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3-D numerical simulation of heat transfer and turbulent flow in a receiver tube of solar parabolic trough concentrator with louvered twisted-tape inserts

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Abstract

High temperature and higher-thermal efficiency for CSP cycles are main goals to improve trough collector's technologies. For a parabolic trough collector the major factor for optimum heat transfer from sun to the heat transfer fluid passing in the absorber tube is to have high convection heat transfer coefficient. Literature shows that absorber tubes with various tape inserts are used and recommended to produce high convection coefficient. Typical twisted-tape (TT) enhances heat exchange between tube surface and working fluid by generating turbulent swirling flow. In this study, enhancement of convection coefficient in the receiver tube of a solar parabolic trough concentrator that the absorber tube is equipped with a new perforated louvered twisted-tape (LTT) is studied numerically. For numerical simulations three different twist ratios (TR), TR=y/W= 2.67, 4, 5.33 (y is the length required for one twist and W is the width of the tape) are used in an experimental laboratory trough collector. Flow is assumed turbulent due to louvered perforated surface and rotational shape of the tape. For thermal boundary condition, non-uniform wall solar heat flux is determined by Sotrace code on the outer surface of the absorber tube. Heat transfer rate and pressure drop are determined for fully developed condition for several Reynolds numbers based on the tube diameter and flow mean velocity. Results show that the heat transfer coefficient and pressure drop increase significantly in comparison with a typical plain twisted-tape in the tube and a plain tube.

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1. Introduction

For solar systems heat transfer augmentation techniques refer to different methods used to increase the rate of heat transfer without affecting much the overall operation of the system. Nithiyesh [1] classified existing heat transfer enhancement techniques into three groups: active techniques, passive techniques and compound techniques. In active techniques, heat transfer enhancements occur because of the external power existence. Passive techniques use geometrical modifications without needing external power input. Finally in compound techniques, any two or more of those techniques are used with each other. Using the modified twisted-tape that is investigated in this study is one of the effective passive techniques for such enhances of heat transfer rate and pressure drop by generating a turbulent swirling flow.

Hejazi et al. [2] carried out an experimental study on heat transfer enhancement and pressure drop changes for a tube with twisted-tape insert. Twisted-tapes with a twist ratio of 6, 9, 12 and 15 are investigated and the results are compared with a plain tube. For all twist ratios heat transfer and pressure drop enhancement are observed. The best heat transfer enhancement and thermal performance occurred for twist ratio of 6 and 9, respectively. Eiamsa-ard et al. [3] carried out numerical investigation on heat transfer in a tube with loose-fit twisted-tape insert. The results showed that the tube with twisted-tape insert without clearance between the edge of the tape and tube wall (tight-fit) had maximum heat transfer enhancement rather than the loose-fit twisted-tape insert and this enhancement increased with decreasing the clearance between the edge of the tape and tube wall. Yadav [4] investigated experimentally heat transfer and pressure drop in a U-Bend double pipe heat exchanger with half-length twisted-tape insert. By comparison with a typical heat exchanger, heat transfer coefficient enhanced 40%. Also thermal performances of it were 1.3-1.5 times better than the half-length twisted-tape. Eiasma-ard et al. [5] studied convective heat transfer in turbulent flow with short-length twisted-tape insert under uniform wall heat flux boundary conditions. The experiments are done at several tape length ratios of 0.29, 0.43, 0.57 and1.0 (full-length twisted-tape). It was found that heat transfer and pressure drop of the tube with full-length twisted-tape has higher convection coefficient than short-length twisted-tape.

Jaisankar et al. [6] carried out an experimental study on heat transfer and friction factor for a solar water heater with spacer and rod at the ending edge of the twisted-tape for several lengths and twist ratios. Reduction of heat transfer coefficient enhancement for twisted-tape with rod and spacer was 17% and 29%, respectively as compared with full-length twisted-tape. It is also observed that use of twisted-tape with rod instead of full-length twisted-tape had low friction factor with less reduction on heat transfer enhancement. Ferroni et al. [7] carried out experimental investigation for isothermal condition for tubes with separated, multiple, short-length twisted-tape inserts in turbulent regime. Results showed that pressure drop with multiple short-length twisted-tapes were at least 50% lower than full-length twisted-tapes. Eiamsa-ard et al. [8] investigated heat transfer enhancement and pressure drop in a single, full-length and regularly-spaced dual twisted-tape under uniform wall heat flux conditions. Result showed that the tube with dual twisted-tapes had higher heat transfer than the plain tube and tube with typical twisted-tape inserts. Their results also showed that the heat transfer of the regularly-spaced twisted-tape decreased with increasing space ratio. Eiamsa-ard [9] investigated experimentally the influences of multiple twisted-tapes on heat transfer and friction factor in a rectangular channel. Results showed that the channel with the smaller twist ratio and more free space between tapes provided higher heat transfer rate and pressure drop than those with the larger twist ratio and less free space between tapes. Seemawute and Eiamsa-ard [10] studied numerically flow in a tube with alternative axis twisted-tape insert. Their numerical results showed that the fluid in the tube with alternative axis twisted-tape insert has more uniform and temperature distribution than the typical twisted-tape.

