarity perpendicular to the plane of incidence at 555 nm. Experimentally measured reflected beam profiles at $\theta = 15^\circ$ (solid) and $\theta = 60^\circ$ (dash-dotted), and numerically simulated beam profiles with $\theta_{n,x} = 0.17$ mrad, $\sigma = 0$ mrad (dashed).

the surface slope error (leftmost slit image in Fig. 7.11). For $m = 1$, the slope error of the front glass surface is overcompensated by refraction, which corresponds to the main slit image. For $m > 1$, the scattering angle keeps increasing with the number of internal reflections (right slit images). By means of the distance between the slit images in Fig. 7.11 and Eq. (7.15), the surface normal error of the glass front surface is determined to be $\theta_{n,x} = 0.17$ mrad. This result is verified by MC ray-tracing where the back-silvered glass is modelled as described in Section 7.5.1 with a front-surface normal error of $\theta_{n,x} = 0.17$ mrad in the plane of incidence and $\sigma = 0$ mrad, i.e. perfect specular reflection at the back surface. The numerically simulated reflected beam profiles at 555 nm and incidence angles of 15° and 60° are also shown in Fig. 7.11. The good agreement at both incidence angles validates the optical modeling approach of the back-silvered glass and confirms the surface slope error of the front surface. Furthermore, since a good agreement with experiments is obtained for $\sigma = 0$ mrad

in the MC simulations, it is likely that angular scattering with the direction of perfect planarity approximately parallel to the plane of incidence as reported in Table 7.3 is also caused by a small angular error of the front surface normal in that direction. This also explains the dependence on incidence angle as the scattering angle is scaled by the factor $[(n_1 / n_0)^2 - \sin^2 \theta]^{1/2} / \cos \theta$, which increases with incidence angle from 1.55 at 15° to 2.5 at 60° (assuming $n_0 = 1$ for air and $n_1 = 1.518$ for N-BK7 glass at 555 nm).

7.5 Application to solar concentrators

In this section, the materials characterized in the previous two sections are employed in solar concentrators to elucidate the impact of the measured optical properties on the optical performance of the solar concentrator.

7.5.1 Optical modeling

Angular errors of solar concentrators

According to [47, 70], the variance of the total angular error of a solar concentrator can be expressed as the sum of variances from different sources of error,

$$\sigma_{s,\text{total}}^2 = \sigma_{s,\text{specularity}}^2 + 4\sigma_{n,\text{slope}}^2 + 4\sigma_{n,\text{shape}}^2 + 4\sigma_{n,\text{alignment}}^2 + \sigma_{n,\text{tracking}}^2 \quad (7.16)$$

where the standard deviations of surface normal errors (subscript n) are multiplied by a factor 2 to transfer from surface slope error to scattering angle (subscript s). For Fresnel reflectors that are actuated independently of the solar receiver, $\sigma_{n,\text{tracking}}$ is also multiplied by 2 [70]. In MC ray-tracing, all effects are conveniently simulated by a single, cumulative error of the surface normal with standard deviation $\sigma_{n,\text{total}} = \sigma_{s,\text{total}} / 2$ [137]. It is noted that the factor 2 between surface normal and reflection angle only holds in the plane of incidence ($\theta_{s,x} = 2\theta_{n,x}$), whereas in the plane perpendicular to the plane of incidence, the scattering angle is attenuated by the cosine factor of the incidence angle ($\theta_{s,y} = 2\theta_{n,y} \cos \theta$) [70]. Accordingly, simulating scattering due to non-specularity of the reflector material by a surface normal error generally yields non-conservative results. However, considering that all errors on the right-hand-side of Eq. (7.16), except for specularity, are described by macroscopic

deviations of the surface normal, and that microscopic non-specularity is usually small compared to the other errors, this approach is considered appropriate for the simulation of solar concentrators.

Reflective materials

The solar reflective material is modelled as a specularly reflecting surface with measured $R_{\text{specular},\lambda}$. Angular surface scattering is simulated by Gaussian surface slope errors in the two main directions of the material with the parameters given in Table 7.3 or Table D.1 [86] for materials that are better described by a superposition of two Gaussian distributions. Note that the reported standard deviations need to be divided by 2 to convert from scattering angle to surface slope error. In the case of a back-silvered glass reflector, the glass layer is included as a refractive, absorptive medium using spectral data of N-BK7HT [138] for the index of refraction and absorption coefficient. The silvered back surface is specularly reflecting with spectral surface reflectivity $\rho_{1,\lambda}$ calculated by

$$\rho_{1,\lambda} = \frac{R_{\text{specular},\lambda} - \rho_{0,\lambda}}{(1 - (2 - R_{\text{specular},\lambda})\rho_{0,\lambda})\tau_{\lambda}^2} \quad (7.17)$$

where $\rho_{0,\lambda}$ is the spectral surface reflectivity of the front surface given by Fresnel's equations, τ_{λ} is the spectral internal transmissivity of the glass layer according to Bouguer's law with the angle of refraction calculated according to Snell's law, and $R_{\text{specular},\lambda}$ is the measured spectral specular reflectance of the sample [54]. Eq. (7.17) guarantees that the simulated reflectance of the back-silvered glass matches the measured specular reflectance. To calculate the standard deviation of surface scattering at the silvered back-surface ($\sigma_{s,1}$) from the experimentally measured standard deviation ($\sigma_{s,0}$), refraction at the glass front surface according to Snell's law needs to be taken into account,

$$\sigma_{s,1} = \sigma_{s,0} \frac{\cos \theta}{\sqrt{(n_1/n_0)^2 - \sin^2 \theta}} + O(\sigma_{s,0}^2) \quad (7.18)$$

where n_1 and n_0 denote the indices of refraction of the glass and surrounding air, respectively. No angular deviation between the glass front surface and the

silvered back surface is considered other than that resulting from curvature of the solar reflector (cf. Section 7.4.4).

Semi-transparent materials

Polymeric films are modelled as a refracting, absorbing, and narrow-angle scattering medium. The refractive index and absorption coefficient are determined from measured T_λ . Using a constant refractive index (1.37 for ETFE, 1.33 for FEP) and spectral data for the extinction coefficient determined from the measured normal transmittance spectrum, the measured T_λ is reproduced well at all incidence angles. Angular scattering is modelled as an isotropic Gaussian error of the transmitted ray direction within the refractive medium, with standard deviations calculated according to Eq. (7.18).

7.5.2 Concentrator design

Four different material configurations, listed in Table 7.4, are selected to build a solar concentrator: two designs using outdoor solar reflectors and two designs using indoor reflectors protected by a semi-transparent top film. The designs are applied to two solar concentrators: 1) a parabolic dish concentrator with a flat disk receiver; and 2) a parabolic trough concentrator with a circular tubular receiver. Geometric cross-sections of the parabolic dish and trough collectors are shown in Fig. 7.12 a and b, respectively. The rim angles Φ_{rim} of the solar concentrators are chosen for maximum theoretical geometric concentration. The geometric concentration ratios of a parabolic dish concentrator with a flat disk receiver and a parabolic trough concentrator with a circular tubular receiver are [83, 139, 140]

$$C_{\text{g,dish}} = \frac{\pi(a_i^2 - a_o^2)}{\pi a_o^2} = \left(\frac{\cos \Phi_{\text{rim}} \sin \Phi_{\text{rim}}}{\cos \theta_{\text{acc}} \sin \theta_{\text{acc}}} \right)^2 - 1 \quad (7.19)$$

and

$$C_{\text{g,trough}} = \frac{2a_i}{2\pi r_o} = \frac{\sin \Phi_{\text{rim}}}{\pi \sin \theta_{\text{acc}}} \quad (7.20)$$

Table 7.4: Material configurations for solar concentrators.

