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# Comparison of Effectiveness of Two Different Setups of Double Pipe Heat Exchangers

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***Abstract*— At present cost and performance are the two major factors which plays a vital role in the selection of any augmentation method of heat transfer. Cutting of triangular shaped fins is also very difficult to insert inside smaller annular pipes compared to that of the rectangular shaped fins. Cost spending for all other shaped fins except for rectangular shaped fins are more for cutting and welding purpose with respects their effectiveness. For given dimensions of the double pipe heat exchanger with selected rectangular fins, copper twisted tape is inserted into the inner pipe of the double pipe heat exchanger as first setup of heat exchanger. But second setup of heat exchanger is fabricated with copper twisted tape only without fins at the outside of the inner copper pipe in second heat exchanger. Comparing these two heat exchangers, effectiveness is more for first heat exchanger ie., it can carry out more heat from inner hot fluid to outer cold fluid than that of second heat exchanger which consists only twisted tape without fins. So by using fins in addition to twisted tape effectiveness can be enhanced.**

***Keywords*— Double pipe heat exchangers, Two different setups, Twisted tape as insert, Rectangular fins, Effectiveness**

## I. INTRODUCTION

One of the most commonly used conductive and convective heat transfer device is the concentric tube heat exchanger. These exchangers are made by coaxial tubes located one inside the other. Heat Exchanger is a device used for effecting the process of heat exchange between two working fluids (a hot fluid and a cold fluid). A heat exchanger is a piece of equipment built for efficient heat transfer from one fluid to another. Heat exchangers are useful in many engineering processes like those in refrigerating and air conditioning systems, food processing, power systems, chemical reactors and space or aeronautical applications. Many types of heat exchangers are used in industry, such as shell and tube, double pipe, compact heat exchangers etc., which vary in both application and design. The choice of heat exchanger type directly affects the process performance and also influences the plant size, plant layout, length of pipe runs, and the strength and size of the supporting structures

**Heat exchanger principle** states that heat exchange between two fluids takes place when there exists temperature difference between those two fluids. The greater the temperature difference higher will be the heat transfer rates from cold fluid to hot fluid. This is the principle on which heat exchanger works. Heat exchangers work because heat naturally flows from higher temperature to lower temperatures. Therefore if a hot fluid and a cold fluid are separated by a heat conducting surface heat can be transferred from the hot fluid to the cold fluid.

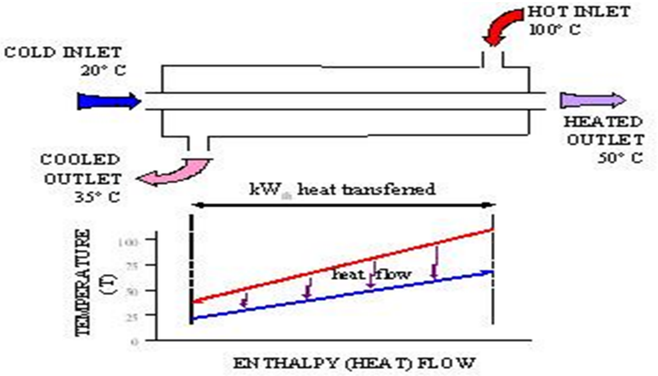


Fig 1: Schematic representation of heat exchanger principle

*Fins:* In the study of heat transfer, a fin is a surface that extends from an object to increase the rate of heat transfer to or from the environment by increasing convection. The amount of conduction, convection, or radiation of an object determines the amount of heat it transfers. Increasing the temperature difference between the object and environment, increasing the convection heat transfer coefficient, or increasing the surface area of the object increases the heat transfer. Sometimes it is not economical or it is not feasible to change the first two options. Adding a fin to an object, however, increases the surface area and can sometimes be an economical solution to heat transfer problems.

## II. LITERATURE SURVEY

P.SIVASANMUGAN AND S.SUNDARAM -The velocity field for the mechanism of flow associated with a tube tested with twisted tape was analysed by Lopina and Bergles. Also they reported a model for the prediction of the heat transfer coefficient and friction factor. Empirical correlations and models are available in many number but there is a lacking in experimental work with respect to performance prediction. By making this as an important note, P.Sivasanmugan and S.Sunadaram made an experimental studies on the improvement in performance of double pipe heat exchanger fitted with turbulence promoter named twisted tape.

S.NAGA SARADA AND A.V. SITA RAMA RAJU-They had worked out to show the results obtained from experimental investigations which involves the augmentation of turbulent flow heat transfer by using varying width twisted tape inserts in a horizontal tube. The working medium they have used is air and correlations are developed which are applicable to full width as well as reduced width twisted tapes for friction factors and Nusselt numbers for fully developed turbulent swirl flow. In these correlations they have used a modified ratio as pitch to width ratio of the tape. They found from the experimental results that as compared to plain tube, heat transfer with twisted tape inserts has increased. It was noticed that the Reynolds number is varied from 6,000 to 13,000. Their experimental results of the Nusselt number and friction factor are correlated in terms of modified twist ratio and Reynolds number. Based on this experimental data, correlations for the friction factor and heat transfer coefficient are proposed. For tape widths of 22 and 10mm, compared to full width twisted tape inserts Nusselt number decreased by a maximum of 8% and 29% respectively

A.DEWAN AND P.MAHANTA -They suggested that passive technique is more advantageous than that of active method to enhance heat transfer rates because it involves usage of inserts in the flow passage. It is easy to fabricate inserts into the heat exchanger. Passive heat transfer technique has an important role if the selection of insert configuration is done properly. In this paper they have done experiment on different types of flows with different types of inserts. For example laminar flow with twisted tape, turbulent flow with twisted tape, wire coil in turbulent flow and also using other inserts such as dimples, baffles, ribs, e.t.c.

