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Thermal Analysis and Performance Evaluation of Triple Concentric Tube Heat Exchanger

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Abstract*: *Triple concentric-tube exchanger (TCTHE) is an improved version of double concentric tube heat exchanger (DCTHE). Introducing an intermediate tube to a DCTHE provides TCTHE and enhances the heat transfer performance. Recognizing the need of experimental results, extremely scarce in the literature and essential to validate theoretical analyses, the aim of this work is to investigate thermal behavior of TCTHE. The present study includes design, development and experimental analysis of TCTHE for oil (ISO VG 22) cooling application required for industrial purposes. It comprises of water (cooling fluid) flowing through innermost tube as well as outer annulus and oil (hot fluid) flows through inner annulus. The experimental studies of the temperature distribution for three fluids along the length and heat transfer characteristics for TCTHE under insulated condition for counter current flow mode are carried out and discussed. The effect of change in oil (hot fluid) temperatures is analyzed keeping water inlet temperatures same at various operating conditions. The experiments have been conducted by varying flow rate of one of the fluids at a time and keeping other two fluid flow rates constant. The results are expressed in terms of temperature variation for all three fluids along the length. The effect of change in hot fluid inlet temperature is expressed in terms of heat transfer rate variation with respect to Reynolds number. The variation of non-dimensional parameters as temperature effectiveness and thermal conductance with respect to Reynolds number is also presented in this paper. Theoretical studies are carried out for evaluation of heat transfer rate using empirical correlations. Experimental validation is carried out for degree of cooling at different Reynolds numbers with theoretical analysis.

***Index Terms*: Triple concentric tube heat exchanger (TCTHE), Temperature variation, Temperature effectiveness, Thermal conductance.**

I. INTRODUCTION

The heat exchangers are widely used for industrial as well as residential heating and cooling applications. The need to improve performance and to reduce space required for heat exchanger has been a motivation for researchers to understand, suggest and analyze newer and newer designs.

In Industry, [1]the most common heat exchanger is double concentric tube heat exchanger (DCTHE) and Triple concentric-tube exchanger (TCTHE) is an improved version resulting improvement in thermal performance. TCTHEs find the applications in different sections as dairy, food, beverage and pharmaceutical industries. In triple tube heat exchangers, a thermal fluid is passed through inner annular space and heat transfer media are passed through the central pipe and outer annular space.

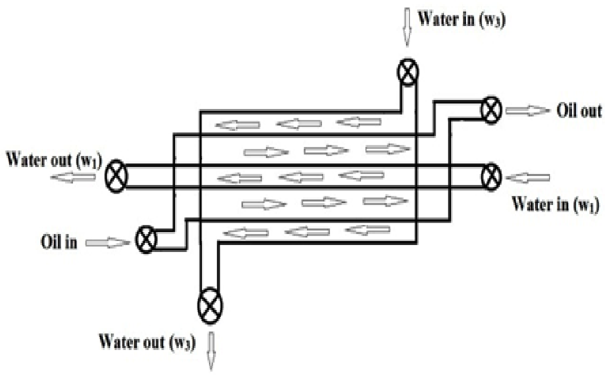

Figure 1. Schematic of counter flow mode in the TCTHE.

The performance of the heat exchanger depends on the various thermo physical properties of fluids and the material. There is need to enhance the effectiveness and compactness of double tube heat exchangers. One of the ways to enhance effectiveness and compactness is Triple Concentric Tube Heat Exchangers (TCTHE). These types of heat exchangers can be extensively useful for the process of pasteurization of food products, viz. dairy products, fruit juice, liquid egg storages and sauces. TCTHE provides an additional flow passage and a larger heat transfer area per unit exchanger length compared to a double concentric-tube heat exchanger. This ultimately enhances heat transfer performance. As the fluid velocities are higher for the flows through annular regions, there is improvement in overall heat transfer coefficients in TCTHE which enhances performance and compactness of heat exchanger. The performance analysis of a triple concentric tube heat exchanger under steady state conditions for insulated and non-insulated conditions was undertaken by G.A. Quadir et al.[1], [2]. The experimentation was carried out using hot water, cold water and normal tap water for different flow arrangements.

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Results of temperature variation along the length for all three fluids indicate pinch points at high volume flow rates of hot fluid. Early pinch points are observed in non-insulated condition. The work was extended for numerical investigation using finite element method (FEM) for same arrangement and the results were expressed in the form of dimensionless temperature variations along the length for all three fluids.

Unidirectional steady and transient numerical simulation of the thermal and fluid-dynamic behavior of TCTHE has been analyzed by Garcia-Valladares [3]. Different numerical techniques are employed and results are compared with those available in open literature. Valerio Giovannoni et al. [4] have developed a numerical model for concentric triple tube combustion chamber used in Ultra-Micro Gas Turbines. The results are represented as a temperature and Nusselt number variation along the length at different mass flow rates and thermal conductivities of walls for all three fluids. The performance of DTHE and TCTHE are compared for adiabatic as well as non-adiabatic case.