Configuration	Protective film	Solar reflector
Design 1	none	AgGlass4mm
Design 2	none	AgFilm#1
Design 3	ETFE100 μ m	AlFilm
Design 4	FEP100 μ m	AgSheet#2

respectively, where a_i is the inlet aperture half-width, a_o the outlet aperture half-width of the flat disk receiver, r_o the radius of the circular tubular receiver, and θ_{acc} the acceptance half-angle of the solar concentrator, as indicated in Fig. 7.12. Accordingly, $\Phi_{rim,dish} = 45^\circ$ is chosen for the parabolic dish and $\Phi_{rim,trough} = 90^\circ$ is chosen for the trough concentrator. The acceptance half-angle of the parabolic dish concentrator is set to $\theta_{acc,dish} = 17.5$ mrad, which corresponds to the acceptance half-angle of the detector used in the specular reflectance measurements. For the parabolic trough concentrator, $\theta_{acc,trough} = 12.0$ mrad [9]. This acceptance angle applies at the rim where the distance to the focus is maximal. The corresponding angles calculated at the vertex and at the mean focus distance are 23.9 and 17.9 mrad, respectively. Accordingly, the detector acceptance angle used in the specular reflectance measurements closely resembles the average conditions in state-of-the-art parabolic trough concentrators. a_i is chosen in both cases as 3 m, which is in the range of current parabolic trough concentrators [9]. The semi-transparent top film, if present, is considered to form a convex shape of circular cross-section with radius of curvature of 8.4 m and 4.6 m for 100 μ m films of ETFE and FEP, respectively.

The sun is modelled as a solar disk with a half-subtended angle of 4.65 mrad and spectral irradiance distribution according to the ASTM G173 – 03 AM 1.5 reference spectrum [46]. Thus, the source divergence angle $\theta_{src} = 4.65$ mrad. The concentrator is assumed to be free of surface slope, shape, alignment and tracking errors to simulate the maximum attainable concentration with the different materials. For the parabolic trough concentrator, either the parabolic shape or an array of tangentially adjacent circular arc segments is applied [13], as the latter can reach the theoretical geometric concentration of a parabolic shape [83]. For

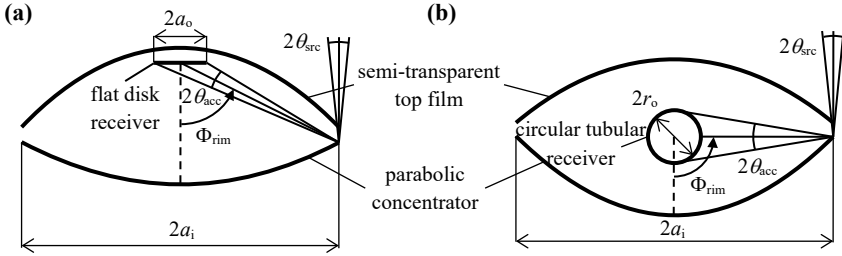


Fig. 7.12: Cross-sectional geometry of a) a parabolic dish concentrator with a flat disk receiver and b) a parabolic trough concentrator with a circular tubular receiver covered by a protective top film.

the parabolic dish concentrator, optical properties measured at $\theta = 15^\circ$ are used because θ varies between 0° and 22.5° and the measured reflective properties are generally weakly dependent on incidence angle. For the parabolic trough concentrator, properties measured at the skew angle, i.e. the incidence angle of sunrays on the concentrator inlet aperture $\theta = \theta_{\text{skew}}$, are used, where for 0° and 30° reflective data measured at 15° and interpolated data between 15° and 45° are used, respectively. The solar receiver is modelled as a blackbody absorber. The optical efficiency is defined as the solar radiative power absorbed by the solar receiver divided by the solar radiation incident on the inlet aperture of the solar concentrator,

$$\eta_{\text{optical}} = \frac{\Phi_{\text{receiver}}}{A_i E_{\text{DNI}} \cos \theta_{\text{skew}}} \quad (7.21)$$

where A_i is the inlet aperture area of the concentrator and $\theta_{\text{skew}} = 0^\circ$ for the parabolic dish. The average solar concentration ratio is defined as

$$C = \frac{\Phi_{\text{receiver}}}{A_o E_{\text{DNI}}} = \cos \theta_{\text{skew}} \eta_{\text{optical}} C_g \quad (7.22)$$

7.5.3 Results

Parabolic dish concentrator

The numerically simulated, circumferentially averaged solar flux distribution on the flat disk receiver as a function of the radial coordinate is shown in Fig. 7.13

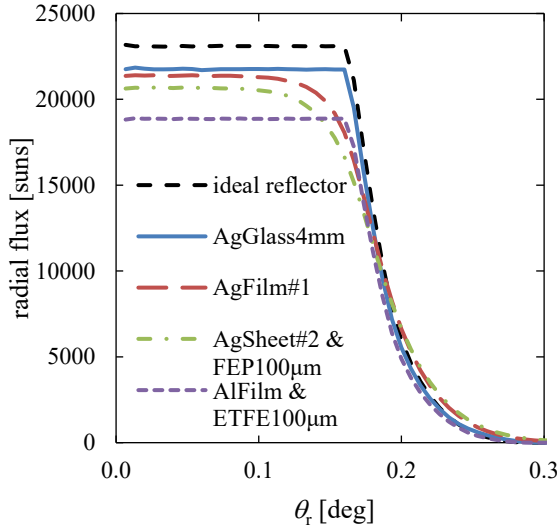


Fig. 7.13: Numerically simulated, circumferentially averaged solar flux distribution on the flat disk receiver of a parabolic dish collector as a function of the radial coordinate expressed in terms of the receiver acceptance angle for an ideal reflector and four different concentrator material configurations.

for the four different material configurations and for an ideal parabolic dish concentrator with unity reflectivity and perfect specularity (reference). For convenience, the radial coordinate, which may be written from Eq. (7.19) as,

$$r = a_i \frac{\cos \theta_r \sin \theta_r}{\cos \Phi_{\text{rim}} \sin \Phi_{\text{rim}}} = a_i \frac{\sin 2\theta_r}{2 \cos \Phi_{\text{rim}} \sin \Phi_{\text{rim}}} \quad (7.23)$$

is represented in Fig. 7.11 as the radial acceptance angle of the solar receiver,

$$\theta_r = \frac{1}{2} \arcsin \left(2 \cos \Phi_{\text{rim}} \sin \Phi_{\text{rim}} \frac{r}{a_i} \right) \approx \frac{r}{a_i} \cos \Phi_{\text{rim}} \sin \Phi_{\text{rim}} \quad (7.24)$$

For all material configurations, the concentrated solar radiation is absorbed within the design acceptance angle of 1° . Consequently, the intercept factor, defined as the ratio of solar radiation intercepted by the receiver aperture to the solar radiation reflected from the concentrator, is equal to 1 in all cases. Additionally, the flux distributions created by the highly specular materials

(AgGlass4mm, AlFilm) differ from the flux profile of an ideal reflector only in terms of the absolute value of solar concentration due to absorption, whereas the flux distribution shapes are nearly identical. Accordingly, angular scattering is negligible for these materials. The solar flux distributions of the less specular reflector materials, namely AgFilm#1 and AgSheet#2, show some angular dispersion at the edge of the distribution, indicating that angular scattering should be taken into account for these materials to accurately simulate the solar flux distribution on a receiver. The average solar concentration ratio (in suns) is 772 for AgGlass4mm, 759 for AgFilm#1, 670 for AlFilm & ETFE100 μ m, and 733 for AlSheet#2 & FEP100 μ m, compared to the theoretical geometric concentration of 820 with an ideal reflector. The highly specular AlFilm potentially allows surpassing the average solar concentration achieved by AgFilm#1 and AgSheet#2 at a given intercept factor.