PADMAKSHI AGARWAL, ADHIRATH SIKAND AND SHANTHI **V**-The usage of heat exchangers in bioprocess industry changes over a wide range, from the thermal treatment of fluid foods to the making of juices to obtain optimum concentration to the cooling of stirred yoghurt or freezing of liquids. It is also being applied in cryogenic processes, preparation of ethanol and techniques of sterilization. To liquefy gases, such as, liquid carbon, dioxide liquid nitrogen and liquid helium , the cryogenic process is taken place in many industries. Packaged juice manufacturers follow a Flash Pasteurization procedure for the sterilization of juices when need to packed. High temperature treatment principle is used to preserve food items which is known as pasteurization in bio-process industries.

S.MITTAL AND A. RAGHUVANSHI -Many of engineering structures involve bodies with bluff nature that experience wind loads unsteadily which may have a significant influence on their design. Reduction in the unsteady forces acting on the bluff bodies can be effected by vortex control shedding. By using numerical simulation it can be noted that the vortex shedding from the lower surface of the main cylinder in the presence of control cylinder is absolutely similar to that for single cylinder. To study the effect of the control cylinder location in the near wake of the main cylinder, a numerical study has been carried out for flows at low Reynolds numbers.

PROF. ARVIND S. SORATHIYA, MANANKUMAR B. JOSHI, PROF. (DR.) PRAVIN P. RATHOD -At low Reynolds number regime before the previous case transitions to turbulence, the later case with twice the fin pitch in the former case configuration develops flow incapability at ReH=900. Based on his review work, it is to be noted that heat transfer from the fin can be improved by modifying geometry, fin pitches shape, and material and wind velocity. As per available literature surveyed there is a little work available on the wavy fins geometry pertaining to current research area to till date. So there is a scope of delve on wavy fins on cylinder head –block assembly of 4 stroke SI engine in the field of heat transfer study.

PANKAJ N. SHRIRAO, RAJESHKUMAR U.SAMBHE, PRADIP R.BODADE -An experimental study this work studied on the friction factor mean Nusselt number, and thermal augmentation factor characteristics in a circular tube with different types of internal threads of 120 mm pitch under uniform wall heat flux boundary conditions. In the experiments, deliberate readings are taken at Reynolds number in range of 7,050 to 14,500 with air as the test fluid. The experiments were conducted on circular tube with three different types of internal threads viz. acme, buttress and knuckle threads of constant pitch. The friction factor and heat transfer data obtained is compared with the data obtained from a plain circular tube under similar geometric and flow conditions. The variations of heat transfer and pressure loss in the form of Nusselt number (Nu) and friction factor (f) respectively is determined and depicted graphically. It is observed that at all Reynolds number, the Nusselt number and thermal performance increases for a circular tube with buttress threads as compared with a circular tube with acme and knuckle threads. These are all due to increase in intensity and strength of vortices evicted from the buttress threads. An empirical correlation is also formulated consequently to contest with experimental results.

MAZIN ALI A. ALI -The effect of atmospheric turbulence on transmission of wavelengths in free space is discussed in this paper. For spherical and plane waves the Rytov variance is calculated, where Rytov variance of plane waves is greater than spherical waves, on the other hand in spherical waves the values of wavelength 1550 nm less than from the other wavelengths. Scintillation attenuation was calculated depends on Rytov approximation for different wavelengths (1550, 850, 633, 532) nm, scintillation attenuation values of the wavelength 1550 nm less than from the other wavelengths. The wavelength with value 1550nm has a good signal to noise ratio from other wavelengths. The length taken between the receiver and transmitter links was (0-1000) m, the refractive index structure parameter is taken into account at different turbulence ($10^{14}$ high, $10^{15}$ medium, $10^{16}$ low) m for all calculations.

SALILA RANJAN DIXIT, DRTARINICHARANA PANDA -For horizontal rectangular fin array a chimney flow pattern is developed due to density difference. Since orientation and Geometry plays a vital role in natural convection heat transfer, this flow pattern generates a stagnant zone near central bottom region. That area does not donate much towards heat dissipation. This area is isolated from fins and they converted to inverted notched fins. This adapted geometry leads to reduction in material weight, material cost without get reduction in heat transfer rate. This exploration is also extended upon different types of notches and the comparison of their effectiveness. Numerical models are prepaid to research on heat transfer characteristics in inverted notched fins plane fins. Fin height, fin spacing, fin length, heat input, percent of area ejected in the form of inverted notch are the parameters under contemplation. This similar analysis is done numerically but using Fluent in CFD package. It is found that the convective heat transfer coefficient of inverted notch fin array is 25 percent to 35 percent higher as compared with normal fin arrangement. Also we found that the triangular shaped notch results better than trapezoidal and rectangular shaped notches.