C. A. Zuritz [5] has given the macroscopic thermal energy balance equations in differential form and are solved using Laplace transforms. The equation for calculation of bulk temperatures at any location along the length is derived. Flow patterns for TCTHE are numerically and experimentally investigated and comparative study of the performance is done by Abdalla Gomma et al. [6]. The correlations are presented for Nusselt number, friction factor and effectiveness.

The expressions for -NTU relations of TCTHE for parallel and counter flow arrangements have been derived by Ahmet Unal [7]. The variation of effectiveness for individual fluid streams and effectiveness of whole heat exchanger with respect to number of transfer units (NTU) is presented. Ediz Batmaz et al. [8] have suggested a new method for calculating overall heat transfer coefficients in TCTHE without using correlations. This method calculates U values using energy balance equations on a control volume. Ahmet Unal [9] has derived the governing equations for insulated TCTHE which can be solved using advanced mathematical techniques enabling to perform the design and performance calculations and also get bulk temperature variation along the length. Several case studies related to first part are conducted and performance as well as design calculations are discussed by Ahmet Unal [10]. The shell and tube heat exchanger with cross flow design is replaced by triple tube heat exchanger and overall effectiveness is calculated by Subramanian Arun S. [11]. Optimum diameters for minimum cost which includes running as well as fabrication cost are determined through numerical study by Achour Touatit et al. [12] for triple concentric tube heat exchanger. The heat transfer correlations are used from S. Kakac et al. [13] while designing the heat exchanger. The exit temperatures for three fluids in TCTHEx are determined by using method proposed by D. P. Seculic et al. [14].

II. DESIGN OF TCTHE

There are two types of design problems of TCTHE as rating and sizing problem. For an already selected construction type and flow arrangement (including the way of coupling the fluid streams), the sizing problem reduces to the determination of a physical size (length, of the heat exchanger). The tube diameters are selected as per standard and length of heat exchanger is calculated.

The Design Considerations

The heat exchanger is designed to cool oil from 85 to 55 with following considerations:

1. The inlet temperatures of water are same (30).
2. Flow arrangement is W-O-W in a counter current mode.
3. Thermo physical properties of fluids are considered as constant taken at bulk mean temperature.
4. Heat Exchanger operates under insulated condition, i.e. insulation is provided on the outermost surface.
5. The fouling in the heat exchanger is negligible.
6. The radiation heat loss from test-section to ambient is negligible.

Table 1. Geometric Parameters.

Parameter	***Symbol***	***Value***
Inner tube diameter	d_1	0.0381 m
Intermediate diameter	d_2	0.0508 m
Outer diameter	d_3	0.0762 m
Thickness of tube	t	0.0015 m

Table 2. Operating Parameters

Parameter	***Symbol***	***Value***
Load	Q	14.3 kW
Water1 inlet temperature	$T_{w1,in}$	30 °C
Oil (ISO VG 22) inlet temperature	$T_{o,in}$	85 °C
Water 3 inlet temperature	$T_{w3,in}$	30 °C
Oil outlet temperature	T_{oo}	55 °C
Volumetric flow rate of water	V_w	4 m^3/h
Volumetric flow rate of oil	V_o	1 m^3/h
Specific heat of water	C_{pw}	4180 J/kgK
Specific heat of oil	C_{po}	2040 J/kgK
Thermal conductivity of steel	k_{steel}	16 W/mK
Thermal conductivity of water	k_w	0.61 W/mK
Thermal conductivity of oil	k_o	0.12 W/mK
Dynamic viscosity of water	μ_w	0.0008 kg/ms
Dynamic viscosity of oil	μ_o	0.008 kg/ms

Logarithmic Mean Temperature Difference (LMTD) method is used for design of triple concentric tube heat exchanger. LMTD method uses the following steps:

1. Using energy balance, the unknown outlet temperatures of water through innermost tube and outer annulus are calculated. The ratio of hydraulic diameter for innermost tube is 1.5 times greater than the hydraulic diameter for outer annulus. So the flow rate of water (w_1) through innermost tube is taken 1.5 times the flow rate of water (w_3) through outer annulus while doing designing the heat exchanger for a fixed oil flow rate through inner annulus.

$$(\dot{m}C_p)_o(T_{o,in} - T_{o,out}) = (\dot{m}C_p)_{w1}(T_{w\,1,out} - T_{w1,in}) + (\dot{m}C_p)_{w3}(T_{w3,out} - T_{w3,in}) \quad (1)$$

2. The convective heat transfer coefficients for water are determined from physical properties of fluids using Gnielinsiki correlation [13],

$$Nu = \frac{\left(\frac{f}{8}\right)(\mathrm{Re} - 1000)\mathrm{Pr}}{1 + 12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} \quad (2)$$

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This correlation can also be used to estimate heat transfer coefficients for the range **Error! Reference source not found.** , especially when Reynolds number is closer to 10,000 than it is to 2300.
The convective heat transfer coefficient for developing laminar flow of oil is determined from correlation by Sieder and Tate (1936),

