Review of Recent Techniques of Heat Transfer Enhancement and Validation of Heat Exchanger

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***Abstract*—Heat exchangers have several industrial and engineering applications. The design procedure of heat exchangers is quite complicated, as it needs exact analysis of heat transfer rate and pressure drop estimations apart from issues such as long-term performance and the economic aspect of the equipment. Whenever inserts are used for the heat transfer enhancement, along with the increase in the heat transfer rate, the pressure drop also increases. This increase in pressure drop increases the pumping cost. Therefore any augmentation device should optimize between the benefits due to the increased heat transfer coefficient and the higher cost involved because of the increased frictional losses. Heat transfer augmentation techniques (passive, active or a combination of passive and active methods) are commonly used in areas such as process industries, heating and cooling in evaporators, thermal power plants, air-conditioning equipment, refrigerators, radiators for space vehicles, automobiles, etc. Passive techniques, where inserts are used in the flow passage to augment the heat transfer rate, are advantageous compared with active techniques, because the insert manufacturing process is simple and these techniques can be easily employed in an existing heat exchanger**

***Index Terms*— Active Technique, Passive Technique, Compound Technique, Heat exchanger**

I. INTRODUCTION

Heat exchangers have several industrial and engineering applications. The design procedure of heat exchangers is quite complicated, as it needs exact analysis of heat transfer rate and pressure drop estimations apart from issues such as long-term performance and the economic aspect of the equipment. To make the equipment compact and achieve a high heat transfer rate using minimum pumping power.

Techniques for heat transfer augmentation are relevant to several engineering applications. In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment.
Furthermore, sometimes there is a need for miniaturization of a heat exchanger in specific applications, such as space application, through an augmentation of heat transfer. Therefore, an increase in the efficiency of the heat exchanger through an augmentation technique may result in a considerable saving in the material cost.

In some specific applications, such as heat exchangers dealing with fluids of low thermal conductivity (gases and oils) and desalination plants, there is a need to Increase the heat transfer rate. The heat transfer rate can be improved by introducing a disturbance in the fluid flow (breaking the viscous and thermal boundary layers), but in the process pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, several techniques have been proposed in recent years and are discussed in the following sections.

II. IMPORTANT DEFINATIONS

In this section a few important terms commonly used in heat transfer augmentation work are defined.

A. Thermohydraulic performance-

For a particular Reynolds number, the thermohydraulic performance of an insert is said to be good if the heat transfer coefficient increases significantly with a minimum increase in friction factor. Thermohydraulic performance estimation is generally used to compare the performance of different inserts such as twisted tape, wire coil, etc., under a particular fluid flow condition.

B. Overall enhancement ratio-

The overall enhancement ratio is defined as the ratio of the heat transfer enhancement ratio to the friction factor ratio. This parameter is also used to compare different passive techniques and enables a comparison of two different methods for the same pressure drop. The friction factor is a measure of head loss or pumping power.

C. Nusselt number-

The Nusselt number is a measure of the convective heat transfer occurring at the surface and is defined as hd/k, where h is the convective heat transfer coefficient, d is the diameter of the tube and k is the thermal conductivity

D. Prandtl number-

The Prandtl number is defined as the ratio of the molecular diffusivity of momentum to the molecular diffusivity of heat.

E. Pitch

Pitch is defined as the distance between two points that are on the same plane, measured parallel to the axis of a twisted tape.

F. Twist ratio, y

The twist ratio is defined as the ratio of pitch to inside diameter of the tube y ¼ H/di, where H is the twist pitch length and d is the inside diameter of the tube

III. HEAT TRANSFER AUGMENTATION

Classification of Augmentation Techniques:

They are broadly classified into three different categories:

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

1. Passive Techniques:

These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop. They 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 behavior except for extended surfaces. Heat transfer augmentation by these techniques can be achieved by using;

i. Treated Surfaces: Such surfaces have a fine scale alteration to their finish or coating which may be continuous or discontinuous. They are primarily used for boiling and condensing duties.

ii. Rough surfaces: These are the surface modifications that promote turbulence in the flow field in the wall region, primarily in single phase flows, without increase in heat transfer surface area.

iii. Extended surfaces: They provide effective heat transfer enlargement. The newer developments have led to modified finned surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.

iv. Displaced enhancement devices: These are the inserts that are used primarily in confined forced convection, and they improve energy transport indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.

v. Swirl flow devices: They produce and superimpose swirl flow or secondary recirculation on the axial flow in a channel. These include helical strip or cored screw type tube inserts, twisted tapes. They can be used for single phase and two-phase flows.

vi. Coiled tubes: These lead to relatively more compact heat exchangers. It produces secondary flows and vortices which promote higher heat transfer coefficients in single phase flows as well as in most regions of boiling.

vii. Surface tension devices: These consist of wicking or grooved surfaces, which direct and improve the flow of liquid to boiling surfaces and from condensing surfaces.

viii. Additives for liquids: These include the addition of solid particles, soluble trace additives and gas bubbles in single phase flows and trace additives which usually depress the surface tension of the liquid for boiling systems.

ix. 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).

2. Active Techniques:

In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer.

Augmentation of heat transfer by this method can be achieved by

(i) Mechanical Aids: Such instruments stir the fluid by mechanical means or by rotating the surface. These include rotating tube heat exchangers and scrapped surface heat and mass exchangers.

(ii) Surface vibration: They have been applied in single phase flows to obtain higher heat transfer coefficients.

(iii) Fluid vibration: These are primarily used in single phase flows and are considered to be perhaps the most practical type of vibration enhancement technique.

(iv) Electrostatic fields: It can be in the form of electric or magnetic fields or a combination of the two from dc or ac sources, which can be applied in heat exchange systems involving dielectric fluids. Depending on the application, it can also produce greater bulk mixing and induce forced convection or electromagnetic pumping to enhance heat transfer.

(v) Injection: Such a technique is used in single phase flow and pertains to the method of injecting the same or a different fluid into the main bulk fluid either through a porous heat transfer interface or upstream of the heat transfer section.

(vi)Suction: It involves either vapor removal through a porous heated surface in nucleate or film boiling, or fluid withdrawal through a porous heated surface in single-phase flow.