A. D. YADAV, V. M. KIRPLANI -Heat transfer enhancement techniques like active, passive or a combination of passive and active methods are generally used in many areas such as bio-process industries, heating and cooling in evaporators, thermal power plants, air-conditioning devices, refrigerators, radiators for space vehicles, automobiles, etc. In design of compact heat exchangers, passive techniques of heat transfer augmentation can play an important role if a proper passive insert configuration can be selected according to the heat exchanger working condition (both flow and heat transfer conditions). In passive heat transfer techniques inserts are used in the flow passage to improve the heat transfer rate and are beneficial when compared with active heat transfer techniques, because the production of inserts is simple and these techniques can be easily employed in an available heat exchanger. In the past ten years, several studies on the passive techniques of heat transfer augmentation have been reported. Twisted tapes, wire coils, ribs, fins, dimples, etc., are the most generally used passive heat transfer enhancement tools. In this paper accent is given to manipulate with different shapes of ribs, and there arrangement because, regarding to recent studies, they are called to be economic heat transfer augmentation tools. The present outlook is ordered in four different sections: circular ribs staggered at 45º; circular ribs staggered at 90º, triangular ribs staggered at 45ºand triangular ribs staggered at 90º.

SREEJITH K, BASIL VARGHESE- A plate heat exchanger is one type of heat transfer device that utilizes metal plates to transport heat between two fluids. This has a major advantage over a conventional heat exchanger in that the fluids are exposed to a much larger surface area because the fluids spread out over the plates. This can facilitates the exchange of heat, and hugely increases the speed of the temperature variation. The PHE (plate heat exchanger) is a expert design perfectly suited for transferring heat between medium- and low-pressure fluids. Brazed, Welded, and semi-welded heat exchangers are used for heat transfer between high-pressure fluids or where a more compact product is required. At alternating chambers the hot fluid flows in one direction where as the cold fluid flows in real counter-current flow in the other alternating chambers. The heat transfer surface area consists of a number of thin ridged plates compressed out of a high grade metal. The compressed pattern on each plate surface induces turbulence and minimizes stagnant areas and fouling. Unlike tube and shell heat exchangers, which can be custom-built to encounter almost any capacity and operating conditions, the plates for plate and frame heat exchangers are mass-produced using expensive dies and presses. In this paper they designed the PHE for the required operating conditions. In the design it was calculated the PHE’s overall heat transfer coefficient . The number of plates required and the heat transfer rate for the PHE were also calculated. Optimization of cost for the fabricated PHE was carried out and it has been observed that there is a substantial drop in the heat exchanger cost.

PROF. ALPESH MEHTA, DINESH K TANTIA- Heat exchanger using nano fluid is a device in which the heat transfer takes place by using nano fluid. In this the working fluid is nano fluid. This paper reviews the research work on heat exchanger with nano fluid as working medium. In this paper they are using compact heat exchanger as heat transferring device while $Al_2O_3$ as a nano fluid. Nano fluid's effect on compact heat exchanger is estimated by using ε –NTU method on turbo-charged diesel engine Comparative study on nano fluid $Al_2O_3^+$ and water as coolant is carried out.

By suspending nano particles in the fluid like ethylene glycol, water & oil, fluorocarbons etc., nano fluid is produced

## III. EXPERIMENTAL PROCEDURE

Hot water is employed as hot fluid, which is heated in an electric geyser and it flows through the inner tube. Cold water is employed as cold fluid which flows through the annulus that is, to the outer side of the inner copper tube. The hot water flows always in one direction and the cold water can be made to flow as required in both directions, that is parallel flow and counter flow.

One end of the inner copper tube is provided a coupling which is connected to a T-joint by means of a stud. Thermometer is placed in this T-joint for measuring the temperature of the fluid flowing inside the tube at that position. Then, it is connected to an adjusting valve, through which flow rate can be adjusted as required, by means of another stud which is connected to an inlet pipe through which water is allowed into the heat exchanger. Other end of this inner copper tube is provided with a coupling which is then connected with a T-joint by means of a stud and then to an L-bend. This L-bend allows water to get stagnated in the tube. This is then connected to a pipe through which water leaves from the heat exchanger.

Thus, water entering into the inlet pipe is initially made to adjust its flow rate by means of the adjusting valve and then its inlet temperature is measured by means of the thermometer provided in the T-joint and then is traversed through the heat exchanger and its outlet temperature is measured by means of a thermometer provided at the other end in the T-joint and then flows out through the exit pipe provided. The outer mild steel tube of the concentric tube exchanger is attached with various components as shown in figure. At the inlet end, a coupling is provided which is connected to an L-bend by means of a stud and this in turn is connected to another L-bend by means of another stud. Then another L-bend is provided to this arrangement by means of a stud and then connected with a T-joint thus making provision for inserting thermometer which enables to measure the temperature. An adjustable valve is provided to this T-joint by means of a stud, the other side of which is connected to a pipe that allows the flow of water into heat exchanger.

Fig 2: Experimental unit

The outlet end is provided with a coupling and then with an L-bend to which T-joint is attached, into which is inserted and then connected to a pipe through which water leaves from the heat exchanger.

Thus water entering into the outer tube is initially made to adjust its flow rate by means of an adjusting valve and then this inlet temperature is measured by means of an adjusting valve provided in the T-joint. Then this water is traversed through the heat exchanger and then its outlet temperature is measured at the other end by means of another thermometer provided in the T-joint and then water flows out through the exit pipe.

Thus, when the hot water and the cold water get traversed through the heat exchanger the heat exchanger, heat exchange occurs between two fluids. The hot water flows always in one direction and cold water can be admitted at one of the ends enabling the heat exchanger to run as a parallel flow apparatus. And flow rates are adjusted in both the cases by means of valve operations.