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

(3)

Where all properties are calculated at bulk mean temperature, except for**Error! Reference source not found.**, which is evaluated at the surface temperature

3. Two overall heat transfer coefficients are calculated based on outside area of the inner tube (U_1) and on inside area of intermediate tube (U_2).
4. Calculating the two LMTDs, the length of heat exchanger is calculated by using equation,

$$(\dot{m}C_p)_o(T_{o,in} - T_{o,out}) = U_1\pi d_{1,outside}\, L\Delta T_{lm,1} + U_2\pi d_{2,inside}\, L\Delta T_{lm,2} \quad (4)$$

5. By adding the entrance length to respective tube lengths calculated from energy balance, the final lengths for all three pipes are calculated and the Triple Tube Concentric Tube Heat Exchanger of length 2.7 m is fabricated with test section of 2 m length.

The present work further follows a process as mentioned in Figure 2:

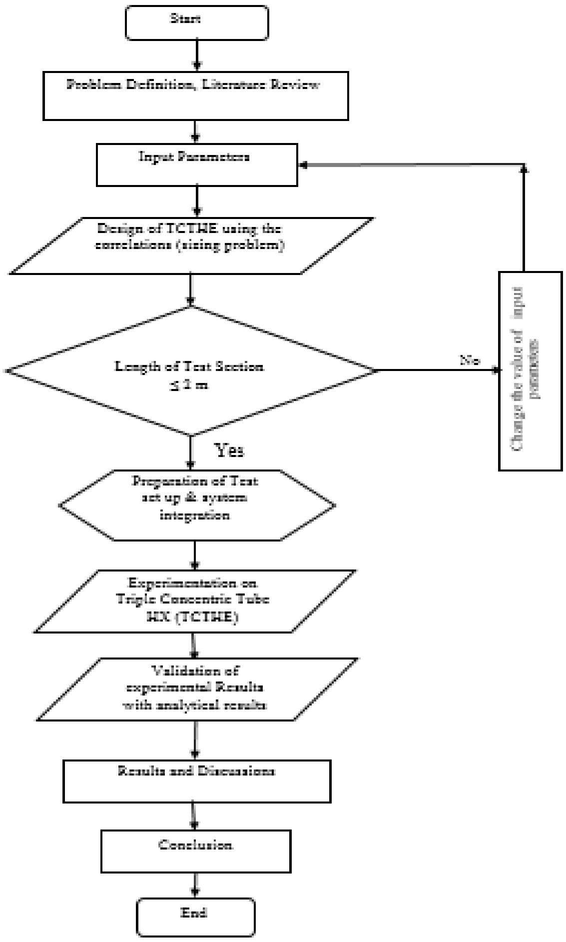

Figure 2: Methodology for Performance Evaluation of TCTHE

III. EXPERIMENTAL INVESTIGATIONS.

A triple concentric tube heat exchanger (TCTHE) is fabricated to cool oil (ISO VG 22) from 85 **Error! Reference source not found.** to 55**Error! Reference source not found.**using water as a coolant. The water supplied at ambient temperature (30℃), flows as a cooling fluid through inner tube as well as outer annulus and oil flows through inner annulus of the heat exchanger. The direction of flow of oil is opposite to that of water so as to achieve counter current flow condition. The fluids, oil and water are supplied from different tanks. Oil is heated initially to 85 **Error! Reference source not found.**by using oil heater of 6 kW capacity.

A thermostat and a controller unit are used to maintain the temperature of oil at 85**Error! Reference source not found.**. After attaining the required oil temperature, both the fluids are pumped separately through the specified flow passages with separate pump from respective tanks. The arrangement of flow of different fluids is called W-O-W (Water-Oil-Water) configuration of the heat exchanger as shown in Figure 1and Figure 3. Rotameter are used for measuring the volume flow rates of fluids entering the heat exchanger inlet. The flow rate is adjusted using the ball valve and by-pass mechanism and temperature readings are recorded at all the intermediate locations on the pipes along the length. Figure 4 and 5 shows schematic and actual experimental setup respectively. Total fifteen temperature sensors of PT100 class A type are used. Five temperature sensors are used along the length for each fluid as indicated in Table 3.

Table 3. Position of Thermocouple

Length (x) m	0	0.5	1	1.5	2
Position of Temperature sensor	1	2	3	4	5

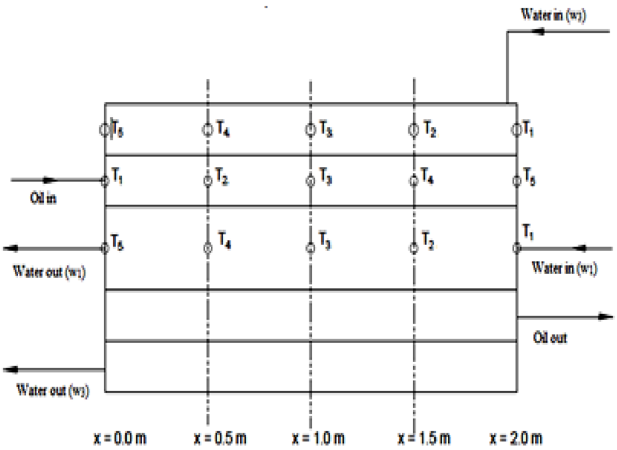

Figure 3. Position of Temperature sensors in TCTHE.

