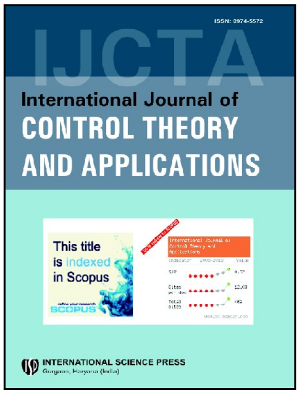

International Journal of Control Theory and Applications

ISSN : 0974-5572

Volume 10 • Number 10 • 2017

Design Validation through Testing of a Double-tube Counter-flowheat Exchanger

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Abstract: To examine the validity of empirical relations for design, a comparative study is made considering the test data of a double-tube counter-flow heat exchanger. The outlet temperatures of the cold and hot fluids in the heat exchangerhave beenevaluated directly by adoptingeffectivenessmethod.The overall heat transfer coefficientis evaluated considering the dimensions and material properties of tubes, properties of hot and cold fluids, inlet temperatures, mass-flow rates and the fouling factors.The estimates of outlet temperatures are matching well with measured ones. This study confirms the validation of empirical relations for use in the design as well as in the performance evaluation of double-tube counter-flow heat exchangers in service.

Keywords: Double-tube counter-flow heat exchanger; effectiveness; friction factor; inlet and outlet temperatures;mass flow rate; Nusselt number; overall heat transfer coefficient;Prandtl number; Reynolds number.

1. INTRODUCTION

Heat exchangers are classified according to transfer process, construction, flow arrangement (viz.,parallel flow, counter flow, single-pass cross flow and multi-pass cross flow.), surface compactness, number of fluids and heat transfer mechanisms [1]. They have been developed for use in the steam power plants, chemical processing plants, heating and air conditioning in buildings, household refrigerators, car radiators, radiators for space vehicles and so on. The design of heat exchangers involves cost, size, weight and economic considerations [2]. In steam and chemical power plants, cost of heat exchanger is the main concern, whereas, the size and weight are the dominant factors in radiators. High efficiency gasket plate heat exchangers (PHE) are also being used in free cooling, cooling tower isolation, water heaters, waste heat recovery, heat pump isolation and thermal (ice) storage systems. These are cost effective, simple and compact in size, which can be cleaned easily and no extra space requirement for dismantling.The heat exchanger life assessment system has been developed to estimate the ultrasonic immersion length for conversion to the corrosion depth inside cooling water/air fin type tubes [3]. Heat exchangers are of expendable type having limited life, which are usually designed for 10 years. Fouling is the most common problem that is very difficult to identify. It is preferred to have the maximum flow rate with increased turbulence within the channel to retard the rate of fouling. It is well known fact that the actual design of heat exchangers is involved when compared to the heat transfer analysis alone due to cost, weight, size and economic considerations [4].Experiments have been performed by several researchers and examined the performance of heat exchangers utilizing various empirical relations for friction factor and Nusselt number [5-8].

The computational fluid dynamics (CFD) codes in use are FLUENT, CFX, STAR CD, FIDAP, ADINA, CFD2000 and PHOENICS [9]. Compared to the correlation based methods, the use of CFD in heat exchanger design is limited due to the requirement of large amounts of computer power, computer memory and computational time [10-12]. The CFD developmental activities are progressing rapidly to become an integral part of all design processes. However, there is a need for comparison of CFD simulations with experimental data due to lack of universally applicable boundary conditions and turbulent models [13-16] and design validation through testing is unavoidable.

To design or to assess the performance of a double-tube counter-flow heat exchanger, the total heat transfer rate has to be related to the inlet and outlet fluid temperatures, the overall heat transfer coefficient, and the heat transfer surface area. There is a need for identifying the appropriate empirical relations from the existing various relations. Potrascioiu and Radulescu [17] have performed experiments on the double-tube heat exchangers. For the specified constant inlet temperatures of cold and hot fluids, they measured the outlet temperatures of the cold and hot fluids in a double-tube heat exchanger and compared the test data with complex iterative numerical simulations. Motivated by the work of above researchers, this paper follows a simplified approach for direct evaluation of outlet temperatures of hot and cold fluids in a double-tube counter-flow heat exchanger (see Figure 1) using effectivenessmethod. From the inlet parameters of a typical double-tube counter-flow heat exchanger, heat transfer coefficient (h), overall heat transfer coefficient (U_m), pressure drop (ΔP) and effectiveness (ε) are evaluated using the identified empirical relations. The estimated outlet temperatures are found to be in good agreement with existing test data [17].