Parabolic trough concentrator

The optical efficiency of the parabolic trough concentrator for the four different material configurations is shown in Fig. 7.14 as a function of the skew angle. Similar to the parabolic dish concentrator, the circular tubular receiver intercepts all rays reflected from the parabolic trough concentrator, i.e. an intercept factor of unity is achieved at all incidence angles. Solar concentrators comprising a semi-transparent top film and an indoor reflector material are generally outperformed by outdoor solar reflectors in terms of η_{optical} particularly at large incidence angles, since the latter do not suffer from Fresnel losses. However, the protective environment created by the semi-transparent top film has some practical advantages such as no soiling of reflector/receiver, reduced material degradation, and reduced sensitivity to wind.

7.6 Conclusion

The optical properties of various materials for solar concentrators were measured over the solar spectrum and as a function of the incidence angle. The spectrally and directionally resolved values were used to determine the solar-weighted specular reflectance and narrow-angle transmittance, as well as the angular scattering. These were incorporated in Monte Carlo ray-tracing simulations to

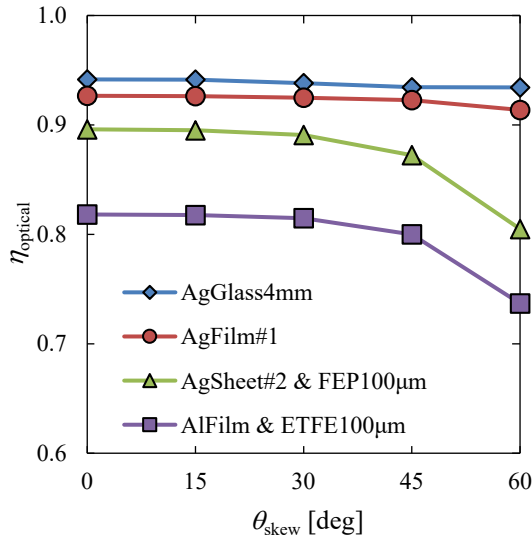


Fig. 7.14: Optical efficiency as a function of skew angle for a parabolic trough concentrator with a circular tubular receiver and four different material configurations.

elucidate their effect in the optical performance of solar concentrators. It is found that the optical efficiency and the solar concentration ratio are strongly dependent on the reflectance and transmittance of materials, while the effect of non-specularity is secondary and affects rather the solar flux distribution. For a parabolic dish concentrator of 45° rim angle and ideal geometry, the highest solar concentration ratio is obtained for back-silvered glass, followed by silvered polymer film, silvered aluminum sheet covered by FEP film, and aluminized boPET film covered by ETFE film. For a parabolic trough concentrator of 90° rim angle and ideal geometry, the percentage decrease of optical efficiency with incident skew angle from 0° to 60° is 1% for outdoor reflectors and 10% for indoor reflectors protected by semi-transparent film covers.

8 Summary and outlook

8.1 Summary

A new solar receiver for parabolic trough concentrators using air as heat transfer fluid (HTF) at operating temperatures up to 600 °C has been developed from scratch to a 1.2 MW_{th} solar collector system.

The proof of concept was delivered by a 1 m-long solar receiver prototype comprising 7 coiled absorber tubes that was designed, fabricated, tested, and numerically simulated. Model validation was accomplished by comparison to experimental results in terms of the temperature distributions on the absorber tube and the external surface of the window. On-sun experiments were performed on a 1.2 m-long, 4.85 m-aperture parabolic trough segment. With feeding rates in the range of 5 – 20 l_n/min to each absorber tube, air outlet temperatures of 621 – 449 °C and receiver efficiencies of 25 – 64% were achieved, while the pressure drop and pumping power demand remained below 10 mbar and 1% of the expected electric power output, respectively. Major heat losses occur at the primary mirror by absorption and spillage of sunlight, at the window by solar reflection, thermal re-radiation, and natural convection, and by other loss mechanisms including conduction through the insulation and chimney effect.

The validated model was further applied to a parametric study and yearly performance predictions of the new solar receiver design. For a typical high DNI place with 2400 kWh/m²/yr and air inlet and outlet temperatures of 250 °C and 570 °C, respectively, the maximum attainable annual mean DNI-to-thermal efficiency with an array of coiled absorber tubes is restricted to below 0.45.

Demonstration on an industrial scale was accomplished by a 212 m-long parabolic trough concentrator and associated solar receiver mounted on a concrete support structure and enclosed by a transparent envelope. The solar concentrator was constructed from a stack of inflatable polymeric films whose topmost layer is aluminized, and the solar receiver was designed for operation

with air at 500 °C. A numerical optical and heat transfer analysis of the complete solar collector unit was formulated and experimentally validated by comparison to measured data collected during the commissioning phase of the first-of-its-kind solar pilot plant in Ait Baha, Morocco. For a yearly DNI of 2400 kWh/m², model simulations predict an annual thermal energy output at 500 °C of 926 kWh/m² of primary solar concentrator aperture.

Multiple parallel straight tubes were investigated as solar absorbers in a receiver for line-focus solar radiation using air as heat transfer fluid. The thermodynamics of this receiver design in terms of pressure drop and temperature difference between the tube walls and HTF were studied and solar receivers were designed for two different operating temperatures. The steady-state energy conservation was formulated and solved numerically using the finite volume technique. For a yearly DNI of 2400 kWh/m² and realistic performance of the thermal insulation, annual mean DNI-to-thermal efficiencies of 0.37 and 0.42 for electricity generation ($T_{\text{in}} = 250\text{ °C}$, $T_{\text{out}} = 570\text{ °C}$) and industrial process heat ($T_{\text{in}} = 25\text{ °C}$, $T_{\text{out}} = 600\text{ °C}$) are predicted.

The theoretical optical and mechanical design of the pneumatic solar trough concentrator was established including the film length regulation mechanism based on inflatable sleeves and gravity effects. Experiments on the industrial scale solar concentrator revealed intercept factor variations of ± 0.14 within a single collector module due to deflection, and reductions by up to 0.30 during solar tracking due to temperature and gravity. The theory captured the behavior of the real system and was applied to the preliminary design of a control system for automatic pressure regulation.

The optical properties of various materials for solar concentrators were measured over the solar spectrum and as a function of the incidence angle. The spectrally and directionally resolved values were used to determine the solar-weighted specular reflectance and narrow-angle transmittance, as well as the angular scattering. These were incorporated in Monte Carlo ray-tracing simulations to elucidate their effect in the optical performance of solar concentrators in general, and were specifically applied to the numerical simulation of the industrial scale solar collector unit.

8.2 Outlook

8.2.1 Cross-flow solar receivers

Of all the solar receivers investigated in this thesis, the cross-flow concept based on an array of coiled absorber tubes shows the lowest heat losses through the windowed aperture at high solar DNI and near-normal incidence. Unlike conductive heat losses through the insulation, radiative and convective heat losses from the window are inevitable, which ascribes a high potential to this receiver design. However, the current technical implementation based on coiled tubes is rather sensitive to the solar incidence angle on the trough concentrator's aperture. This problem could be mitigated by using alternative cross-flow absorbers that preserve the 2D groove geometry of the incoming solar radiation. One possible solution based on U-duct absorbers is sketched in Fig. 8.1. By feeding the cold air at the bottom and extracting the hot air from the top, a positive temperature gradient may be established from the receiver window to the top, similar to the helically coiled tubes (cf. Fig. 2.7). Further potential cross-flow absorbers could be explored with the emphasis on low cost, ease of manufacturing and assembly, and the most promising designs could be tested by building small-scale receiver prototypes.