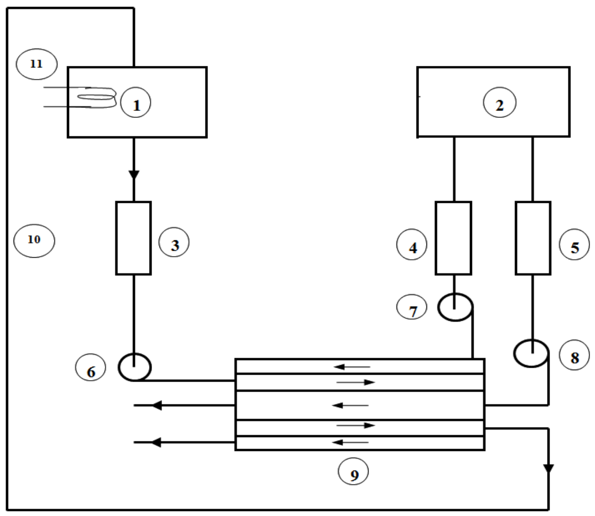

Figure 4. Schematic of Experimental Setup.

1. Oil tank, 2. Water tank, 3. Oil Rotameter, 4. Water (w_3) Rotameter, 5. Water (w_1) Rotameter, 6. Oil Pump, 7. Water (w_3) Pump, 8. Water (w_1) Pump, 9. Triple Concentric Tube Heat Exchanger (TCTHE), 10. Oil recirculation line, 11. Oil heating arrangement.

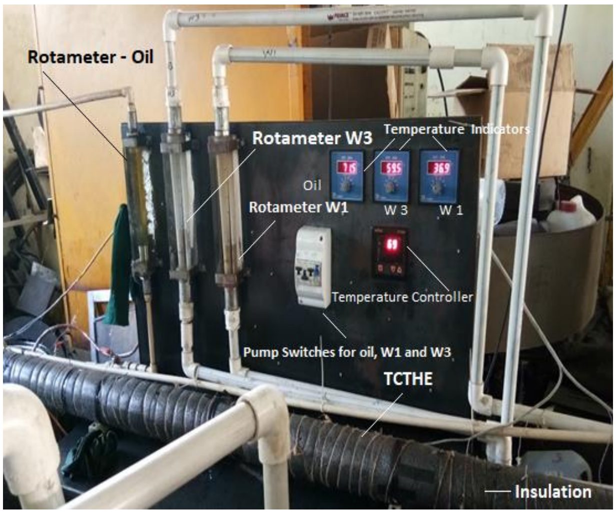

Figure 5. Actual experimental Setup.

IV. RESULTS AND DISCUSSION

The experimental analysis is carried out for TCTHE operating in counter current mode. In the analysis, temperature variation along the length of heat exchanger for three fluids is considered.

Figure 6 represents the theoretical and experimental temperature drop of oil in a total length of heat exchanger. Theoretical temperatures are determined by ε-NTU method explained by D. P. Seculic et al. [14].

Experimental results are validated by comparing values of difference between inlet and outlet temperature of hot fluid with theoretical temperature drop. Experimental results agree with theoretically calculated values with maximum variation of 2.6 %.

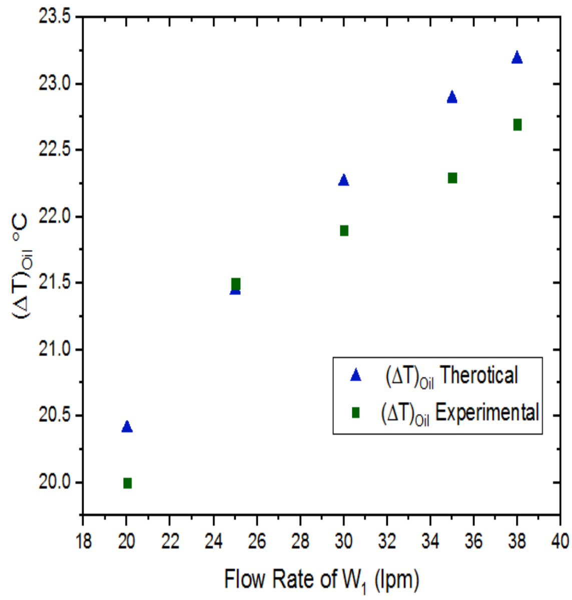

Figure 6. Degree of cooling for oil, @ oil at 85^{0}C and 16.5 *lpm*, w_3 at 10 *lpm*

