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Gnielinski method in calculating the heat transfer coefficient for metallic solar tower absorber

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Abstract

This work is done to calculate the heat transfer coefficient of metallic wire mesh to air in an open volumetric thermal absorber. It is aimed to replace the actual ceramic with metallic which latter has better thermal properties in order to increase the efficiency of the concentrating solar power plant. The calculation of the heat transfer coefficient from porous wire structure to the air has been conducted to pursue the purpose. The structure that has been chosen is a set of metallic wires with aligned and shifted configurations. The variations of wire diameter together with various porosities from 0.10 to 0.50 have been calculated to obtain the best configuration for the absorber. The flow characteristic within the mesh structure in terms of Reynolds number and its relationship with the heat transfer coefficient has been obtained. The condition on mass flow rate of heat transfer medium and other aspects which influence the heat transfer are also discussed.

Nomenclature

$\dot{m}_0$	*Mass flow rate of air [kg/s]*
θ_{in}	*Air inlet temperature [°C]*
θ_{out}	*Air outlet temperature [°C]*
θ_m	*Mid temperature [°C]*
θ_w	*Front absorber temperature [°C]*
θ_A	*Tail absorber temperature [°C]*
w	*Flow velocity in the matrix [m/s]*

Dimensionless character

Re_{xx}	*Reynolds number*

Greek symbols

α	*Heat transfer coefficient [W/m²K]*
ψ	*Fraction of hollow space*

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Latin alphabets
da — *Wire diameter [m]*
$s1$ — *Wire pitch [m]*
L — *Length of absorber cup [m]*
w — *Width of absorber cup [m]*
d — *Depth of absorber cup [m]*
a — *Ratio s1/da*
z — *Wire/row*
$s2$ — *Distance between rows [m]*
b — *Ratio s2/da*
S — *Stream cross-sectional area in front of absorber [m²]*
l — *Length of wire facing stream [m]*
P — *Porosity*
n — *Number of layers of metal mesh above one another*
P_t — *Transverse pitch [m]*
r_h — *Hydraulic radius / effective length across wire matrix [m]*
A — *Heat transfer area [m²]*
a_v — *Surface area density [1/m]*
m_{mesh} — *Mass of wire mesh [kg]*

1. Introduction

Concentrating Solar Tower (CST) is an advanced solar energy collecting system used to harness solar thermal energy. Solar energy is reflected by a field of heliostat and the concentrated sun ray which may reach to a factor of 600 to 1000 hits the collector or receiver located at the top of the tower [1]. This dense energy heats up the receiver rapidly and thus the most suitable material for the receiver or absorber is ceramic-based. Ceramic is a low thermal conductivity material which hinders higher power plant thermal efficiency. Action has been taken to replace the ceramic to a metallic material which has better heat transfer capability. This new metal-based absorber will be deployed at the circumference of the gigantic absorber frame where operating temperature is the lowest and suitable for metal to survive. The idea is to use wire mesh packed in the absorber cup to maximize the surface area but at an optimal porosity to avoid unnecessary load on blower unit [2-3].

Sintered porous structure which behaves like a micro mesh has been successfully used in a low temperature cryogenic application [4] while Kempers et. al. [5] applied the wire screen mesh for heat exchanger. As wire mesh is widely used in heat transfer application this research has been carefully conducted to examine whether it is also suitable for the absorber structure in CST application. The porous wire screen has been chosen as a candidate for the absorber of the CST. The work is started with measuring the size of the absorber cup, absorber structure and other parameters as shown in Table 1:

Table 1. Absorber cup and absorber dimension (source: Solar Institute Juelich, SIJ)

Dimension of cup and absorber structure	
Length, L [m]	0.14
Width, w [m]	0.14
Depth, d [m]	0.03
Surface area, A [m^2]	0.0196
Volume, V [m^3]	0.000588
Total absorber surface area, A_f(tot) [m^2]	22
Air intake, m(tot) [kg/s]	12

2. Selection of wire mesh

Calculation is started with two possible wire configurations as stated by Gnielinski in the GG section which are aligned and shifted configurations [6]. Porosity for both configurations can be calculated using Eq. (1) [7]:

$$P = 1 - \frac{\pi * n * da^2}{2 * s1 * d}\left(1 + \frac{da^2}{s1^2}\right)^{\frac{1}{2}} \tag{1}$$

where n is number of layer of screens, s1 is transverse pitch of screens, da is the wire diameter and d is the depth of matrix bed.