An alternative to changing the geometry of the solar receiver consists of matching the incoming solar radiation to the array of 3D coiled tubes. This could be achieved by line-to-point (LTP) secondary concentrators, based on which the array of coiled tubes was invented. Besides the decreased sensitivity to the skew angle, LTP focus would allow also for higher solar concentration ratios exceeding the theoretical limit of $215\times$ of line-focus systems [14, 78] and thus pave the way to higher operating temperatures. The major practical obstacle to the utilization of LTP secondary concentrators in solar thermal receivers are the elevated temperatures, which complicates the optical, mechanical, and thermal coupling of the hot solar receiver and actively cooled secondary optics. A further possible route of investigation is the application of the array of coiled tubes in alternative types of solar collectors such as a cross-linear solar concentration system [141] that produces an array of point foci in one line, or in point-focusing solar towers.

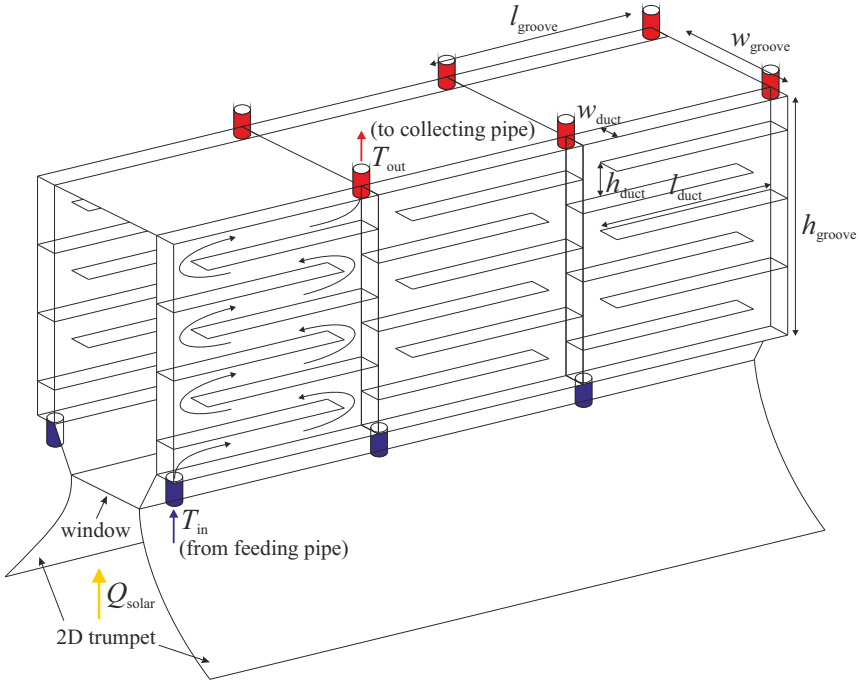


Fig. 8.1: Cross-flow receiver based on U-duct solar absorbers.

The design and construction of the thermal insulation is a critical task for any type of solar thermal receiver, as was underlined by the high conductive heat losses observed in both experimental campaigns. The most delicate part concerns the insulation between the hot solar absorber and the water-cooled secondary concentrator, which requires a careful assessment and re-engineering. In addition, cross-flow receivers are particularly sensitive to heat losses occurring in the collecting manifold pipe because these require the cross-flow absorbers to operate at temperatures above the receiver outlet temperature. While heat losses from the collecting pipe were acceptably low during the operation of the industrial scale solar collector unit, this needs to be verified at higher operating temperatures. Furthermore, thermal inertia, cost and weight of the solar receiver should be minimized by replacing solid insulation materials on the collecting pipe by radiation shields and decreasing the wall thickness of tubes where

possible. Moreover, a rigid and low-cost support structure of the solar receiver should be designed.

8.2.2 Straight tubular receivers

Straight tubular solar receivers offer a potentially simpler design, construction, and assembly with less components than cross-flow receivers and — according to numerical simulations — comparable annual mean efficiencies as the array of coiled tubes. As a next step, a receiver prototype should be designed, fabricated and tested to validate the numerical models experimentally. One challenge of straight tubular receivers is their need for either testing on full scale, which is time- and resource-intensive, or downsizing, which generally affects the thermodynamics because of different scaling of the various heat transfer mechanisms. This is probably a key advantage of the cross-flow receiver concept, whose solar absorbers can be characterized under the real operating conditions on short receiver sections.

8.2.3 Solar trough concentrators

The operation of the pneumatic arcspline concentrator in the industrial scale solar collector unit revealed several challenges. Manufacturing tolerances and beam deflection of the concrete support structure as well as weight of the pneumatic sleeves and longitudinal tension in the mirror films should be minimized. An automatic control system needs to be implemented that regulates the inflation pressures. The starting point is a feedforward controller that adjusts the pressures in the arcspline chambers to the actual temperature (and tracking angle, if needed) of the solar collector. For higher accuracy, a feedback control system is required that regulates the local sleeve pressures based on concentrator shape measurements, as outlined in Section 6.2. For testing such a model-based control system at a certain axial position on the solar collector, the recognition of circular arcs in the mirror shape scans needs to be automatized and suitable initial pressure have to be identified. The implementation on an industrial scale would require a 2-axis mirror shape measuring system that autonomously scans the full concentrator's aperture in the transversal and longitudinal directions during solar tracking while keeping the camera aligned with the direction of the sun.

Alternatively, the use of rigid solar reflector materials such as back-silvered glasses or metallized aluminum sheets could be considered, which eliminate the need for pressure control and auxiliary equipment, and generally have a higher solar-weighted reflectance than metallized films. While the relevant optical properties of the most common materials for solar concentrators have been characterized under lab conditions in this work, it is recommended to perform accelerated aging tests including thermal cycling before using them in the enclosed solar collector. Given the rough environment of solar collectors particularly for process heat next to factories that produce emissions, a protective transparent envelope appears favorable despite the additional optical losses. This allows for lightweight and low-cost concentrator support structures and innovative tracking mechanisms that would be worth investigating. Furthermore, alternative greenhouse concepts could be explored, aiming at minimum costs per square meter by minimizing material and energy contents, construction time, and maintenance.

Appendix A

Discretization of heat conduction equation¹

The 3D steady-state heat conduction equation (2.2) in arbitrary orthogonal coordinates (u_1, u_2, u_3) is given by

$$\nabla \cdot (k \nabla T) = 0 = \frac{1}{h_1 h_2 h_3} \left[\frac{\partial}{\partial u_1} \left(\frac{h_2 h_3}{h_1} k \frac{\partial T}{\partial u_1} \right) + \frac{\partial}{\partial u_2} \left(\frac{h_1 h_3}{h_2} k \frac{\partial T}{\partial u_2} \right) + \frac{\partial}{\partial u_3} \left(\frac{h_1 h_2}{h_3} k \frac{\partial T}{\partial u_3} \right) \right] \quad (\text{A.1})$$

where the scale factors h_i relate an infinitesimal change in coordinate u_i to the displacement of the position vector $\mathbf{r}$

$$h_i = \left| \frac{\partial \mathbf{r}}{\partial u_i} \right| = \left| \frac{\partial x}{\partial u_i} \hat{\mathbf{e}}_x + \frac{\partial y}{\partial u_i} \hat{\mathbf{e}}_y + \frac{\partial z}{\partial u_i} \hat{\mathbf{e}}_z \right|, \quad i = 1, 2, 3 \quad (\text{A.2})$$

with (x, y, z) denoting Cartesian coordinates. A useful coordinate transformation for the integration over a volume element of a torus is $(x, y, z) \rightarrow (r, \zeta, \phi)$

$$\begin{aligned} x &= (R + r \cos \phi) \cos \zeta \\ y &= (R + r \cos \phi) \sin \zeta \\ z &= r \sin \phi \end{aligned} \quad (\text{A.3})$$

where ζ and ϕ are the toroidal and poloidal angles, respectively, r is the minor radius, and the major radius R is a constant corresponding to the centerline radius of the coil ($R = R_{\text{coil}}$). The 3D steady state heat conduction equation (A.1) in the right-handed, orthogonal coordinate system (r, ζ, ϕ) reads

