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Design and Testing of a Spiral Plate Heat Exchanger for Textile Industry

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Abstract — A heat exchanger is a system designed to transfer heat between two fluids to control the temperature of one of the fluids. In textile industries, heat exchangers are used to reduce the temperature of the dye liquor to be discharged. To meet this requirement, plate type and shell and tube type of heat exchangers are used. The problem with these heat exchangers is clogging, space requirement and cost. The most feasible solution to these problems is offered by spiral plate heat exchanger. Process industries have many applications where heat transfer across the multiple streams and fluids takes place, thus most effective heat exchanger is the need and making it affordable to the end user is still very important. A test rig is modified in order to carry out performance evaluation of spiral plate heat exchanger. The effect of various parameters like mass flow rate of hot fluid, mass flow rate of cold fluid, inlet temperature of hot fluid and outlet temperature of hot fluid on overall heat transfer coefficient of spiral plate heat exchanger is studied. The Spiral Plate Heat exchanger is designed and fabricated as per the design specifications. Hydro testing is done after the completion of fabrication to check if there are some leakages. The performance testing of the spiral plate heat exchanger is done, based on the testing results obtained experimental validation is carried out to check the accuracy of the experimentation. Various graphs are plotted and their trends are studied. The Spiral Plate Heat Exchanger proves to be good alternative due to higher values of overall heat transfer coefficient and self-cleaning properties.

Key words — Dye liquor, Hydro testing, Overall heat transfer coefficient, Spiral Plate Heat Exchanger.

1. INTRODUCTION

Heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. Common examples of heat exchangers are plate type, shell and tube, spiral plate heat exchanger. Plate type and shell and tube type heat exchangers causes high pressure drop and are very costly. They have low overall heat transfer coefficient and occupy more space. Choking of impurities present in dye liquor is observed in these heat exchangers. Spiral Plate heat exchanger can be a good alternative in this case owing to its self-cleaning properties and compactness.

Spiral plate heat exchangers consist of two long plates rolled together, forming a spiral. Studs welded to the plates fix the spacing between the plates and provide mechanical strength. An important feature of spiral plate exchangers with respect to other exchanger technologies is its capacity to handle high viscosity and dirty fluids, exhibiting lower tendency to fouling. This is due to the particular geometry that creates a constant change in direction thus increasing local turbulence that eliminates fluid stagnant zones. In addition its geometrical features make it suitable to accommodate a large heat transfer area in a relatively small volume [4].

Fig. 1. Inside of spiral plate heat exchanger.

Spiral heat exchangers do not suffer from fouling and clogging. These heat exchangers are circular units containing two concentric spiral flow channels, one for each fluid. The different media flow counter-currently in these channels with no risk of intermixing. One fluid enters the centre of the unit and flows towards the periphery. The other fluid enters the unit at the periphery and moves towards the centre. Its single-flow passages induce high shear rates that scrub away deposits as they form. This self-cleaning effect reduces fouling and makes spiral heat exchangers ideal for handling tough fluids such as process slurries, sludge, and media with suspended solids or fibres [6]. From application point of view, spiral heat exchanger is an optimum solution for various reasons such as less space requirement i.e. compact design, no choking of threads i.e. no fouling, it can withstand higher pressure and temperatures, overall heat transfer coefficient is high, etc.

2. METHODOLOGY

1. Identification:- Spiral plate heat exchanger is identified as the most suitable heat exchanger for the application.
2. Selection of heating source:- The readily available sources for heat generation in company are electricity and steam. The heat generation through electricity is effective but not feasible because of its cost. Since, Boiler is readily available in the company, steam will be preferred.
3. Selection of components:- According to the layout of test rig the components will be selected of the suitable range and the same will be subsequently procured.
4. Procurement of components:- Order will be issued after component selection. After procurement of components, fabrication of test rig will be done.
5. Modification of test rig:- The test rig modification will be done according to the requirement and as per the specification of the test rig. The pressure gauges and temperature gauges will be replaced by pressure transmitter and temperature transmitter for the measurement of temperature and pressure. The magnetic flow meter will be used to measure the flow rate of hot fluid and cold fluid respectively.
6. Actual design of spiral plate heat exchanger:- The thermal design and mechanical design of the spiral plate heat exchanger will be done accordingly.
7. Fabrication of spiral plate heat exchanger:- The material with desired properties will be selected and the fabrication of the spiral plate heat exchanger will be done accordingly.
8. Performance testing of spiral plate heat exchanger:- Process parameters such as mass flow rate of hot fluid, mass flow rate of cold fluid, inlet temperature of hot fluid will be varied and its effect on overall heat transfer coefficient will be studied.
9. Experimental Validation:- Accuracy of the experimentation will be calculated. Various graphs will be plotted and their trends will be studied.