Eiamsa-ard and Promvonge [11] carried out an experimental study of heat transfer enhancement in a circular tube with alternate clockwise and counterclockwise twisted-tape inserts in turbulent flow. Their results indicate that heat transfer enhancement in the tube with alternate clockwise and counterclockwise twisted-tape inserts are higher than those with the typical twisted-tape inserts and plain tube by 12.8–41.9% and 27.3–90.5%, respectively. Guo et al. [12] investigated numerically effect of center-cleared twisted-tape and the short-width twisted-tape in laminar flows. Their results showed that friction factor reduced for both of them but the center-cleared twisted-tape was more suitable for having better overall performance. Thianpong et al. [13] experimentally investigated the influences of the twisted-tape with perforation on heat transfer under uniform wall heat flux condition. Their results indicate that
twisted-tape with bigger holes diameter, more space between the holes and smaller twist ratios had maximum heat transfer with respect to plain tube and tube with typical twisted-tape insert around 27.4 and 86.7%, respectively. 

Eiamsa-ardet al. [14] investigated influences of the twisted-tape insert with peripherally-cut on heat transfer rate for laminar and turbulent flow regimes. The tube with the peripherally-cut twisted-tape insert had higher heat transfer enhancement than the tube with typical twisted-tape insert and plain tubes. Chang et al. [15] carried out an experimental study on heat transfer characteristics in a tube with serrated twisted-tape insert. Heat transfer was about 1.25–1.67 times higher than a tube with smooth twisted-tape insert and 250-480% higher than plain tubes. They [16] also measured heat transfer of the tube with a broken twisted-tape insert. Heat transfer coefficients in the tube with the broken twisted-tape insert enhanced to 1.28-2.4 times of the tube with the smooth twisted-tape insert. Rahimi et al. [17] presented experimental and numerical studies on Nusselt number, friction factor for a typical tube with three twisted-tape inserts form. Their results showed that the tube with jagged twisted-tape insert had higher Nusselt number than other ones. Maximum enlargement in Nusselt number was 31% higher than a tube with typical twisted-tape insert. Based on the above literature no studies are observed for an absorber tube of a parabolic collector with perforated louvered twisted-tape insert.

2. Numerical simulation

2.1. Physical method

In this study, 3-D numerical investigation is carry out on a laboratory type receiver tube of a parabolic trough collector with inner diameter of 17 mm and 1000 mm length. As it is common in heat transfer studies and the literature, the important metric for comparing different twisted-tape is the twist ratio. With each twist, different swirling flow will be generated and with changing the twist ratio, the number and intensity of these swirling flows will changed along the flow direction. The width and thickness of the twisted-tapes of the present study have 15 mm and 0.9 mm length, respectively. Three different twist ratios, y/W= 2.67, 4, 5.33 (y is twist length, and W is the wide of tape) are tested. With decreasing the clearance between the edge of a tape and tube wall and increasing the length of a tape better heat transfer enhancement are observed [3, 5]. Fig.1 shows an example of a louvered twisted-tape (LTT) with perforation. The absorber tube and twisted-tape are AISI304 and Aluminum, respectively. The LTT perforations have 30° angle attack and the length and width of 10 mm, with 10 mm distance between each fin. The flow is assumed in the louvered-fins direction. The glass inner and outer diameters are 36 mm and 40 mm, respectively and vacuum side thickness is 8 mm. Also the vacuum and glass length are 860 mm.

Several meshes are tested to find independent heat transfer and pressure drop of grid size. For this model the best results of meshes found to be 200,000 – 300,000 cells for different tapes in the receiver tube. Typical meshing is shown in Fig.2. Both tetrahedral and hexahedral cells are generated based on the geometry of the tape.
2.2. Boundary conditions

In this study, Behran thermal oil is used as the working fluid with inlet temperature of 353.15 K. Its properties are temperature dependent; these properties are shown in Table 1. Turbulent flow due to perforation and louvered surfaces, and rotational shape of the tape is assumed and RNG version of $k-\varepsilon$ turbulence model is used. Heat transfer rate and friction factor are determined for fully developed condition for Reynolds numbers between 5000 - 25000 based on the tube diameter and flow mean velocity. For thermal boundary condition, non-uniform wall solar heat flux boundary condition is considered on the outer surface of the absorber tube, and the tube wall ends are assumed adiabatic. Absorbed solar radiation flux on the absorber and glass side are determined for the noon-hour of the summer and winter Solstice days by Soltrace software, and also conduction and convection in the vacuum side are assumed negligible.

