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Experimental Investigation of the Effect of Integrated Fins on Heat Transfer Rate of Double Pipe Heat Exchanger

Praveenkumara B M[*], B Sadashive Gowda[a] and Naveen H V[b]

Dept. of Mechanical Engineering, Vidyavardhaka College of Engineering Mysuru 570002, India.
E-mail: [*]praveenkumarabm@vvce.ac.in, [a]sadshivegowdab@vvce.ac.in, [b]naveenhv200030@gmail.com

Abstract

In this paper, the effects of integrated fins on the thermal performance of the concentric tube heat exchanger with a variable flow rate of hot and cold water are discussed. The inner pipe is made of copper pipe of length = 800 mm, OD=30 mm, ID=26 mm, t=2 mm. the integrated fines of the geometry 1.25 mm and 1.5 mm pitch length, 1 mm of the depth of cut are considered in the study. The thermal performance of the heat exchanger with a flow rate of hot and cold fluids ranges from 0.014 kg s^{-1} to 0.07 kg s^{-1} and the effects of parallel and counter flow direction are also discussed in detail. The experimental result confirms that the thermal performance of the concentric tube heat exchanger is more in the case of the tube with integrated fins and counter flow direction with the highest flow rate of water in the pipe and annulus of the heat exchanger.

Keywords: *Mass flow rate, Coefficient of heat transfer, Heat transfer rate, LMTD, Counterflow.*

1.0 Introduction

"Heat exchangers are devices which are used to exchange heat energy from hot liquid to cold liquid with the highest rate, low investment and maintenance costs". Utilizing the energy more effectively with minimum investment in the heat exchanger is one of the major challenges faced by an engineer. Heat transfer enhancement techniques are frequently employed in thermal power plants, refrigerators, air conditioners, vehicles, and other technical applications. Over the past few years, various heat transfer augmentation practices are employed in engineering applications [1-14]. Heat transfer augmentation (enhancement) techniques are classified as: (i) Active techniques, (ii) Passive techniques, and (iii) Compound techniques. Active heat transfer enhancement techniques require some external power sources such as electric power, surface vibrations or any mechanical devices which consume power. But in the case of passive techniques make use of design modifications and using inserts to create resistance to the flow field as abstraction materials such as twisted tapes, vertex generators, threaded pipes, fins, dimples, additives etc., are used. The majority of these strategies have been utilised to improve the heat energy transfer of single-phase heat-exchanging devices [15]. Among the aforementioned processes, passive and combination techniques seem to be the most encouraging for heat transfer augmentation of heat-exchanging devices [16]. Specific types of inserts of different shapes and geometry are employed to create turbulence fluid flow and the formation of a thermal boundary layer, in addition, to inducing longitudinal vortices, which improve heat energy transfer between two fluids of different temperatures [17-26]. In this present work, the passive technique is discussed.

[*]Corresponding Author

2.0 Literature Review and Objective

The number of investigations that have been conducted by using various passive augmentation (enhancement) techniques to increase the heat transfer coefficient in heat exchangers is as follows:

Andaç Batur Çolak et al [1] used an ANN model to determine the thermophysical performance of the DPHE. Pipe and annulus side pressure decreases, and overall cost were evaluated using Model-one with deviations of 0.16%, 0.23%, 0.02%, and 0.003%, and Model-two with variances of 0.02%, 0.18%, 0.16%, and 0.15%, respectively. Furthermore, Model-one gives the best average squared deviation for annulus side pressure difference and average cost, with values of 0.000254 and 0.000193, respectively, whereas Model-two yields the best values for U and pipe side pressure change, with values of 0.000111 and 0.000190 respectively.

Tejaraju R et al [2] investigate a new form of an insert, called the Para-Winglet Thermal and the flow behaviour of tape inserts is examined with 6000d “Red” 30000. For the inserts, three unique sets of pitches and para-inclinations are investigated. The application of para-winglet tape (PWT) to the tube enhanced turbulent kinetic energy, leading to recirculation between the inserts. The PWT configuration is superior to the simple tube. Case-9 at Re=30000 had the greatest Nu with an enhancement of 407%, while Case-7 at Re=6000 had the lowest with an augmentation of 88% when compared to plain DPHE. Case-3 had the greatest ff at Re= 30000, with an augmentation of 846%. Case-7 got the lowest at Re=24000, with a 286% improvement over standard DPHE. At Re=30000 and 6000, the highest and lowest performing optimization index values were 02.69 and 01.09, respectively.

B K Dandoutiya and Arvind Kumar [3] examined the impact of w-cut TT on DPHE hydrothermal performance numerically. In comparison with Re, the rate of heat transfer is linearly increased. The maximum enhancement of the heat transfer rate was seen at Re=15300. The friction factor reduces as Re rises, but increases dramatically when w-cut TT is used. The depth of w-cut in increased, the thermal performance also increased. At lower Re, the greatest thermal performance factor was found. When the thermal performance factor was compared to earlier studies, it was discovered that the current studies have a higher thermal performance factor than other inserts.