The experiments are conducted by keeping the identical flow rates while running the units as a parallel flow exchanger and counter flow exchanger. Also, experiments are conducted by keeping the identical flow rates for the two models. The temperatures are measured by thermometers and flow rates by a graduated measuring flask and stop watch. The readings are recorded when steady state is reached. Also, the outer tube is provided with adequate thread insulation to minimize the heat losses.

*SPECIFICATIONS:*

| Particulars | Inner dia (mm) | Outer dia (mm) | Length (cm) |
|---|---|---|---|
| Cu pipe | $d_1$=25 | $d_2$=28 | L=180 |
| MS pipe | $D_1$=50 | $D_2$=56 | l=140 |

Table 1: Specifications of heat exchanger pipes

| Particulars | Length (mm) | Width (mm) | Thickness (cm) |
|---|---|---|---|
| Rectangular fins | 25 | 28 | 180 |
| Twisted tape | 50 | 56 | 140 |

Table 2: Specifications of Inserts

Number of Rectangular fins attached on each side= 7
So total number of rectangular fins on both the sides = 14

### *A. Model I: Heat Exchanger with Rectangular Fins and Twisted Tape As Inserts*

Orifices or other temporary restrictions to flow often induce turbulence. Our approach, therefore, was periodically to restrict the flow with twisted tape rectangular fins. A schematic of the heat exchanger with rectangular fins and twisted tape is shown in Fig. The gap between fins and outer pipe forces all the fluid to pass through a narrow opening near the surface of the outer tube. Because this opening is much smaller than the rest of the annulus, velocity is high through the opening.

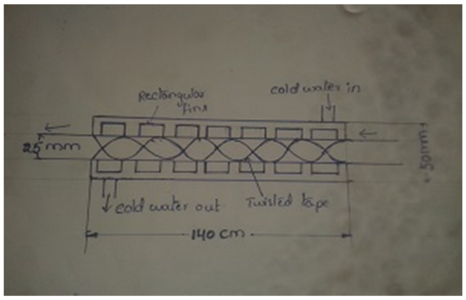


Fig 3: Experimental setup-1 with both twisted tape and fins

Also, the rapid expansion on the opposite side of the opening triggers the transition from laminar to turbulent flow. The outside edges of the fins are sealed to the copper pipe to force the fluid through the narrow opening near the mild steel tube. The resulting high liquid velocity and flow turbulence tend to increase the heat transfer coefficient.

The problem is then how to install and support the rectangular fins outside the copper pipe, twisted tape inside the copper pipe. The method that is chosen to insert the twisted tape is that all the 7 fins with 3mm thickness and 15cm length were welded at a distance of 4mm between each of them. The specified twisted tape is inserted into the copper pipe by manual force. This assembly is put inside the larger mild steel pipe where cold water flows. This assembly was fitted inside the annular pipe, resulting in the flow channel shown in cross-section in the previous figure.

### *B. Model II: Heat Exchanger with Only Twisted Tape As Insert*

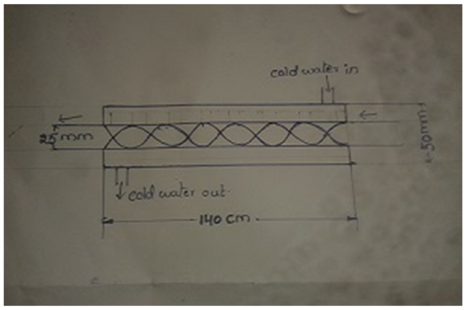


Fig 4: Experimental setup-2 with only twisted tape

The thin twisted helical tape was made of a copper sheet (2 mm thick). The tape was twisted eight times. There was a small clearance between the twisted tape and the walls of the copper tube to allow easy insertion of the tape. The blockage caused by the finite tape thickness increases the average velocity. Heat transfer enhancement may occur for two reasons.
Firstly, the tape reduces the hydraulic diameter of the inner copper tube; this tends to increase the heat transfer coefficient, even for zero tape twist. Secondly, the twist of the tape causes a tangential velocity component. This causes the speed of the flow to increase, particularly near the wall. The enhancement in the heat transfer is a result of the mixing by the secondary flow and the increased shear stress at the wall.

## IV. RESULTS AND DISCUSSIONS

In this experiment, comparision of the effectiveness and other non-dimensional numbers between two different setups of double pipe heat exchangers takes place. readings of the experiment are listed below.

### *A. Double Pipe Heat Exchanger Setup- 1*

The readings from heat exchanger setup- 1 are as follows:

| S.No | Cold water | | | Hot water | | |
|---|---|---|---|---|---|---|
| | Mass flow rate(kg/s) | $T_{hi}$(°C) | $T_{ho}$(°C) | Mass flow rate(kg/s) | $T_{ci}$(°C) | $T_{co}$(°C) |
| 1 | 0.3 | 59 | 49 | 0.2 | 38 | 46 |
| 2 | 0.3 | 58 | 48 | 0.2 | 37 | 45 |
| 3 | 0.3 | 59 | 49 | 0.2 | 38 | 46 |
| 4 | 0.3 | 57 | 48 | 0.2 | 38 | 45 |
| Average | 0.3 | 58.25 | 48.5 | 0.2 | 37.75 | 45.5 |