¹ Material in this appendix has been published in P. Good, G. Ambrosetti, A. Pedretti, and A. Steinfeld, "An array of coiled absorber tubes for solar trough concentrators operating with air at 600 °C and above," *Solar Energy*, vol. 111, pp. 378-395, 2015.

$$\begin{aligned} & \frac{1}{r(R+r\cos\phi)} \left[\frac{\partial}{\partial r} \left(r(R+r\cos\phi) k \frac{\partial T}{\partial r} \right) \right. \\ & \left. + \frac{\partial}{\partial \zeta} \left(\frac{r}{R+r\cos\phi} k \frac{\partial T}{\partial \zeta} \right) + \frac{\partial}{\partial \phi} \left(\frac{R+r\cos\phi}{r} k \frac{\partial T}{\partial \phi} \right) \right] = 0 \end{aligned} \quad (\text{A.4})$$

Integration of Eq. (A.4) over the control volume of a torus using the differential volume element $dV = r(R+r\cos\phi)d\phi d\zeta dr$ and considering first order terms only yields

$$\begin{aligned} \iiint_{\Delta V_{i,j,k}} \nabla \cdot (k \nabla T) dV &= \int_{r_{i-}}^{r_{i+}} \int_{\zeta_{j-}}^{\zeta_{j+}} \int_{\phi_{k-}}^{\phi_{k+}} \nabla \cdot (k \nabla T) r(R+r\cos\phi) d\phi d\zeta dr = \\ \Delta \zeta_j \Delta \phi_k \left[r(R+r\cos\phi_k) k \frac{\partial T}{\partial r} \right]_{r_{i-}}^{r_{i+}} &+ \Delta \phi_k \Delta r_i \left[\frac{r_i}{R+r_i\cos\phi_k} k \frac{\partial T}{\partial \zeta} \right]_{\zeta_{j-}}^{\zeta_{j+}} \\ &+ \Delta r_i \Delta \zeta_j \left[\frac{R+r_i\cos\phi}{r_i} k \frac{\partial T}{\partial \phi} \right]_{\phi_{k-}}^{\phi_{k+}} = 0 \end{aligned} \quad (\text{A.5})$$

where Δr_i , $\Delta \zeta_j$, $\Delta \phi_k$ are the dimensions of the control volume $\Delta V_{i,j,k}$ around node (i, j, k) . Further assuming a piecewise linear temperature profile between the nodes such that the gradients at the control volume interfaces are expressed by nodal temperature differences [45] yields

$$\begin{aligned} \Delta \zeta_j \Delta \phi_k \left[r_{i+} (R+r_{i+}\cos\phi_k) k_{i+,j,k} \frac{T_{i+1,j,k} - T_{i,j,k}}{\delta r_{i+}} \right. \\ \left. - r_{i-} (R+r_{i-}\cos\phi_k) k_{i-,j,k} \frac{T_{i,j,k} - T_{i-1,j,k}}{\delta r_{i-}} \right] \\ + \frac{r_i \Delta \phi_k \Delta r_i}{R+r_i\cos\phi_k} \left[k_{i,j+,k} \frac{T_{i,j+,k} - T_{i,j,k}}{\delta \zeta_{j+}} - k_{i,j-,k} \frac{T_{i,j,k} - T_{i,j-,k}}{\delta \zeta_{j-}} \right] \\ + \frac{\Delta r_i \Delta \zeta_j}{r_i} \left[(R+r_i\cos\phi_{k+}) k_{i,j,k+} \frac{T_{i,j,k+1} - T_{i,j,k}}{\delta \phi_{k+}} \right. \\ \left. - (R+r_i\cos\phi_{k-}) k_{i,j,k-} \frac{T_{i,j,k} - T_{i,j,k-1}}{\delta \phi_{k-}} \right] = 0 \end{aligned} \quad (\text{A.6})$$

where $\delta r_{i\pm}$, $\delta \zeta_{j\pm}$, and $\delta \phi_{k\pm}$ are the distances between nodes in the $\pm r$ -, $\pm \zeta$ -, and $\pm \phi$ -directions, respectively. The coefficients a of Eq. (2.4) are then obtained as

$$\begin{aligned}
 a_{i\pm 1,j,k} &= k_{i\pm,j,k} \frac{r_{i\pm} (R + r_{i\pm} \cos \phi_k) \Delta \zeta_j \Delta \phi_k}{\delta r_{i\pm}} \\
 a_{i,j\pm 1,k} &= k_{i,j\pm,k} \frac{r_i \Delta \phi_k \Delta r_i}{(R + r_i \cos \phi_k) \delta \zeta_{j\pm}} \\
 a_{i,j,k\pm 1} &= k_{i,j,k\pm} \frac{(R + r_i \cos \phi_{k\pm}) \Delta r_i \Delta \zeta_j}{r_i \delta \phi_{k\pm}}
 \end{aligned} \tag{A.7}$$

For completeness, the coefficients for a finite control volume $\Delta V_{i,j,k} = \Delta x_i \Delta y_j \Delta z_k$ of the rectangular window and groove walls in Cartesian coordinates are also given,

$$\begin{aligned}
 a_{i\pm 1,j,k} &= \frac{k_{i\pm,j,k} \Delta y_j \Delta z_k}{\delta x_{i\pm}} \\
 a_{i,j\pm 1,k} &= \frac{k_{i,j\pm,k} \Delta x_i \Delta z_k}{\delta y_{j\pm}} \\
 a_{i,j,k\pm 1} &= \frac{k_{i,j,k\pm} \Delta x_i \Delta y_j}{\delta z_{k\pm}}
 \end{aligned} \tag{A.8}$$

and the coefficient $a_{i,j,k}$ of temperature $T_{i,j,k}$ at node (i,j,k) is simply given by the sum

$$a_{i,j,k} = a_{i-1,j,k} + a_{i+1,j,k} + a_{i,j-1,k} + a_{i,j+1,k} + a_{i,j,k-1} + a_{i,j,k+1} \tag{A.9}$$

Fig. A.1 shows the finite volume discretization in a cross-section of the window, groove walls, and one loop of the coiled absorber tube. The numbers of segments used for each domain in the numerical simulations are listed in Table A.1, where the x -, y - and z -coordinates of the Cartesian domains are defined in the directions perpendicular to the surface, parallel to the surface in trough direction, and parallel to the surface perpendicular to trough direction, respectively.

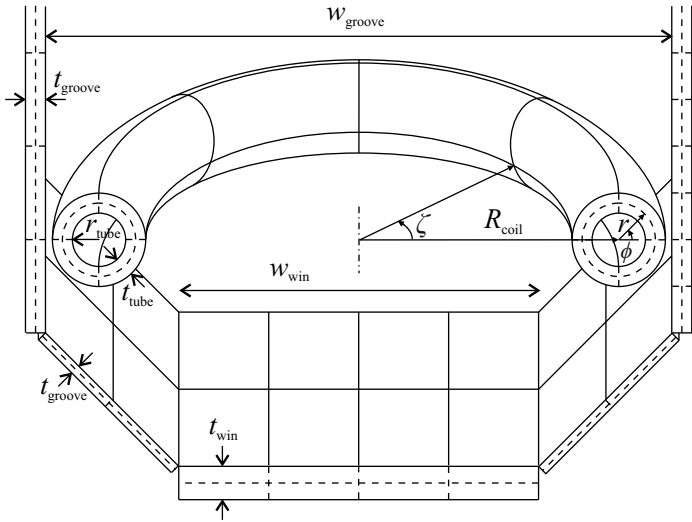


Fig. A.1: Cross-section of the discretized window, groove walls, and one loop of absorber tube. The coordinates (r, ζ, ϕ) used for the integration over a finite volume of a torus are also indicated.