3. MODELLING AND CALCULATION

3.1 Modelling of Spiral Plate Heat Exchanger

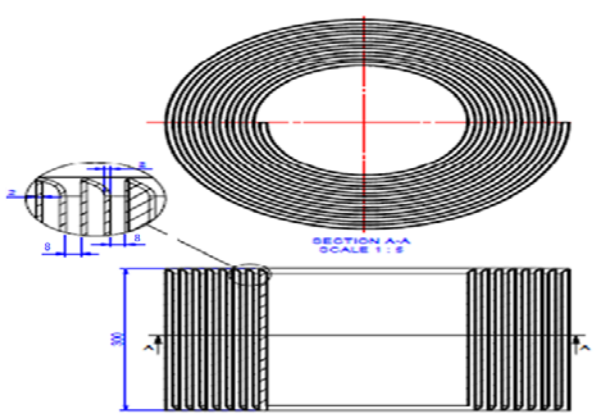

Fig. 2. 2D view of spiral plate

Table 1. Dimensions of spiral plate

No of turns of spiral	10
Channel spacing	8 mm
Pitch	20 mm
Core diameter	150 mm
Start diameter of inner spiral	170 mm
Start diameter of outer spiral	192 mm
Spiral thickness	2 mm
Length of spiral	8364 mm
Spiral outer diameter	485 mm
Width of plate	300 mm

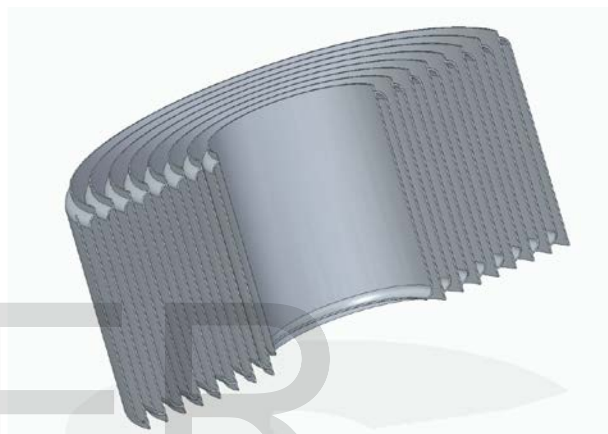

Fig. 3. 3D model of spiral plate

Figure 2 shows the 2 D view of the spiral plate along with its dimensions. The dimensions of the spiral plate which are obtained from the thermal design calculations are used as input parameters for making the geometry of the spiral plate. Based on the 2 D view of the spiral plate along with its specified dimensions, 3 D model of spiral plate is created in solid edge ST 9 as shown in the figure 3. The spiral plate is designed accordingly and the dimensions which are obtained in the thermal design calculations are considered. Based on this model with its specified dimensions the fabrication of the spiral plate heat exchanger is done. After fabrication of the spiral plate heat exchanger testing is done to check if there are some leakages. Further the performance testing of the heat exchanger is done and based on the theoretical and experimental results obtained the experimental validation is carried out.

3.2. Thermal Design Calculations

Hot fluid inlet temperature (T_{hi}) = 70˚C
Hot fluid outlet temperature (T_{ho}) = 47˚C
Cold fluid inlet temperature (T_{ci}) = 29˚C
Cold fluid outlet temperature (T_{co}) = 51˚C
$C_{ph} = 4.1824 \frac{KJ}{kg*K}$

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$C_{pc} = 4.179 \frac{KJ}{kg*K}$

$m_h = 3000 \frac{kg}{hr} = 0.8333 \frac{kg}{sec}$

$m_c = 3000 \frac{kg}{hr} = 0.8333 \frac{kg}{sec}$

Viscosity of hot fluid (μ_h) = $0.0004774 \frac{kg}{m-sec}$

Viscosity of cold fluid (μ_c) = $0.000653 \frac{kg}{m-sec}$

Conductivity of water hot (k_h) = $0.6525 \frac{W}{m*K}$

Conductivity of water cold (k_c) = $0.631 \frac{W}{m*K}$

Density of hot fluid (ρ_h) = $983.96 \frac{kg}{m^3}$

Density of cold fluid (ρ_c)= $992.22 \frac{kg}{m^3}$

Amount of heat transfer (Q)

