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Numerical modeling of the impact of regenerator housing on the determination of Nusselt numbers

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ABSTRACT

It is suggested that the housing of regenerators may have a significant impact when experimentally determining Nusselt numbers at low Reynolds and large Prandtl numbers. In this paper, a numerical model that takes the regenerator housing into account as a domain that is thermally coupled to the regenerator fluid is developed. The model is applied to a range of cases and it is shown that at low Reynolds numbers (well below 100) and at Prandtl numbers appropriate to liquids (7 for water) the regenerator housing may influence the experimental determination of Nusselt numbers significantly.

The impact of the housing on the performance during cyclic steady-state regenerator operation is quantified by comparing the regenerator effectiveness for cases where the wall is ignored and with cases where it is included. It is shown that the effectiveness may be decreased by as much as 18% for the cases considered here. A reduced number of transfer units (NTU_{eff}) is proposed based on the calculated regenerator effectiveness that accounts for the effect of the housing heat capacity.
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1. Introduction

The heat transfer characteristics of packed beds operating at low Reynolds numbers (below 100) and using aqueous heat transfer fluids (with high Prandtl number) are not abundant in literature. Generally, the behavior of regenerators using gases and high Reynolds numbers are the focus of research due to their applications in regenerative cryogenic refrigeration cycles and energy storage. However, certain research areas including near room temperature magnetic refrigeration rely on highly efficient regenerators operating at relatively low Reynolds numbers (ranging between 1 and 100, approximately) using high Prandtl number heat transfer fluids.

It is well known that the heat transfer coefficient, h, is a function of the Reynolds number and Prandtl number. It is also apparent that for regenerator geometries based on packed particles (and other similar geometries) the heat transfer coefficient, or Nusselt number, increases as a function of Reynolds number following some power law. However, the experimental determination of the Nusselt number at low Reynolds numbers and using high Prandtl number fluids is experimentally difficult as shown below and not available in detail in the literature.

It is non-trivial to derive accurate heat transfer coefficients from experiments at low Reynolds numbers [1]. Thermal interaction with the ambient (i.e., parasitic losses), axial conduction and the housing of the heat exchanger are all issues that can significantly affect the measurements under these conditions. In Ref. [2] a factor as a function of the non-dimensional wall thickness is suggested as a correction for the thermal lag caused by the thermal interaction between the heat exchanger and the surrounding housing/wall.

It is common to apply a numerical model where the heat transfer coefficient may be adjusted in order to match predicted behavior with the experimental data (typically in the form of fluid outlet temperature as a function of time). Techniques for doing this have been applied for decades (see, e.g., Ref. [3]). If the applied numerical model is not sufficiently accurate or if it ignores important physical effects then the resulting Nu–Re correlation may become inaccurate and unphysical; this would be the case if the Nusselt number goes to zero or even becomes negative in the limit when that the Reynolds number approaches zero.

In this paper we propose that the regenerator wall/housing may have an influence on the experimental determination of the heat transfer properties at low Reynolds numbers. That is, the apparent (or measured) heat transfer coefficient m be substantially different from the actual heat transfer coefficient. We also propose that the regenerator wall/housing may have a significant influence on the performance of a regenerator at low Reynolds number. That is, the effectiveness of the regenerator under periodic steady-state operating conditions may be substantially reduced. The wall may act as a passive regenerator surrounding the actual regenerator matrix since heat must be transferred to and from the wall from

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Nomenclature

Greek letters

α_{w}	thermal diffusivity of the wall [$\mathrm{m^2/s}$]
ϵ	regenerator effectiveness
μ_{f}	fluid dynamic viscosity [Pa s]
ψ	thermal mass ratio of the wall and the regenerator solid
ρ	mass density [$\mathrm{kg/m^3}$]
τ	total cycle time [s]
ε	bed porosity
φ	thermal utilization