Table A.1: Numbers of segments used for discretization of solar receiver.

Number of segments	n_i	n_j	n_k
Cartesian domains	x	y	z
window	2	4	4
groove walls	2	4	32
torus domains	r	ζ	ϕ
1 absorber tube loop	2	20	4
1 fluid flow loop	1	20	1

Appendix B

Solar flux measurements¹

The optical performance of the 5.8 m²-aperture primary trough concentrator (one wing) is characterized by flux gauge measurements at the focal plane coupled to CCD-Lambertian target imaging. The flux gauge measurement system comprises an array of 5 Gardon-type flux gauges (Vatell TG1000-0, flux range: 0 – 400 kW/m², accuracy: $\pm 3\%$, diameter 25.3 mm), which are mounted in one row on a rotatable, water-cooled copper disk (diameter: 60 cm) in radial intervals of 40 mm with the innermost flux gauge in the center of the disk of rotation. The disk is actuated by a remotely controlled step motor that allows for a 360° scan of the focal area in discrete angular steps [142]. The front surface of the copper disk is sprayed with alumina-based, diffuse white paint to serve as a Lambertian target. Images of the solar flux distribution are recorded by a CCD camera (Basler scA640-70gm, resolution: 659 × 494 pixels, frame rate: 70 fps) mounted on one edge of the primary concentrator halfway between the vertex and rim. The solar DNI is measured by the Kipp & Zonen CHP-1 pyrliometer (cf. section 2.4). Fig. B.1 shows the perspective-corrected images of the circular Lambertian target accommodating the 5 flux gauges during defocusing (a) and of the focal line during solar tracking (b). The dark spots in the image of the solar flux distribution caused by the flux gauges are corrected by interpolation from the surrounding non-disturbed regions.

Absolute flux gauge measurements are acquired over a full turn of the disk at 24 angular orientations in 15° intervals, which yields 120 data points. The calibration constant of the Zynolyte coated flux gauges — obtained by the manufacturer using hemispherical blackbody radiation from a source at 850 °C — is corrected for the gauge's spectral and angular response by a factor of

¹ Material in this appendix has been published in P. Good, G. Ambrosetti, A. Pedretti, and A. Steinfeld, "An array of coiled absorber tubes for solar trough concentrators operating with air at 600 °C and above," *Solar Energy*, vol. 111, pp. 378-395, 2015.

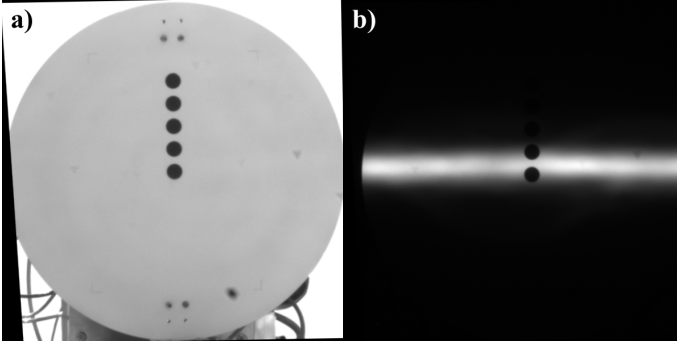


Fig. B.1: Images of the Lambertian target during defocusing (a) and solar tracking (b) at a turning angle of 0° .

$F_{\text{calibration}} = 0.906$, as recommended by Ulmer et al. [143]. Furthermore, the flux gauge measurements are normalized by the time-synchronized DNI measurements to obtain solar concentration,

$$C = \frac{\dot{q}_{\text{solar}}''}{E_{\text{DNI}}} = \frac{F_{\text{calibration}} \dot{q}_{\text{measured}}''}{E_{\text{DNI}}} \quad (\text{B.1})$$

The relative image of the solar flux distribution is scaled to the 120 absolute flux gauge measurements by a constant factor using the least-squares technique. The scaled, absolute flux map and the flux gauge measurements are shown in Fig. B.2. The average absolute and root-mean-square (RMS) errors between the image and flux gauge data points are 1.6 and 2.4 suns, respectively.

A representative solar flux distribution across the focal line (x -direction) is determined by averaging the 2D flux map in the trough direction over the y -interval of the central absorber tube. The specular reflectivity ρ_{primary} and standard deviation of the angular scattering σ_s of the primary concentrator are then fitted, such that the integrated solar power intercepted by the secondary inlet aperture calculated by the Monte Carlo simulation matches the experimentally measured solar power and the RMS error between the two distributions is minimized. This yields $\rho_{\text{primary}} = 83\%$, $\sigma_s = 3.5$ mrad, with average absolute and RMS errors of 1.8 and 2.1 suns, respectively. The experimentally measured and numerically simulated solar flux distributions at the tilted focal plane ($\Phi_{\text{tilt}} =$

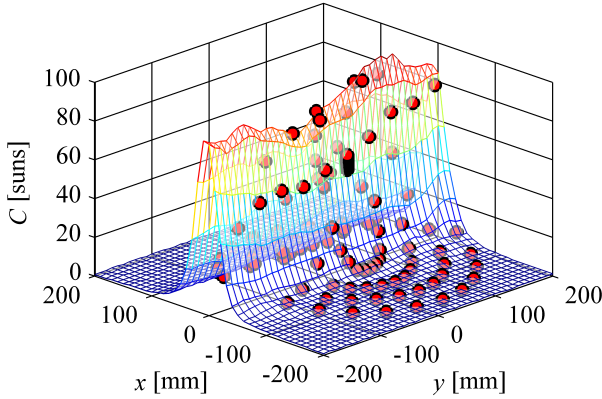


Fig. B.2: Scaled 2D solar flux distribution captured by the CCD camera (down-sampled to a 10 mm grid for clarity) and flux gauge measurements (filled circles).

42.5°) are shown in Fig. B.3. The inlet aperture of the secondary concentrator is also indicated by vertical dashed lines. The intercept factor amounts to $\gamma_{\text{primary}} = 92\%$ and the primary optical efficiency is $\eta_{\text{optical,primary}} = 77\%$. The lower surface reflectivity (83% compared to 88% stated by the manufacturer) is attributed to soiling and outdoor exposure of the reflective surface.

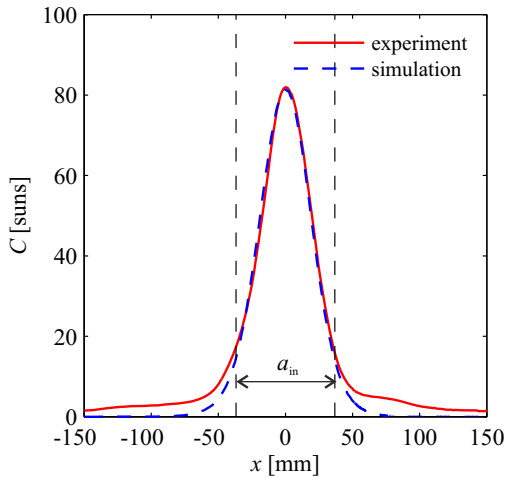


Fig. B.3: Experimentally measured (solid) and numerically simulated (dashed) solar flux distributions incident on the secondary inlet aperture (dashed, vertical) at the tilted focal plane.

