ASIAN JOURNAL OF PHARMACEUTICAL AND CLINICAL RESEARCH

INNOVARE ACADEMIC SCIENCES
Knowledge to Innovation

Special Issue (April)

Online - 2455-3891
Print - 0974-2441

Review Article

EXPERIMENTAL STUDY ON IN-POND HEAT EXCHANGER PROVIDED WITH TWISTED TAPES

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Received: 19 January 2017, Revised and Accepted: 20 February 2017

ABSTRACT

In-pond heat exchangers (IHE) are used in salinity gradient solar ponds (SGSP), to transfer heat from hot saline water of the lower convective zone to the heat transfer medium. The performance of the SGSP depends on the performance of the IHE, which in turn depends on the method of heat transfer augmentation. A laboratory model IHE was fabricated to augment the heat transfer using helical twisted tapes (HTTs). The effects of varying twist ratio and double TTs on pressure drop and heat transfer were studied in this experimental work. Conventional HTTs of three twist ratios (Y=7, 9, 11), twin TTs of co-swirl flow (CoTT), and counter-swirl TTs (CTT) with Y=7 were provided in the flow path of the heat exchanger tube for analysis. From the results, it is observed that TTs with less twist ratio yielded higher rate of heat transfer by sacrificing the pressure drop. Compared to single TT, double TT provided better thermal performance. The pressure drop and rate of heat transfer for CTT are more compared to all other cases, including double TT with co-swirl flow.

Keywords: In-pond heat exchanger, Heat transfer augmentation, Twisted tapes, Twist ratio.

 DOI: http://dx.doi.org/10.22159/ajpcr.2017.v10s1.20515

INTRODUCTION

Augmenting the thermal performance of the heat exchange process is the need of the hour, especially in the heat exchanger industry. This reduces the bulkiness of the heat exchanger, thereby saves energy, and material and cost of the system as well. Solar thermal systems like solar ponds are not popular due to their poor thermal efficiency. Improving the thermal performance of the heat exchangers used in solar ponds will make the solar pond technology a viable solution to meet the increasing global energy demand. Enhancing the heat transfer in a thermal system by various active and passive methods was dealt [1]. The thermohydraulic performance of circular pipe provided with or without V-cut helical twisted tape (HTT) inserts was studied on concentric tube heat exchanger, and empirical correlations were suggested to determine the friction factor and Nusselt number [2]. The performance analysis of an in-pond heat exchanger (IHE) of a salt gradient solar pond was carried out both experimentally and computationally and observed that the computational outputs were in agreement with the experimental results [3]. Experimental study on heat transfer and pressure drop was performed on a double pipe heat exchanger with an inner corrugated tube, this inner tube was fitted with different types of tape inserts from simple HTTs to modified TTs such as perforated, U-cut, and V-cut types and it was observed that higher Nusselt number and pressure drop were reported for modified TTs as compared to conventional TTs [4]. Use of modified TTs resulted with heat transfer enhancement in a simulated laboratory model solar pond [5]. In an experimental work on tubular heat exchanger, double V-ribbed TTs were used to enhance the rate of heat transfer [6]. Heat transfer, effectiveness, and pressure drop across a corrugated (both inner and outer) double tube heat exchanger were studied and the results indicated that corrugations on the outer tube have more significance on thermal and frictional characteristics [7]. Small pipe inserts with arc radii of 5 mm and spacer length of 100 mm showed 2.61-3.33 times the maximum rate of heat transfer and 1.6-1.8 times of friction factors when compared plain tube [8]. Regularly spaced TT elements were used in a heat transfer test section and it was found that the difference between of isothermal friction factor and heated friction factor for the periodic swirl flow is lower compared to plain tube [9]. The different active and passive methods of heat transfer augmentation used in the processes of recovering heat, air conditioning, and refrigeration are discussed in detail [10]. A passive and economical technique of heat transfer augmentation using tape insert and wire coil insert was reviewed in detail, the outcome of the review is TTs, and wire coil inserts are suitable for laminar flow and turbulent flow, respectively [11]. Experimental investigation on characteristics such as friction factor and heat transfer for a circular flow passage provided with various TT was studied, and empirical relations were made for the friction factor and Nusselt number. HTTs provided with center wing and alternate axis placed in a round tube was analyzed for its thermohydraulic characteristics for Reynolds number ranging between 5200 and 22,000 [12]. HTT of twist ratio, Y=3 provided better thermal performance compared to plain tube and other twist ratios. Study was also performed on HTTs with square cut and V-cuts on a solar water heater (of V-trough profile) to decrease the size of the solar collector [13]. Friction factor and heat transfer of a heat exchanger using nanofluid were determined. HTT of twist ratios 5, 10, 15, and 20 and nanofluid concentrations of 0.01% and 0.03% (on volume basis) were used in a U-tube heat exchanger. It was found that corresponding to twist ratio of 5 and volume concentration of 0.03%, the Nusselt number was 32.91% and friction factor was 1.38 times greater than water [14].