Subscripts

f	fluid index
s	solid index
w	wall index

Variables

Δp	pressure drop [Pa]
Δr_j	radial extent of the jth cell [m]
$\Delta V_{i,j}$	volume of the grid cell with indices i, j [$\mathrm{m^3}$]
Δx	axial extent of the cells [m]
$\dot{m}$	mass flow rate [kg/s]
$\dot{q}_{\mathrm{fw,\ i}}$	the heat flux across the boundary between the fluid and the wall domains at node i [W]
$\mathrm{Re_f}$	Reynolds number based on the hydraulic diameter
$\mathrm{Re_p}$	Reynolds number based on the sphere diameter
$n_{r,\mathrm{sf}}$	number of grid points in the r-direction in the solid and fluid domains
T_{hot}	fluid inlet temperature at the hot end [K]
A=180	constant in the Ergun equation
A_{c}	bed cross sectional area [$\mathrm{m^2}$]
A_{HT}	total heat transfer surface area of the bed [$\mathrm{m^2}$]
a_s	specific surface area [$\mathrm{m^{-1}}$]
B=1.8	constant in the Ergun equation
c	specific heat [J/kg K]
d_h	hydraulic diameter [m]
d_p	sphere diameter [m]
f	operating frequency [Hz]
h	convective heat transfer coefficient [W/(m K)]
i, j	axial and radial indices, respectively
k	thermal conductivity [W/(m K)]
k_{disp}	thermal dispersion [W/(m K)]
k_{stat}	static thermal conductivity of the bed [W/(m K)]
L	length of the regenerator bed [m]
n	index for the timestep
n_x	number of grid points in the x-direction
$n_{r,\mathrm{w}}$	number of grid points in the r-direction in the wall
R	radius of the regenerator bed [m]
r	radial direction
r_j	radial center coordinate of the jth cell [m]
S_{HT}	total heat transfer surface area of a grid cell [$\mathrm{m^2}$]
T	temperature [K]
t	time [s]
$T_{\mathrm{f,cold\,out}}$	fluid outlet temperature at the cold end [K]
u	pore fluid velocity [m/s]
W	width of the wall bed [m]
x	axial direction
CFL	criterium for the timestep
NTU	number of transfer units
Nu	Nusselt number
h_{w}	inverse thermal resistance between wall and fluid [W/$\mathrm{m^2}$ K]

the regenerator solid and the heat transfer fluid. For housing materials with sufficient thermal diffusivity, axial conduction in the wall may also have an impact on the regenerator performance/apparent heat transfer coefficient.

In order to investigate the effect of the regenerator housing, a detailed numerical model is derived, described, validated and applied to a range of relevant cases. The model is two-dimensional resolving the flow direction (denoted x) and the transverse direction (denoted r) while assuming azimuthal symmetry. Three domains are included in the model: the regenerator solid, the heat transfer fluid and the regenerator housing/wall. The appropriate heat transfer equations are solved in all three domains and evolved forward in time. The model is designed so that it may be applied in a single-blow mode, which is relevant if it is used to understand the impact of the housing on the derivation of accurate Nusselt numbers from experimental data of this type. The model can also be used for periodic steady-state operation, i.e. having a periodic (balanced and symmetric) fluid flow. The latter mode is relevant when probing the impact of the housing on regenerator performance, or effectiveness, as a function of operating conditions and wall properties.

The remainder of this paper is outlined as follows. In Section 2 the numerical model is derived and presented. In Section 3 the results are presented. Finally, in Section 4 the results are discussed and the paper is concluded.

2. Numerical model

The modeled geometry is cylindrical and assumes symmetry around the center axis. The axial (x) and the radial (r) directions are spatially resolved in all three domains: fluid, solid regenerator matrix and the solid wall, respectively. Fig. 1 shows a schematic of the model geometry and defines the coordinate system. In the following section the governing equations for each of the three domains are written out in discretized form using finite differences.

2.1. Governing regenerator equations

The model solves the transient partial differential equations describing heat transfer via conduction and convection in a porous regenerator:

$$\rho_{\mathrm{f}} c_{\mathrm{f}} \varepsilon \left(\frac{\partial T_{\mathrm{f}}}{\partial t} + u \frac{\partial T_{\mathrm{f}}}{\partial x} \right) = \frac{1}{r} \frac{\partial}{\partial r} \left(k_{\mathrm{disp},r} r \frac{\partial T_{\mathrm{f}}}{\partial r} \right) + \frac{\partial}{\partial x} \left(k_{\mathrm{disp},x} \frac{\partial T_{\mathrm{f}}}{\partial x} \right) - h a_s (T_{\mathrm{f}} - T_{\mathrm{s}}) + \left| \frac{\Delta p \dot{m}}{\rho_{\mathrm{f}} L A_{\mathrm{c}}} \right|, \tag{1}$$

$$\rho_{\mathrm{s}} c_{\mathrm{s}} (1-\varepsilon) \frac{\partial T_{\mathrm{s}}}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(k_{\mathrm{stat}} r \frac{\partial T_{\mathrm{s}}}{\partial r} \right) + \frac{\partial}{\partial x} \left(k_{\mathrm{\ stat}} \frac{\partial T_{\mathrm{s}}}{\partial x} \right) + h a_s (T_{\mathrm{f}} - T_{\mathrm{s}}), \tag{2}$$

$$\rho_{\mathrm{w}} c_{\mathrm{w}} \frac{\partial T_{\mathrm{w}}}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(k_{\mathrm{w}} r \frac{\partial T_{\mathrm{w}}}{\partial r} \right) + \frac{\partial}{\partial x} \left(k_{\mathrm{w}} \frac{\partial T_{\mathrm{w}}}{\partial x} \right). \tag{3}$$

The temperature fields (T) are solved for on the three domains (fluid, regenerator solid and wall, respectively denoted by subscripts f, s and w). The fluid and solid domains are coupled through the convective heat transfer coefficient, h and the specific heat transfer surface area, a_s, of the solid regenerator material.

The above given equations for the fluid and the solid (1) and (2) are volume averaged since the actual porous medium is not

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