A Lattice Boltzmann simulation on fin enhanced water cooling of a solar CPV-T receiver

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Abstract. A fin enhanced cooling of a Solar Concentrated Photovoltaic Thermal (CPV-T) receiver is studied with the new computational method, i.e., an improved Lattice Boltzmann Method (LBM). Transient flow patterns and temperature fields have been obtained by the mesoscopic scale simulations with Non-dimensional Lattice Boltzmann Methods (NDLBM). The results show that the application of metal fins such as material copper will enhance the cooling slightly, while glass fins are not friendly to enhance heat transfer although they can keep high light transportation ratio. Geometry factors such as fin heights, gap distance, and thicknesses are also compared. It is found out that shorter smaller gap thicker fins are better designs because they reduce the impeding effects on fluid flow and increase the heat transfer areas between solid and fluid phases.

1. Introduction

Recently, one of new computational methods, Lattice Boltzmann Method (LBM), has been developed quickly to increase the computation speed and also to adapt to simulations of fluid flow and heat transfer in various engineering domains [1]. Various LBM simulations for natural and mixed convection inside a closed enclosure driven by top lid and buoyancy force have been carried out by introducing a body force term into the momentum equation for the macroscopic Rayleigh number up to $10^6$ [2]. Also conjugate heat transfer can be solved easily with LBM method by modification of the relaxation time in the energy equation [2]. A novel Non-Dimensional lattice Boltzmann method (NDLBM) based on the mesoscopic length scale has been developed by Su et al [3,4] to solve coupled flow and heat transfer fields for 240 tube bundle heat exchangers. A time step adjustable independent on the geometry mesh has been developed by Su and Davidson [5], which expanded the potential application fields. In the present study, flow and heat transfer fields inside a Solar Concentrated Photovoltaic Thermal (CPV-T) receiver with inlet flow cooling will be simulated. The fins used to enhance the heat transfer will also be compared for different fin materials.

2. Number model

The system picture for flow and heat transfer of a solar CPV-T receiver is shown in figure 1(a). The numerical models for the solar receiver with cold water cooling without and with designs of enhancement of fins are shown in figures 1(b) and 1(c), respectively. As shown in figure 1(a), the system is composed by a solar concentrator, a solar receiver. The water tank under the concentrator will provide the cooling water as the cooling fluid. The solar radiation will heat the top glass panel with normal heat flux equal to direct solar irradiance. A concentrated solar irradiance with about 10 times stronger heat fluxes are concentrated to inject on the bottom PV panel. As shown in the water
cooling model of figure 1(b), the inlet fluid is at lower temperature inject into the CPV-T solar receiver, and the fluid flow out through the outlet at the same flow rate due to the low compressibility of water.

Various types of designs of using fins to enhance the water cooling effects are shown in figure 1(c). The fins shown in figure 1(c) are designed to enhance the convective heat transfer by increasing the contact areas are attached on the CPV behind the CPV along the solar irradiance injection directions. As shown in figure 1(c), we studied different types of fins. The top designs of figure 1(c) will compare three different heights of fins. For the top designs, attached fins behind the bottom PV panels have dimensionless fins heights of 64, 96, and 128, respectively, where the dimensionless length is scaled by the mesh size. The mid designs will compare three different gaps of fins with dimensionless fin gaps of 40, 80, and 160, respectively. And the bottom designs compare three different fin thicknesses of 4, 8, and 16, respectively.

Two kinds of fin materials are also been compared for the same geometry design of fin with height =64, fins separated space =80, fins thickness=16. The comparison of materials will show if the effects of thermal conductivities of the fin materials on the results of introducing of the fins to CPV-T cooling system. Effective designs of fins are expected to cold down the bottom PV panel in relatively low temperature. The outside enclosure of the traditional CPV-T receiver is assumed to be glass material to

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Figure 1. A solar CPV-T receiver: (a) system picture, (b) heat transfer model without fins, and (c) heat transfer model with enhancement of different types of fins.
effective transfer the solar light. The fluid water is used to be the medium for convective cooling, because forced convective water cooling heat transfer coefficients are much higher than conductive or natural convective air cooling. Table 1 collected the thermal properties of all the materials, which will be applied in the present simulation model.

Table 1. Material properties for the present numerical simulations.

| Material        | Thermal conductivity (W/(m·K)) | Density (kg/m³) | Specific heat (J/kg·K) | Thermal diffusivity (m²/s) |
|-----------------|-------------------------------|-----------------|------------------------|---------------------------|
| Glass Fins      | 0.04                          | 2700            | 0.84                   | 1.764e-5                  |
| Copper Fins     | 401                           | 8960            | 0.385                  | 0.116                     |
| Liquid Water    | 0.6                           | 1000            | 4.2                    | 1.429e-4                  |

The whole enclosure of the traditional CPV-T receiver is assumed to be glass material filled with liquid water as figure 1, and liquid water is in relatively high value of thermal diffusivity than glass. And we proposed to improve the design of traditional PV-T receiver by attaching some fins which materials in excellent thermal diffusivity. This conference select one two different material fins for fins to investigate if this design of fins cooling PV-T receiver can enhance heat transfer to effectively cooling the CPV panel. Velocity amplitudes for both inlet and outlet are equal and assumed to be fully developed velocity to satisfy the rule of mass conservation. Equivalent flow rates are used in the present studied cases to compare the effects of geometry and materials for fins. The objective of the comparative study is to obtain the optimized cooling performance.

