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Heat Transfer Enhancement Using Variable Pitch Swirl Flow Device in Single Tube Heat Exchanger

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ABSTRACT

Energy and material saving considerations, as well as economic incentives, have led toward making effort for producing more efficient heat exchange equipment such as solar air heaters, heat exchangers. The present study explored the effect of variable pitch swirl flow device on convective heat transfer. The experiments were performed with swirl floe device using different pitch of 150 mm, 100 mm and 50 mm using water as the testing tube. The Reynolds number is varied by varying the flow rate of the water. The experimental results demonstrated that convective heat transfer coefficient increases by using swirl flow device. As the pitch is decreased from 150 to 100 to 50 mm the heat transfer coefficient increased. Performance evaluation criteria is maximum for 50 m pitch of swirl flow device. 4. As the Reynolds number increases performance evaluation criteria for a particular pitch decreases which indicated that at higher Reynolds number effect of turbulence element becomes less significant.

Keywords: Heat exchanger, Swirl flow device, Reynolds number, heat transfer coefficient, performance evaluation criteria

I. INTRODUCTION

Heat exchangers are used in different processes ranging from conversion, utilisation & recovery of thermal energy in various industrial, commercial & domestic applications. Some common examples include steam generation & condensation in power & cogeneration plants; sensible heating & cooling in thermal processing of chemical, pharmaceutical & agricultural products; fluid heating in manufacturing & waste heat recovery etc. In these applications, thermal-hydraulics and energy usage play dominant roles. Energy, space, and materials saving considerations, as well as the present-day global economics, have led to the expansion of efforts to produce more efficient heat exchange equipment for minimizing cost, which is to reduce the physical size of heat exchange equipment for a given heat duty. Therefore the main thermal-hydraulic objectives are to reduce the size of a heat exchanger required for a specific heat duty, to upgrade the capacity of an available heat exchanger [1].

The need to increase the thermal performance of heat exchangers, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as “Heat transfer Augmentation”. These techniques are also referred as “Heat transfer Enhancement”. Augmentation techniques increases convective heat transfer by reducing the thermal resistance in a heat exchanger. Existing enhancement techniques can be broadly classified into three different categories:

1. Passive Techniques
2. Active Techniques
3. Compound Techniques

PASSIVE TECHNIQUES [2]

These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power.

Heat transfer augmentation by these techniques can be achieved by using:

a) Treated Surfaces
b) Rough surfaces
c) Extended surfaces
d) Displaced enhancement devices
e) Swirl flow devices
f) Coiled tubes
g) Additives for liquids

ACTIVE TECHNIQUES [2]

These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. Various active techniques are as follows:

a) Mechanical Aids
b) Surface vibration
c) Fluid vibration
d) Electrostatic fields
e) Injection
f) Suction
g) Jet impingement

COMPOUND TECHNIQUES

A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

In the past few decades, numerous heat transfer enhancement techniques for tubes and ducts have been proposed. Wen-Chieh Huang et al. [3] investigated heat transfer of repeated ring type ribs in circular tubes. Tu Wenbin et al. [4] studied heat transfer characteristics of turbulent flow through a circular tube with small pipe insert. They varied space between inserts and arc radius and readings were taken at Reynolds number between 4000 and 18,000 with tap water as working fluid. Smita Agrawal et al. [5] measured heat transfer and velocity in the presence of agitation and channel wall surface augmentation with pin fins in a Large Scale Mock Up (LSMU) test facility. The LSMU facility allows resolution both in time and space for these measurements. Subhashis Ray et al. [6] studied the possibility of using wire-loop structures on the active plate of a parallel plate channel for efficient heat transfer augmentation. They observed substantial heat transfer augmentation with wire-loop structures as compared to the empty parallel plate channel under the condition of identical pressure gradient, the thermal-hydraulic performance improved significantly with the increase in loop-density. M.M.K. Bhuiya et al. [7] explored the effects of the double counter twisted tapes on heat transfer and fluid friction characteristics in a heat exchanger tube. The double counter twisted tapes were used as counter-swirl flow generators in the test section. They varied Reynolds number from 6950 to 50050. The heat transfer rate in the tube fitted with double counter twisted tape was significantly increased. Mohsen Sheikholeslami et al. [8] reviewed various turbulators like coiled tubes, extended surfaces (fin, louvered strip, winglet), rough surfaces (Corrugated tube, Rib) and swirl flow devices such as twisted tape, conical ring, snail entry turbulator, vortex rings, coiled wire for enhancing heat transfer in heat exchangers.