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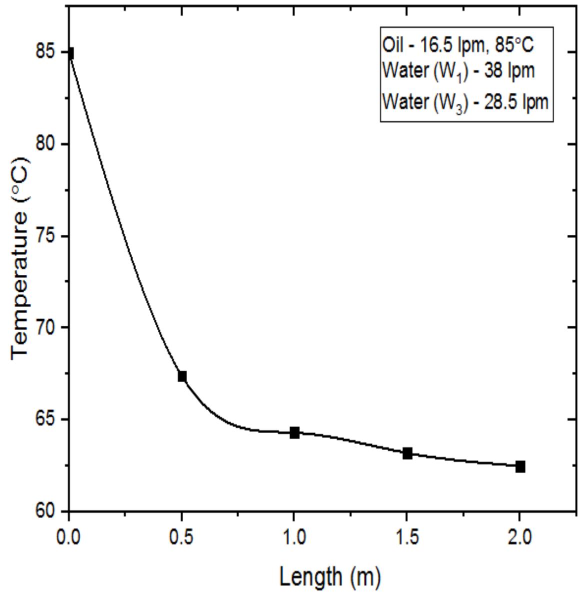

Figure 7. Temperature variation of oil along length.

Figure 7 represents the temperature variation of oil along length of heat exchanger. In the analysis, it is observed that 77% of oil temperature drops in first one fourth part of the heat exchanger whereas later part contributes to 23% temperature drop. This shows that high temperature oil performance is predominant in initial section.

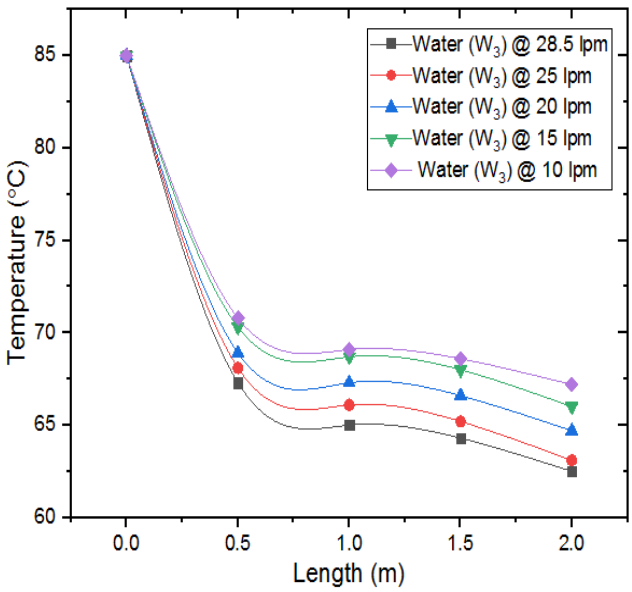

Figure 8. Temperature variation along length of oil (16.5 *lpm*) for w_1 @ 38 *lpm* and w_3 varied from 10 *lpm* to 28.5 *lpm*.

It is observed from figure 8 that as flow rate of w_3 decreases keeping oil flow rate and w_1 flow rate 16.5 lpm and 38 lpm respectively, the outlet *temperature* of oil increases indicating lesser degree of cooling. The drop in oil temperature is higher in first section of heat exchanger irrespective of flow rates of cooling fluids and keeping the same trend only change in slope of curve is observed. This indicates that the temperature variation for oil follows similar trend irrespective of flow rates of water. Similar trend is observed during experimental investigation by varying w_1 flow rates and keeping flow rates of w_3 and oil constant.

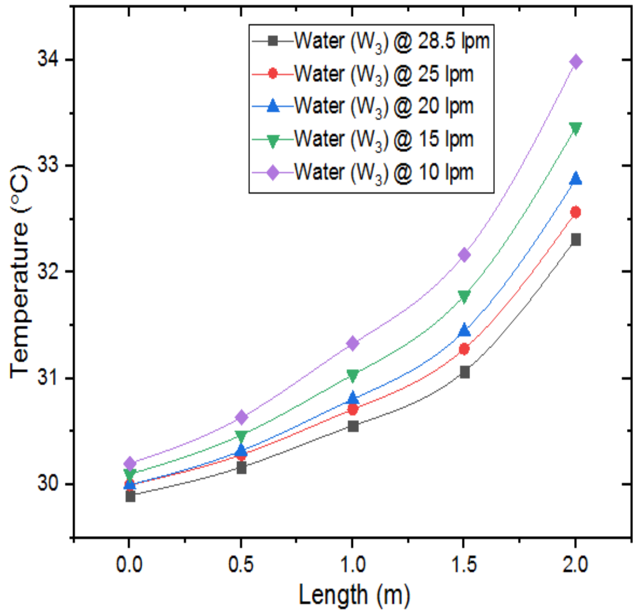

Figure 9. Temperature variation for water through innermost tube along length (in the direction of water flow).

Figure 9 shows that rise in temperature of water along the length (in the direction of water flow) is function of mass flow rate of water. At lower flow rates, rate of temperature rise for water (w_3) is more. This degree as well as rate of temperature rise decreases for increased flow rates of water (w_3). The degree of cooling is at rate 1.2 °C/ m in the initial section and 4 °C/ m for the last quarter length because of large temperature difference between hot and cold fluid in this section. Similar trend is recorded for change in temperature of water (w_1) with respect to varying flow rates of w_1 while oil as well as w_3 flow rates are kept constant.