Marwa A.M et al [4] did a numerical investigation on self-rotating eccentric DPHE. The rotation affects Nu, which increases by 491% at lower speed and 120% at high speed when compared to the resting condition. The heat transfer rate is increased up to 223% due to 40 mm eccentricity change and inner pipe rotating at 500 rpm. The pressure change across the test section increases by about 53% in the rotational speed of the inner tube up to 500 rpm and eccentricity of 40 mm.

Anas El Maakoul et al [5] investigated using a CFD package the effects of the finned tube in DPHE. The research is carried out with a fluid with a high Pr (engine oil) and changing thermo-physical parameters. For set ups with fin split intervals ranging from 0.333 to 0.166 m, 3D CFD simulations in laminar flow are done. The split longitudinal fin (SLF) designs are compared to the reference longitudinal fin (LF) arrangement for different mass flow rate and pumping power. Heat transfer rates in annuli fitted with SLF are 31%-48% greater than in normal LF annuli for the same unit weight and pumping power.

Do Huu-Quan et al [6] investigate using a 3D numerical method the effects of the flat inner tube in a turbulent flow boundary condition on the DPHE. The impact of the inner tube shape on thermal characteristics was revealed to be substantially impacted by the value of Re. Using flat core pipes with a low aspect ratio (AR) improves U, thermal effectiveness, and the Re=7000 performance index. A 0.37 aspect ratio flat inner tube increases the thermal effectiveness, total convective heat transfer coefficient, and performance index by 2.7%, 2.9%, and 16.8%, respectively. When 7000≤ Re, however, circular inner tubes outperform flat inner pipes significantly.

SMS Mousavi and SMA Alavi [7] described a tube-in-tube heat exchanger with symmetrical 4-digit NACA airfoils with 0^o angles of attack as a vertex generator is optimised for heat energy transmission and pressure drop. First time in the research, these airfoils were used in a DPHE to decrease pressure drop due to their insignificant producing vortex. Through experimental work and CFD package, the thermodynamic performance of four different shaped aerofoil turbulators with varying settings of the Re, pitch (P) and thickness (t) was evaluated. The correlation of the Nu and ff has been calculated using the experimental method in this context. The findings shows improvement under various Re. Furthermore, an increase in t and a decrease in PR result in an increase in ff and Nu, which boosts overall. Finally, the maximum = 1.91 was found with PR = 1.11, Re = 6000, and t = 0.3.

Smith Eiamsa-ard, Chinaruk Thianpong [8] utilized a two-fold funnel heat exchanger with various plans of louvred strip embeds, trial results indicated that the forward louvred strip game plans can advance the warmth move rate by roughly 150 per cent to 284 per cent, while the regressive courses of action could improve the warmth move by around 133 per cent to 264 per cent. It was seen that louvred-strip additions can be utilized proficiently to increase the heat move rate because the disturbance force actuated could improve the warmth move.

A T Wijayanta et al [9] investigated two-sided delta-wing tape (TSDW) inserts devised in this work to improve convective heat transmission in a DPHE. By varying the Re from 5300 to 14500, the effect of the wing width ratio = 0.63, 0.47, and 0.31 on the hydrothermal performance of heat exchangers is described. The TSDW tape inserts with a wing-width ratio of 0.63 yield the maximum Nu, which is 177% more than the smooth pipe. Despite the considerable increase in heat transmission, the ff is 11.6 times that of a plain tube, illustrating that the presence of TSDW causes more severe friction loss.

The following are the primary goals of this work:

i. To find out the overall coefficient of transfer (U) for the variable flow rate of hot and cold water.
ii. To determine the rate of heat transfer at varying hot and cold water flow rates.
iii. To compare the values of heat transfer coefficient and heat transfer rate for smooth pipe and pipe with integrated fins of 1.25 mm and 1.5 mm pitch at the same flow rate.
iv. To calculate the percentage of increasing the heat transfer rate and sheated transfer 2 coefficient use of integrated finned pipe by comparing the smooth pipe at the same flow rate.
v. Select the suitable integrated fined pipe which gives the highest thermal performance.