EXPERIMENTAL FACILITY

The experimental setup consists of a collecting tank, a laboratory model shallow salinity gradient solar ponds, circular pipe heat exchanger, TTs, submersible pump, digital manometer, and pipe fittings. The details of the devices used are provided in Table 1 and the schematic diagram is represented in Fig. 1.

A laboratory model pond containing water was heated using two electric immersion heaters of 1000 W each and the temperature of the pond was maintained at 75°C. Water from the collecting tank is circulated through the heat exchanger using a submersible pump with a flow rate of 0.02 kg/s, to achieve this flow rate, a bypass line and bypass valve is provided. The Reynolds number corresponding to this flow is 2800.

Full-length helical tapes of twist ratios (Y) 7, 9, and 11 were used to augment the rate of heat transfer. Figs. 2a-c show a typical HTT, co-swirl TTs, and CTTs. The temperatures of laboratory model pond, inlet water, and outlet water of the heat exchanger are measured using "J" type thermocouple connected to a digital temperature indicator.

Table 1:

Equipment/device	Description
Laboratory model pond	1.2 m (L) ×0.3 m (W) ×0.4 m (H) ×1.5 mm (T), GI sheet
Heat exchanger tube	Copper tube of ½ N.B (16 mm O.D, 15 mm I.D and 1.3 m long)
Pump	Submersible pump of 1.5 lit/minutes
Digital manometer	Pressure sensor with pressure transducer
Collecting tank	0.6 m (L) ×0.5 m (W) ×0.3 m (H)
Thermocouple	J type(iron-constantan), range (−40-750°C)
Temperature indicator	12 channel, digital and SELEC make
Heating source	Electric immersion heater of 1000 W of 2 nos

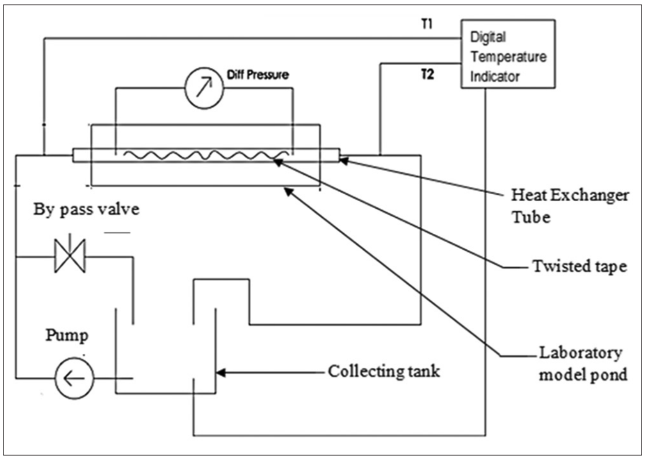

Fig. 1: Experimental setup

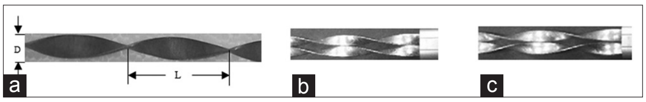

Fig. 2: (a) Helical twisted tape, (b) co-swirl flow twisted tapes, (c) counter-swirl twisted tapes

The difference in pressure along the flow passage was measure using a pressure sensor connected to a pressure transducer. The performance parameters of the IHE such as difference in temperature, rate of heat transfer, and heat exchanger effectiveness were determined corresponding to various temperatures of the pond. Nusselt number, friction factor, and pressure drop were determined for various TTs for maximum pond temperature of 75°C.

