

CONVECTIVE HEAT TRANSFER ENHANCEMENTS IN TUBE USING LOUVERED STRIP INSERT

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Abstract— The heat transfer and friction factor were experimentally investigated in a louvered strip inserted tube in turbulent flow region. A copper tube of (I.D.=28mm, O.D.=32mm) and 900mm length was used. A louvered strip insert with different geometrical configuration was inserted into the smooth tube. A uniform heat flux condition was created by wrapping heating tape of 2500 watt around the test section. Fibre glass cloth was used as a thermal insulator which surrounds the heating tape. Outer surface temperature of the tube were measured at five different equally spaced points of test section by k-type thermocouples. Two thermocouples were used to measure the inlet and outlet temperature of water. The Reynolds numbers were varied in the range of 2500 to 4000 with constant heat flux of 24 kw/m² for smooth tube and louvered strip inserted. Nusselt number and friction factor obtained for louvered strip (with forward backward arrangement) > Nusselt number and friction factor for louvered strip (with semi-forward semi-backward arrangement) > Nusselt number and friction factor for louvered strip (with forward arrangement).

Key words— passive heat transfer, twisted tape, wire coil, swirl flow.

I. INTRODUCTION

Heat exchangers are popular used in industrial and engineering applications. The design procedure of heat exchangers is quite complicated, as it needs exact analysis of heat transfer rate, efficiency and pressure drop apart from issues such as long-term.

performance and the economic aspect of the equipment. Whenever inserts technologies are used for the heat transfer enhancement, along with the improvement in the heat transfer rate, the pressure drop also increases, which induces the higher pumping cost. Therefore any augmentation device or methods utilized into the heat exchanger should be optimized between the benefits of heat transfer coefficient and the higher pumping cost owing to the increased frictional losses. In general, heat transfer augmentation methods are classified into three broad categories

A. Active method

This method involves some external power input for the enhancement of heat transfer. Some examples of active methods include induced pulsation by cams and reciprocating plungers, the use of a magnetic field to disturb the seeded light particles in a flowing stream, mechanical aids, surface vibration, fluid vibration, electrostatic fields, suction or injection and jet impingement requires an external activator/power supply to bring about the enhancement.

B. Passive method

This method generally uses surface or geometrical modifications to the flow channel by incorporating inserts or

additional devices. For example, inserts extra component, swirl flow devices, treated surface, rough surfaces, extended surfaces, displaced enhancement devices, coiled tubes, surface tension devices and additives for fluids.

C. Compound method

Combination of the above two methods, such as rough surface with a twisted tape swirl flow device, or rough surface with fluid vibration, rough surface with twisted tapes. This paper focuses on reviewing the passive methods in pipe heat exchanger. The passive heat transfer augmentation methods as stated earlier do not need any external power input. For the convective heat transfer, one of the ways to enhance heat transfer rate is to increase the effective surface area and residence time of the heat transfer fluids. The passive methods are based on this principle, by employing several techniques to generate the swirl in the bulk of the fluids and disturb the actual boundary layer so as to increase effective surface area, residence time and consequently heat transfer coefficient in existing system. Although there are hundreds of passive methods to enhance the heat transfer performance, the following nine are most popular used in different aspects:

- **Treated Surfaces:** They are heat transfer surfaces that have a fine-scale alteration to their finish or coating. The alteration could be continuous or discontinuous, where the roughness is much smaller than what affects single-phase heat transfer, and they are used primarily for boiling and condensing duties.
- **Rough surfaces:** They are generally surface modifications that promote turbulence in the flow field, primarily in single-phase flows, and do not increase the heat transfer surface area. Their geometric features range from random sand-grain roughness to discrete three-dimensional surface protuberances.
- **Extended surfaces:** They provide effective heat transfer enlargement. The newer developments have led to modified fin surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.
- **Displaced enhancement devices:** These are the insert techniques that are used primarily in confined force convection. These devices improve the energy transfer indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct/pipe with bulk fluid to the core flow.
- **Swirl flow devices:** They produce and superimpose swirl flow or secondary recirculation on the axial flow in a channel. These devices include helical strip or cored screw type tube inserts, twisted tapes. They can

be used for single phase or two-phase flows heat exchanger.