Table 3: Readings of setup-1 heat exchanger with parallel flow

| S.No | Cold water | | | Hot water | | |
|---|---|---|---|---|---|---|
| | Mass flow rate(kg/s) | $T_{hi}$(°C) | $T_{ho}$(°C) | Mass flow rate(kg/s) | $T_{ci}$(°C) | $T_{co}$(°C) |
| 1 | 0.3 | 65 | 51 | 0.2 | 37 | 46 |
| 2 | 0.3 | 63 | 49 | 0.2 | 36 | 44 |
| 3 | 0.3 | 64 | 50 | 0.2 | 36 | 45 |
| 4 | 0.3 | 65 | 50 | 0.2 | 37 | 46 |
| Average | 0.3 | 64.3 | 50 | 0.2 | 36.5 | 76.75 |

Table 4: Readings of setup-1 heat exchanger with counter flow

### *B. Double Pipe Heat Exchanger Setup- 2*

*The readings from heat exchanger setup- 2 are as follows:*

| S.No | Cold water | | | Hot water | | |
|---|---|---|---|---|---|---|
| | Mass flow rate(kg/s) | $T_{hi}$(°C) | $T_{ho}$(°C) | Mass flow rate(kg/s) | $T_{ci}$(°C) | $T_{co}$(°C) |
| 1 | 0.3 | 48 | 44.5 | 0.2 | 37 | 42 |
| 2 | 0.3 | 49 | 44.5 | 0.2 | 37 | 42 |
| 3 | 0.3 | 49 | 45 | 0.2 | 37 | 43 |
| 4 | 0.3 | 47 | 44 | 0.2 | 38 | 42 |
| Average | 0.3 | 48.25 | 44.5 | 0.2 | 37.25 | 42.25 |

Table 5: Readings of setup-2 heat exchanger with parallel flow

| S.No | Cold water | | | Hot water | | |
|---|---|---|---|---|---|---|
| | Mass flow rate(kg/s) | $T_{hi}$(°C) | $T_{ho}$ (°C) | Mass flow rate(kg/s) | $T_{ci}$(°C) | $T_{co}$(°C) |
| 1 | 0.3 | 59 | 50 | 0.2 | 39 | 49 |
| 2 | 0.3 | 57 | 49 | 0.2 | 38 | 48 |
| 3 | 0.3 | 58 | 50 | 0.2 | 38 | 49 |
| 4 | 0.3 | 57 | 49 | 0.2 | 38 | 48 |
| Average | 0.3 | 57.75 | 49.5 | 0.2 | 38.25 | 48.5 |

Table 6: Readings of setup-2 heat exchanger with counter flow

### *C. Sample Calculations*

We know that mass flow rate of working hot fluid is given by the formula:

$$\dot{m} = \rho A V_h = \rho \frac{\pi d^2}{4} V_h$$

$$0.03 = 986*(\frac{\pi}{4})*(\frac{8}{1000})^2 \ V_h$$

$$V_h = 0.6 \text{ m/s}$$

Where ρ is the density of hot fluid at mean temperature, A is the cross sectional area, Mass flow rate of cold water through annulus is given by

$$\dot{m} = \rho A V_c = \rho \frac{\pi d^2}{4} V_c$$

$$0.02 = \frac{990*\pi*8^2}{1000*4} * V_c$$

$$V_c = 0.4 \text{m/s}$$

Where 'd' is the diameter of water discharge through hose pipe for both hot water and cold water.

Annulus characteristic length $L_c = D_1 - d_2 = \frac{50-28}{1000} = 0.022\text{m}$

### *D. Heat Exchanger 1 – Parallel Flow*

$T_{hi}$=59 $T_{ci}$= 39 $T_{ho}$= 50 $T_{co}$=49

$T_{hi}$, $T_{ho}$ are inlet and outlet temperatures of hot water respectively and $T_{ci}$, $T_{co}$ are inlet and outlet temperatures of cold water respectively

Substitute the above temperature values in the given below equation (1)

Logarithmic mean temperature difference (LMTD):

$$\text{LMTD} = \frac{(T_{hi}-T_{ci})-(T_{ho}-T_{co})}{log_e[(T_{hi}-T_{ci})/(T_{ho}-T_{co})]} = \frac{(59-38)-(49-46)}{log_e[(59-38)/(49-46)]} = 9.25 \quad \longrightarrow \quad (1)$$

$$\text{Effectiveness } (\epsilon) = \frac{\text{actual heat transfer}}{maximum \text{ possible heat transfer}} = \frac{C_h (T_{hi}-T_{ho})}{C_{min}(T_{hi}-T_{ci})} = \frac{125.22(59-49)}{83.48(59-38)} = 0.71$$

### *E. Hot Water Side*

Mean temperature for inner fluid flow is given by

$$T_m = \frac{(T_{hi}+T_{ho})}{2} = \frac{58.25+48.5}{2} = 53.4°C$$

Properties of hot fluid at mean temperature are taken below from thermal engineering data hand book as:

ρ – 986Kg/m$^3$  $\mu$ – 5.13*10$^{-4}$Kg/ms  k – 0.641w/m℃  C$_p$ – 4179 J/kg°C  Pr$_1$ – 3.3