(vii) Jet impingement: It involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface.

3) Compound Techniques:

When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement. This technique involves complex design and hence has limited applications.

1] PASSIVE METHODS-

These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop.

Brief descriptions of these methods follow:

a)*TREATED SURFACES*:

These are primarily applicable in two phase heat transfer and they consist of a variety of structured surfaces (continuous or discontinuous integral surface roughness or alterations) and coatings. Though the treatment provides a roughness to the surface, it is not large enough to influence single phase heat transfer.

Boiling: Different types of treated surfaces used are

- Machined or grooved surfaces
- Formed or modified low-fin surfaces
- Multilayered surfaces
- Coated surfaces

The principle of providing treated surfaces for enhanced boiling is to produce a large number of stable vapor traps or nucleation sites on the surface. This is applicable for highly wetting fluids like refrigerants, organic liquids, cryogens and alkali liquid metals where the normal cavities present on the heated surfaces tend to experience sub-cooled liquid flooding. For less wetting or relatively higher surface tension fluids, coatings of non-wetting material (eg.teflon) on either the heated surface or its pits and cavities have been found to improve stable nucleation and reduce the required wall super

When the stainless steel surface along with Teflon is spread to create spots of the no-wetting material on the heated surface it was found to promote nucleate boiling in water with relatively low wall super heat and three to four times higher heat transfer coefficients, was proposed by Young and Hummel(1965). In a more recent study of boiling of alcohols (methanol, ethanol and isopropanol) at atmospheric and sub-atmospheric pressures on a horizontal brass tube coated with poly-tetra-fluoro-ethylene (PTFE), Vijaya Vittala et al. (2001), found a significant enhancement in heat transfer.

Condensing*:*

Vapor space condensation heat transfer coefficients can be enhanced primarily by treated surfaces that promote drop wise condensation. The intent here is to prevent surface wetting and break up the condensate film into droplets which lends to better drainage and more effective vapor renewal at the cold heat transfer interface. This technique had been found to enhance the heat transfer by a factor of 10 to 100 in comparison with that in film wise condensation proposed by Bergles, (1998). Non-wetting coatings of an inorganic compound or a noble metals or an organic polymer have been used effectively. Among these, organic coatings have been used considerably in steam systems. Glicksman et al. (1973) have been found out that, by placing strips of Teflon or other non-wetting material in a helical or axial arrangement around the circumference of horizontal tubes, the average condensation heat transfer coefficients of steam on horizontal tubes can be improved by 20 to 50%. The application of hydrophobic coatings of self-assembled monolayers, formed by chemisorptions of alkylthiols on metallic surfaces, to promote drop wise condensation has been proposed by Das et al. (2000). It was found that steam condensation on coated corrugated tubes with gold and copper-nickel alloy surfaces under atmospheric and subatmospheric pressure conditions with wall sub-cooling of about 16°C and 6°C respectively showed that condensation heat transfer coefficients increased by factors of 2.3 to 3.6 compared to those for un-coated tubes.

b)ROUGH SURFACES:

Single Phase Flow:

The use of surface roughness in turbulent single phase flow is one of the simplest and highly effective techniques; small scale roughness has little effect in laminar flows. It essentially disturbs the viscous laminar sub-layer near the wall to promote higher momentum and heat transport. Surface roughness can be introduced in the form of wire-coiled type inserts or it may be integral to the surface.

Rough surfaces have been employed to enhance heat transfer in single phase flows both inside tubes and outside tubes. Dong et al. (2001) developed a new set of analogy based friction factor and Nusselt number correlations for turbulent flows of water and oil in spirally corrugated tubes. Adopting an empirical approach, combined with a statistical analysis of a fairly large database for heat transfer coefficients and friction factors for various roughness shown above, Ravigururajan and Bergles (1996) proposed correlations for Nusselt number and fanning factor as: By Bergles, (1998) found to increase the heat transfer coefficient and critical heat flux (CHF) in once through boiling of water. Commercially structured rough surfaces in the form of corrugated tubing have extensively employed in refrigerant evaporators. Withers and Habdas (1974) have reported up to 100% increase in the heat transfer coefficient and up to 200% enhancement in critical heat flux (CHF) in bulk boiling, in helically corrugated tubes. Artificial roughness in the form of longitudinal ribs or grooves has been applied in gravity driven, horizontal tube evaporators. Though these types of surfaces promote turbulence, it tends to impede film drainage proposed by Bergles (1998).Cox et al. (1969) found that three-dimensional rough surfaces tend to promote turbulence as well as the liquid spreading thereby increasing the heat transfer coefficient as much as 100%.

Condensing:

Corrugated tubes have been extensively used for enhancement of vapor space condensation. Rough surfaces also improve in-tube forced convective condensation. The overall heat transfer coefficient with forced convection condensation inside and spray film evaporation outside could be improved by using spirally indented and V-grooved tubes. Thomas (1967) attached axial wires around the periphery of vertical smooth tubes, which facilitated better surface tension driven condensate film drainage to produce three to four fold enhancement in heat transfer. He further proposed that square profiled wires were more effective than circular ones of the same roughness height. In steam condensation on horizontal helically corrugated tubes, Mehta and Raja

Rao (1979) and Zimparov et. al. (1991), Webb (1994), Das et al. (2000) have reported about 1.1 to 1.4 times increase in heat transfer coefficient. Dreitser et al. (1988) have reported 1.8 to 2.0 times higher steam condensation heat transfer coefficients on horizontal tubes with transverse grooves.

c) EXTENDED SURFACES:

Extended or finned surfaces are most widely used techniques which include finned tube for shell & tube exchangers, plate fins for compact heat exchanger and finned heat sinks for electronic cooling.

Single-Phase Flow: Enhanced heat transfer from finned surfaces for buoyancy driven natural or free convection has been considered primarily for cooling of electrical and electronic devices and for hot water baseboard room heaters. The use of extended surfaces for cooling electronic devices is not restricted to the natural convection heat transfer regime but also can be used for forced convective heat transfer. By using segmented or interrupted longitudinal fins inside circular tubes, heat transfer can be increased by periodically disrupting and restarting the boundary layer on the finned surface and perturbing the bulk flow field. Plate fin or tube and plate fin type of compact heat exchangers, where the finned surfaces provide a very large surface area density, are used increasingly in many automotive, waste heat recovery, refrigeration and air conditioning , cryogenic, propulsion system and other heat recuperative applications.