Figure 10 shows the effect of inlet temperature of hot fluid on the heat transfer characteristics of TCTHE. The experiments are carried for different oil inlet temperatures as 85°C, 75°C and 65°C for the flow rate of hot fluid 16.5 *lpm* and by adjusting flow rates of w_1 as well as w_3 in such a way that both cooling liquids will flow with same Reynolds number. It is observed that as the inlet temperature of oil decreases, cooling rate also decreases at same Reynolds number. This shows that inlet temperature of hot fluid has considerable effect on rate of heat transfer which may be due to approached temperature difference between hot fluid (oil) and cold fluids (water) at the particular section

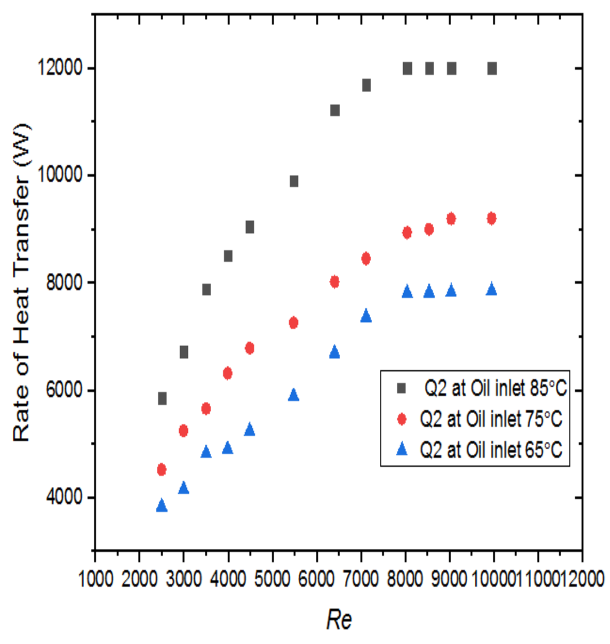

Figure 10. Rate of Heat Transfer from oil (Q_2) Vs Reynolds number for water ($Re_{w1} = Re_{w3}$) at different oil inlet temperatures.

With increase in Reynolds number heat transfer by oil (Q_2) to inner and outer water *also* increases. Slope of graph at lower Reynolds number is higher and it decreases with increase in Reynolds number. The curves become flatter at higher Reynolds numbers, i.e. flow rates of w_1 and w_3 which indicates that there is no predominant effect on heat transfer rate for rise in flow rate at higher values.

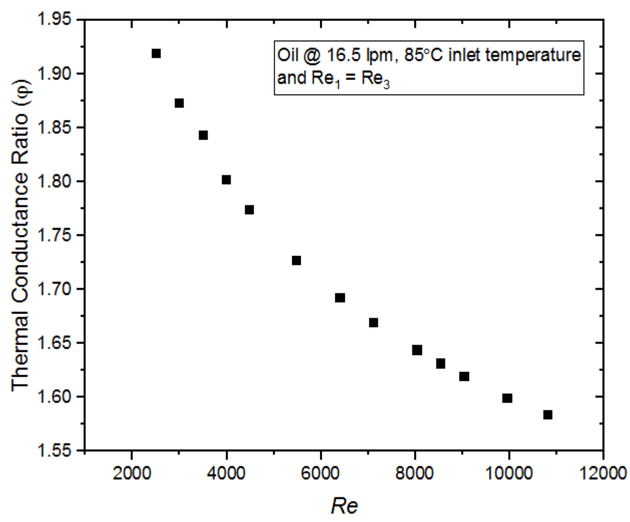

Figure 11. Thermal Conductance ratio Vs Reynolds Number.

Figure 11 represents relation between ratio of thermal conductance (φ) and Reynolds number of cold fluids ($Re_1 = Re_3$) keeping flow rate of hot fluid (oil) same.

Thermal Conductance ratio is expressed as,

$$\varphi = \frac{(UA)_{3,2}}{(UA)_{1,2}} \tag{5}$$

The ratio of thermal conductance (φ) indicates the relative ability of separating surfaces to transfer heat.

This ratio (φ) can be less than, equal to or greater than one. In the experimentation, it is observed that thermal conductance ratio (φ) decreases with increase in Reynolds number. Thermal conductance for outer annulus fluid is seen to be 1.5 to 2 times higher than that *for* fluid flow through innermost tube. It is also observed that the variation at low Reynolds number is significant as compared to that at high Reynolds numbers. This shows that conductance of outer cooling fluid is significant at low Reynolds numbers and the significance reduces as Reynolds number increases. This may be due to approach time for heat transfer for outer cold fluid is higher at low Reynolds numbers and at high Reynolds numbers the approach time of inner cooling fluid is predominant. Similar trend is observed for different oil inlet temperatures and different hot fluid flow rates.