$Q = m_h * C_{ph} * \Delta T$ ……………………(1)

$Q = m_h * C_{ph} * (T_{hi} - T_{ho})$

Q = 0.833 * 4.184 * (70 - 47)

Q = 80.16 KW

Log Mean Temperature Difference (LMTD)

$LMTD = \frac{(T_{hi}-T_{co})-(T_{ho}-T_{ci})}{\ln\frac{(T_{hi}-T_{co})}{(T_{ho}-T_{ci})}}$ …………….(2)

LMTD = 18.49

Assume $U_o = 850 \frac{W}{m^2*K}$

$A_o = \frac{Q}{U_o*LMTD}$ …………………………(3)

$A_o = 5.09\ m^2$

Assume plate width H = 0.3048 m

$L = \frac{A_o}{2*H}$ …………………………..(4)

L = 8.364 m

Channel Spacing for Cold and Hot (d_h=d_c) = 7.938 mm = 0.007938 m

Core diameter, c (d_s) = 150 mm = 0.150 m

Plate Thickness, t = 2 mm = 0.002 m

Outside Spiral Diameter (D_s)

$D_s = \sqrt{(1.28 * L(d_c + d_h + 2t) + c^2)}$ ....(5)

D_s = 0.485 m

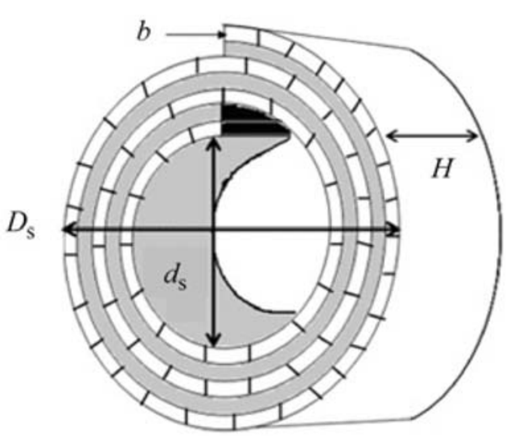

Fig. 4. Geometrical features of spiral plate heat exchanger [4]

Hot side calculations

Equivalent diameter

$D_{eh} = \frac{2*d_h*H}{d_h+H}$ …………………(6)

D_{eh} = 0.01547 m

Mass flux (G)

$G = \frac{m_h}{A_c}$ …………………………..(7)

$G = \frac{m_h}{H*d_h}$

$G = 344.42 \frac{kg}{m^2-sec}$

Reynold number

$R_e = \frac{D_{eh}*G}{\mu_h}$ ……………………..(8)

R_e = 11163.12

Critical Reynold number

$R_{ec} = 20000*(\frac{D_{eh}^{0.32}}{D_s^{0.32}})$ …………………(9)

R_{ec} = 6640.99

$R_e > R_{ec}$

Flow is turbulent.

Prandtl number

$P_r = \frac{\mu_h*C_{ph}}{k_h}$ …………………..(10)

P_r = 3.06

Heat transfer coefficient (h_h)

$h_h = (1+3.54*\frac{D_{eh}}{D_s})*0.023*C_P * G * R_e^{-0.2} * P_r^{-0.67}$ ……………(11)

$\mathbf{h_h = 2702.18 \frac{W}{m^2*K}}$

Cold side calculations

Equivalent diameter

$D_{ec} = \frac{2*d_c*H}{d_c+H}$ ……………………….(12)

D_{ec} = 0.01547 m

Mass flux (G)

$G = \frac{m_c}{A_c}$ ….................................(13)

$G = \frac{m_c}{H*d_c}$

$G = 344.42 \frac{kg}{m^2-sec}$

Reynold number

$R_e = \frac{D_{ec}*G}{\mu_c}$ ………………………(14)

R_e = 8161.21

Critical Reynold number

ISSN 2229-5518

$R_{ec} = 20000*\left(\frac{D_{ec}^{0.32}}{D_s^{0.32}}\right)$ ……………...(15)

$R_{ec} = 6640.99$

Since $R_e > R_{ec}$

Flow is turbulent.