3.0 Experimental Method and Methodology

Figure 1 illustrates a schematic representation of the experimental set up employed in this study. The DPHE test section, hot water reservoir, cold water reservoir, pump, two rotameters, six valves, four J-type thermocouples, and an electric water heater comprise the majority of the test equipment. The DPHE test section is the heart part of the test rig of length 800 mm. This test section consists of a smooth inner copper tube with an outside diameter is 30 mm and inside diameter is 26 mm. For experimentation purposes, the inner copper tube surface texture is changed by providing integrated fins on an outer surface pitch of 1.25 mm and 1.5 mm and the height of the fin 1 mm as shown in Figures 3 and 4 respectively. The inner copper pipe is enclosed with an outer GI pipe with an inside diameter is 46 mm and outside diameter is 50 mm which is externally insulated with cotton threads to avoid heat loss from the pipe to the surroundings. The input and output temperatures of hot and cold water are measured using thermocouples positioned on each side of the test section. The temperature read by the thermocouples is displayed with the help of a digital display. The hot water from the hot water reservoir is pumped through the test section, and the flow rate of hot water is recorded using a rotameter.

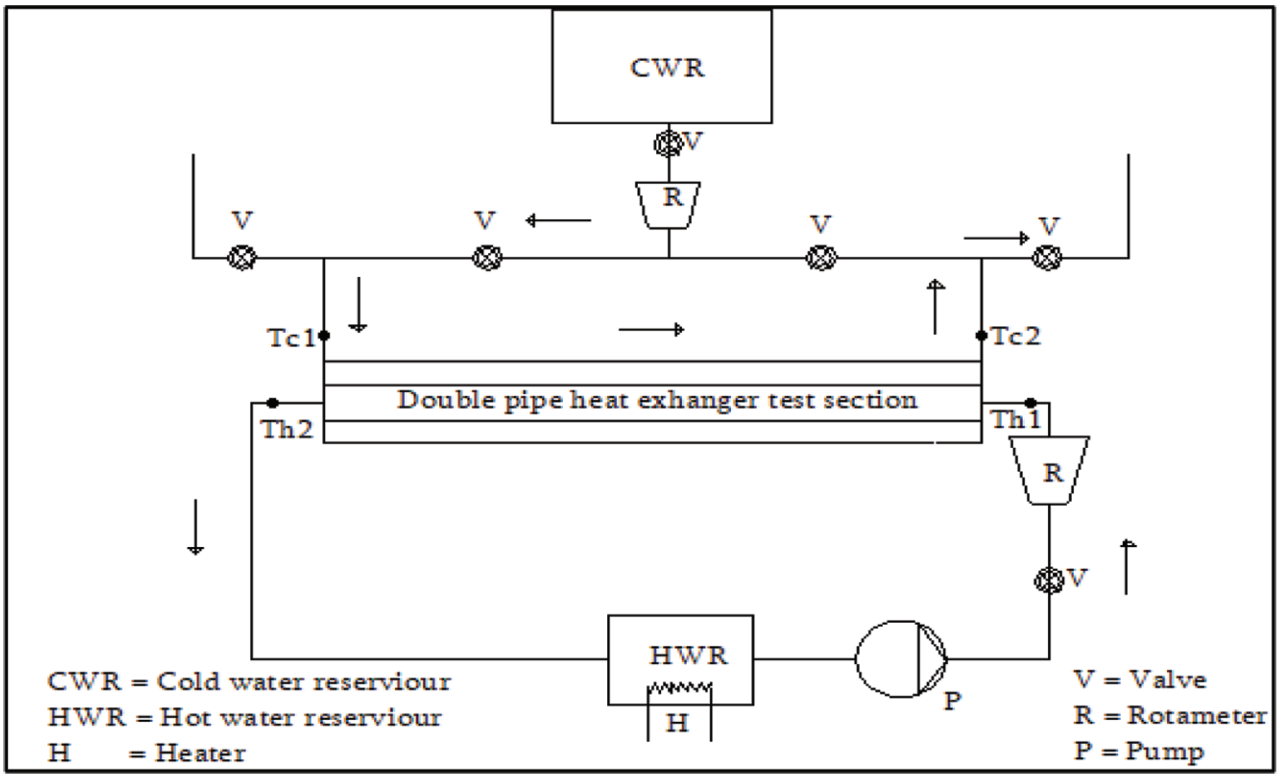

Figure 1: Experimental setup

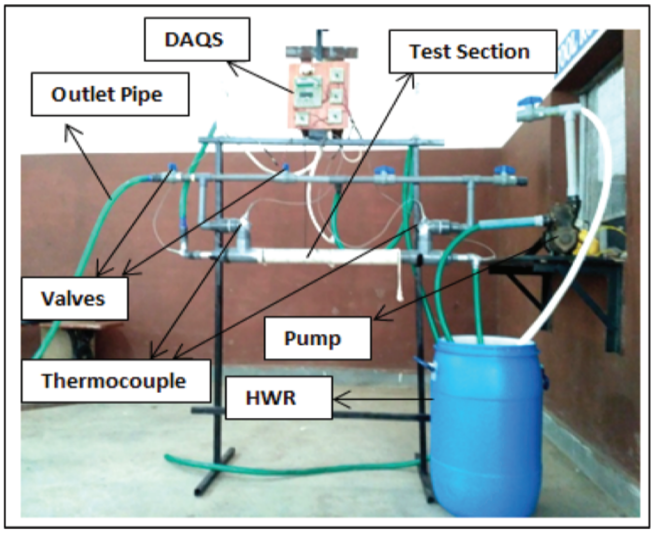

Figure 2: Pictorial view of the test rig

At the same time, cold water is delivered into the annulus side of the test section and measured with another rotameter. The mass flow rate of cold water and hot water is regulated with help of ball valves. The cold water in the annulus absorbs the heat from the hot water flowing inside the tube and drains it. Figure 2 depicts a visual representation of the real experimental set up. Table 1 shows the complete specification of the test section and experimentation circumstances.