$\mu$ is dynamic viscosity, Pr is prandtl number, k is thermal conductivity of respective fluid at corresponding mean temperatures, C$_p$ is specific heat at constant pressure.
Reynolds number is given by:

$$\mathrm{Re}_1 = \frac{d_1 V}{\nu} = \frac{\rho d_1 V}{\mu} = \frac{986*25*0.6}{5.13*10^{-4}*1000} = 28830$$

$$(\mathrm{Re}_1)^{0.8} = (28830)^{0.8} = 3697$$

$$(\ \mathrm{Pr}_1\ )^{0.3} = (3.3)^{0.3} = 1.431$$

Nusselt number is given by:

$$\mathrm{Nu}_1 = \frac{h_1 d_1}{K} = 0.023 * (\mathrm{Re}_1)^{0.8} * \mathrm{Pr}^{0.3}$$

$$\frac{h_1 * 25}{1000*0.649} = 0.023 * 3697 * 1.431$$

$$h_1 = 3159 \text{ w/m}^2\text{-k}$$

$$\mathrm{Nu}_1 = \frac{3159*25}{0.641*1000} = 122$$

*F. Cold Water Side*

$$T_m = \frac{(T_{ci\ avg} + T_{co\ avg})}{2} = \frac{37.75+45.5}{2} = 42℃$$

Properties of cold water at T$_m$ = 42℃

ρ – 990Kg/m$^3$  $\mu$ – 6.61*10$^{-4}$Kg/ms  k – 0.637w/m℃  C$_p$ – 4174 J/kg°C  Pr$_2$ – 4.04

$$\mathrm{Re}_2 = \frac{L_c V}{\nu} = \frac{\rho L_c V}{\mu}$$

$$\mathrm{Re}_2 = \frac{990*0.022*0.4}{6.16*10^{-4}} = 14143$$

Where $\nu$ is the kinematic viscosity.

$$(\mathrm{Re}_2)^{0.8} = (14143)^{0.8} = 2091$$

$$(\mathrm{Pr}_2)^{0.4} = (4.04)^{0.4} = 1.75$$

$$\mathrm{Nu}_2 = \frac{h_2 L_c}{K} = 0.023 * (\mathrm{Re}\ _2)^{0.8} * (\mathrm{Pr}_2)^{0.4}$$

$$\frac{h_2 * 22}{1000*0.628} = 0.023*2403*1.84$$

$$h_2 = 2903 \text{ w/m}^2\text{-k}$$

$$\mathrm{Nu}_2 = \frac{2903*22}{1000*0.637} = 84$$

Overall heat transfer coefficient ‘U’ is given by:

$$\frac{1}{U} = \left(\frac{1}{h_1} * \frac{d_2}{d_1}\right) + \frac{1}{h_2} + \left(\frac{d_2 - d_1}{d_2 + d_1} * \frac{d_2}{k}\right)$$

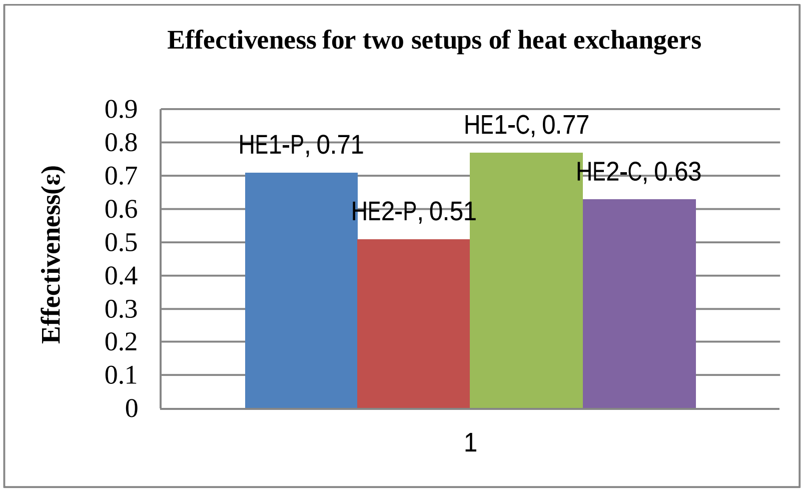
$$\frac{1}{U} = \frac{28}{3159*25} + \frac{1}{2437} + \frac{3*28}{1000*386}$$

$$U = 1300 \text{w/m}^2\text{-k}$$

Similarly all parameters are calculated for counter flow of setup-1 as well as for parallel and counter flow of setup-2 and tabulated in the given below table

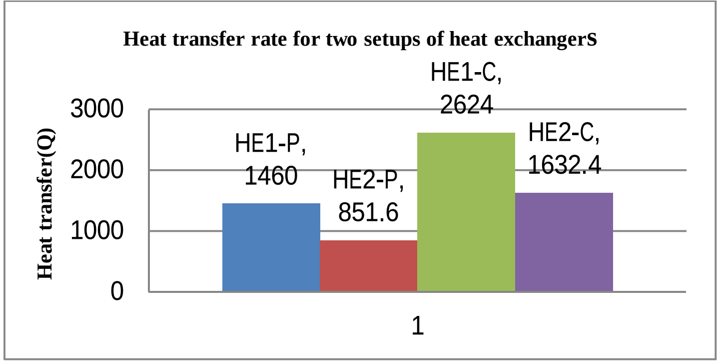
| | Effectivess (ε) | Overall heat transfer coefficient'U' (W/m$^2$-k) | Heat transfer'Q' (W) | Reynolds number (Re$_1$) | Nusselt number (Nu$_1$) | LMTD (℃) |
|---|---|---|---|---|---|---|
| HE1-P | 0.713 | 1300 | 1460 | 28830 | 122 | 9.11 |
| HE1-C | 0.77 | 1331 | 2624 | 31306 | 127 | 16 |
| HE2-P | 0.51 | 1280 | 852 | 26397 | 116.5 | 5.4 |
| HE2-C | 0.63 | 1299 | 1632 | 28801 | 121.5 | 10.2 |