3. Governing equation and numerical method

The concept of the porosity was introduced to present the permeability of the porous medium, which defined as the fraction of the volume of the fluid phase,

\[
\phi = \frac{V_f}{V_s + V_f}
\]

where \( \phi \) represent the global porosity, and distribution of \( \psi \) represent the local porosity among the fins structures.

Denote \( f^* \) as the dimensionless density distribution function and \( g^* \) as the dimensionless temperature distribution function, following discrete dimensionless equations for momentum equation and energy equation can be obtained from equations (1) and (2) respectively.

\[
f_k^*(x^* + c_k^* t^* + 1) = f_k^*(x^*, t^*) - \frac{1}{\tau_f^*} \left( f_k^*(x^*, t^*) - f_k^{eq*}(x^*, t^*) \right) + F_k^*
\]

\[
g_k^*(x^* + c_k^* t^* + 1) = g_k^*(x^*, t^*) - \frac{1}{\tau_g^*} \left( g_k^*(x^*, t^*) - g_k^{eq*}(x^*, t^*) \right) + Q_k^*
\]

The respective dimensionless density, velocity, and temperature can be obtained as,

\[
\rho^* = \sum_k f_k^*
\]

\[
\nu^* = \psi \frac{\sum_k f_k^* c_k^*}{\sum_k f_k^*}
\]

\[
T^* = \sum_k g_k^*
\]
The local equilibrium distribution functions for fluid flow and mass transfer are given by equations (7) and (8), respectively.

\[ f_k^{eq} = w_k \zeta (c_k^*, v^*) \rho^* \] (7)

\[ g_k^{eq} = w_k \zeta (c_k^*, v^*) T^* \] (8)

Function \( \zeta (c_k^*, v^*) \) is obtained based on the Bhatnagar-Gross-Krook (BGK) model as,

\[ \zeta (c_k^*, v^*) = \left[ 1 + \frac{c_k^* \cdot v^*}{c_s^2} + \frac{1}{2} \left( \frac{c_k^* \cdot v^*}{c_s^2} \right)^2 - \frac{1}{2} \left( \frac{v^* \cdot v^*}{c_s^2} \right) \right] \] (9)

To implement the dynamic bounce back boundary condition among the fins structure, transient local porosity based boundary condition developed by Su et al [5] is applied:

\[ f_k^*(x^*, t^*) = (1 - h_k) f_k^*(x^*, t^*) + h_k f_{k,od}^*(x^* + c_{k,od}^*, t^*) \] (10)

where \( h_k \) is a transient interpolation factor related to local porosity as,

\[ h_k(x^*, t^*) = \frac{|\psi(x^*, t^*) - \psi(x^* + c_{k,od}^*, t^*)|}{\max [\psi(x^*, t^*), \psi(x^* + c_{k,od}^*, t^*)]} \] (11)

The porosity based bounce back scheme simplified the simulation of the porous structure in dynamically, and guarantee the mass conservation on the process of bounce back as well.

The boundary conditions for given velocities is presented by the momentum source term based on mass flow rate, which can be expressed in form of dimensionless force as

\[ F_k^* = w_k \rho_{in}^* |\partial u_{in}^*|^2 (c_k^* \cdot e_{u_{in}}) \] (12)

where \( e_{u_{in}} \) is the unit vector in the inlet velocity direction. The fully developed boundary for velocity is given by

\[ f_{k,w}^* = f_{k,w-1}^* \] (13)

For heat transfer, at adiabatic surfaces,

\[ g_{k,w}^* = g_{k,w-1}^* \] (14)

where \( w-1 \) is the mesh layer nearest to the outer boundary. A constant temperature boundary condition is expressed as,

\[ g_k^* = T_w^* (w_k + w_{k,od}) - g_{k,w-1}^* \] (15)

The fully developed boundary for heat transfer is given by

\[ g_{k,w}^* = g_{k,w-1}^* \] (16)

For a constant heat flux boundary \( q_w \) can be modeled as,

\[ Q_k^* = w_k M_d \left[ \frac{q_w}{(\rho c_p) U \Delta T} \right] \] (17)

The present NDLBM code as a coupled solver of the two sets of governing equations are developed in Fortran 90, which is compiled and run on the high performance computing cluster with GNU/Linux operating system (CentOS 5.5 64-bit). The D2Q9 Gaussian quadrature grid is applied in the present study.

4. Numerical results
Figure 2. The transient isotherms and streamlines from $t^* = 50,000$ to 500,000: (a) dimensionless isotherms, and (b) streamlines.