The experiment was conducted by changing the hot and cold water flow rates and parallel and counter flow directions. The direction of the flow is made by operating the appropriate valves provided in the test rig. Initially, the hot water flow rate is kept constant, while the cold water flow rate is modified after each set of data. The cold water flow rate is then kept constant while the hot water flow rate is adjusted. For counterflow arrangements, the same set of procedures is performed.

Table 1: Geometric parameter of the test section and the boundary conditions

	Parameter	Quantity
1.	Test section length (mm)	800
2.	Inside copper pipe diameter (mm)	26
3.	Thickness of inside copper pipe (mm)	2
4.	Outside GI pipe diameter (mm)	50
5.	Thickness of GI pipe (mm)	2
6.	Temperature of inlet Hot water (^{o}C)	87
7.	Temperature of inlet Cold water (^{o}C)	27
8.	Water flow rate (kg/s)	0.014-0.07

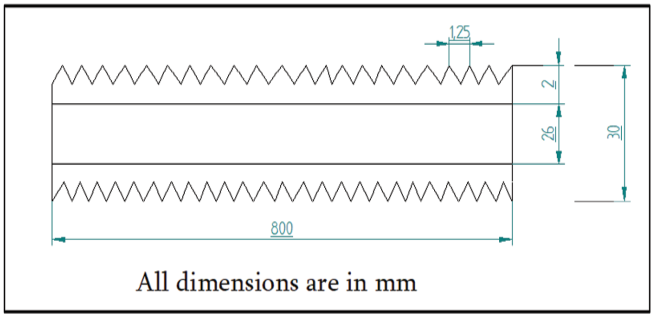

Figure 3: Pipe with Integrated fin of 1.25 mm Pitch

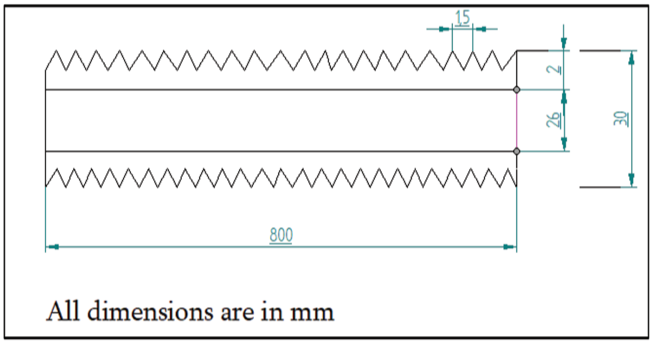

Figure 4: Pipe with Integrated fin of 1.5 mm Pitch

Data Reduction

The experimental results were calculated with the help of the following equations [10-26]

4.1 Heat transfer in hot water

$$Q_h = m_{fh}\, C_{p\,hot}\,(T_{h,in} - T_{h,out}) \quad \text{watts} \qquad \dots (1)$$

4.2 Heat gain by cold water

$$Q_c = m_{fc}\, C_{p,cold}\,(T_{c,out} - T_{c,in}) \quad \text{watts} \qquad \dots (2)$$

4.3 Average heat transfer

$$Q = \frac{Q_h + Q_c}{2} \quad \text{watts} \qquad \dots (3)$$

4.4 Log mean temperature difference

$$\text{LMTD} = \frac{\theta 2 - \theta 1}{\ln^{\theta 2}/_{\theta 1}} \qquad \dots (4)$$

Where:

for parallel flow heat exchanger:

$\Theta_1 = (T_{h,\,in} - T_{c,in})$

$\Theta_2 = (T_{h,\,out} - T_{c,\,out})$

for counter flow heat exchanger:

$\Theta_1 = (T_{h,\,in} - T_{c,\,out})$

$\Theta_2 = (T_{h,\,out} - T_{c,\,in})$

Table 2: Results for counter-flow heat exchanger for varying mass flow rate

m_c (kg/s)	0.014	0.028	0.042	0.056	0.07	m_h (kg/s)
Q (W)	586.152	1201.612	1465.38	1496.688	1699.841	0.014
	1787.764	2110.147	2373.916	2403.223	2139.455	0.028
	2930.76	3428.989	3692.758	4132.372	3751.373	0.042
	4015.141	4278.91	4290.987	3985.834	4015.141	0.056
	5040.907	5187.445	5744.29	5714.982	5861.52	0.07