Appendix C

Specular reflectance and narrow-angle transmittance¹

Spectral data for specular reflectance at 15° incidence angle and narrow-angle transmittance at normal incidence are given in Table C.1 and Table C.2, respectively. Type A measurement uncertainties are calculated from the estimated variance of the averaged voltage signals using the Gaussian error propagation formula. The maximum type A uncertainty occurring at 300 nm and the root-mean-square uncertainty over the measured spectral range are below 0.02 and 0.003, respectively. The type B uncertainty is estimated as 0.004 from spectral transmittance and reflectance measurements of an N-BK7HT glass sample with known optical properties [138]. Accordingly, the combined measurement uncertainty calculated with a confidence factor 2 (95% confidence) is usually within 0.01 for spectral specular reflectance and narrow-angle transmittance.

¹ Data in this appendix has been published in P. Good, T. Cooper, M. Querci, N. Wiik, G. Ambrosetti, and A. Steinfeld, "Spectral data of specular reflectance, narrow-angle transmittance and angle-resolved surface scattering of materials for solar concentrators," *Data in Brief*, vol. 6, pp. 184-188, 2016.

Table C.1: Spectral specular reflectance at 15° incidence angle.

Material λ [nm]	AgGlass 4mm	AgFilms		AlFilm	AgSheets			AlSheet
		#1	#2		#1	#2	#3	
300	0.117	0.043	0.042	0.848	0.194	0.158	0.178	0.099
310	0.075					0.165	0.172	0.049
320	0.090	0.043	0.042	0.854		0.225	0.134	0.060
325					0.439			
330	0.387					0.287	0.054	0.180
340	0.657	0.042	0.041	0.862		0.137	0.470	0.430
350	0.770	0.042	0.040	0.863	0.462	0.410	0.735	0.678
360	0.820	0.042	0.040	0.865		0.620	0.825	0.684
370	0.848	0.042	0.040	0.868		0.710	0.868	0.754
375					0.436			
380	0.867	0.047	0.041	0.871		0.755	0.894	0.766
390	0.885	0.206	0.143	0.872		0.783	0.915	0.748
400	0.902	0.569	0.557	0.872	0.712	0.822	0.928	0.769
425	0.922	0.902	0.841	0.877	0.889	0.915	0.952	0.757
450	0.932	0.939	0.880	0.876	0.931	0.947	0.961	0.816
475	0.943	0.944	0.901	0.881	0.947	0.958	0.966	0.817
500	0.948	0.951	0.915	0.882	0.953	0.964	0.968	0.808
550	0.954	0.958	0.931	0.882	0.960	0.967	0.968	0.864
600	0.956	0.961	0.940	0.881	0.962	0.967	0.968	0.910
650	0.958	0.966	0.947	0.879	0.962	0.965	0.964	0.899
700	0.957	0.970	0.953	0.873	0.963	0.964	0.960	0.881
750	0.958	0.970	0.957	0.862	0.963	0.962	0.957	0.842
800	0.953	0.972	0.960	0.848	0.961	0.959	0.954	0.897
850	0.956	0.974	0.963	0.848	0.962	0.958	0.959	0.812
900	0.949	0.974	0.967	0.873	0.956	0.953	0.952	0.905
950	0.947	0.977	0.970	0.903	0.954	0.951	0.954	0.867
1000	0.952	0.980	0.973	0.924	0.955	0.957	0.960	0.899
1050	0.951	0.979	0.971	0.934	0.953	0.956	0.959	0.932
1100	0.954	0.983	0.970	0.943	0.955	0.961	0.960	0.886
1150	0.953	0.976	0.966	0.947	0.955	0.965	0.961	0.905
1200	0.952	0.977	0.962	0.947	0.959	0.968	0.961	0.942
1250	0.950	0.986	0.970	0.947	0.959	0.968	0.961	0.927
1300	0.955	0.986	0.972	0.949	0.960	0.969	0.963	0.877
1350	0.955	0.973	0.965	0.955	0.963	0.970	0.965	0.896
1400	0.951	0.973	0.959	0.958	0.966	0.973	0.968	0.929
1450	0.953	0.981	0.966	0.956	0.968	0.973	0.965	0.940
1500	0.954	0.987	0.974	0.957	0.969	0.973	0.968	0.932

Table C.1 (continued)

Material λ [nm]	AgGlass 4mm	AgFilms		AlFilm	AgSheets			AlSheet
		#1	#2		#1	#2	#3	
1550	0.957	0.982	0.971	0.960	0.969	0.974	0.966	0.901
1600	0.957	0.981	0.968	0.960	0.973	0.975	0.969	0.887
1650	0.954	0.965	0.941	0.960	0.972	0.978	0.967	0.911
1700	0.951	0.906	0.905	0.966	0.974	0.980	0.967	0.938
1750	0.949	0.927	0.917	0.965	0.973	0.980	0.969	0.948
1800	0.952	0.948	0.941	0.965	0.974	0.982	0.969	0.951
1850	0.945	0.962	0.954	0.967	0.975	0.982	0.966	0.948
1900	0.940	0.931	0.937	0.966	0.976	0.982	0.966	0.935
1950	0.940	0.940	0.941	0.967	0.978	0.984	0.969	0.910
2000	0.936	0.956	0.953	0.968	0.978	0.984	0.970	0.902
2050	0.930	0.953	0.953	0.973	0.980	0.989	0.970	0.904
2100	0.927	0.926	0.936	0.967	0.981	0.985	0.976	0.916
2150	0.915	0.901	0.882	0.970	0.981	0.986	0.973	0.932
2200	0.876	0.868	0.899	0.970	0.981	0.987	0.971	0.943
2250	0.862	0.400	0.590	0.968	0.978	0.988	0.959	0.949
2300	0.874	0.544	0.594	0.975	0.980	0.986	0.964	0.945
2350	0.877	0.659	0.659	0.963	0.985	0.983	0.980	0.955
2400	0.863	0.579	0.660	0.976	0.980	0.988	0.979	0.951
2450	0.842	0.652	0.611	0.974	0.981	0.988	0.964	0.939
2500	0.820	0.747	0.765	0.972	0.980	0.992	0.982	0.942

Table C.2: Spectral narrow-angle transmittance at normal incidence.

λ [nm]	ETFE100 μ m	FEP100 μ m	Borosilicate3.3mm	BorosilicateAR3.3mm
300	0.616	0.784	0.611	0.554
310	0.645	0.807	0.756	0.706
320	0.675	0.825	0.839	0.806
330	0.704	0.846	0.886	0.858
340	0.729	0.853	0.902	0.887
350	0.749	0.866	0.912	0.901
360	0.765	0.872	0.916	0.912
370	0.782	0.884	0.917	0.914
380	0.794	0.890	0.918	0.916
390	0.806	0.899	0.919	0.923
400	0.815	0.902	0.922	0.929
425	0.839	0.915	0.922	0.936
450	0.859	0.924	0.919	0.939
475	0.871	0.928	0.922	0.947
500	0.882	0.933	0.921	0.957
550	0.902	0.941	0.924	0.961
600	0.909	0.949	0.926	0.966
650	0.919	0.950	0.923	0.966
700	0.924	0.950	0.923	0.965
750	0.929	0.952	0.923	0.965
800	0.934	0.951	0.920	0.962
850	0.936	0.954	0.921	0.958
900	0.940	0.955	0.920	0.952
950	0.936	0.956	0.920	0.952
1000	0.943	0.958	0.919	0.949
1050	0.943	0.958	0.920	0.944
1100	0.944	0.956	0.920	0.944
1150	0.945	0.957	0.922	0.942
1200	0.942	0.960	0.925	0.941
1250	0.947	0.963	0.923	0.940
1300	0.941	0.963	0.926	0.941
1350	0.943	0.960	0.923	0.936
1400	0.947	0.963	0.911	0.925
1450	0.946	0.964	0.924	0.935
1500	0.946	0.960	0.924	0.937
1550	0.946	0.958	0.926	0.936
1600	0.947	0.958	0.926	0.937
1650	0.945	0.957	0.922	0.936