2. EMPIRICAL RELATIONS FOR DESIGN

The rate of heat transfer $(\dot{Q})$ is given by [18],

$$\dot{Q} = m_h C_{ph} (T_{hi} - T_{ho}) = m_c C_{pc} (T_{co} - T_{ci}) = A_t U_m \Delta T_{\ln} \tag{1}$$

Here, m_c and m_h are the mass flow rates of cold and hot fluids respectively; C_{pc} and C_{ph} are the specific heats of cold and hot fluids respectively; T_{hi} and T_{ci} are the inlet temperatures of hot and cold fluids respectively; T_{ho} and T_{co} are the outlet temperatures of hot and cold fluids respectively; $A_t = \pi L(D_o + d_o)$, is the total heat transfer area.

Figure 1: A typical double-tube counter-flow heat exchanger (A) Inlet of cold fluid; (B) Outlet of hot fluid; (C)Inlet of hot fluid; (D) Outlet of cold fluid

The logarithmic mean temperature difference between the hot and cold fluids,

$$\Delta T_{\ln} = \frac{\Delta T_0 - \Delta T_L}{\ln\left(\dfrac{\Delta T_0}{\Delta T_L}\right)} \tag{2}$$

where $\Delta T_0 = T_{hi} - T_{co}$ and $\Delta T_L = T_{ho} - T_{ci}$ (see Figure 2).

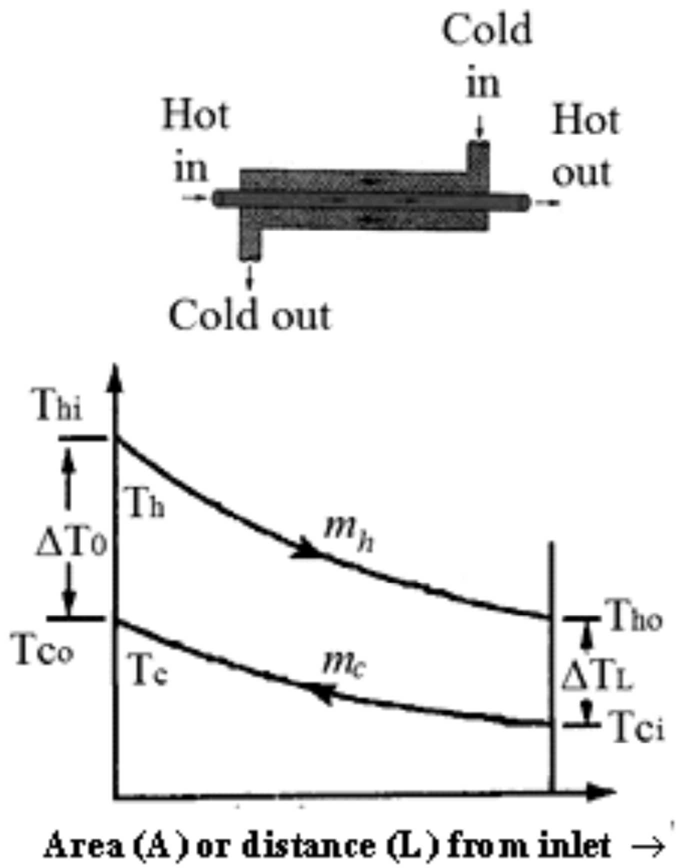

Figure 2: Temperature difference along the length in counter-flow heat exchangers

For the case of equal mass flow rates of the cold and hot fluids, equation (2) yields: $\Delta T_{\ln} = \Delta T_0 = \Delta T_L$. The outlet temperatures of the cold and hot fluids can be estimated directly for the case of equal mass-flow rates from

$$T_{ho} = \left\{T_{ci} + \frac{m_h C_{ph}}{A_t U_m} T_{hi}\right\}\left(1 + \frac{m_h C_{ph}}{A_t U_m}\right)^{-1} \tag{3}$$

$$T_{co} = \left\{T_{hi} + \frac{m_c C_{pc}}{A_t U_m} T_{ci}\right\}\left(1 + \frac{m_c C_{pc}}{A_t U_m}\right)^{-1} \tag{4}$$