4.5 The overall convection heat transfer coefficient established on the outside surface of the inside copper tube

$$U_0 = \frac{Q}{Ao\ LMTD} \quad W/m^2K \tag{5}$$

Where, $A_o = \pi d_o l \quad m^2$

4.0 Results and Discussions

For Smooth pipe

Figures 5 and 6 exhibit the graph plotted for the Q and the m_h by changing the cold-water flow rate (m_c) for the smooth pipe. The nature of the graphs demonstrates that the heat transfer rate increases with an increase in the flow rate of hot and cold water in both counter-flow and parallel-flow set ups. It resulted in the maximum heat transfer of 5861.52 W at the combination of m_h and m_c of 0.07 kg s^{-1}. The heat transfer rate during counter-flow arrangement is 13.2% more than that of the heat transfer rate of parallel flow at the same flow rate. Table 2 summarises the findings obtained for the counter-flow heat exchanger when the m_h and m_c fluid is varied.

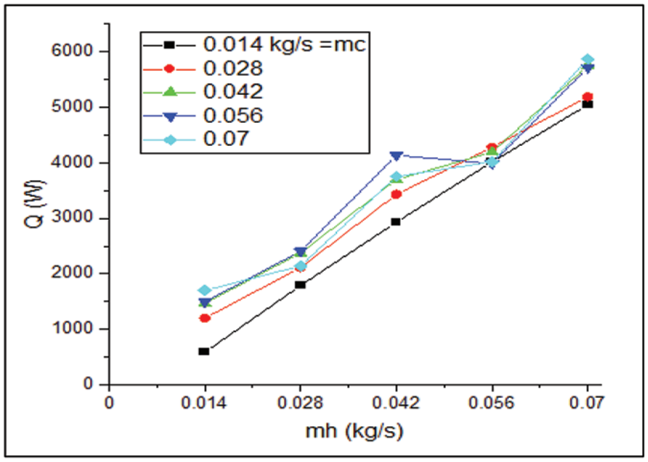

Figure 5: Q for the smooth pipe with counter-flow

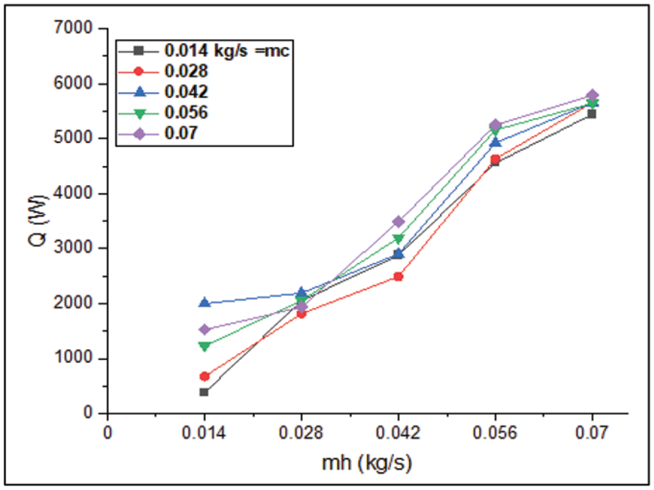

Figure 6: Q for the smooth pipe with parallel-flow

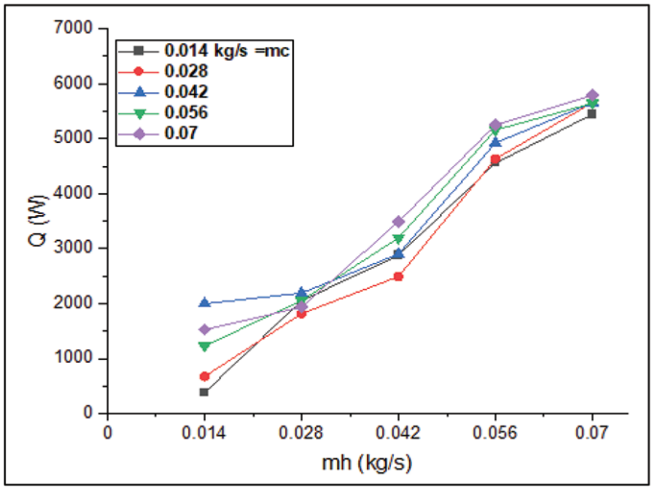

Figure 7: U for smooth pipe with counter-flow

For Pipe with Integrated fins of Pitch 1.25 mm

Figures 9 to12 show the graphs plotted for the pipe which has integrated fins of pitch 1.25 mm for varying hot and cold