As discussed above there are many techniques that can be used to enhance in tube heat transfer. In this paper a new type of swirl flow insert (device) is used and also its pitch can be varied.

II. EXPERIMENTAL SETUP & APPARATUS

The swirl flow device was manufactured from aluminium bar and then it was cut in 120° sections and inserted in pipe. Figure 1 shows the production drawing of swirl flow insert.

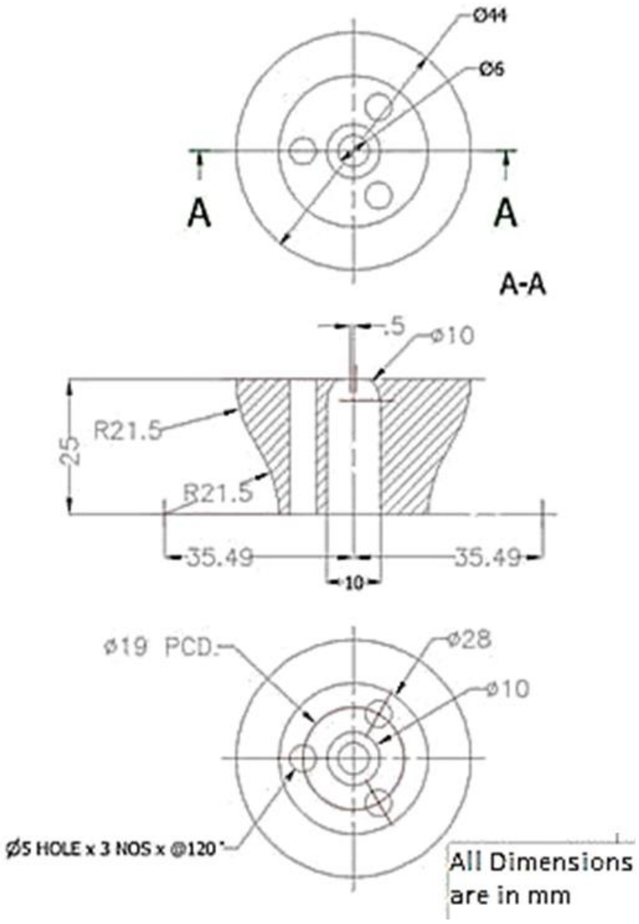

Figure 1. Production drawing of swirl flow insert

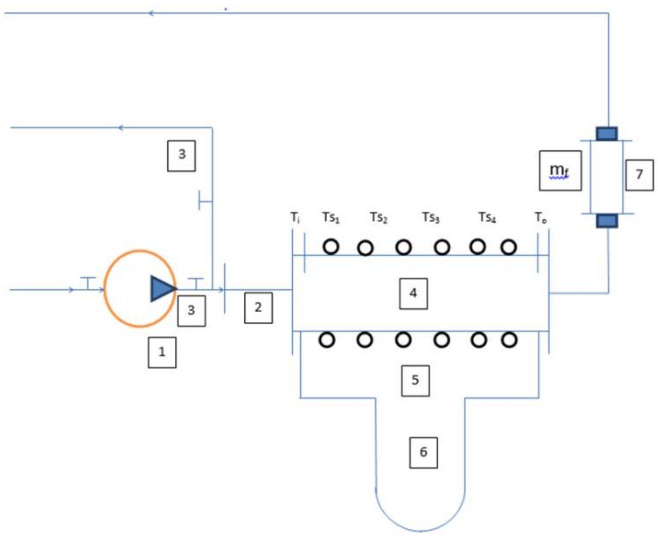

Figure 2. Experimental Setup. (1) Pump, (2) Calming Section, (3) Flow control section, (4) Test Section, (5) Electric Heater, (6) Manometer, (7) Rotametr

