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Numerical analysis of a solar tower receiver tube operated with liquid metals

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ABSTRACT

Computational fluid dynamics is used in the present work to analyze the conjugate heat transfer in the receiver tube of a solar thermal tower operated with a liquid metal. A circumferentially and longitudinally non-uniform heat flux, due to solar irradiation, is applied on half the external surface while the other one is considered as insulated. The heat transfer mechanism of liquid metals differs from that of ordinary fluids. As a consequence, the Reynolds analogy, which assumes a constant turbulent Prandtl number close to unity, cannot be applied to these fluid flows. Therefore two additional equations, namely one for the temperature variance and one for its dissipation rate are additionally solved, in order to determine the turbulent thermal diffusivity. The effects of the wall thickness ratio, the solid-to-fluid thermal conductivity ratio, the Péclet number and the diameter-to-length ratio have been analyzed.

The calculated average Nusselt numbers closely agree with those evaluated with appropriate correlations for liquid metals, valid for uniformly distributed heat flux. Nonetheless these are not suited to evaluate the local Nusselt number and wall temperature distribution.

1. Introduction

Liquid metals are considered as efficient heat transfer media in many processes with exceptionally high thermal loads. They have been already proposed in the past as high temperature heat transfer media in concentrating solar power systems. Indeed, during the 80s, tests have been carried out on a demonstration plant (Plataforma Solar de Almeria) operated with a liquid sodium-cooled central receiver [1,2]. Unfortunately, the experiments have been stopped after a fire, caused by a sodium leakage during improper maintenance work. Recently, after a period of reduced interest in that approach, several new efforts have been reported, such as thermodynamic evaluation of candidate liquid alloys [3], as well as corrosion and heat transfer tests [4]. Further works, as e.g. those reported in Refs. [5–10] have recently highlighted the attractive properties of liquid metals for Concentrated Solar Power (CSP) applications. Presently a small concentrating solar power system operated with lead-bismuth eutectic ($Pr = 0.025$) is under construction at the Karlsruhe Institute of Technology in Germany, as discussed in Ref. [11].

In a solar tower plant, a field of sun-tracking mirrors, called heliostats, concentrate the sunlight onto a tower-mounted, centrally located receiver. The incident solar energy heats a fluid which, in case of a liquid metal, subsequently transfers its energy to the operating fluid of the power cycle. Compared to other technologies such as parabolic through, Fresnel and dish collectors, solar tower thermal plants are expected to reach lower levelized electricity costs [12,13]. The central receiver is a key component and accounts for about 15% of the total investment costs [14]. One typical arrangement for the receiver is based on arrays of parallel tubes, which are cooled from the inside by a heat transfer fluid and heated from the outside by the concentrated sunlight. In this design, only half of the tubes' surface is exposed to solar irradiance, resulting in a strongly non-uniform heat flux on the outer wall. This can result in high thermal stresses in the tube walls, whose magnitude depends on the cooling effect of the heat transfer fluid. Thus, for a proper thermo-hydraulic, as well as mechanical design of the receiver, good knowledge of the local wall temperatures and convective heat transfer coefficients is required.