Firstly, the two different material fins with the same geometry design of fin with height =64, fins separated space =80, fins thickness=16 are compared. The two fin materials are glass and copper. The simulation results of the two materials are compared together with the pure water cooling without fins. Figure 2(a) and 2(b) compare the transient temperature and streamline distributions of no fins, glass fins and copper fins cases. Both glass and copper fins present the similar transient temperature distribution and streamline during the simulation. And the no fins case present the typical fully developed distribution of both temperature and streamline distributions. By comparing cases of the no fins and fins, both recirculations have been generated on the left hand side, and become increasing the regions gradually. But the attaching fins will cause it splitting into some small recirculation on the bottom, and will cause the temperature distribution more tilted on the top half side of the container. Major heat transfer occurs on the top half inside the CPV-T receiver, due to that the existence of the fins will block the injection of the cooling fluid directly from the inlet to the outlet.
Figure 3 shows the comparison of the transient average temperatures of CPV plates. After long time of running, copper fin design is slightly better than the original pure water design without fins with slightly lower CPV plate temperature. The glass fin design is obviously worse than the other two types of design due to lower thermal conductivity of glass. So it can be concluded that the introduction of the fins into the traditional CPV-T system can only be effective when the fin material has high thermal conductivity.

Then different designs of copper fins with various geometries (shown in figure 1(c)) are simulated to obtain the optimal geometry designs. We change one geometry factor with other parameters fixed to make the comparison in a standard way. In the following parts we will study the effects of fin heights, fin gaps, and fin thicknesses, respectively.

For geometry parameters, we start from changing the heights of fins. Figure 4(a) and 4(b) compare the transient temperature and streamline distributions of cases with the three dimensionless fin heights of 64, 96, and 128, respectively. By comparing cases of the different fin heights, both big recirculations have been generated on the left hand side, and several small recirculations have been generated on the bottom side. Higher attaching fins will cause the temperature distribution more tilted on the top half side of the container, because the more longer fins will block the fluid flow from the inlet to outlet, cause the faster fluid flow within the traveling path.
Figure 5 shows the comparison of the transient average temperature of the CPV panel for three fin heights. From figure 5, we can see that increasing the height of the fins with fixed gaps and thickness, the transient averaged CPV plate temperature will increase. This is reasonable, because higher fins will impede the flow and thus to reduce the mean fluid velocity. The lower velocity will reduce the convective heat transfer accordingly.

![Figure 5. Transient temperatures of the CPV plate for cases with different fin heights.](image)

Then the geometry factors of fin gaps are studied. The three different dimensionless fin gaps of 40, 80, and 160 are simulated. The comparison of the transient average temperature of the CPV panel for three fin gaps is shown in figure 6. As shown in figure 6, the dimensionless gap distances in range of 40 to 160 show small effects on transient averaged temperatures of CPV plates. The smaller gas distance of 40 shows slightly better performance comparing to gap distance 80 and 160, due to better heat conduction from the CPV plate to the fluid water. This is caused by larger heat transfer areas between the fluid and solid interfaces.

![Figure 6. Transient temperatures of CPV plates for cases with different fin gaps.](image)

The third geometry factor of fin thickness is also studied. The three different dimensionless fin thicknesses of 4, 8, and 16 are simulated. The comparison of the transient average temperature of the CPV panel for three cases is shown in figure 7. As shown in figure 7, the dimensionless fin thickness in range of 4 to 16 show stronger effects on transient averaged temperatures of CPV plates comparing
to the effects of gap distances. The larger fin thickness case of 16 shows better performance comparing to the other two thinner fin cases. Similar as the mechanism for the gaps distance effects, this better performance of thicker fins is also caused by better heat conduction from the CPV plate to the fluid water, due to larger heat transfer areas between the fluid and solid interfaces.

![Figure 7. Transient temperatures of CPV plates for cases with different fin thickness.](image)

5. Conclusions
The water cooling of a solar CPV-T receiver enhanced by increasing contact areas of convective heat transfer with fin designs. Two kinds of materials are compared by solving the transient flow and heat transfer fields with LBM simulations. The key geometry parameters of fins, i.e., fin heights, gap distance, and thicknesses are widely studied. The results show that using metal fins such as copper fins can slightly enhance the cooling comparing to water cooling without fin design, while glass fins are not friendly for heat transfer, although they can keep the light transportation. When other geometry parameters are fixed, decreasing the heights, decreasing the gap distances, and increasing the fin thicknesses can reduce the transient average CPV plate temperature. The two key mechanisms are reducing the effects of the flow field so as to keep higher velocity to enhance convective heat transfer, and increasing the solid-fluid interface area to increase conduction between interfaces of solid and fluid phases.

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Appendices
Nomenclature

- $f$: density distribution function
- $g$: temperature distribution function
- $h$: interpolation factor
- $q$: heat flux
- $Q$: heat source
- $t$: time
- $T$: temperature
- $\rho$: density
- $v$: velocity
- $\zeta$: BKG model function
\[ \phi \quad \text{porosity} \]

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