A variety of finned surfaces typically used, include offset strip fins, louvered fins, perforated fins and wavy fins.

Fig. 3.1: (Tubes with Circumferential and strip fins on their outer surface)

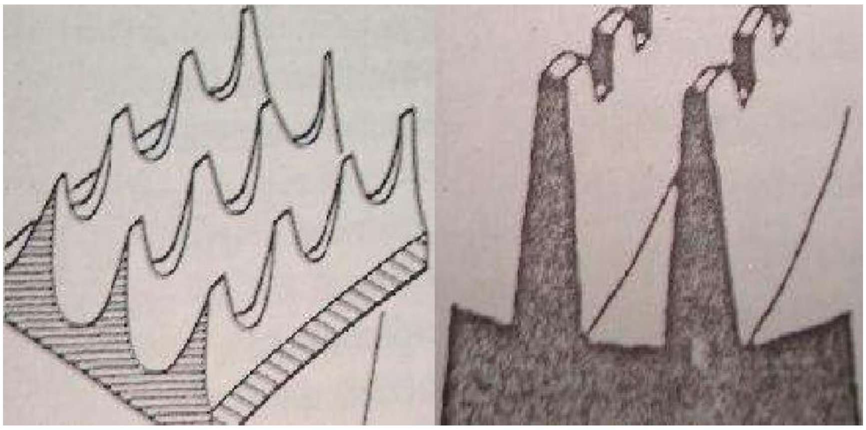
Kelkar and Patankar (1990) considered in-line segmented fins which had half the fin surface area of staggered or continuous fins, were found to perform better with 6% higher Nusselt number and 22% lower friction factor.

Boiling:

Internal finned tubes are widely used in refrigerant evaporators and for flow boiling. In pool-boiling, finned tubes have higher heat transfer coefficients compared with the performance of equivalent smooth tubes. In most heat exchangers for refrigeration and air-conditioning systems, micro finned tubes are extensively used. Bergeles (2000) pointed out that by using fin structures, the heat transfer coefficients can be increased up to 200%.

***Condensing*:**

Extended surfaces that include a variety of large, medium and micro sized fins are used extensively for condensation heat transfer enhancement in power, process, and air conditioning and refrigeration applications. The heat exchangers in these duties involve both horizontal and vertical tube condensers with fins on the inside or outside surfaces of tubes. For integral fin tubes, besides the increased surface area, high heat transfer coefficients are obtained because a relatively thin condensate film tends to be formed near the fin tubes and surface tension forces pull the condensate into the inter fin grooved spaces, there by promoting better drainage and reduction of liquid film resistance.



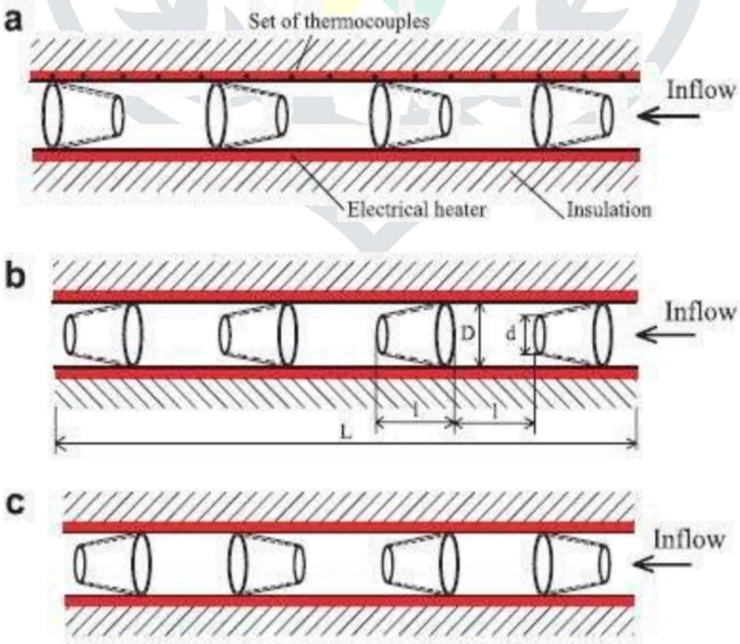
(Notched fins) (Serrated-tip micro fins)

Fig. 3.2: Three-dimensional finned surfaces for enhanced condensation

Chandran and Watson (1976) found that by using circular pin fins, average heat transfer coefficient can be increased to 20% more than those for a smooth tube. Itoh et al. (1997) had shown that micro fins with serrated tips (as shown above) provide 30 to 60% improvements in the heat transfer coefficients over same sized conventional micro fin tubes.

d) DISPLACED ENHANCEMENT DEVICES:

***Single-Phase Flow*:** Several types of inserts which are categorized as displaced enhancement devices include static mixer elements (e.g. Kenics, Sulzer), metallic mesh, discs, wire matrix inserts, rings or balls which tend to displace the fluid from the core of the channel to its heated or cooled wall and vice versa, keeping the heat transfer surface unaltered. Rings and round balls have comparable heat transfer improvements, but the friction factors are exorbitantly high. Most of the devices are effective only in laminar flows, as in turbulent flows, the pressure drop penalties are extremely high as reported by Bergles (1998). The applications of static mixers are generally restricted to chemical processing with heat transfer, where fluid mixing is the primary need. Spiral brush inserts in short channels with turbulent flows and high wall heat flux have been shown by Megerlin et al. (1974) and found out that heat transfer coefficient can be improved as much as 8.5 times that in a smooth tube, but pressure drop was exorbitantly high; which restricted its use in practical applications. P. Promvonge (2007), conducted experiments by inserting several conical rings as turbulators over a test tube. Conical rings with three different diameter ratios of the ring to the diameter (d/D = 0.5, 0.6, 0.7) were introduced in the tests and for each ratio, the rings were placed with three different arrangements (Converging conical Ring-CR, Diverging conical Ring-DR, Converging Diverging conical Ring-CDR). Cold air at ambient air temperature was passed through the tube. He found out that such inserts lead to a higher heat transfer rates than plane tubes and DR yielded better heat transfer than the others. The Nusselt number was found to increase by 197%, 333%, 237% in case of CR, DR and CDR array respectively. It leads to a substantial increase in friction factor.



a:- Diverging Ring
b:- Converging Ring
c:- Converging and Diverging Rings

Fig. 3.3: Conical Ring inserts in circular tubes.

e) SWIRL FLOW DEVICES:

Swirl flow devices generally consist of a variety of tube inserts, geometrically varied flow arrangements and duct geometry modifications that produce flows. These techniques include twisted tape inserts, periodic tangential fluid injection and helically twisted tubes.