DATA REDUCTION

The following are the data reduction equations used:

i. Difference in temperature

$\Delta T = (T_{out} - T_{in})$

Where

ΔT - Difference in temperature (°C)

T_{out} - Outlet temperature (°C)

T_{in} - Inlet temperature (°C).

ii. Heat transfer

$Q_{act} = \dot{m} C \Delta T$

Where

$\dot{m}$ - Mass flow rate of heat transfer medium (kg/s)

Q_{act} - Heat transfer (W)

C - Specific heat capacity (kJ/kgK)

ΔT - Temperature difference (°C)

iii. Overall heat transfer coefficient

$$U = \frac{Q_{act}}{A*LMTD}$$

A - Surface area of heat exchanger = $\pi d_h l$

d_h - hydraulic diameter (m)

l - length of the heat exchanger (m)

iv. Effectiveness of heat exchanger

$\varepsilon = Q_{act}/Q_{max}$

Q_{act} - Actual heat transfer

Q_{max} - Maximum heat transfer

$Q_{max} = M(T_{pond} - T_{in})$

T_{pond} - pond temperature (°C)

v. Nusselt number

The theoretical Nusselt number is calculated as per the equation given below,

For plain tube heat exchanger (Dittus-Boelter Equation) [15]

$Nu = 0.023\ Re^{0.8}\ Pr^{0.3333}$

For heat exchanger with TT [16]

$Nu = 4.162\ [6.413\times10^{-9}\ (SW*Pr^{0.391})^{3.385}]^{0.2}\ (m/m_w)^{0.14}$

Where SW - Swirl parameter = $Re/Y^{0.5}$

Y - Twist ratio.

For heat exchanger with co-swirl TTs [17]

$Nu = 0.209\ Re^{0.72}\ Pr^{0.4}\ Y^{-0.45}$

For heat exchanger with CTTs [17]

$0.268 Re^{0.7}\ Pr^{0.4}\ Y^{-0.4}$

Experimental Nusselt number is calculated as mentioned below

$Nu = (Q/A_s\ DT)\ d_i/k$

Where

Q - Actual rate of heat transfer (W)

As - Surface area of the heat exchanger (m^2)

k - Thermal conductivity of water (W/mK)

vi. Pressure drop and fanning friction factor,

(For plain tube) fanning friction factor is

$f = 0.314\ Re^{-0.25}$ (Blasius equation)

For plain TT [18]

$f = 32.415\ Re^{-0.598}\ Y^{-0.7986}$

For heat exchanger with co-swirl TTs [17]

$f = 24.96\ Re^{-0.59}\ Y^{0.1}$

For heat exchanger with CTTs [17]

$f = 66.7\ Re^{-0.6}\ Y^{-0.54}$

Pressure drop is calculated as

$$\Delta P = \frac{4flV^2}{2gd}$$

Where,

Δp - Pressure drop in (Pa)

f - Friction factor

V - Velocity of flow (m/s)

g - Acceleration due to gravity (m/s^2)

d - Inner diameter of the pipeline (m).

RESULTS AND DISCUSSION

The outputs obtained from the experiment for plain tube heat exchanger are found to be in agreement with correlation of Dittus- Boelter and Blasius equation for the Nusselt number and friction factor, respectively. The performance parameters such as rate of heat transfer, heat exchanger effectiveness, overall heat transfer coefficient, Nusselt number, and friction factor are represented in the Figs. 3-8.

The rate of heat transfer for different pond temperatures with various heat exchangers considered is illustrated in Fig. 3. The trend is same for all temperatures indicating increased thermal performance of the heat exchangers with less twist ratio. The heat exchanger with CTT performed extremely well compared all other heat exchangers. This due to the generation of swirl is in opposite direction, and resulted with the formation of recirculation zone and impingement zone. The thermal performance of heat exchanger with CoTT performed well next to CTT. In the heat exchanger with CoTT, even though the number swirl generated are two, both the swirls are in the same direction. The generation of swirl and recirculation zone in the heat exchanger with CoTT and heat exchanger with CTT are represented in Fig. 4a and b.