Figure 2 shows schematic diagram of the experimental set-up. It consist of a pump, calming pipe, flow control vale, test section, Rotameter, electric heater with variac, U- tube manometer, RTD set for surface temperature measurement and inlet/outlet water temperature. The water at room temperature is circulated by pump. Flow is regulated by flow control valve. Electric heater is used to rise the temperature of water at constant heat flux.

Test section shown in figure 3 is a mild steel pipe, cylindrical surface heater is fixed on outer surface of test section pipe. Total heat load can be varied by variac. Outer surface of heater is insulated by glass wood and asbestos to minimize heat loss. Surface temperature of test section (T1 to T4) and fluid inlet temperature (Ti) / fluid outlet temperature (To) are measured by multipoint temperature indicator. Variable pitch turbulence element can be fixed inside test section.

Figure 3. Test Section

Variable pitch swirl flow element is made from aluminium alloy. Receiving surface is made concave for smooth reception of fluid. Exit surface is made convex for guiding the fluid in the flow direction. Each element covers 120° of a full ring. Three M4 stainless steel studs with nuts are provided to adjust the pitch of turbulence element. Studs are locked at end (figure 4).

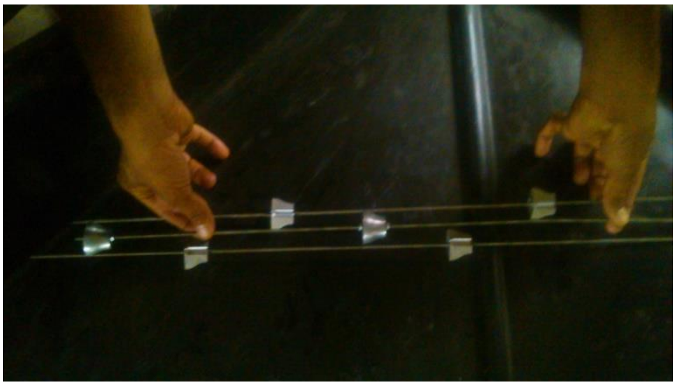

Figure 4. Swirl flow elements set

Geometry of the test section fitted with swirl flow element on stud is shown in figure 3. The plain tube was made of mild steel having 52 mm inside diameter, 61 mm outside diameter and 930 mm length. The calming section is also having same ID and OD but its length is 1800 mm. Heater used with capacity of 3.75 kW, 61 mm inside diameter and 450 mm length. The electrical output power was controlled by a variac transformer to obtain a constant heat flux condition throughout the entire test section (0-260 V). Circulating pump is of 0.5 HP centrifugal pump. RTD with range 0-400 ℃ were used to monitor local wall temperature as well as fluid temperature. The flow of water was measured by Rotameter (50-1000 Lph). The SS stud on which turbulence element were fitted had length 930 mm and the maximum and minimum pitch that can be adjusted on stud were 287.5 mm and 27.5 mm.

The experiments were conducted for pitch length of 50 mm, 100 mm and 150 mm at three different flow rates 300, 450 and 600 lph.

CALUCLATIONS:

1. Theoretical heat transfer co-efficient

a) Laminar Flow:

For Re < 2100

Nu = f(Gz)

Where $Gz = \frac{\mathrm{Re} \times \mathrm{Pr} \times d_i}{L}$

i. For Gz < 100, Hausen Equation is used.

$$Nu = 3.66 + \frac{0.085Gz}{1 + 0.45Gz^{0.67}} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

ii. For Gz > 100, Seider Tate equation is used

$$Nu = 1.86Gz^{\frac{1}{3}} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

b) Transition Zone:

For 2100 <Re< 10000, Hausen equation is used

$$Nu = 0.116 \left(\mathrm{Re}^{\frac{2}{3}} - 125 \right) \times \mathrm{Pr}^{\frac{1}{3}} \times \left(1 + \left(\frac{D}{l} \right)^{\frac{2}{3}} \left(\frac{\mu_b}{\mu_w} \right)^{0.14} \right)$$

c) Turbulent Zone:

For Re > 10000, Seider- Tate equation is used

$$Nu = 0.23 \times \mathrm{Re}^{0.8} \times \mathrm{Pr}^{\frac{1}{3}} \times \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

Viscosity correction Factor is assumed to be equal to 1 for all calculations as this value for water in present case will be very close to 1 & the data for wall temperatures is not measured.

2. Experimental heat transfer co-efficient

$$Q = mC_p \left(T_o - T_i \right) = h_a A \left(T_s - T_m \right)$$

III. RESULTS & DISCUSSION

The performance evaluation criterion (PEC) was used to evaluate the performance of the tube fitted with swirl flow device and is expressed as follows:

Performance evaluation criteria

$$R = \frac{h_a \textit{ for particular pitch}}{h_a \textit{ for plain pipe}}$$

Theoretical and actual convective heat transfer co-efficient was calculate for the plain tube first at different Reynolds number. Then the same is calculated for the tube fitted with swirl flow elements with pitch 150 mm, 100 mm and 50 mm respectively.

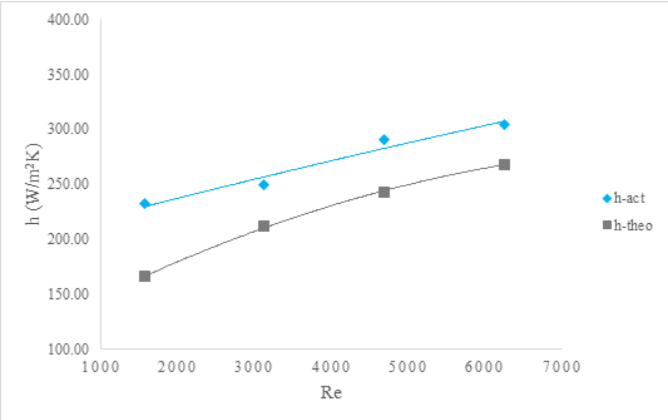

Figure 5. h_{th} & h_a vs. Re for plain tube

Figure 4.1 represents variations of actual and theoretical heat transfer co-efficient with respect to Reynolds number for a plane pipe. Higher deviation is observed between h_a and h_{th} at lower Reynolds number because of natural convection taking place along with forced convection. As Reynolds number increases natural convection becomes less significant. So difference between h_a and h_{th} decreases.