2.1. Aligned and shifted configurations

The calculation started by calculating the fraction of hollow space, ψ (Greek: Psi) and it is dependent on the value of b which is defined as ratio of distance between layers over wire diameter, s2/da.

for b≥1
$$\psi = 1 - \pi /(4a) \tag{2}$$

for b≤1
$$\psi = 1 - \pi /(4ab) \tag{3}$$

The arrangement of tubes is then divided into two groups. One is aligned and the other is shifted. The distance between tubes in a row, s1 and the distance between rows of layers, s2 are constants and defined as:
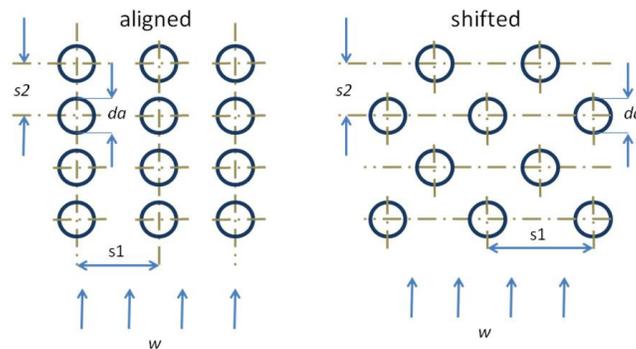
$$a = s1/ da \tag{4}$$

and

$$b = s2/ da \tag{5}$$

Fig. 1. Proportionally aligned and shifted configuration of bundle of tubes

Once the parameters are programmed correctly in the software the values of *s1*, *s2* and *da* are systematically manipulated and simulated using actual mass flow rate and operating temperature to get the best structure for the absorber. Shifted configuration will be calculated further as this configuration will be the most relevant configuration during production. The small wire diameter used in the application also will eventually bend creating shifted configuration when exposed to cyclic thermal loads.

2.2. Presetting the shifted configuration

The operation temperature and also the set of wire mesh of the absorber are set primarily to follow the values given by Prasad et. al. [2] in order to examine the formula. After confirming that the formulas are giving the correct values the parameter of the wire mesh is then changed to the desired value for the CST absorber.

The best heat transfer occurs in the least porous structure. Therefore, calculations started at fraction of hollow space ψ of only 0.1 which means 90 percent of the cup volume is empty. Relative mass flow rate G0 of 0.5455 kg/m.s² as well as other parameters listed on the left of the table below are the actual air parameters during operation. The corresponding air properties are listed on the right (Table 2).

Table 2. Operating condition and the corresponding air properties

G0	0.5455	rel. mass flow (kg/m².s)	v	6.45E-05	by ϑm (m²/s)
m	0.01069	mass flow (kg/s)	λ	0.04996	by ϑm W/m.K
ϑin	20	inlet air temp. (°C)	ρ	0.517	by ϑm (kg/m³)
ϑout	780	outlet air temp. (°C)	Cp	1069	by ϑm (J/kg.K)
ϑm	400	average air temp. (°C)	Pr	0.7137	by ϑm
ϑw	800	average absorber temp (°C)	PrW	0.7342	by ϑw

2.3. Other setting

Flow characteristics like Reynolds number, Nusselt number and also Prandtl number have been inserted into the program to calculate and simulate the real condition.