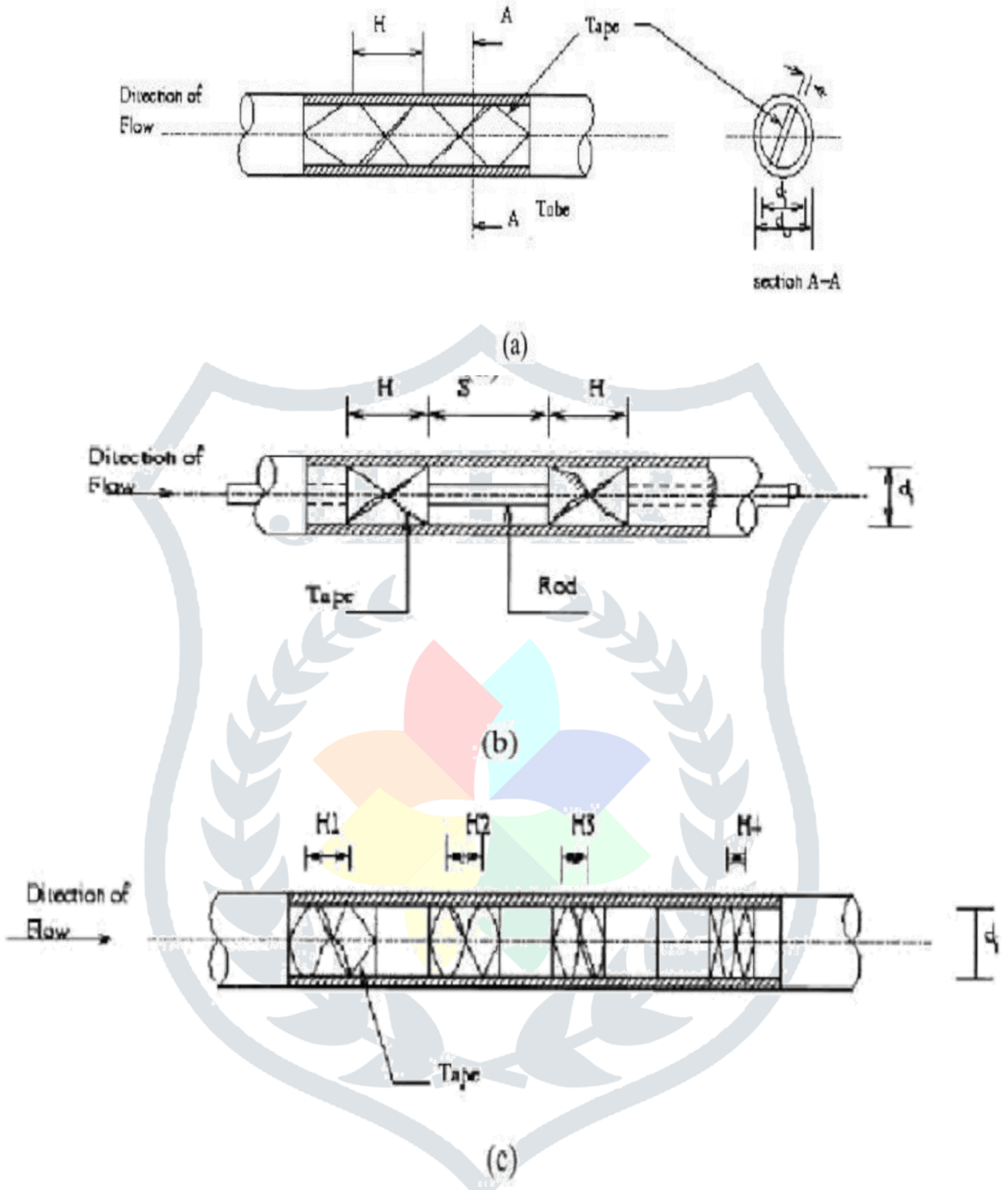


Fig 3.4: Example of (a) full-length twisted tape, (b) regularly spaced twisted tape and (c) smoothly varying (gradually decreasing) pitch full-length twisted tape

***Single-Phase flows*:**

Twisted tape inserts are the most widely used swirl flow device for single-phase flows. These inserts increase the heat transfer coefficient significantly with a relatively small pressure drop penalty as reported by Smithberg and Landis (1964); Lopina and Bergles (1969); Date and Singham (1972); Manglik and Bergles (1992); Manglik and Yera (2002). Twisted tapes can be used in the existing shell and tube heat exchangers to upgrade their heat duties or when employed in a new exchanger for a specified heat duty, significant reduction in size can be achieved. The ease of fitting multiple bundles with tape inserts and their removal makes them useful in fouling situations, where frequent tube-side cleaning may be required. When swirl flow devices are placed inside a circular tube, the flow field gets altered in several ways like an increase in axial velocity and wetted perimeter due to the blockage and partitioning of the flow cross-section, longer effective flow length in the helically twisting partitioned duct and tape's helical curvature induces secondary fluid circulation or swirl. Swirl generation is the most dominant mechanism which effects transverse fluid transport across the tape partitioned duct, thereby promoting greater fluid mixing and higher heat transfer. P.Sivashanmugam and S.Suresh (2006) conducted experimental investigation of heat transfer and friction factor characteristics of circular tube fitted with full length helical screw element of different twist ratio (1.95, 2.93, 3.91, 4.89), and helical screw inserts with spacer length 100, 200, 300 and 400 mm as shown above with uniform heat flux under turbulent flow condition. The friction factor for helical twist insert with spacer length 100 mm was found to be very close to the value of that of full length helical twist for all Reynolds number and decreases by 5% for each 100 mm increment space length indicating that there is no much reduction in pumping power. The increase in Nusselt number from twist ratio 4.89 to 1.95 is nearly 30 to 40% for all Reynolds number for full length helical twist whereas the decrease in friction factor is about 40 to 45% for various spacer lengths. They developed empirical equations for Nusselt number and friction factor:

$Nu = 0.258\,(Re)^{0.254}\,(Pr)\,(Y)^{-0.242}(1+S/Dh)^{-0.042}$

$f = (Re)^{-0.384}(Y)^{-0.852}(1+S/Dh)^{-0.047}$

Boiling:

The heat transfer enhancement due to the tape inserts is reflected in the reduced wall temperature along the tube length in a single phase liquid, sub-cooled boiling, bulk boiling and dispersed film boiling. The primary enhancement mechanism is the tape induced swirl, which tend to increase vapor removal and wetting of the heated surface.

f) ADDITIVES FOR LIQUIDS:

Single-Phase flow: This technique for single-phase liquid flows has focused primarily on drag reducing consequences on the additives. The lowering of frictional losses has the indirect effect of providing heat transfer enhancement when evaluated on a fixed pressure drop or pumping power basis. In the case of soluble polymeric additives in water, where the solution has a shear thinning rheology, the non-Newtonian effects lead to a significant reduction in frictional loss as well as a modest increase in the heat transfer coefficient as reported by Joshi and Bergles (1982), Prusa and Manglk (1995), Hartnett and Cho (1998), Chhabra and Richardson (1999), Manglik and Fang (2002). With polymeric additives that imparts a viscoelastic character to the solution, the heat transfer has been found to be further enhanced in rectangular ducts due to a viscoelasticity driven secondary circulation that is imposed over the bulk flow (Hartnett and Kostic, 1985; Hartnett, 1992; Hartnett and Cho, 1998). Some of the additives used are polystyrene spheres suspension in oil and injection of gas bubbles. By injecting air bubbles at the base of a heated vertical wall, Tamari and Nishikawa (1976) found up to 400% higher free convection heat transfer coefficient in water and ethylene glycol. In a turbulent flow of water, Kenning and Kao (1972) obtained upto 50% increase in heat transfer by injecting nitrogen bubbles.

Boiling: The use of various additives like surfactants, polymers, etc. that lower the surface tension of the solution and binary mixtures of liquid (wetting agents, alcohols) have been found to enhance pool boiling substantially. Nucleate boiling heat transfer coefficient increases up to 20 to 160% in surfactant solutions depending on their concentrations (Tzan and Yang, 1990; Ammerman and You, 1996; Wu et al., 1998; Manglik, 1998; Hetsroni et al., 2000; Wasekar and Manglik, 2002) and 20 to 40% in binary liquid mixtures with wetting agents or alcohols.The improved thermal performance is strongly depended on the type and concentration of the surfactant additive, its chemistry (ionic nature, molecular and chemical composition and structure) and the diffusion kinetics at the dynamic liquid interface. The lowering of the solution's surface tension promotes nucleation of smaller bubbles, with a clustered activation of nucleation sites which depart at much higher frequencies than seen in pure water.

2} COMPOUND ENHANCEMENT:

Compound techniques are slowly emerging area of enhancement that hold promise for practical applications, since heat transfer coefficients can usually be increased above each of the several techniques acting alone. Some examples that have been studied are as follows:

1. Rough tube wall with twisted-tape inserts
2. Rough cylinder with acoustic vibrations
3. Internally finned tube with twisted-tape insert
4. Finned tubes in fluidized beds
5. Externally finned tubes subjected to vibrations
6. Rib-roughened passage being rotated
7. Gas-solid suspension with an electrical field
8. Fluidized bed with pulsations of air
9. Rib-roughened channel with longitudinal vortex regeneration.

One may consider the use of augmentation techniques to satisfy any of the following thermal-hydraulic objectives: (1) to reduce prime surface area, (2) to increase heat transfer capacity, (3) to reduce the approach temperature difference for the process streams, or (4) to reduce pumping power.

IV. EXPERIMENTAL SETUP

The setup of heat exchanger is used validation purpose. The setup was suitable for the water as a working fluid.

The basic idea for experimental is taken from Omkar Shewale's thesis.

Specifications--

Inner pipe ID = 22mm
Inner pipe OD=25mm
Outer pipe ID =53mm
Outer pipe OD=61mm
Heat transfer length= 2.43m

Procedure:

i. Heaters are switched "ON" and fluid is heated to required temperature.

ii. After achieving the desired temperature level of the hot fluid, valve of the cold fluid is opened. The flow of cold fluid is adjusted by using the valve and Rota meter which is calibrated.

iii. Pump is switched "ON" to circulate the hot fluid and flow rate is adjusted by using the ball valve and by-pass valve arrangement provided to the pump assembly also Rota meter which is calibrated.

iv. Inlet and outlet temperatures of hot and cold fluids are recorded as steady state is reached.

v. After completion of first set of readings, the procedure is repeated for the different flow rate of the cold fluid.

SAMPLE CALCULATIONS

A. Pressure Drop and Friction Factor Calculation-

$Ac = \pi/4 \times di^2$

$V = m / (Ac \times \rho_w)$

$\Delta P = (\rho_{ccl4} - \rho_w) \times g \times h$

$f_a = \Delta P \times d_i / (2 \times \rho \times L_p \times V^2)$

$\mu = 0.85$ cP

$Re = 4 \times m / (\pi \times d_i \times \mu)$

$f_o = 0.046 \times Re^{-0.2}$

B. Heat Transfer Coefficient Calculation

For Y=3.127 (Twisted tape)

$m_1 = 0.2806$ kg/s (hot water)

$m_2 = 0.1376$ kg/s (cold water)

	Actual Temp	Corrected Temp	
T1 = 30.6°C		30.1	30.6
T2 = 35.2°C		35.2	35.2
T3 = 49.4°C		49.4	49.4
T4 = 47.1°C		47.0	47.1