Table C.2 (continued)

λ [nm]	ETFE100 μ m	FEP100 μ m	Borosilicate3.3mm	BorosilicateAR3.3mm
1700	0.920	0.957	0.923	0.939
1750	0.932	0.957	0.927	0.936
1800	0.941	0.957	0.925	0.935
1850	0.946	0.956	0.926	0.934
1900	0.947	0.956	0.925	0.936
1950	0.945	0.954	0.927	0.935
2000	0.946	0.957	0.923	0.929
2050	0.941	0.955	0.923	0.930
2100	0.945	0.954	0.918	0.926
2150	0.944	0.958	0.905	0.908
2200	0.943	0.955	0.892	0.891
2250	0.840	0.958	0.893	0.893
2300	0.843	0.955	0.897	0.900
2350	0.847	0.963	0.886	0.888
2400	0.848	0.953	0.856	0.856
2450	0.883	0.966	0.843	0.847
2500	0.907	0.961	0.822	0.829

Appendix D

Superimposed Gaussian scattering distributions¹

The identified parameters of a superposition of two Gaussian distributions for the description of angular scattering by reflection from silvered polymer films and aluminum sheets are listed in Table D.1.

¹ Data in this appendix has been published in P. Good, T. Cooper, M. Querci, N. Wiik, G. Ambrosetti, and A. Steinfeld, "Spectral data of specular reflectance, narrow-angle transmittance and angle-resolved surface scattering of materials for solar concentrators," *Data in Brief*, vol. 6, pp. 184-188, 2016.

Table D.1: Fractions and standard deviations (in mrad) of superposition of two Gaussian angular scattering distributions as a function of wavelength and incidence angle for AgFilms#1 and #2 and AgSheets#1 and #2.

Material	θ [deg]	15°			45°			60°		
	λ [nm]	F_1	σ_1	σ_2	F_1	σ_1	σ_2	F_1	σ_1	σ_2
AgFilm#1 marks parallel to plane of incidence	400	0.71	0.26	0.76	0.57	0.27	0.67	0.71	0.40	0.89
	500	0.46	0.18	0.51	0.54	0.25	0.60	0.76	0.43	0.93
	555	0.37	0.16	0.47	0.57	0.25	0.62	0.79	0.44	0.96
	600	0.29	0.11	0.43	0.62	0.27	0.62	0.78	0.43	0.95
	700	0.30	0.10	0.43	0.64	0.27	0.63	0.85	0.45	1.09
	800	0.27	0.09	0.41	0.58	0.24	0.62	0.74	0.39	0.96
	950	0.34	0.10	0.42	0.20	0.07	0.43	0.80	0.40	1.01
	1050	0.32	0.09	0.41	0.70	0.26	0.66	0.84	0.42	1.09
AgFilm#1 marks perp. to plane of incidence	400	0.58	0.56	1.28	0.68	0.69	1.62	0.43	0.60	1.58
	500	0.49	0.47	1.08	0.43	0.52	1.21	0.30	0.47	1.37
	555	0.40	0.42	1.02	0.36	0.45	1.09	0.25	0.36	1.28
	600	0.28	0.32	0.90	0.21	0.26	0.97	0.28	0.40	1.30
	700	0.34	0.30	0.89	0.23	0.27	0.96	0.35	0.45	1.36
	800	0.31	0.25	0.88	0.38	0.34	1.05	0.50	0.48	1.53
	950	0.40	0.27	0.90	0.51	0.38	1.19	0.59	0.48	1.64
	1050	0.48	0.35	1.00	0.54	0.34	1.18	0.65	0.55	1.91
AgFilm#2 marks parallel to plane of incidence	400	0.32	0.35	0.87	0.35	0.36	0.99	0.35	0.30	1.16
	500	0.38	0.32	0.89	0.36	0.29	0.97	0.39	0.24	1.18
	555	0.41	0.31	0.90	0.38	0.25	0.97	0.41	0.20	1.16
	600	0.42	0.29	0.90	0.39	0.23	0.96	0.43	0.18	1.16
	700	0.42	0.23	0.86	0.39	0.16	0.91	0.47	0.14	1.13
	800	0.42	0.19	0.86	0.47	0.16	0.99	0.54	0.15	1.24
	950	0.46	0.15	0.85	0.50	0.10	0.93	0.57	0.08	1.08
	1050	0.50	0.12	0.87	0.55	0.09	0.95	0.59	0.07	1.08
AgFilm#2 marks perp. to plane of incidence	400	0.18	0.30	0.86	0.16	0.32	1.01	0.83	1.01	2.43
	500	0.28	0.31	0.89	0.25	0.30	1.02	0.21	0.38	1.27
	555	0.32	0.31	0.90	0.31	0.30	1.06	0.24	0.33	1.26
	600	0.35	0.31	0.91	0.32	0.27	1.05	0.26	0.31	1.25
	700	0.39	0.27	0.93	0.39	0.26	1.07	0.28	0.23	1.20
	800	0.45	0.26	0.96	0.36	0.22	1.05	0.22	0.16	1.20
	950	0.46	0.22	0.93	0.40	0.15	0.94	0.30	0.17	1.16
	1050	0.44	0.18	0.89	0.41	0.12	0.97	0.40	0.15	1.22

Table D.1 (continued)

Material	θ [deg]	15°			45°			60°		
	λ [nm]	F_1	σ_1	σ_2	F_1	σ_1	σ_2	F_1	σ_1	σ_2
AgSheet#1 marks perp. to plane of incidence	350	0.65	0.27	1.01	0.66	0.20	0.89	0.67	0.15	0.80
	400	0.66	0.24	0.98	0.68	0.18	0.88	0.69	0.13	0.77
	500	0.68	0.20	0.90	0.73	0.15	0.86	0.74	0.12	0.75
	555	0.69	0.18	0.85	0.73	0.13	0.82	0.76	0.11	0.73
	600	0.68	0.16	0.80	0.74	0.12	0.78	0.77	0.11	0.72
	700	0.65	0.11	0.66	0.74	0.09	0.67	0.81	0.10	0.72
	800	0.61	0.06	0.53	0.74	0.07	0.57	0.82	0.09	0.69
	950	0.63	0.03	0.48	0.75	0.06	0.49	0.84	0.09	0.64
	1050	0.67	0.02	0.49	0.76	0.04	0.47	0.85	0.07	0.61
AgSheet#2 marks perp. to plane of incidence	350	0.60	0.22	1.12	0.72	0.23	1.14	0.86	0.28	1.39
	400	0.63	0.20	1.08	0.76	0.22	1.15	0.88	0.28	1.50
	500	0.69	0.16	1.00	0.81	0.21	1.14	0.92	0.26	1.58
	555	0.71	0.15	0.98	0.83	0.20	1.17	0.93	0.26	1.65
	600	0.73	0.15	0.95	0.85	0.20	1.17	0.94	0.26	1.69
	700	0.77	0.14	0.93	0.88	0.20	1.21	0.96	0.26	1.82
	800	0.79	0.13	0.90	0.90	0.19	1.20	0.97	0.26	1.90
	950	0.81	0.11	0.78	0.91	0.19	1.09	0.98	0.25	1.92
	1050	0.80	0.10	0.66	0.91	0.18	0.99	0.98	0.25	2.10

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List of publications

Journal articles

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