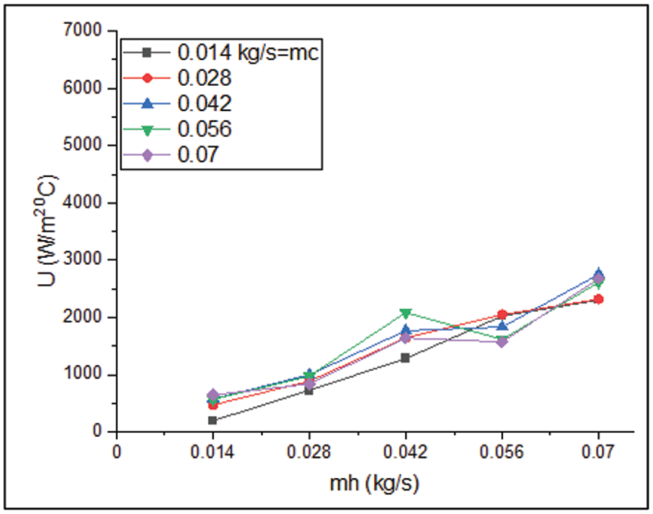

Figure 8: U for smooth pipe with parallel-flow

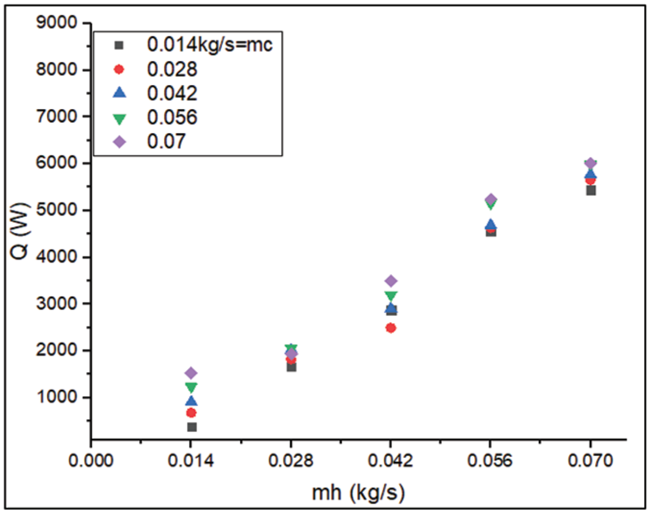

Figure 10: Q for pipe with integrated fins of pitch 1.25 mm with parallel flow

water from 0.014 kg/s to 0.07 kg/s in counter and parallel flow arrangements.

Figures 9 and 10 show the effect of the m_h and m_c on the Q. From the nature of the graph, one can infer the Q value is boosted by an increase in the m_h and m_c. Since the integrated fins enhance the surface area of heat transfer on the outside surface area of the inner pipe simultaneously acts as a vortex generator. These fins produce turbulence in the flow field.

Due to the turbulence, the heat transfer rate (Q) will be maximum in counter-flow arrangement and is 34.15 % more than that of parallel flow, at the flow rate of 0.07 kg s^{-1}. The heat transfer rate in an integrated fins pipe of pitch 1.25 mm is 37.5 % more than the smooth pipe at the same combination of flow. Figures 11 and 12 display the impact of the mass flow rate of hot and cold water on the overall heat transfer coefficient. From the graphs, it is clear that the U increased with increasing the mass flow rate of water. Here counter flow arrangement resulted in the highest heat transfer coefficient of 4332.46 W/m^{2o}C which is 54.9 % more than the parallel flow arrangement. When compared to a smooth pipe, it is greater by 10 %.