- **Coiled tubes:** These techniques are suitable for relatively more compact heat exchangers. Coiled tubes produce secondary flows and vortices which promote higher heat transfer coefficient in single phase flow as well as in most boiling regions.
- **Surface tension devices:** These consist of wicking or grooved surfaces, which directly improve the boiling and condensing surface. These devices are most used for heat exchanger occurring phase transformation.
- **Additives for liquids:** These include the addition of solid particles, soluble trace additives and gas bubbles into single phase flows and trace additives which usually depress the surface tension of the liquid for boiling systems.
- **Additives for gases:** These include liquid droplets or solid particles, which are introduced in single-phase gas flows either as dilute phase (gas–solid suspensions) or as dense phase (fluidized beds).

II. EXPERIMENTAL SET UP

A. Test section

The schematic diagram is shown in fig.2. Test section consist of tube (I.D=28mm, O.D=32mm) of length 900mm. Five k type thermocouples were soldered at five equally spaced points which were separated by 150mm distance. This copper tube were wrapped by flexible heating tape of 2500watt in order to maintain constant heat flux. This flexible heating tape were surrounded by fibre glass cloth which were acting like heat insulator. The heating tape was connected to 270 volt main. A U-tube manometer was used to measure the pressure drop across the tube. The distance between two pressure tapping was 1000mm.

B. Control panel

It consist of following measuring instruments.

- Dimmerstat-(Range 0 to 270 volt).
- Voltmeter-(Range 0 to 270 volt).
- Ammeter-(Range 0 to 10 ampere).
- Temperature indicator – (Range 0 to 10000c).



Fig1: Control Panel



Fig.2:- Photographic view of experimental set up

C. Experimental procedure

Inlet section of set up is connected to the gate valve of water tank which takes water and pumped through test section. The flow rate of water is controlled by gate valve and was measured manually using stop watch and beaker. The flow rate varied using gate valve for different values of Reynold number and kept constant during experiment. After switching on the heating power the sufficient time was given to attain the steady state condition. In each run data were taken for water flow rate, water inlet, outlet and tube outer surface temperature and pressure drop readings.

D. Insert Type

- Louvered strip (with forward arrangement)



A 2mm strip of 1.5cm× 1.5cm (cone angle=300) was welded to iron rod at equally spaced points (pitch=100mm) in cyclic order which was separated by 900.

- Louvered strip (with semi-forward semi-backward arrangement)



Half side of rod was welded with forward directed strip and remaining side was welded with backward directed strip so that mouth of strip facing opposite to each other.

- Louvered strip with (forward backward arrangement)



In this case one strip was welded in forward direction, the next successive strip was welded in backward direction.

III. DATA REDUCTION

Heat transfer rate from heating tape to water was calculated by measuring heat added to the water. Heat added to water was calculated by,

$$Q = mcp(T_{out} - T_{in}) \quad (1)$$

Heat transfer coefficient was calculated from,

$$h = \frac{q}{(T_{wi} - T_b)} \quad (2)$$

and heat flux was obtained from,

$$q = Q/A \quad (3)$$

where, $A = \pi diL$ (4)

The bulk temperature was obtained from the average of inlet and outlet temperatures,

$$T_b = \frac{T_{in} + T_{out}}{2} \quad (5)$$

Tube inner surface temperature was calculated from one dimensional radial conduction equation,

$$T_{wi} = T_{wo} - Q \cdot \ln(do/di) / (2\pi k_w L) \quad (6)$$

Tube outer surface temperature was calculated from the average of five local tube outer surface temperatures,

$$T_{wo} = \frac{\sum_{i=1}^5 T_{wo,i}}{5} \quad (7)$$

Theoretical Nusselt number was calculated from Gnielinski, 1976, correlation,

$$N_{uth} = \frac{\left(\frac{L}{S}\right) (Re - 1000) pr}{1 + 12.7 \left(\frac{L}{S}\right)^{0.5} (pr^{0.66} - 1)}$$

where from Petukhov, 1970,

$$f = (0.79 \ln Re - 1.64) - 2 \quad (9)$$

$$Re = \frac{\rho U m di}{\mu} \quad (10)$$

Mean water velocity was obtained from,

$$U_m = m / (\pi / 4 d^2) \quad (11)$$

Friction factor, f can be calculated from

$$f = \frac{\Delta p}{\left(\frac{L}{di}\right) (\rho U m^2 / 2)} \quad (12)$$

Δp is the pressure drop across tappings. All the fluid properties were evaluated at bulk temperature.