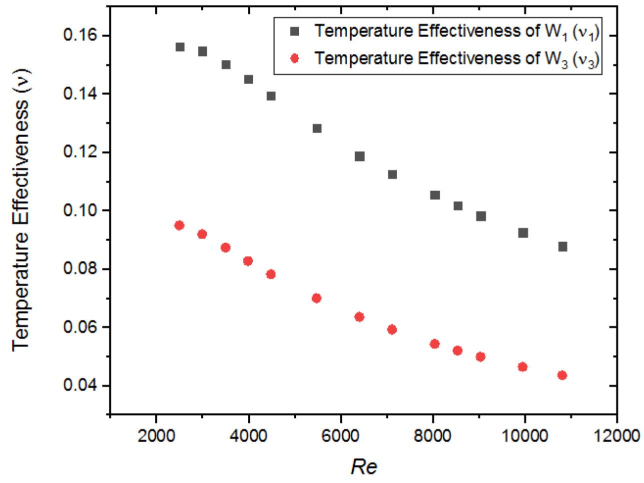

Figure 12. Temperature Effectiveness Vs Reynolds Number.

Triple Concentric Tube Heat Exchangers are analyzed with a parameter known as temperature effectiveness of inner and outer cooling fluid. It is represented by following expressions,

$$\nu_1 = \frac{(T_{out} - T_{in})_{w1}}{(T_{in})_{oil} - (T_{in})_{w1}} \tag{6}$$

And

$$\nu_3 = \frac{(T_{out} - T_{in})_{w3}}{(T_{in})_{oil} - (T_{in})_{w3}} \tag{7}$$

Figure 12 shows the variation of temperature effectiveness of cold fluid flows with respect to Reynolds number (Re_1=Re_3). From above graph, we can conclude that Temperature effectiveness of inner fluid as well as fluid through outer annulus flow decreases with increase in Reynolds number because of rising flow rate. At same Reynolds number, temperature effectiveness of inner cooling fluid is higher than temperature effectiveness of outer cooling fluid because of lower flow rate resulting in lower velocity and higher retainention time for a fluid passing through inner most tube than that of outer annulus for all Reynolds number. This gives more change in temperature for water (w_1) than that for water (w_3).

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V. UNCERTAINTY ANALYSIS

The uncertainty of the different parameters is calculated in the analysis of TCTHE on the basis of least count and the sensitivities of the measuring instruments used in the present investigation. Coleman et al. [15] and "Measurement uncertainty" ANSI/ASME [16] has proposed the uncertainty analysis for all experiments and by referring Kannadasan et al. (2012). The uncertainty for different geometric and operating parameters is shown in table no. 4.

Table 4. Uncertainty Calculated for measurement parameters.

Sr. No.	Quantity	Uncertainty	% Uncertainty
1	Area (A_1)	0.0104	1.04
2	Area (A_2)	0.00786	0.786
3	Area (A_3)	0.0052	0.52
4	Volume flow rate hot fluid	0.0166	1.66
5	Volume flow rate cold fluid	0.006	0.6
6	Overall Heat Transfer coefficient (U_1)	0.0697	6.97
7	Overall Heat Transfer coefficient (U_2)	0.0694	6.94
8	Reynolds Number (Re_1)	0.0120	1.2
9	Reynolds Number (Re_3)	0.00796	0.7

VI. CONCLUSION

This experimental investigation helps to understand thermal behavior of TCTHE under counter current flow, insulated conditions for two thermal communications. Oil cools drastically in first 0.5 m length (from x=0 m to x=0.5 m). Slope of cooling curve in this section is 36°C/m and it decreases along the length as 3.33 °C/m in the later part. In the analysis, it is observed that 77% of oil temperature drops in the initial quarter part of the heat exchanger whereas later part contributes to 23% temperature drop. Trend of oil cooling for different flow rates of water and oil remains same along the length. This may be due to sufficiently high temperature gradient between hot and cold fluids in the initial section of TCTHE. Water temperature increases in last 0.5 m section along the direction of water flow considerably because of counter flow arrangement. The heat transfer rates increase, with increase in flow rates of water (w_1 and w_3). Similar trend is observed for rate of heat transfer from oil to water with increase in Reynolds number of oil.

Temperature effectiveness of fluids decrease with increase in Reynolds number, as fluid retention time is inversely proportional to fluid flow rate. At same Reynolds number, more change in temperature for water (w_1) than that for water (w_3) is recorded resulting in temperature effectiveness of inner cooling fluid higher than temperature effectiveness of outer cooling fluid. The reason behind this is lower flow rate resulting in lower velocity and higher retention time for a fluid passing through inner most tube than that for outer annulus at all Reynolds numbers. Thermal conductance for outer annulus fluid is seen to be 1.5 to 2 times higher than that for fluid flow through innermost tube due to fact that the fluid contact surface area for outer annulus fluid is greater than inner tube fluid.