Table 1. Behran oil properties

<table>
<thead>
<tr>
<th>Properties</th>
<th>Equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific heat(C_p)</td>
<td>$C_p = 0.8132 + 0.003706 \times (T(°C) + 273.15)$, Kj/kg°C</td>
</tr>
<tr>
<td>Thermal conductivity(k)</td>
<td>$k = 0.1882 - 0.00008304 \times (T(°C) + 273.15)$, W/m°C</td>
</tr>
<tr>
<td>Density(\rho)</td>
<td>$\rho = 1071 - 0.72 \times (T(°C) + 273.15)$, Kg/m³</td>
</tr>
<tr>
<td>Prandtl(Fr)</td>
<td>$Pr = (T(°C) + 273.15)^{-7.7127}$</td>
</tr>
</tbody>
</table>

2.3. Governing equations

Thermal model for heat transfer losses in parabolic trough is obtained as explained in Odeh et al. [18]. In this modelling, thermal loss from the absorber tube outer wall to the evacuated glass tube (surrounding the absorber) occurs by radiation. Due to the high vacuum in the absorber element, convection and conduction are negligible, and heat loss from the glass cover tube occurs by radiation to the sky and by convection to the surrounding air.

$$Q_{Sol} = Q_{Sol-g} + Q_{Sol-a}$$ (1)

$$Q_{Sol-g} = Q_{conv-g-sky} + Q_{rad-g-sky}$$ (2)

$$Q_{conv-g-sky} = h_c(T_g - T_a)A_g$$ (3)
\[ Q_{\text{rad-g-sky}} = F_{g\text{-sky}} \cdot \sigma_g \cdot \varepsilon_g \cdot \left( T_g^4 - T_{\text{sky}}^4 \right) \cdot A_g \] 

(4)

Where, \( Q_{\text{Sol-g}} \) is the absorbed heat from radiation on the glass tube, \( Q_{\text{Sol-a}} \) is the absorbed heat by the absorber tube, \( Q_{\text{conv-g-sky}} \) is the heat loss by convection and \( Q_{\text{rad-g-sky}} \) is radiation from the glass tube outer surface to surrounding air; \( h_c \) is the external convection heat transfer coefficient due to forced convection on the outside of the tube, \( T_g \) is outer diameter of glass, \( A_g \) is outer area of glass, \( T_a \) is air temperature and \( F_{g\text{-Sky}} \) is the glass and sky view factor. Also,

\[ Q_{\text{Sol-a}} = Q_{\text{rad-a-g}} + Q_{\text{cond-a}} \] 

(5)

\[ Q_{\text{cond-a}} = Q_{\text{conv-fluid}} \] 

(6)

\[ Q_{\text{rad-g-a}} = \frac{\sigma (T_a^4 - T_g^4)}{\frac{1}{\varepsilon_a} + \frac{d_{a,o}}{d_g \left( \frac{T_g + T_a}{T_g - T_a} \right)}} \cdot A_a \] 

(7)

Where \( Q_{\text{cond-a}} \) is the conduction heat transfer inside the absorber wall and \( Q_{\text{conv-fluid}} \) is the useful energy gained by the working. \( Q_{\text{rad-g-a}} \) is the radiation heat transfer between absorber tube and glass, \( A_a \) is the absorber outer area, \( d_{a,o} \) and \( d_g \) are absorber outer diameter and glass inner diameter, respectively. \( \varepsilon_a \) and \( \varepsilon_g \) are emissivity of absorber and glass respectively.

The absorber temperature is calculated in terms of fluid temperature and internal film coefficient as follows:

\[ T_{\text{wall}} = T_{\text{bulk}} + \frac{Q_{\text{conv-fluid}}}{2\pi U L} \] 

(8)

Where \( U \) is the overall heat transfer coefficient between working fluid and outer surface of the absorber tube (internal convection and wall conduction), \( T_{\text{wall}} \) and \( T_{\text{bulk}} \) are the absorber and fluid temperatures, respectively. Nusselt number, friction factor and thermal performance factor are parameters that used for expressing the effectiveness of this enhancing technique. Nusselt number, with heat flux on the absorber tube is defined by Newton law as:

\[ Nu = \frac{d_{a,i} \cdot Q_{\text{conv-fluid}}}{(T_{\text{wall}} - T_{\text{bulk}}) \cdot k_{\text{bulk}}} \] 

(9)

Where \( d_{a,i} \) is inner diameter of absorber.