Liquid metals are characterized by a high molecular thermal

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Nomenclature

Roman letters

C_f	Fanning friction factor $(-)$
c_p	specific heat capacity at constant pressure (J/kgK)
C_μ, $C_{1\varepsilon}$, $C_{2\varepsilon}$	constants in momentum turbulence model, Table 1 $(-)$
C_θ, C_{d1}, C_{p1}, C_{p2}, $Pr_{t\infty}$	constants in heat turbulence model, Table 2 $(-)$
C_{d2}	function in heat turbulence model, Eq. (32) $(-)$
D	internal diameter $D = 2r_i$ (m)
$f_{1\mu}$, $f_{2\mu}$, f_ε	functions in momentum turbulence model, Eqs. (16)–(18) $(-)$
$f_{1\theta}$, $f_{2\theta}$, $f_{2a\theta}$, $f_{2b\theta}$	functions in heat turbulence model, Eqs. (27) – (30) $(-)$
Gr	Grashof number $Gr = g\beta q'' D^4/(\lambda\nu^2)$ $(-)$
$\boldsymbol{I}$	identity vector $(-)$
k	turbulent kinetic energy (m^2/s^3)
k_θ	variance of temperature fluctuations, Eq. (9) (K^2)
L	tube's length (m)
Nu	Nusselt number $(-)$
P	average static pressure (Pa)
P_θ	production term in heat turbulence model, Eq. (25) (K^2/s)
P_k	turbulent kinetic energy production, Eq. (14) (m^2/s^3)
Pe	Péclet number $Pe = RePr$ $(-)$
Pr	Prandtl number $(-)$
Pr_t	turbulent Prandtl number $Pr_t = \nu_t/\alpha_t$
$\tilde{q}''_w$	non-dimensional wall heat flux $(-)$
q''	heat flux (W/m^2)
R	ratio between thermal and dynamical characteristic time $R = \tau_\theta/\tau_u$, Eq. (31) $(-)$
R_δ	Kolmogorov Reynolds number, Eq. (20) $(-)$
R_t	turbulent Reynolds number, Eq. (19) $(-)$
Re	Reynolds number $Re = u_b D/\nu$ $(-)$
Ri	Richardson number $Ri = Gr/Re^2$ $(-)$
r	tube radius (m)
r^*	outer-to-inner radius ratio $r^* = r_o/r_i$ non-dimensional radial coordinate $r^* = r/r_i$ $(-)$
r^+	non-dimensional wall distance $r^+ = u_\tau r/\nu$ $(-)$
$\boldsymbol{S}$	mean strain rate tensor (s^{-1})
T	temperature (K)
T'	temperature fluctuations (K)
T^+	non-dimensional temperature $T^+ = (T_w - T)/T_\tau$ $(-)$
T_τ	friction temperature $T_\tau = q''/\rho u_\tau c_p$ (K)
T_{b0}	inlet bulk temperature (K)
t	time (s)
$\boldsymbol{u}$	velocity vector (m/s)
$\boldsymbol{u}'$	vector of mean velocity fluctuations (m/s)
u'_r	fluctuating radial velocity (m/s)
$\overline{\boldsymbol{u}'T'}$	turbulent heat flux vector (mK/s)
u_τ	friction velocity $u_\tau = (\tau_w/\rho)^{0.5}$ (m/s)
$\overline{u'_r T'}^+$	non-dimensional radial turbulent heat flux $\frac{\overline{u'_r T'}}{u_\tau T_\tau}$ $(-)$
$\overline{\boldsymbol{u}' \otimes \boldsymbol{u}'}$	Reynolds stress tensor (m^2/s^2)
Δx	dimensional axial mesh spacing (m)
Δx^+	non-dimensional axial mesh spacing $\Delta x^+ = u_\tau \Delta x/\nu$ $(-)$
$\tilde{x}$	non-dimensional axial coordinate $\tilde{x} = x/L$ $(-)$
y^+	non-dimensional wall distance $y^+ = u_\tau\delta/\nu$ $(-)$

Greek letters

α	molecular thermal diffusivity (m^2/s)
α_t	turbulent thermal diffusivity (m^2/s)
δ	distance from the wall (m)
ε	turbulent kinetic energy dissipation rate (m^2/s^3)
ε_θ	dissipation rate of temperature fluctuations, Eq. (9) (K^2/s)
θ_{iw}, θ_{ow}, θ_b	dimensionless inner Eq. (38), outer Eq. (39) and bulk Eq. (40) temperature
λ	molecular thermal conductivity (W/mK)
λ^*	solid-to-fluid thermal conductivity ratio $\lambda^* = \lambda_s/\lambda_f$
ν	kinematic viscosity (m^2/s)
ν_t	turbulent kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
σ_k, σ_ε	constants in momentum turbulence model, Table 1
$\sigma_{k\theta}$, $\sigma_{\varepsilon\theta}$	constants in heat turbulence model, Table 2
τ_w	wall shear stress (N/m^2)
$\tau_{l\theta}$	local thermal characteristic time, Eq. (26) (s)
τ_{lu}	local dynamical characteristic time, Eq. (15) (s)
τ_θ	thermal turbulent characteristic time $\tau_\theta = k_\theta/\varepsilon_\theta$ (s)
τ_u	dynamical turbulent characteristic time $\tau_u = k/\varepsilon$ (s)

Operators

$\overline{(\bullet)}$	Reynolds-averaged values
$\langle\bullet\rangle$	circumferentially averaged value cross-section averaged
$\langle\bullet\rangle_L$	circumferentially and longitudinally averaged value
$(\bullet)$	scalar product
∇	gradient
$(\bullet)^T$	transposed matrix
$(\bullet)\vert_w$	value at the wall
$\otimes$	outer product

Subscripts

b	bulk
f	fluid
fd	fully developed
i	inner
o	outer
s	solid
iw	inner wall
ow	outer wall

conductivity and a low Prandtl number. This results in a heat transfer mechanism different from that of fluids with medium-to-high Pr numbers. Indeed, the thickness of the thermal viscous sublayer is considerably greater than that of the hydrodynamic viscous sublayer [15]. As a consequence, neither the Reynolds analogy [16], which assumes a constant turbulent Prandtl number close to unity, nor the well established Gnielinski or Dittus–Boelter correlations [17] can be applied to liquid metal flows. A recent review of the available Nusselt number correlations appropriate for fully developed forced convection to liquid metal flows in tubes can be found in Ref. [18]. Moreover, the above mentioned circumferentially non-uniform heat flux distribution of the concentrated sunlight creates some doubt about the applicability of correlations developed for uniformly heated tubes.