Table 7: Calculated values for parallel and counter flows of HE-1 and HE-2



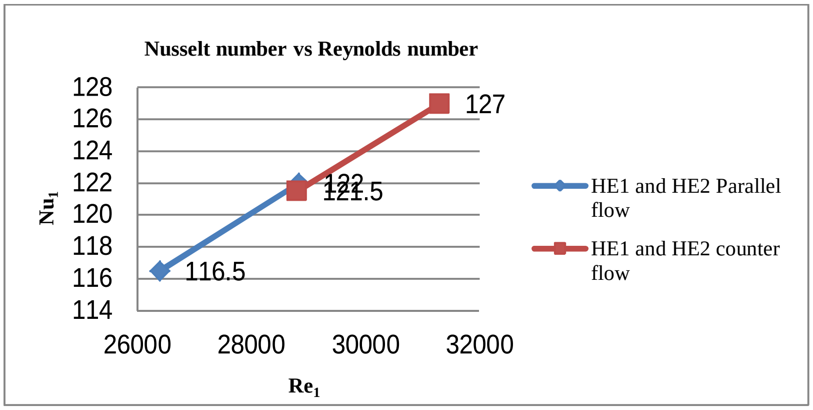
Graph 1: Variation of effectiveness for HE-1 and HE-2

From the graph it is noted that effectiveness is greater for counter flow than that of parallel flow for both setups of heat exchangers. But compared to one particular flow ie., either parallel flow or counter flow in both the setups, effectiveness is higher for first setup of heat exchanger than second setup of heat exchanger. Effectiveness is 20% higher for setup-1 than that of setup-2 with parallel flow and 14% higher for setup-1 than that of setup-2 with counter flow.



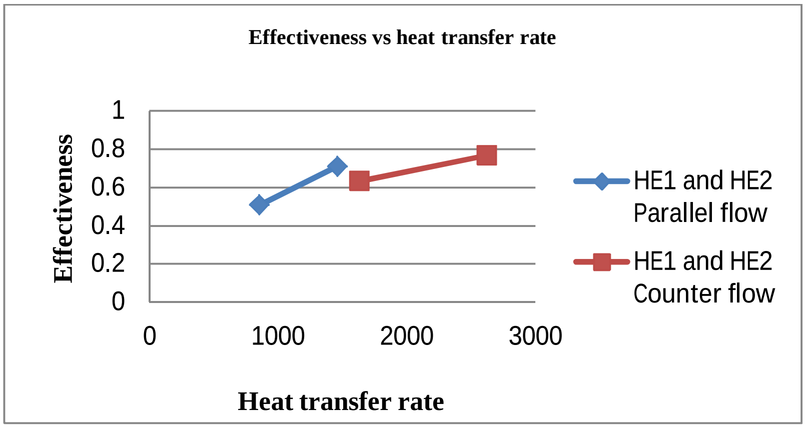
Graph 2: Variation of Heat transfer for HE-1 and HE-2

The temperature difference for inlet working fluids is higher in first setup of heat exchanger than that of second setup. So in parallel flow first setup has greater LMTD value. Similarly the difference in temperature of the inlet hot fluid and outlet cold fluid temperature difference with outlet hot fluid and inlet cold fluid temperature difference is higher for first setup heat exchanger than that of second. Hence it is concluded from the graph that heat transfer rate (Q) which is directly proportional to logarithmic mean temperature difference is also greater for first setup of heat exchanger than that of second heat exchanger.



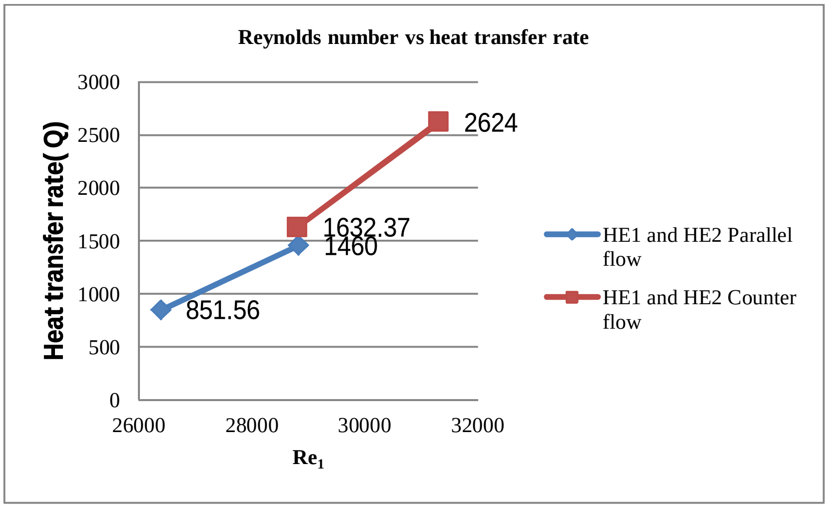
Graph 3: Variation of Reynolds number with Nusselt number for HE-1 and HE-2

Graph represents that turbulence generated inside the copper pipe is higher for first setup of heat exchanger when compared to second setup of heat exchanger regarding to corresponding flows. This is because the temperature gradients at the inner surface of the copper pipe increases due to the presence of fins at the outside of the copper pipe and twisted tape as insert at the inner side of the copper pipe. As a result of this, nusselt number increases linearly with increase in Reynolds number. Absence of fins in second heat exchanger results in less occurrence of temperature gradient and then leads to less turbulence.