Friction factor (\( f \)) is calculated from the pressure drop (\( \Delta P \)), across two pressure taps with absorber length (\( L \)) and Nusselt number for plain tube with Dittus-Boelterand and friction factor with Blasius equations are determined for turbulent flow as follows

\[ Nu_p = 0.023 Re^{0.8} Pr^{1/3} \] 

(11)

\[ f_p = 0.079 Re^{-0.25} \] 

(12)

Eiamsa-ard et al. [19] determined Nusselt number and friction factor for tube with TT under a uniform heat flux condition as follows where \( y/w \) is defined as twist ratio of tape.
\[ Nu_{TT} = Re^{0.66} Pr^{0.4} \left( \frac{Y}{W} \right)^{-0.6} \] (13)

\[ f_{TT} = 65.4 Re^{-0.52} \left( \frac{Y}{W} \right)^{-1.31} \] (14)

The thermal performance (\( \eta \)) of the test tube with TT and LTT inserts, are defined by Eq. (15). This expression is the most common method to find effectiveness of any new scheme of enhancing heat transfer in literature.

\[ \eta = \frac{Nu}{Nu_p} \left( \frac{f}{f_p} \right)^{1/3} \] (15)

2.4. Validation

To validate the results of present study, computation of the plain tube and the tube with typical twisted-tape inserts (TT) are compared with correlations of Eiamasa-ard et al. [19]. The results are shown in Fig.3. Comparison shows acceptable with \( \pm 5\% \) and \( \pm 3\% \) deviation of Nu for plain tube and tube with TT, respectively; and \( +6\% \) and \( \pm 9\% \) deviation of friction factor for plain tube and tube with TT.

![Fig.3. Validation test for plain tube and tube with typical twisted-tape insert with twist ratio of the 2.67(a) Nusselt number (b) Friction factor](image)

3. Results and discussion

Results of the Nusselt number variation of typical twisted-tape (TT) with louvered twisted-tape (LTT) is shown in Fig.4. Nu for LTT with different Reynold numbers (Re) is 37\% and 150\% higher than those with (TT) and plain tube, respectively, and they increased by increasing Re. This remarkable growing in heat transfer is because of the combination of the swirling flow of the twisted-tape and the generated vortex due to louvered-fins on it. In fact, because of additional turbulence that louvered-fins imposed to the flow, heat transfer coefficient is improved. Fig.4 also shows that \( \frac{Nu}{Nu_p} \) for LTT is higher than TT. This ratio decreases with increasing Re and this expresses the fact that the twisted-tape is better for low turbulence flows. Also by decreasing the twist ratio, the value of Nu is increased.
Fig. 4. Variation of Nu with Re: (a) Nusselt number (b) Different twisted-tape Nusselt number to plain tube Nusselt number

Fig. 5 shows that the friction factor in tube with both, LTT and TT is higher than the plain tube. Maximum value of friction factor for different Re are 72% and 210% higher than those with (TT) and plain tube, respectively, and it decreases with increasing Re. The reason for higher pressure drop is due to swirling flow of the twisted-tape and the additional vortex of the louvered-fins that act as an obstacle.

Fig. 5. Variation of Friction factor: (a) Friction factor (b) Different twisted-tape friction factor to plain tube friction factor

The thermal performance is the ratio of the dimensionless Nusselt number and the dimensionless friction factor and this ratio shows the amount of the energy is saved. As it is common in heat transfer research and literature the thermal performance (\( \eta \)) is shown in Fig. 6. Results illustrate that \( \eta \) decreases with increasing Re and this show better performance of the tube with louvered-fin for lower Re and turbulence. As is shown in Fig.6, \( \eta \) for LTT is 26% higher than TT.

Fig. 6. Thermal performances for different twist tape
Conclusion

In present study effect of a new perforated louvered twisted-tape on the heat transfer coefficient and friction factor for an absorber tube of a solar parabolic trough collector is determined numerically. It is found that:
1-High Nusselt number and friction factor are observed for LTT with respect to plain tube. For typical twisted-tape a maximum of 150% and 210%, are observed for Nusselt number and friction factor respectively.
2-Application of the new twisted-tape resulted higher thermal performance especially for low Reynold numbers.
3-With decreasing the value of the Reynolds number and twist ratio, heat transfer coefficient increased, so that for the twist ratio of 2.67 and Re=5000 the best result are observed.

References