Prandtl number

$P_r = \frac{\mu_c * C_{pc}}{k_c}$ ……………………….(16)

$P_r = 4.3247$

Heat transfer coefficient (h_c)

$h_c = (1+3.54*\frac{D_{ec}}{D_s})*0.023*C_P * G * R_e^{-0.2} * P_r^{-0.67}$…………….(17)

$\mathbf{h_c = 2279.89\ \frac{W}{m^2*K}}$

Overall heat transfer coefficient (U)

$\frac{1}{h_h} = 0.0003700$

$\frac{1}{h_c} = 0.0004386$

Stainless steel conductivity (k) = 16.26 $\frac{W}{m^2*K}$

t = 2 mm = 0.002 m

$\frac{t}{k} = 0.0001230$

$\frac{1}{15000} = 0.00006666$

$\frac{1}{15000} = 0.00006666$

$U = \frac{1}{\frac{1}{h_h}+\frac{1}{h_c}+\frac{t}{k}+0.00006666+0.00006666}$ ………………(18)

$\mathbf{U = 938.94\ \frac{W}{m^2*K}}$

Heat transfer area required (A)

$A = \frac{Q}{U*LMTD}$ ……………………………(19)

$A = 4.10\ m^2$

$A_o = 4.61\ m^2$

Excess area = $\frac{A_o - A}{A_o}$ ……………...............(20)

Excess area = 0.09473*100

Excess area provided = 9.473 %

Number of turns of spiral (N)

$N = \frac{-\left(d-\frac{d_{h,c}}{2}\right)+\sqrt{\left(d-\frac{d_{h,c}}{2}\right)^2+\left(\frac{4*d_{h,c}*L}{\pi}\right)}}{2*d_{h,c}}$ ………………..(21)

Outer core diameter (d) = 0.17 m

Channel spacing for hot fluid and cold fluid ($d_{h,c}$) = 0.007938 m

Length of the plate (L) = 8.364 m

Number of turns (N) = 10

Hot side pressure drop calculations

$\Delta P = \{(0.0789*(\frac{L}{\rho_h}))*(\frac{m_h}{H*d_h})*\{\frac{[1.3*(\mu_h)^{0.33}]}{(d_h+0.032)}\} *(\frac{H}{m_h})^{0.33}+1.5+\frac{16}{L}\}$ ……(22)

Length (L) = 8.364 m

Density (ρ_h) = 983.96 $\frac{kg}{m^3}$

Mass flow rate of hot fluid (m_h) = 0.8333 $\frac{kg}{sec}$

Width of plate (H) = 0.3048 m

Channel spacing for hot fluid (d_h) = 0.007938 m

Viscosity of hot fluid (μ_h) = 0.0004774 $\frac{kg}{m-sec}$

ΔP hot = 0.00805 bar

Cold side pressure drop calculations

$\Delta P = \{(0.0789*(\frac{L}{\rho_c}))*(\frac{m_c}{H*d_c})*\{\frac{[1.3*(\mu_c)^{0.33}]}{(d_c+0.032)}\} *(\frac{H}{m_c})^{0.33}+1.5+\frac{16}{L}\}$ ……..(23)