NOMENCLATURE

A	*Area, m^2*
$d_{1,inside}$	*Inner tube inside diameter, m*
$d_{1,outside}$	*Inner tube outside diameter, m*
$d_{2,inside}$	*Intermediate tube inside diameter, m*
$d_{2,outside}$	*Intermediate tube outside diameter, m*
$d_{3,inside}$	*Outer tube inside diameter, m*
$d_{3,outside}$	*Outer tube outside diameter, m*
t	*Thickness of tube, m*
w_1	*water flowing through inner tube*
w_3	*water flowing through outer annulus*
f	*Friction factor*
h	*Heat transfer coefficient, W/m^2K*
k	*Thermal conductivity, W/mK*
L	*Length, m*
lpm	*Liters per minute*
$\dot{m}$	*Mass flow rate, Kg/s*
NTU	*Number of heat transfer units*
Nu	*Nusselt number*
Pr	*Prandtl number*
Re	*Reynolds number*
T	*Temperature,℃*
U_1	*Overall heat transfer coefficient between water through inner tube and oil, W/m^2K*
U_2	*Overall heat transfer coefficient between water through outer annulus and oil, W/m^2K*
	Temperature effectiveness
C_p	*Specific heat at constant pressure, J/kgK*

Greek Letters

Δ	*Difference*
ε	*Effectiveness of Heat exchanger*
φ	*Thermal conductance ratio*
μ	*Dynamic viscosity,*
ρ	*Density,*

Subscripts

i	*Inner diameter*
in	*Inlet*
lm	*Logarithmic mean*
out	*Outlet*
w	*Water*
o	*Oil*
$w_{1,in}$	*Water at inlet through inner tube*
$w_{1,out}$	*Water at outlet through inner tube*
$w_{3,in}$	*Water at inlet through outer annulus*
$w_{3,out}$	*Water at outlet through outer annulus*

REFERENCES

1. G.A. Quadir, N. J. Salman Ahmed and Irfan Anjum Badruddin, "Experimental investigation of the performance of a triple concentric pipe heat exchanger." International Journal of Heat and Mass Transfer, 2014, Vol. 62, pp.562-566.
2. G.A. Quadir, Saqab S. Jarallah, N. J. Salman Ahmed and Irfan Anjum Badruddin, "Numerical investigation of the performance of a triple concentric pipe heat exchanger" International Journal of Heat and Mass Transfer, 2014, Vol. 75, pp. 165-172.
3. O. García-Valladares, "Numerical simulation of triple concentric-tube heat exchangers" International Journal of Thermal Sciences, 2004, Vol. 43, pp. 979-991.
4. Valerio Giovannoni, Rajnish N. Sharma and Robert R. Raine, "Numerical prediction of thermal performances in a concentric triple tube heat exchanger" International Journal of Thermal Sciences, 2017, Vol. 120, pp. 86-105.
5. Carlos A. Zuritz "On the design of triple concentric-tube heat exchangers" Journal of Food Process Engineering, 1990, Vol. 12, pp. 113-130.
6. Abdulla Gomaa, M. A. Halim and Ashraf Mimi Elsaid , "Experimental and numerical investigations of triple concentric-tube heat exchanger" Applied Thermal Engineering, 2015, Vol. 7467, pp. 1-32.
7. Ahmet Unal, "Effectiveness-NTU relations for triple concentric-tube heat exchanger" International Communications in Heat Mass Transfer, 2003, 30, Vol. No. 2, pp. 261-272.
8. Ediz Batmaz and K. P. Sandeep, "Calculation of overall heat transfer coefficients in a triple tube heat exchanger", Heat Mass Transfer, 2005, Vol. 41, pp. 271–279.
9. Ahmet Unal, "Theoretical analysis of triple concentric-tube heat exchangers Part 1: Mathematical Modelling" International Communications in Heat Mass Transfer, 1998, Vol. 25 No. 7, pp. 949-958.
10. Ahmet Unal, "Theoretical analysis of triple concentric-tube heat exchangers Part 2: Case Studies", International Communications in Heat Mass Transfer, 2001, Vol. 28, No. 2, pp. 243-256.
11. Arun Subramanian S., "Design and Fabrication of Automotive Triple Tube Heat Exchanger (with Guided Sinusoidal Motion)", International Journal for Research in Applied Science and Technology, 2017, Volume 5, Issue XII, pp. 695- 704.
12. Achour Touatit and Cherif Bougriou, "Optimal Diameters of triple concentric-tube heat exchangers", International Journal of Heat and Technology, 2018, Vol. 36, No. 1, pp. 367-375.
13. S. Kakac and H. Liu, "Heat Exchangers, Selection, Rating, and Thermal Design", CRC Press, New York, pp. 283-328, 1998.
14. D. P. Seculic, R. K. Shah, "Thermal Design Theory of Three-Fluid Heat Exchangers", Advances in Heat Transfer, Volume 26, pp. 219-328, 1995.
15. H.W. Coleman, W.G. Steele, "Experimental and Uncertainty Analysis for Engineers", Wiley, New York, 1989.
16. ANSI/ASME, "Measurement uncertainty", PTC 19, pp. 1–1985, 1986.

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