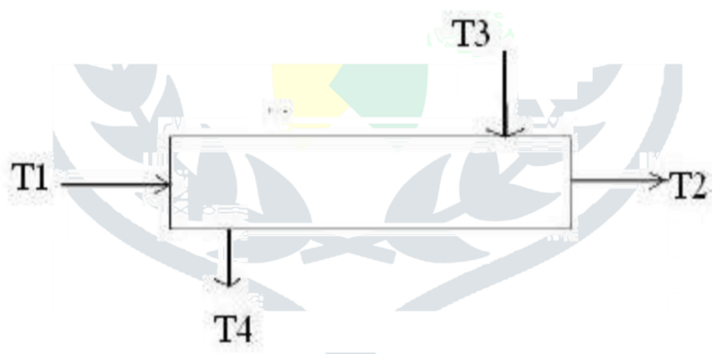


Fig. 4.1 Basic Representation of Counter Current Flow

$Q_1 = m_1 \times C_p \times (T3 - T4) = 0.2806 \times 4187 \times (49.4 - 47.1)$

$= 2702.21$ W

$Q_2 = m_2 \times C_p \times (T2 - T1) = 0.1376 \times 4187 \times (35.2 - 30.6)$

$= 2650.20$ W

$Q_{avg} = (Q_1 + Q_2)/2 = 2676.20$ W

$\%diff = (Q_1 - Q_2) \times 100 / Q_{avg} = 1.94$

$T4 - T1 = 47.1 - 30.6 = 16.5$°C

$T3 - T2 = 49.4 - 35.2 = 14.2$°C

L.M.T.D. = $\{(T4 - T1) - (T3 - T2)\} / \ln \{(T4 - T1) / (T3 - T2)\} = 15.32$°C

$Ai = \pi \times d_i \times L_h = \pi \times 0.022 \times 2.43 = 0.168\ m^2$

$U_i = Q / (A_i \times LMTD) = 2676.20 / (0.168 \times 15.32) = 1040\ W/m^2/°C$

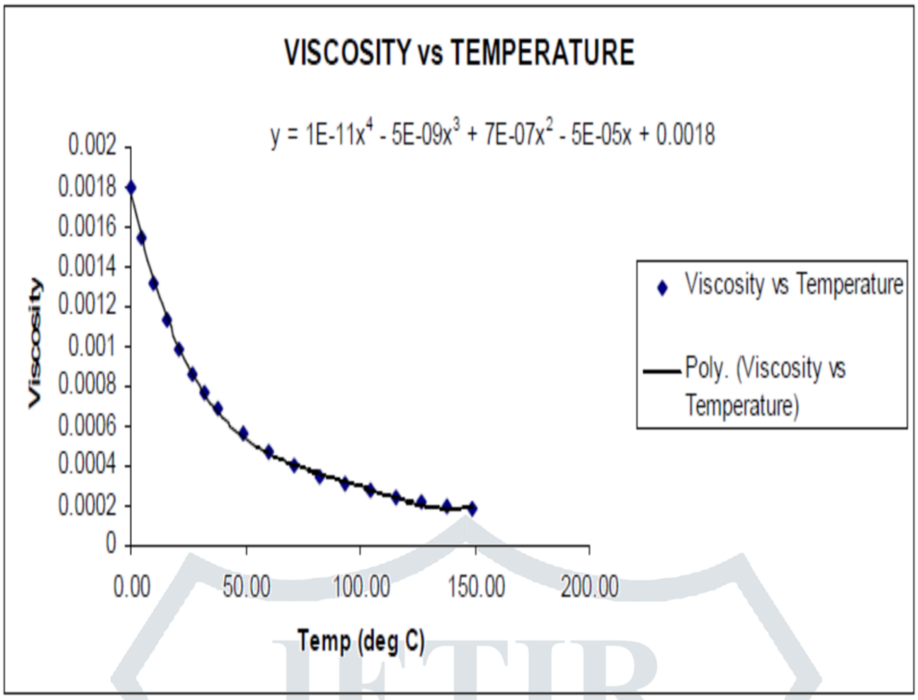


Fig. 4.2: Relationship between Viscosity & Temperature

$\mu = 10^{-11} \times T4 - 5 \times 10^{-09} \times T3 + 7 \times 10^{-07} \times T2 - 5 \times 10^{-05} \times T + 0.0018 = 0.000746$
Where, T = (T1 + T2)/2 = 32.90°C
Re = 4 × m / (π× di × μ) = 10672

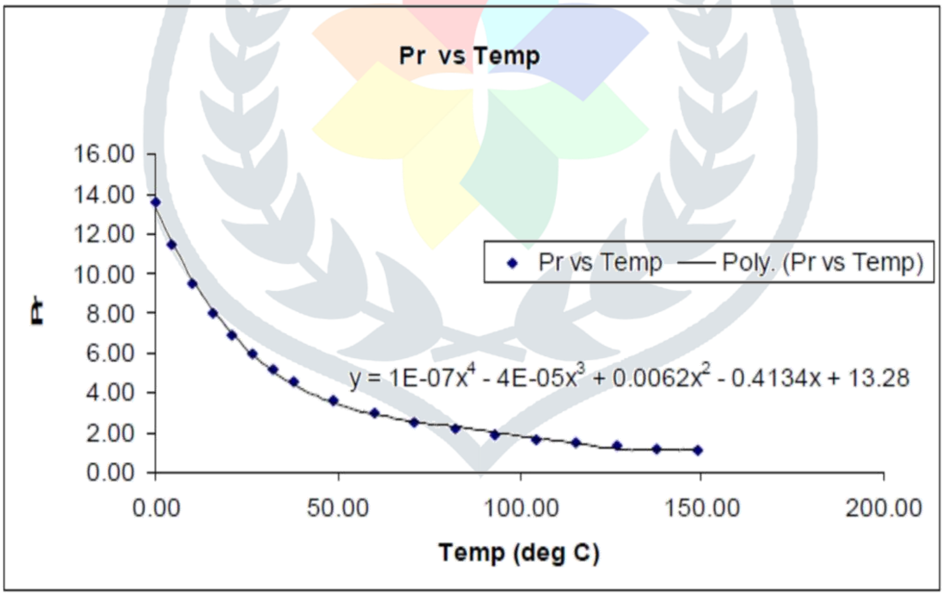


Fig. 4.3: Relationship between Pr & Temperature

$Pr = 10^{-07} \times T4 - 4 \times 10^{-05} \times T3 + 0.0062 \times T2 - 0.4134T + 13.28$
$= 5.0828$

$$\frac{1}{Ui} = \frac{1}{hi} + \frac{di}{do \times ho} + \frac{Xw \times di}{kw \times dL} + Rm$$