Length (L) = 8.364 m

Density (ρ_c) = 992.22 $\frac{kg}{m^3}$

Mass flow rate of cold fluid (m_c) = 0.8333 $\frac{kg}{sec}$

Width of plate (H) = 0.3048 m

Channel spacing for cold fluid (d_h) = 0.007938 m

Viscosity of cold fluid (μ_c) = 0.000653 $\frac{kg}{m-sec}$

ΔP cold = 0.00856 bar

4. FABRICATION OF SPIRAL PLATE HEAT EXCHANGER

Fig. 5. Spiral plate heat exchanger

ISSN 2229-5518

Fig. 6. Spiral plate heat exchanger

Figure 5 shows the front view of the spiral plate heat exchanger. Figure 6 shows the top view of the spiral plate heat exchanger. Spiral plate heat exchanger is built by rolling two parallel long sheets around a central bar to make a spiral shape. The two plates are rolled together and the alternate edges of the plates are welded for the flow of hot and cold fluid and to prevent intermixing between the fluids. The distance between the sheets is kept using studs that are welded to the sheets. The length of the studs is 8 mm. The material used for the plate is stainless steel. Based on the prototype made the actual spiral plate along with its dimensions mentioned in the design is fabricated. The rolling of the plate is carried out on special purpose machine and accordingly the machining and the welding of the plate is done. The one end of the spiral plate is welded on the outer side of the core cylinder and the other end is welded on the inner side of the main cylinder into which the rolled spiral plate is located. The two open sides of the plates are closed with the two cover plates. The cover plates are welded to the main cylinder on both sides of the plate so that the two sides are closed. The core cylinder, main cylinder and the cover plates is fabricated as per the design done. The fluid flow in the heat exchanger is counter-current and accordingly the inlet and outlet of the fluid flow are mentioned in the design of the spiral plate heat exchanger.

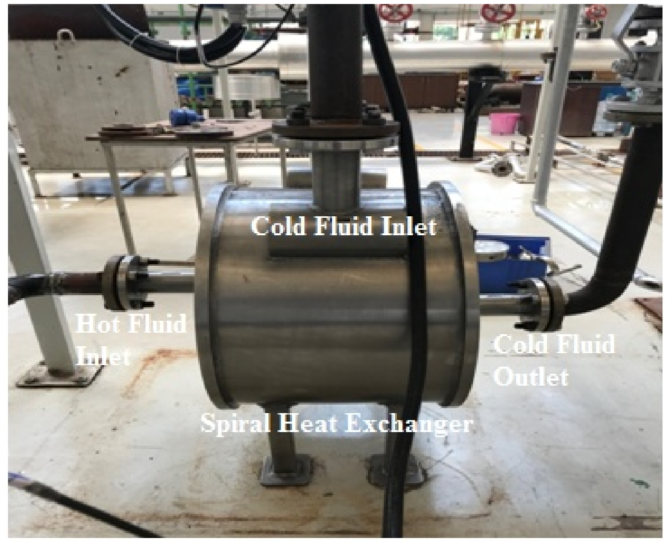

Fig. 7. Spiral plate heat exchanger setup

Fig. 8. Spiral plate heat exchanger setup

Figure 7,8 shows the actual setup for spiral plate heat exchanger. The hot fluid enters the heat exchanger from the centre and leaves from the periphery and the cold fluid enters from the periphery and leaves from the centre. Therefore the fluid flow in the heat exchanger is purely counter-current flow. Accordingly the pipes are connected to the inlet and outlet of the hot fluid and cold fluid respectively with the flanges joining the open end of the pipes and the inlet and outlet of hot fluid and cold fluid respectively.