IV. RESULTS AND DISCUSSION

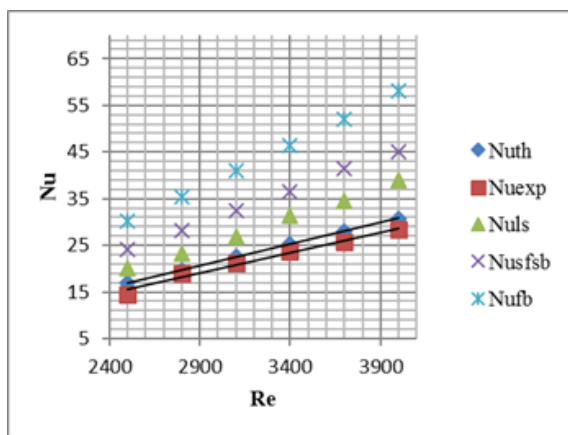


Fig3. The variation of Nusselt number with Reynolds number.

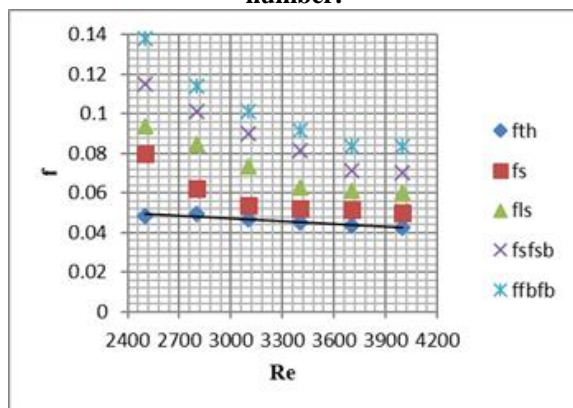


Fig4. The variation of friction factor with Reynolds number.

Notation of fig.3 and 4:

Nuth and fth = Theoretical Nusselt number and friction factor.

Nus and fs = Nusselt number and friction factor for smooth tube.

Nuls and fls = Nusselt number and friction factor for louvered strip (with forward arrangement).

Nusfsb and fsfsb = Nusselt number and friction factor for louvered strip (with semi forward semi backward arrangement).

Nufb and ffbfb = Nusselt number and friction factor for louvered strip (with forward backward arrangement).

From fig.3 it was found that Nusselt number increases with increase of Reynold number, Nusselt number obtained for tube with louvered strip insert was more than that of plain tube and heat transfer rate is higher for tube fitted with louvered strip than the plain tube, this is because of mixing of fluid at centre and fluid at surface region. This mixing of surface fluid and centre fluid has ability to enhance heat transfer rate.

At fixed value of Reynold number, Nusselt number is highest for tube fitted with louvered strip (with forward backward arrangement) than the louvered strip (with semi forward semi backward arrangement) and Nusselt number for tube fitted with (semi forward semi backward arrangement) is more than Nusselt number for tube fitted with louvered strip (forward arrangement). This is due to the fact that louvered strip (with forward backward arrangements) creates more turbulence which results in better mixing of fluid at centre and fluid at surface, which enhances heat transfer rate.

From fig.4. shows the variation of friction factor with Reynold number for different arrangements of louvered strip. It can be clearly seen that the friction factor continues to decrease with increased of Reynold number. The friction factor obtained for tube fitted with louvered strip (with forward backward arrangements) is higher than louvered strip (with semi forward semi backward arrangements). This is due to high contact of fluid with wall region.