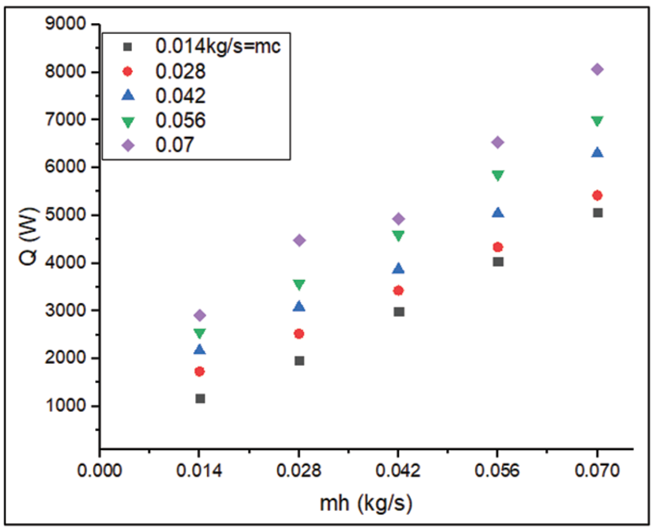

Figure 9: Q for pipe with integrated fins of pitch 1.25 mm with counter-flow

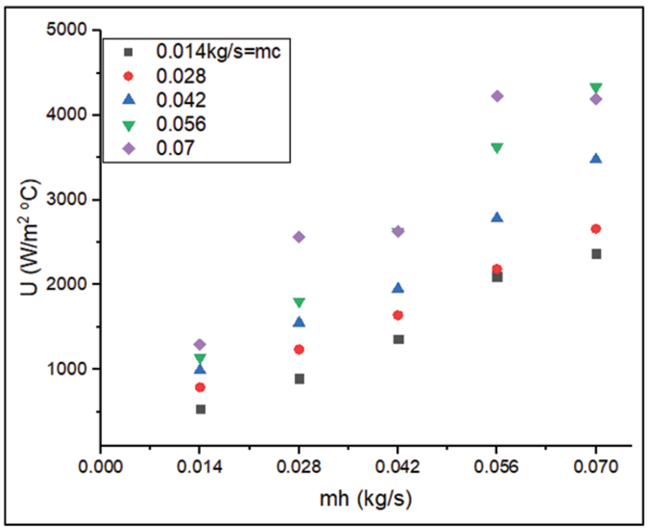

Figure 11: U for pipe with integrated fins of pitch 1.25 mm with counter-flow

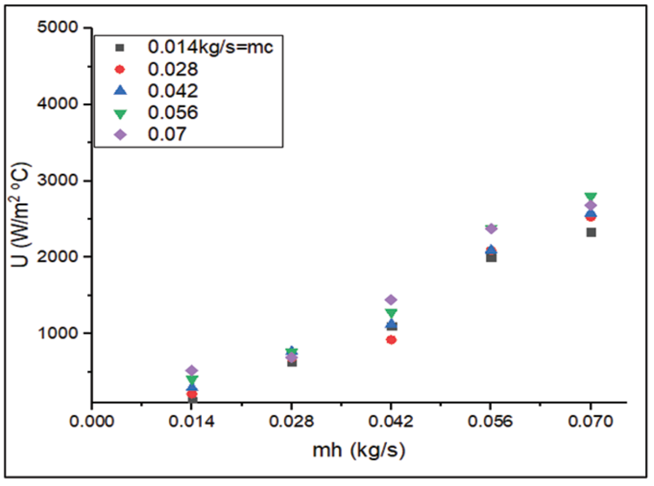

Figure 12: U for pipe with integrated fins of pitch 1.25 mm with parallel flow

For Pipe with Integrated fins of Pitch 1.5 mm

Figures 13 to 16 show that the graph is plotted for mass flow rate versus the Q and U of heat exchangers for the integrated fin pipe of pitch 1.5 mm. From Figures 13 and 14, it is clear that the Q has increased with increasing the mass flow rate of fluid, got maximum heat transfer of 8939.818 W in counter flow arrangement at the combination of m_h and m_c is 0.07 kg s^{-1} and 0.042 kg s^{-1} respectively. By comparing with the pipe of a 1.25 mm pitch, the heat transfer rate is more in the case of a pitch of 1.5 mm by 10%.

Figures 15 and 16 are plotted for mass flow rate versus the overall heat transfer coefficient. In the counter-flow arrangement, the highest U of 4837.32 W/m^2 °C is 21.1% higher than the parallel flow at the same flow rate.