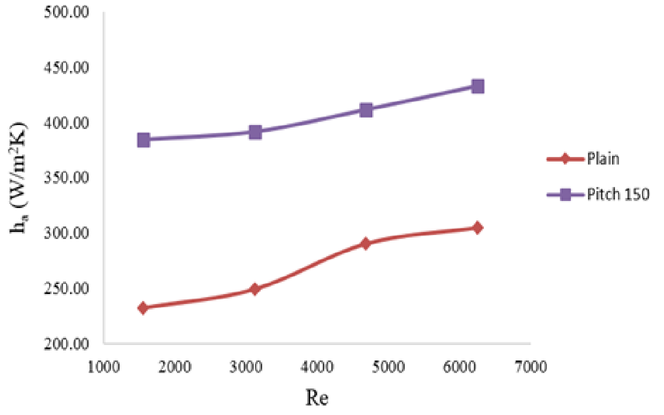

Figure 6. h_a vs. Re for 150 mm pitch

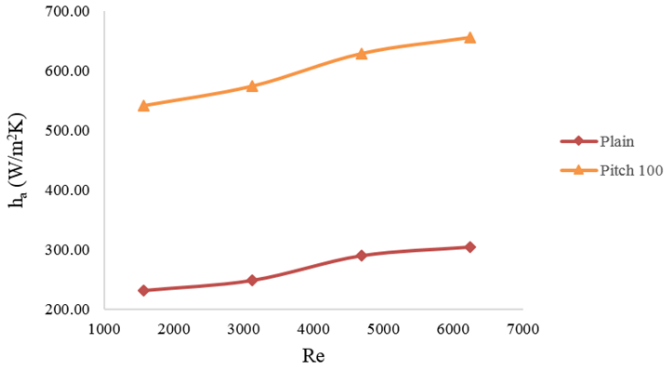

Figure 7. h_a vs. Re for 100 mm pitch

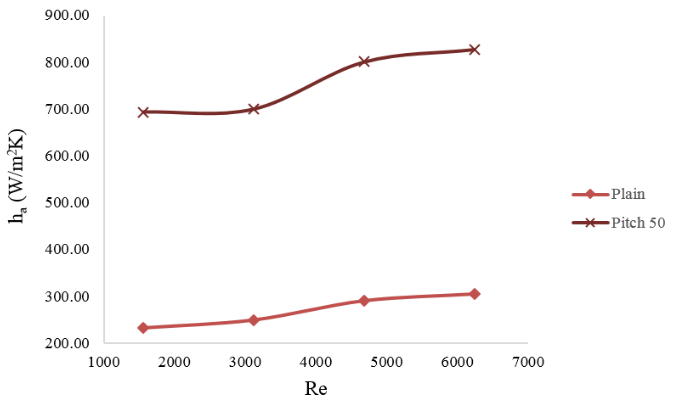

Figure 8. h_a vs Re for 50 mm pitch

Figure 6, 7 and 8 represents heat transfer results for 150 mm, 100 mm, and 50 mm pitch elements respectively.

As pitch decreases more turbulence and secondary fluid motion is generated so heat transfer enhancement increases. At 150 mm pitch maximum rise of heat transfer co-efficient is 65.74% and minimum rise of heat transfer co-efficient is 41.88 % with respect to plane tube. These values are 133.29% and 115.32% for 100 mm pitch and 198.74% and 171.52% for 50 mm pitch element.

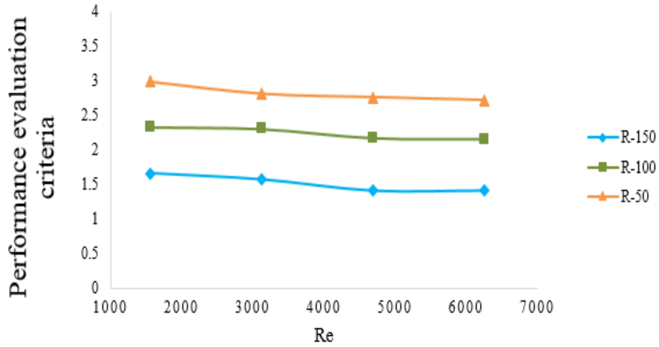

Figure 9. Performance evaluation criteria vs. Re

Figures 9 represents variation of performance evaluation criteria with respect to Reynolds number at different pitches. Performance evaluation criteria is minimum for 150 mm pitch for respective Reynolds number. At lower Reynolds number influence of swirl flow element on heat transfer enhancement is higher so performance evaluation criteria is higher. Visa-versa at higher Reynolds number and at higher pitch performance evaluation criteria decreases.

IV.CONCLUSION

Following conclusions were made after experimental investigations:

1. As the pitch decreases higher degree of swirl is generated. For the same Reynolds number swirl flow element at 50 mm pitch gives maximum heat transfer co-efficient with respect to 100 mm and 150 mm pitch.
2. For same pitch of swirl flow elements of Reynolds number increases heat transfer co-efficient increases.
3. Variation of heat flux have no significant effect on heat transfer co-efficient.
4. As the Reynolds number increases performance evaluation criteria for a particular pitch decreases which indicated that at higher Reynolds number effect of swirl flow elements becomes less significant.
5. By introducing swirl flow elements average surface temperature deceases considerably for same heat load.

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