3. Results and discussion

(a)   (b)
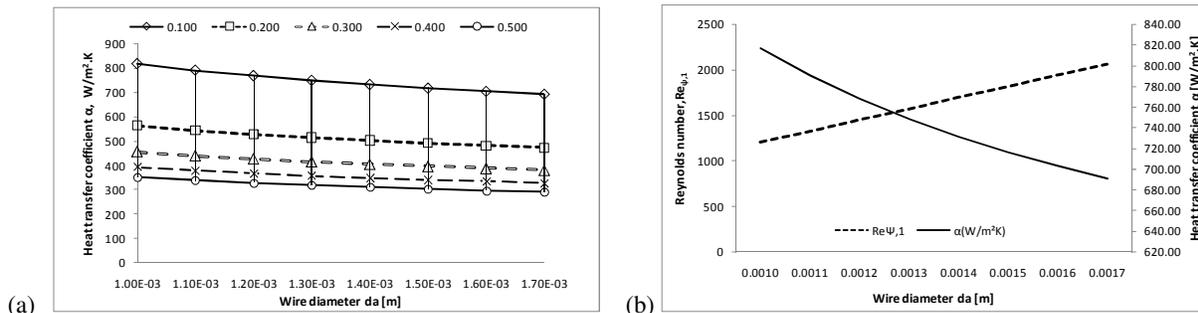
Fig. 2. Results using six layers of wire mesh with porosity from 0.1 to 0.5 for (a) wire diameter versus heat transfer coefficient α and (b) wire diameter versus Reynolds number with corresponding α.

Fig 2(a) shows that the increasing amount of wire inside the absorber cup leads to a less porous structure which reduces the heat transfer ability. Increasing the layer leads to a higher α which is good for the absorber cup. The best α of 1111 W/m². K has been recorded for 1mm wire diameter with 24 mesh layers and ψ equals to 0.1, see Fig 3.

Reynolds number is recorded increasing with the diameter of the wire mesh used. Bigger wire reduces the porosity ψ and thus intensifies the air-to-wall friction creating turbulence (Fig 2(b)). This friction increases the pressure inside the absorber and reducing the air flow rate through the cup. This situation will put unnecessary burden to the pump, create hotspot on the absorber surface and further eliminate a smooth heat transfer from the mesh to the air.
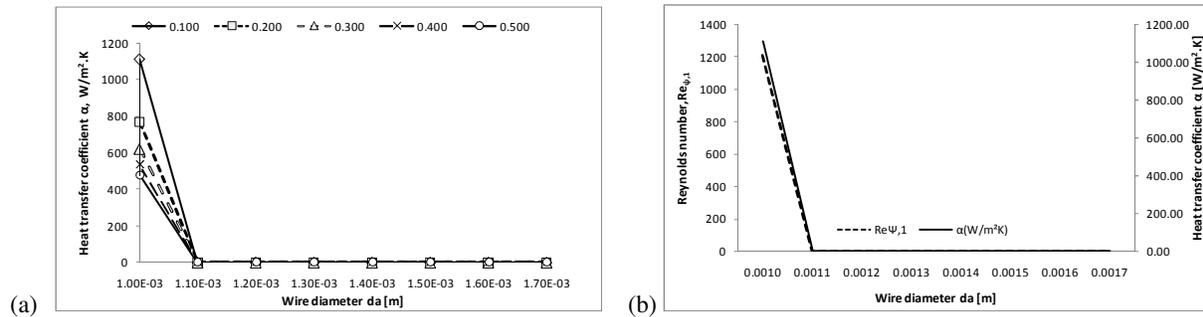
Fig. 3. Results using 24 layers of wire mesh with porosity from 0.1 to 0.5 for (a) wire diameter versus heat transfer coefficient α and (b) wire diameter versus Reynolds number with corresponding α.

4. Conclusion

There is a small possibility that the wire mesh will be placed unevenly during production but this is not entertained in this task as the deviation is neglected at the early stage. The calculations to obtain the relationship of wire mesh candidates with Prandtl number, volumetric heat transfer and also effect of cyclic load on the elongation of the wire itself must be conducted.

As we can see the best heat transfer coefficient α comes with the smallest wire diameter and also the least porous structure. Small wire tends to bend more easily under load compares to the bigger wire. Bending of wire will eventually clog the entrance and destroy the absorber. The application itself which involves cyclic thermal loads and thermal shock will just accelerate the process. Therefore more calculations and experiments are necessary in order to choose the best material, wire size and coordination which can withstand the harsh working condition. The wise choice will make the structure to run at the optimum performance and simultaneously keep the maintenance at the longest interval.

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