Graph 4: Variation of Reynolds number with Nusselt number for HE-1 and HE-2

Logarithmic mean temperature difference is also varies linearly with effectiveness for parallel flows of both the setup of heat exchangers. Also varies linearly with the effectiveness for counter flows of both the heat exchangers. As we know that heat transfer rate (Q) is directly proportional to log mean temperature difference, it also varies linearly with the effectiveness for the corresponding flows in two heat exchangers. It is even proved from the above graph.



Graph 2: Variation of Reynolds number with Nusselt number for HE-1 and HE-2

From the above graph it is very clear that with increase in Reynolds number, heat transfer rate also get increased. But this variation is heavy in first setup of heat exchanger than that of second. This is because turbulence generate rapid collisions between fluid molecules which increases pressure and leads to increase in temperature.

## V. CONCLUSIONS

In this experiment Effectiveness is higher for setup-1 heat exchanger than setup-2 heat exchanger when compared to corresponding parallel and counter flows. Nearly 20% increased in parallel flow and 14% increase in counter flow.

Heat transfer rate , LMTD, and Turbulence also increases for setup-1 heat exchanger than setup-2 heat exchanger which is due to presence of rectangular fins and twisted tape in setup-1 where as only twisted tape and with no fins in setup-2.

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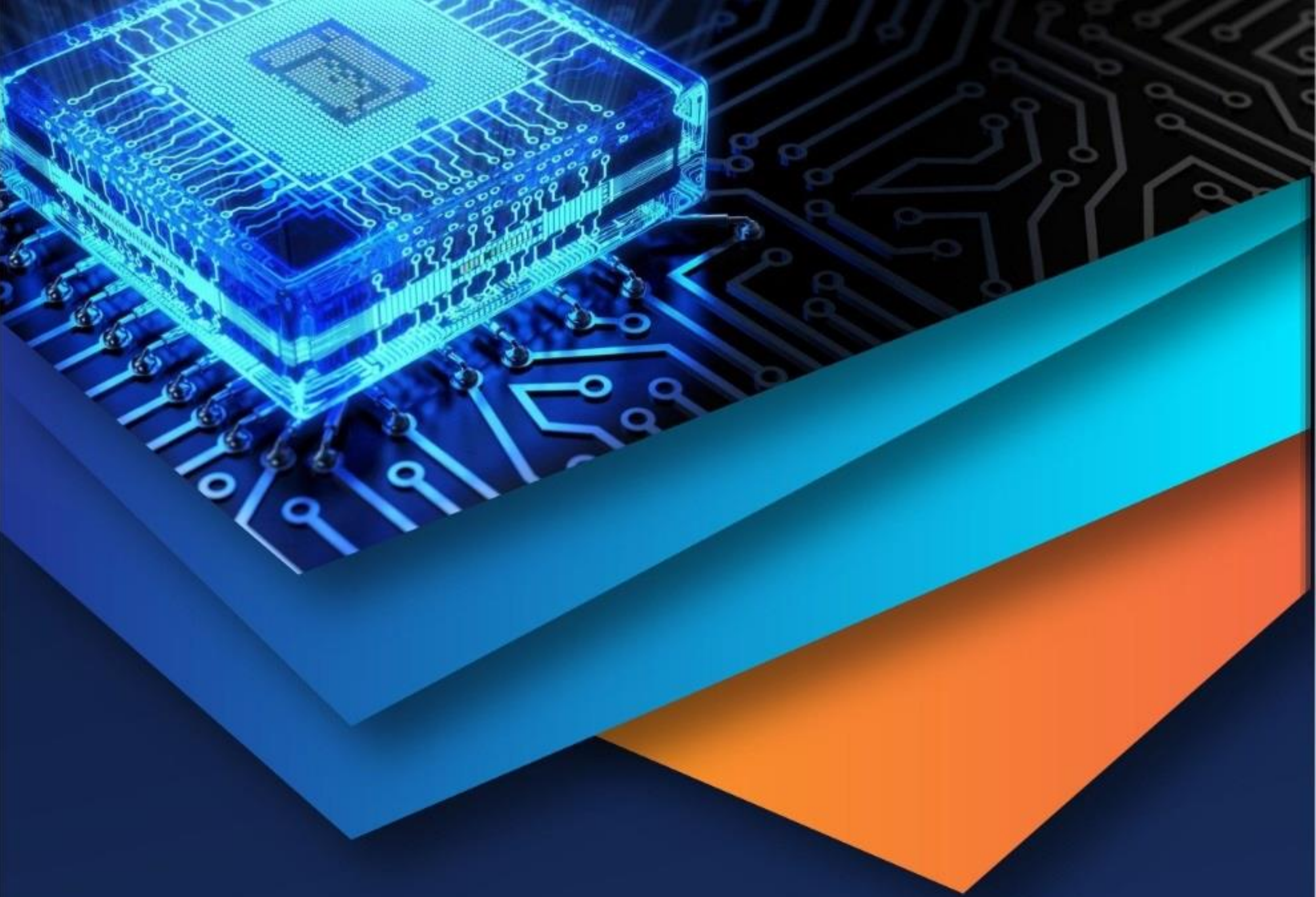

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