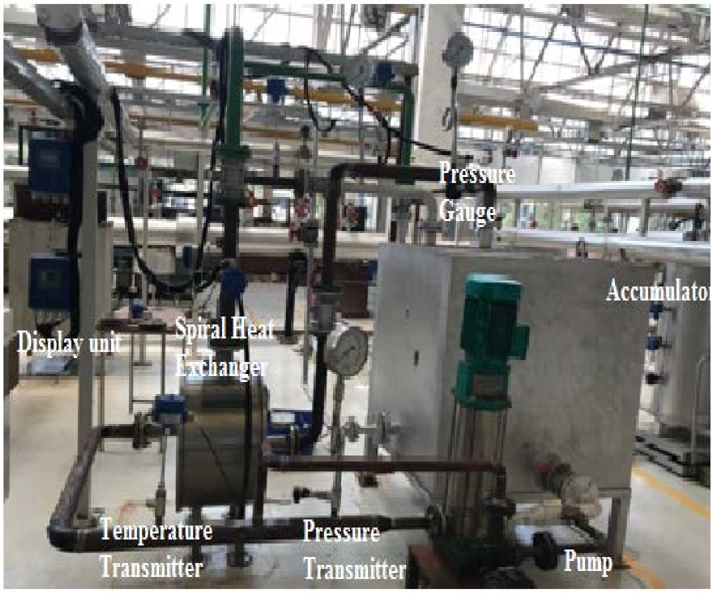

Fig. 9. Actual test rig setup for experimentation

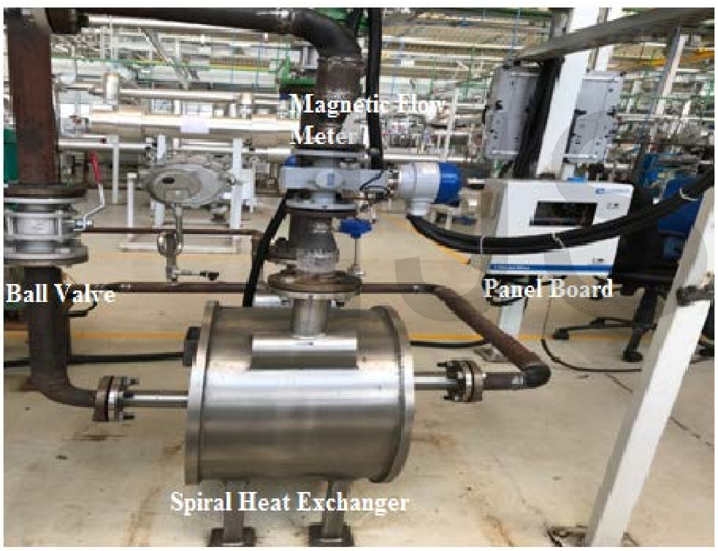

Fig. 10. Actual test rig setup for experimentation

Figure 9,10 shows the actual test rig setup for experimentation. In this set up various sensors are used to measure the temperatures, pressures and mass flow rate of the fluid. The temperature gauges and the pressure gauges are replaced by temperature transmitters and pressure transmitters, magnetic flow meter is used to measure the mass flow rate of the hot fluid and cold fluid respectively. There are four temperature transmitters which are used to measure the inlet temperature of hot fluid, outlet temperature of hot fluid, inlet temperature of cold fluid and the outlet temperature of the cold, four pressure transmitters which are used to measure inlet pressure of hot fluid, outlet pressure of hot fluid, inlet pressure of cold fluid and outlet pressure of cold fluid. There are two magnetic flow meter which are used to measure mass flow rate of the hot and cold fluid. All these modern measurement sensors are connected to a panel board. The input of all these sensors is given to a software known as Ever sense through a panel board and a converter which is connected to the CPU of the computer. When the set up is switched ON for testing all the data i.e. (temperature, pressure and mass flow rate) is displayed on the software. The software stores all the data and gives the output for each and every data that is stored in it. The temperature transmitters and pressure transmitters are more accurate and reliable compared to the pressure gauges and temperature gauges. The gauges contains error more than 5 % and the readings which are noted during testing are also not accurate containing error.

5. RESULTS & DISCUSSION

5.1 Experimental Procedure

Following schematic shows the modified test rig used for experimental validation. The water stored in the accumulator is heated using steam up to 75˚C - 80˚C. This hot water is fed to the heat exchanger using a pump. Hot water enters axially to the heat exchanger and goes radially out. The cold water, on the other hand, enters the heat exchanger from the periphery and is taken out axially. Temperature transmitters are used to measure the inlet and outlet temperature of hot and cold fluid. Pressure transmitters are used to measure the inlet and outlet pressure of hot and cold fluid. Magnetic flow meters are used to measure the mass flow rate of hot and cold fluid. Ball valves are used to control the mass flow rates of hot fluid and cold fluid. The testing procedure is carried out by taking number of iterations. Various sets of readings are obtained by keeping the mass flow rates constant and varying the inlet temperature of hot fluid.