This reduces to

$$\frac{1}{Ui} = \frac{1}{(hi)exp} + K = \frac{1}{c(Re)^{\wedge}0.8} + K$$

K is to be found from the Wilson Chart as the intercept on the y-axis

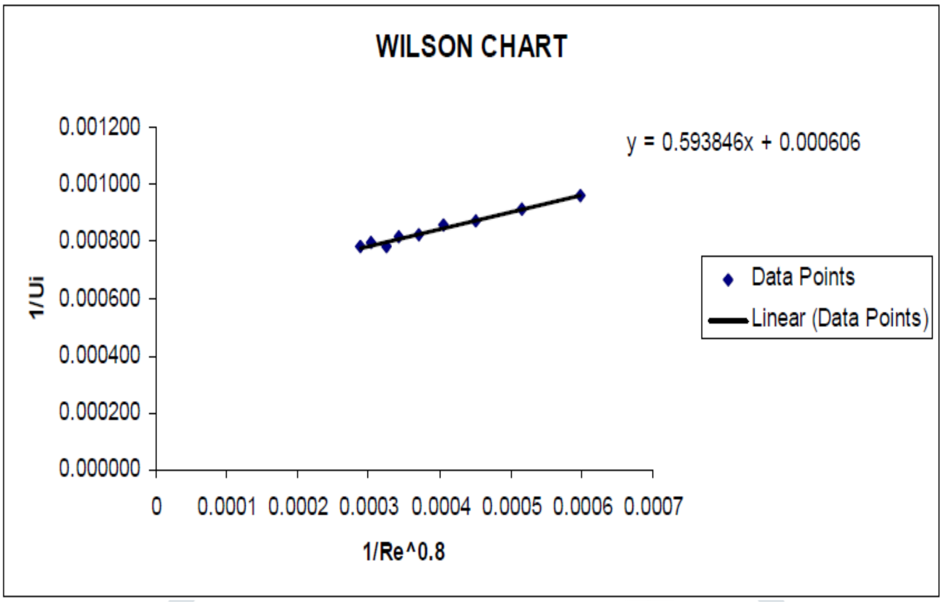


Fig. 4.4: Wilson chart for twisted tape insert

K=0.000606 m °C/W

$1/h_a = (1/U_i) - K = (1/1040) - 0.000606 = 0.0003556$

$h_a = 2811$ W/m^2/°C

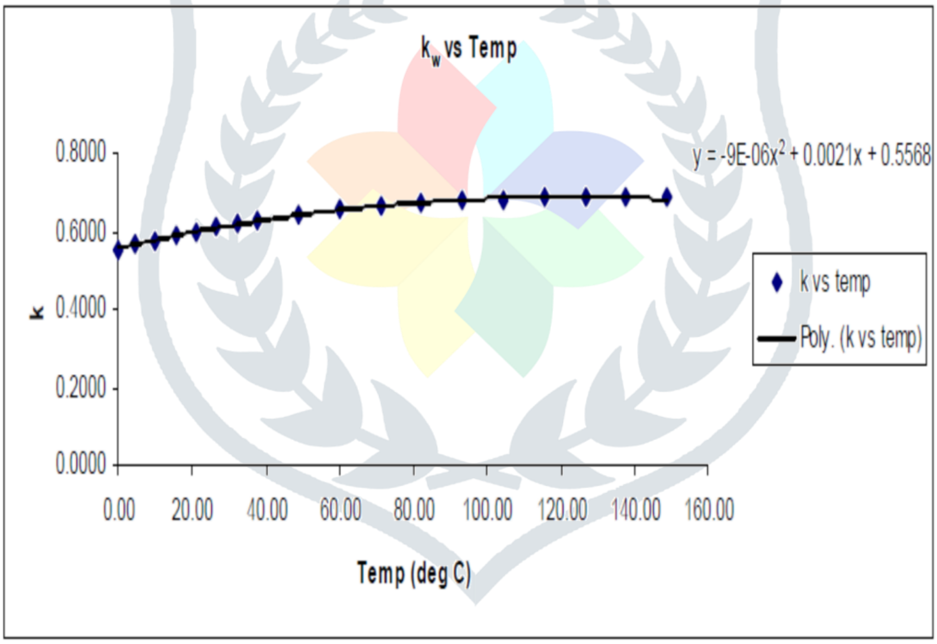


Fig. 4.5: Relationship between k_w and Temperature

k_w = -9 ×10-6 × T2+.0021 T+.5568=0.6161 W/m/°C (at T = 32.9°C)

$h_o = 0.023 \times Re^{0.8} \times Pr^{0.4} \times (k/d_i) = 2061$ W/m^2/°C

$R1 = h_a/ h_o = 2811/2061 = 1.36$

V. RESULTS AND DISCUSSION

From Fig. 5.1 it is observed that hi (expt) and hi (theo) are almost the same in turbulent region. It is also found that from Fig. 5.2, Nu increases with an increase in Re.