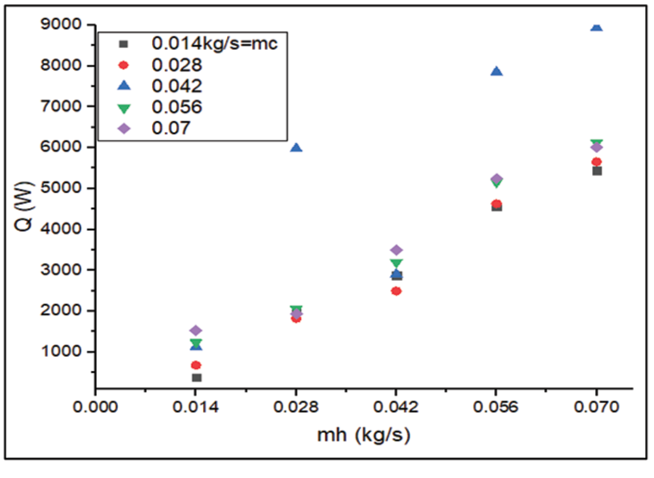

Figure 13: Q for pipe with integrated fins of pitch 1.5 mm with counter-flow

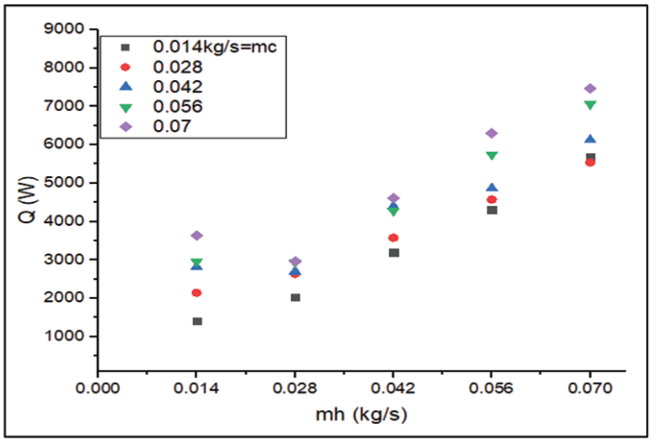

Figure 14: Q for pipe with integrated fins of pitch 1.5 mm with parallel-flow

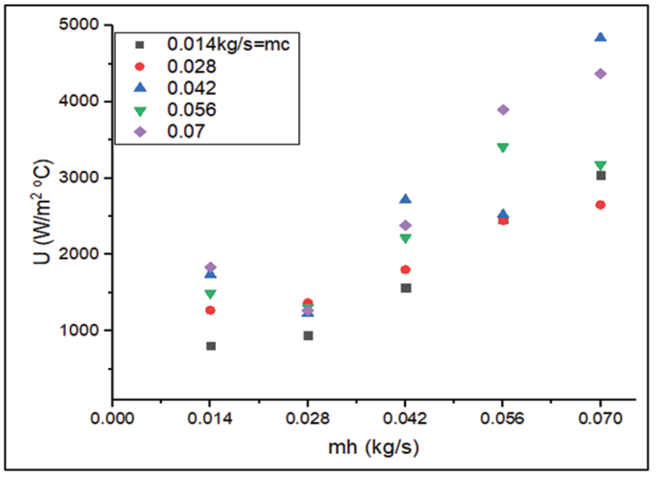

Figure 15: U for pipe with integrated fins of pitch 1.5 mm with counter-flow

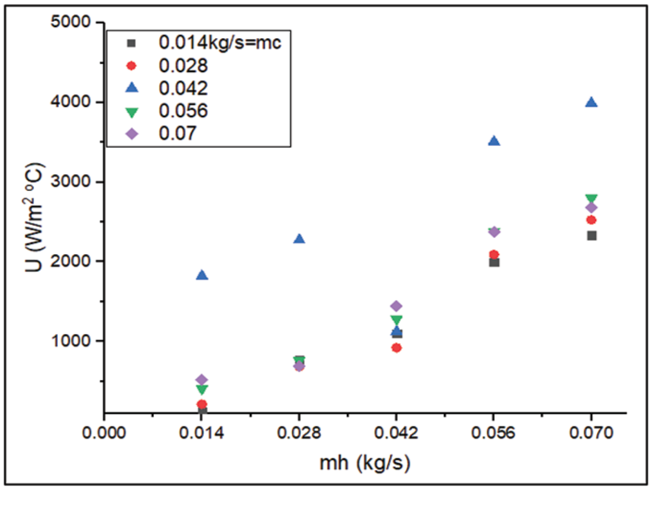

Figure 16: U for pipe with integrated fins of pitch 1.5 mm with parallel-flow

5.0 Conclusions and Future Work

Conclusions

The current work is conducted to understand the outcome of the flow rate of water and pitch of threads on heat transfer rate and overall heat transfer coefficient. The results obtained from this experiment are concluded as follows:

i. The Q and U increase with the increase in the rate of flow of water.

ii. The Q and U are more in the pipe with integrated fins compared to the smooth pipe.

iii. The Q and U are more in the counter-flow direction than in the parallel flow direction.

iv. The Q in the counter-flow heat exchanger with integrated fins of pitch 1.5 mm is 10.9 % higher than that of the fins of pitch 1.25 mm.

v. The U in the counter-flow heat exchanger with integrated fins of pitch 1.5 mm is 11.6 % higher than that of a pipe with a thread pitch of 1.25 mm.

Future work

A few works have been conducted on "Experimental Investigation of the Effect of Integrated Fins on Heat Transfer Rate of Double Pipe Heat Exchanger". Based on these experimental outcomes some of the future works are identified, and they are as follows:

i. These experimental data are helpful to study the influence of different shapes of integrated fins on the Q.

ii. These data are useful to conduct similar experiments by changing the pitch length and height of the fins.

iii. These results are utilized to study and monitor the effect of many passive techniques on the Q and U of double-pipe heat exchangers.

Nomenclature

Q_h&Q_c	Heat transfer in hot and cold water	[W]
Q	Average heat transfer	[W]
T_h&T_c	Temperatures of hot and cold water	[°C]
C_{pw}	Specific heat of water	[kJ/kg K]
m_h&m_c	Mass flow rate of hot and cold water	[kg/s]
A_0	Outer surface area of inner pipe	[m^2]
l	Length of pipe	[m]
Θ_1 & Θ_2	Temperature differences	[°C]

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