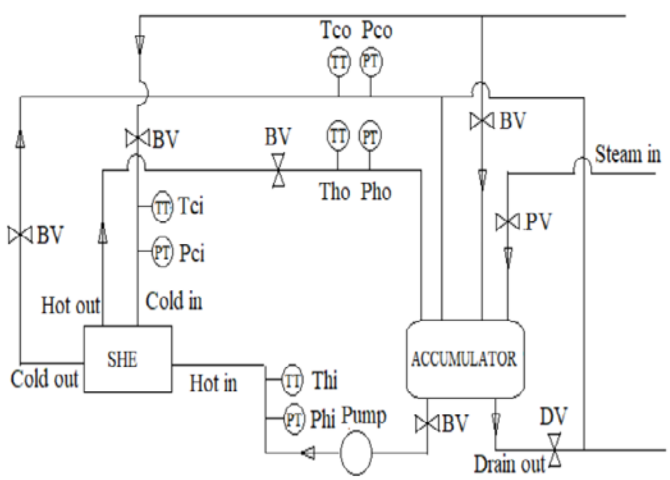

Fig. 11. Layout of Test Rig Setup for Spiral Heat Exchanger

Table 2. Test rig components

SR NO	Symbol	Name
1	BV	Ball Valve

2	PV	Piston Valve
3	DV	Drain Valve
4	TT	Temperature Transmitter
5	PT	Pressure Transmitter
6	SHE	Spiral Heat Exchanger

5.2 Observations

Table 3. Set of observations

T_{hi} ˚C	T_{ho} ˚C	T_{ci} ˚C	T_{co} ˚C	m_h kg/hr	m_c kg/hr
42	38	31	33	3175	2971
49	43	31	34	3018	2942
55	47	29	34	3000	3276
60	52	29	35	2971	3060
64	55	29	36	2962	3041
68	59	30	37	3026	3026
72	61	30	38	2959	3236
75	64	30	39	3024	3000
77	65	30	39	2933	3177
79	67	29	40	2926	2739
82	69	30	41	2838	3153

Table 4. Set of observations

T_{hi} ˚C	h_h ($W/m^{2*}K$)	h_c ($W/m^{2*}K$)	U ($W/m^{2*}K$)
42	1210	1204	523
49	1232	1243	534
55	1244	1303	547
60	1256	1290	547
64	1265	1299	550
68	1282	1304	555
72	1279	1340	560
75	1393	1315	558
77	1285	1343	562
79	1288	1283	552
82	1281	1349	562