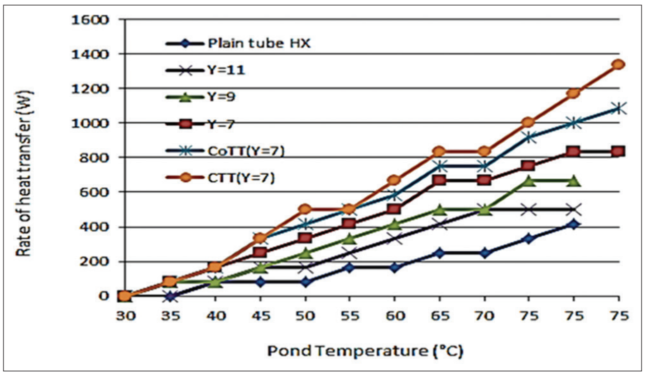

Fig. 3: Rate of heat transfer

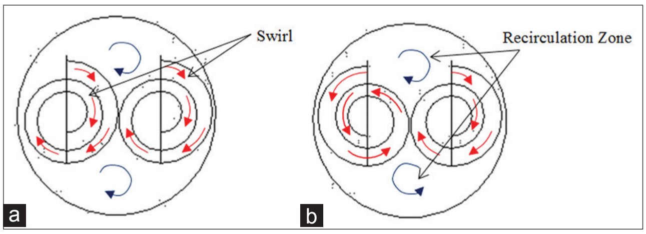

Fig. 4L (a) Direction of swirl and recirculation for co-swirl flow twisted tapes, (b) direction of swirl and recirculation for counter-swirl twisted tapes

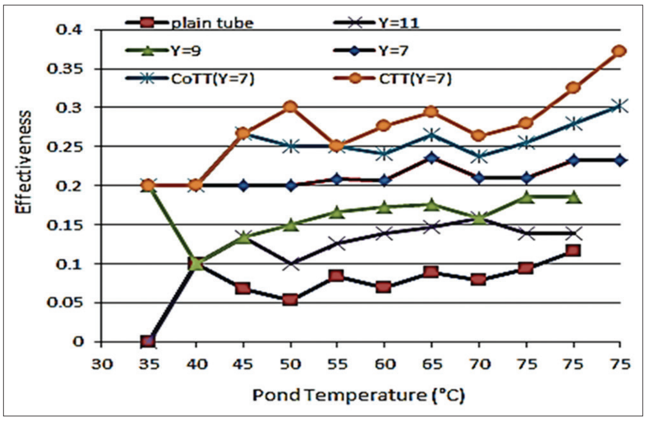

Fig. 5: Effectiveness of IHE

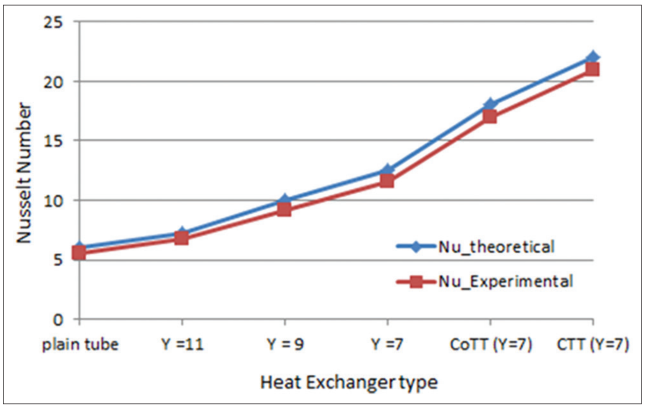

Fig. 6: Nusselt number

The extent to which the outlet water temperature reaches the pond temperature is illustrated in Fig. 5. Placing TTs in the flow passage increases the flow resistance, thereby the period of exposure of the heat transfer fluid to pond temperature is more. Thus, the performance parameter of the heat exchanger "effectiveness" is higher when the flow resistance is higher. Over the range of heat exchangers considered, heat exchanger with CTT is having an effectiveness of above 0.35 when the maximum pond temperature of 75°C is attained.

The Nusselt number corresponding to heat exchange process for various heat exchangers is determined experimentally and from the correlations available from the literature. The experimental values are in good agreement with the empirical correlations. Greater Nusselt number was reported for heat exchanger with CTT, indicating augmented rate of heat transfer as in Fig. 6. This maximum Nusselt number is obtained when the pond temperature was at 75°C.

CONCLUSION

An experimental study in the laboratory model IHE fitted with TTs of twist ratios 7, 9, and 11 under a constant flow (Re = 2800) condition was conducted to assess the performance. The use of TTs provided the additional swirl in the principal flow and enhanced the thermal performance. The important findings are:

- Providing TTs in the flow path resulted with increased rate of heat transfer and pressure drop. This is due to better mixing of the fluid in the flow passage.
- Heat exchanger with less twist ratio provided better rate of heat transfer and more resistance to flow, compared to other heat exchangers as the swirl in the flow creates secondary flow. The better rate of heat transfer is due to secondary flow as it increases the flow length and the residence time of the heat transfer fluid in the flow passage.
- Twin TTs performed well compared to single TTs in the flow passage, this is because of existence of recirculation zone.
- Among the twin TTs, CTTs provided better rate of heat transfer and higher friction factor. In co-swirl TTs, recirculation zone occurs at the top and bottom of the pipeline, whereas recirculation and impingement zone occurs at the top and bottom of the pipe.

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