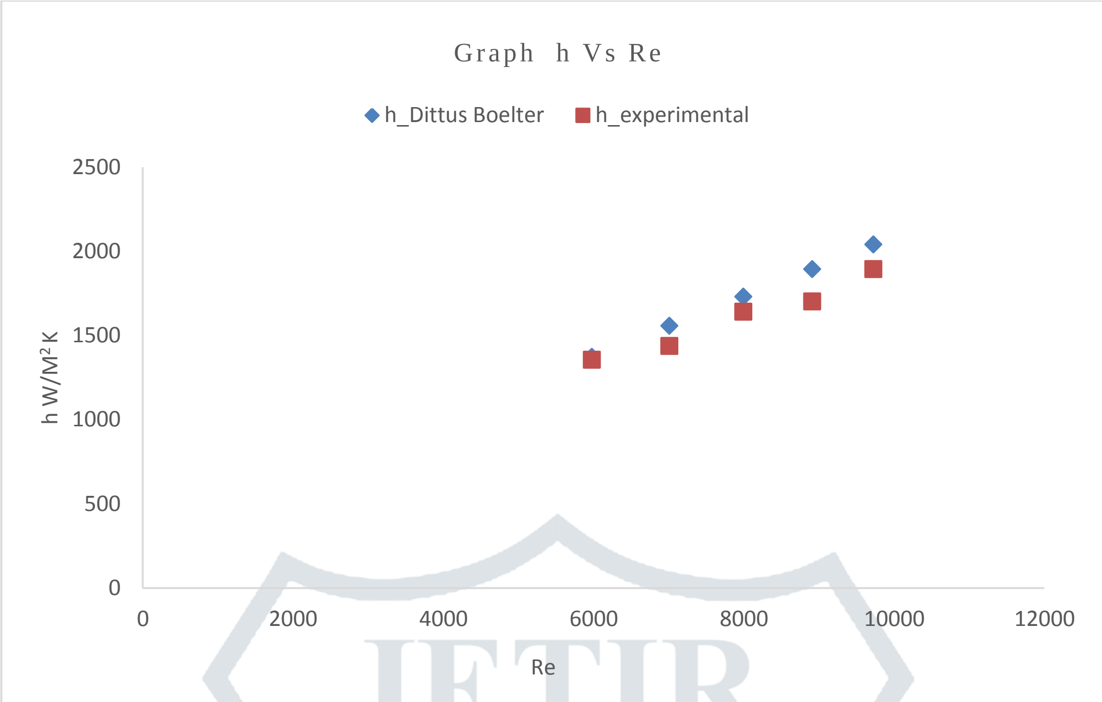


Fig 5.1 Heat Transfer Coefficient

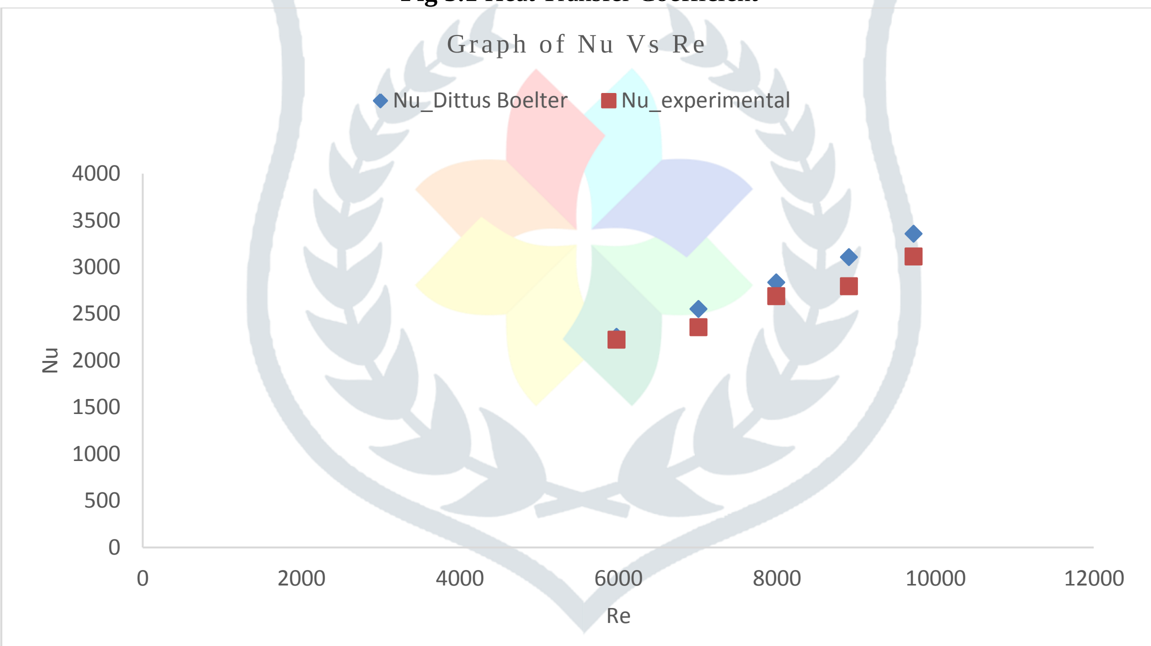


Fig 5.2 Nusselt Number

CONCLUSIONS

1. The experimental values of the Nu and h are compared with that of obtained from standard Dittus-Boelter equation. The graphs shows that, as Re increases the Nu and h increases.
2. Comparing experimental and Dittus-Boelter values of heat transfer coefficient it finds that the difference between these two values goes on increasing with Reynolds number because at higher flow rate less time is available for heat transfer.

VI. NOMENCLATURE:-

A_i -Inside heat transfer surface area, m^2
A_c -Cross sectional area, m^2
C_p -Specific heat of fluid, J/kg K
d_i -Inside diameter of the tube, m

Gz- Graetz number, dimensionless
h -Difference in level of CCl^4 in the manometer, m
h_i -Inside htc, W / m^2 ^{o}C
h_a -(expt) Experimental inside htc, W / m^{2o} C
h_i -(theo) Theoretical inside htc, W / m^2.^{o} C
h_o -Heat transfer coefficient for smooth tube, W / m^2 ^{o} C
h_a -Augmented value of heat transfer coefficient, W / m^2 ^{o} C
H- Linear distance of the tape for 180° rotation, m.
k_w -Thermal conductivity of the tube wall, w/ m °C
L_h -Heat transfer length, m
L.M.T.D- Log mean temperature difference, K
m -Mass flow rate, kg/s
Nu -Nusselt number, dimensionless
Pr -Prandtl number, dimensionless
ΔP -Pressure drop, N/m^2
Q -Heat transfer rate, W
Re -Reynolds number
R1- Performance evaluation criteria (ratio of augmented value of heat transfer coefficient to smooth tube heat transfer coefficient i.e., h_a/h_o), Dimensionless T Temperature in °C
U_i -Overall htc based on the inside surface area, W / m^2. ^{o} C
V -Velocity of water, m/s
W_t- Weight of water taken, kg
Greek letters
ρ- Fluid density in kg/m^3

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