5.3 Experimental Validation

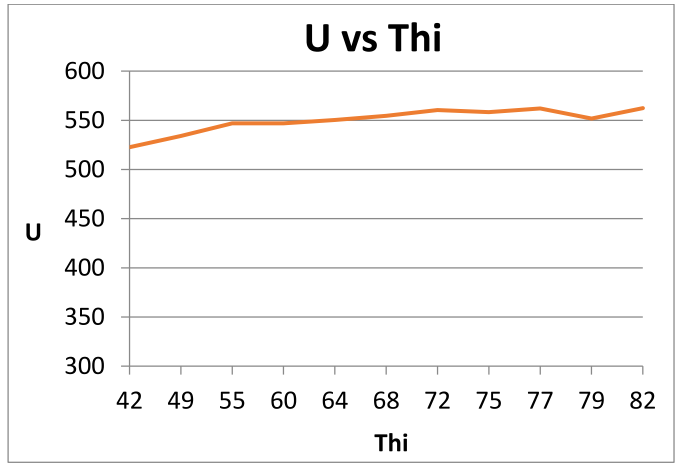

Fig. 12. Overall heat transfer coefficient v/s Inlet temp of hot fluid

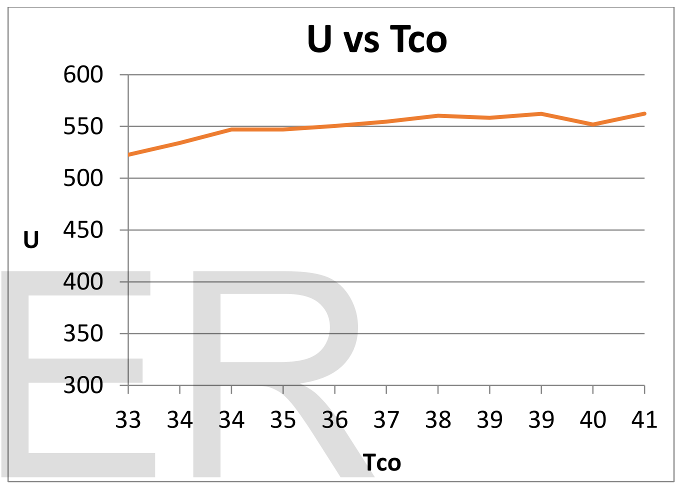

Fig. 13. Overall heat transfer coefficient v/s Outlet temp of cold fluid

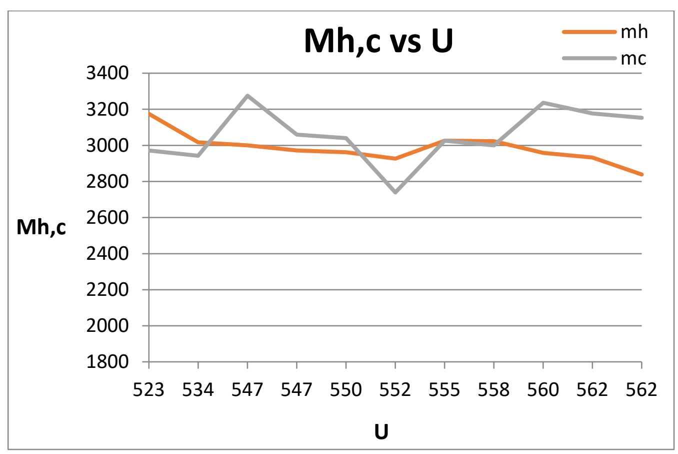

Fig. 14. Mass flow rate of hot, cold fluid v/s overall heat transfer coefficient

The graph of overall heat transfer coefficient versus inlet temperature of hot fluid is shown in figure 12. From the graph it is clear that as the inlet temperature of the hot fluid increases the overall heat transfer coefficient also increases. There is a slight decrease in the value of overall heat transfer coefficient in the range of 42 ˚C to 82 ˚C, when the temperature is 79 ˚C and the overall heat transfer coefficient is maximum when the temperature reaches 77 ˚C. This is due to the decrease in mass flow rate of the hot fluid or cold fluid when the temperature is 79 ˚C and when the mass flow rate of hot fluid and cold fluid is 3000 kg/hr at 75 ˚C, the overall heat transfer coefficient is 558 W/m^2K. The graph of overall heat transfer coefficient versus outlet temperature of cold fluid is shown in figure 13. From the graph it is clear that as the outlet temperature of the cold fluid increases the overall heat transfer coefficient also increases. There is a slight decrease in the value of overall heat transfer coefficient in the range of 33 ˚C to 41 ˚C, when the temperature is 40 ˚C and the overall heat transfer coefficient is maximum when the temperature reaches 39 ˚C. This is due to the decrease in mass flow rate of the hot fluid or cold fluid when the temperature is 40 ˚C and when the mass flow rate of hot fluid and cold fluid is 3000 kg/hr at 39 ˚C, the overall heat transfer coefficient is 558 W/m^2K. The graph of hot fluid and cold fluid mass flow rate versus overall heat transfer coefficient is shown in figure 14. As the mass flow rate of hot fluid and cold fluid increases, the overall heat transfer coefficient also increases. From the graph it is clear that when the mass flow rate of hot fluid is 2933 kg/hr and cold fluid is 3177 kg/hr at 77 ˚C, the overall heat transfer coefficient is maximum. Also when the mass flow rate of hot fluid or cold fluid decreases below 3000 kg/hr, there is a slight decrease in the value of overall heat transfer coefficient. The small fluctuations in the value of overall heat transfer coefficient are due to the increase or decrease in the mass flow rate of hot fluid and cold fluid respectively. The overall heat transfer coefficient obtained is 558 W/m^2K for mass flow rate of 3024 kg/hr for hot fluid and 3000 kg/hr for cold fluid respectively. The overall heat transfer coefficient varies from 523 W/m^2K to 562 W/m^2K when the mass flow rate of hot fluid ranges from 2838 kg/hr to 3175 kg/hr and cold fluid ranges from 2739 kg/hr to 3276 kg/hr respectively.

6. CONCLUSION

The testing is done and the following observations are noted down when the mass flow rate of the hot fluid and cold fluid is 3000 kg/hr and the inlet temperature is varied from 42 degree celcuis to 82 degree celcuis. The results shows that as the inlet temperature is increased the overall heat transfer coefficient also increases. As discussed in the above the overall heat transfer coefficient changes when the mass flow rate of hot fluid and cold fluid varies. When both the mass flow rate of hot fluid and cold fluid is maximum, higher is the overall heat transfer coefficient. The experiment is performed by keeping the mass flow rate of hot fluid and cold fluid constant (i.e) 3000 kg/hr and varying the inlet temperature of hot fluid from 42 degree celcuis to 82 degree celcuis and the corresponding outlet temperature of hot fluid, inlet and outlet temperature of cold fluid is noted down. It is found that when the mass flow rate of hot fluid is 3024 kg/hr and cold fluid is 3000 kg/hr, the overall heat transfer coefficient obtained is 558 $W/m^{2*}K$. Even when the mass flow rate of hot fluid and cold fluid is increased above 3000 kg/hr, the overall heat transfer coefficient remains constant as the heat exchanger is designed for mass flow rate capacity of 3000 kg/hr.

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