V. LITERATURE REVIEW

Large number of experimental work are carried out by researchers to investigate the thermohydraulic performance of various twisted tapes including the traditional simple twisted tapes, regularly spaced twisted tapes, varying length twisted tapes, tapes with different cut shapes, tapes with baffles and tapes with different surface modifications. The followings content will detail these reaches and display the finds from different researchers.

Kumar and Prasad [1] started to investigate the impact of the twist ratio on the enhancement efficiency for a solar water heater. When changed the twist ratios from 3.0 to 12.0 the heat transfer rate inside the solar collectors have been found increased by 18–70%, whereas the pressure drop increased by 87–132%. Synthetically consider the increase of heat transfer and pressure drop it is concluded that the twisted tape enhanced collectors would be preferable for higher grade energy collection to balance the pressure drop rather than for the solar collectors.

Murugesan et al.[2] carried out a study of the heat transfer and pressure drop characteristics of turbulent flow in a tube fitted with a full length twisted tape coupled with trapezoidal-cut. The results. show that for the twist ratio of 6.0, the mean Nusselt number and fanning friction factor for the trapezoidal-cut twisted tape are 1.37 and 1.97 times over the plain tube,

respectively. When the twist ratio reduces to 4.4, the corresponding Nusselt number and fanning friction factor will increased to 1.72 to 2.85 times. This indicate that trapezoidal – cut induces significant enhancement of heat transfer coefficient and friction factor ,in addition the impact will be heavier for a lower twist ratio.

Sivashanmugam and Suresh [3] studied circular tube fitted with full-length helical screw element of different twist ratio there is no much change of the heat transfer coefficient enhancement by increasing or decreasing the twist ratio, as the magnitude of swirl generated at the inlet or at the outlet is the same in the both of two cases.

Yakut and Sahin [4]. The vortex characteristics of tabulators, heat transfer rate and friction characteristics were considered as the criterions to evaluate the enhancement performance of coiled wire. Garcia et al. experimentally studied the helical-wire-coils fitted inside a round tube in order to characterize their thermohydraulic behaviour in laminar, transition and turbulent flows. Results have shown that

- In laminar flow, wire coils behave as a smooth tube but accelerate transition to critical Reynolds numbers down to 700.
- At the low Reynolds numbers about $Re \approx 700$, transition from laminar to turbulent flow occurs in a gradual way.
- Within the transition region, heat transfer rate can be increased up to 200% when keep the pumping power constant.
- Wire coils have a predictable behaviour within the transition region since they show continuous curves of friction factor and
- Nusselt number, which involves a considerable advantage over other enhancement techniques.
- In turbulent flow, wire coils cause a high pressure drop which depends mainly on the pitch to wire-diameter ratio (p/e).
- In turbulent flow, the pressure drop and heat transfer are both increased by e up to nine times and four times respectively, compared to the empty smooth tube.

Gunes et al.[5] also investigated the thermohydraulic behaviour of coiled wires in tube and pipe heat exchangers in 2010. Also experimentally investigated the coiled wire inserted in a tube for a turbulent flow regime. The coiled wire has equilateral triangular cross section and was inserted separately from the tube wall. They discovered that the Nusselt number rises with the increase of Reynolds number and wire thickness, and the decrease of pitch ratio; the best operating regime of all coiled wire inserts is detected at low Reynolds number, which leads to more compact heat exchanger; the pitch increases, the vortex shedding frequencies decrease and the maximum amplitudes of pressure fluctuation of vortices produced by coiled wire turbulators occur with small pitches.

VI.CONCLUSION

An experimental investigation was carried out for measuring tube-side heat transfer coefficient, friction factor of water for turbulent flow in a circular tube fitted with louvered strip insert. The results can be summarized as,

1. Nusselt number increases with increase of Reynolds number and friction factor decreases with increase of Reynold number.

2. At particular value of Reynold number, Nusselt number and friction factor of tube fitted with louvered strip (with forward backward arrangement)> Nusselt number and friction factor of tube fitted with louvered strip (with semi forward semi backward arrangement)> Nusselt number and friction factor of tube fitted with louvered strip (with forward arrangement).

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