The overall heat transfer coefficient (U_m) can be obtained from

$$\frac{1}{U_m} = \frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o}{2\kappa}\ln\left(\frac{d_o}{d_i}\right) + R_{fo} + \frac{1}{h_o}, \tag{5}$$

in which d_i and d_o are the inner and outer diameters of the inner tube respectively; $R_{fi} = 0.000172 m^2K/W$ and $R_{fo} = 0.000352 m^2K/W$ are the fouling resistances corresponding to inner and outer tubes; $\kappa = 52 W/m.K$ is the thermal conductivity of the tube material (AISI 316 Stainless steel) [19]; $h_i = \frac{Nu_h \kappa_f}{d_i}$ and $h_o = \frac{Nu_c \kappa_f}{D_e}$ are the heat transfer coefficients for hot and cold fluids respectively. The equivalent diameter, $D_e = \frac{D_i^2 - d_o^2}{d_o}$. D_i is the inner diameter of the outer tube. The Nusselt numbers Nu_h and Nu_c corresponding to the hot and cold fluids are obtained from

$$Nu = \frac{f \operatorname{Re} \Pr}{2 + 12.3037\sqrt{f}(\Pr - 1)} \tag{6}$$

Defining the average velocity, $u_{mh} = \frac{\dot{m}_h}{\rho_h A_c}$ and the cross-sectional area, $A_c = \frac{\pi}{4} d_i^2$, the Reynolds number for hot side fluid, $\operatorname{Re} = \frac{\rho u_{mh} d_i}{\mu} = \frac{4\dot{m}_h}{\mu \pi d_i}$. Similarly, definingthe hydraulic diameter, $D_h = D_i - d_o$, average velocity, $u_{mc} = \frac{\dot{m}_c}{\rho_c A_c}$ and the cross-sectional area, $A_c = \frac{\pi}{4}\left(D_i^2 - d_o^2\right)$, the Reynolds number for cold side fluid, $\operatorname{Re} = \frac{\rho u_{mc} D_h}{\mu}$. The friction factor ($f$), for hot and cold fluids: $f = \frac{16}{\operatorname{Re}}$ for Re<2000 and $f = (1.56 \ln \operatorname{Re} - 3)^{-2}$ for Re>2000. The Prandtl number (Pr) is at film temperature.

In order to simplify the complex iterative evaluation procedure for the outlet temperatures of the cold and hot fluids, the effectiveness (ε) is defined as the ratio of actual heat transfer rate $(\dot{Q})$ to the maximum possible heat transfer rate $(\dot{Q})_{\max}$, which can be written as [20-24]

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{m_h C_{ph}(T_{hi} - T_{ho})}{(mC_p)_{\min}(T_{hi} - T_{ci})} = \frac{m_c C_{pc}(T_{co} - T_{ci})}{(mC_p)_{\min}(T_{hi} - T_{ci})} \tag{7}$$

From equation (7), one can write

$$T_{ho} = T_{hi} - \varepsilon \frac{(mC_p)_{\min}}{m_h C_{ph}} (T_{hi} - T_{ci}) \tag{8}$$

$$T_{co} = T_{ci} + \varepsilon \frac{(mC_p)_{\min}}{m_c C_{pc}} (T_{hi} - T_{ci}) \tag{9}$$

Here $(mC_p)_{\min}$ is the smaller of $m_h C_{ph}$ and $m_c C_{pc}$ for the hot and cold fluids.

Using equations (8) and (9), one can find

$$\Delta T_0 = \left[1 - \frac{\varepsilon (mC_p)_{\min}}{m_c C_{pc}}\right] (T_{hi} - T_{ci}) \tag{10}$$

$$\Delta T_L = \left[1 - \frac{\varepsilon (mC_p)_{\min}}{m_h C_{ph}}\right] (T_{hi} - T_{ci}) \tag{11}$$

From Ref. [18], it is noted that

$$\frac{\Delta T_L}{\Delta T_0} = \exp(-M A_t U_m) \tag{12}$$

Using equations (10) and (11) in equation (12), one gets the effectiveness,

$$\varepsilon = \frac{1 - \exp(-M A_t U_m)}{(mC_p)_{\min} \left\{ \frac{1}{m_C C_{pc}} - \frac{\exp(-M A_t U_m)}{m_h C_{ph}} \right\}} \; ; M = \frac{1}{m_c C_{pc}} - \frac{1}{m_h C_{ph}} \tag{13}$$

The effectiveness (ε) in equation (13) is defined in terms of the parameters A_t, U_m, $m_h C_{ph}$ and $m_c C_{pc}$, which can be found easily for the specified inlet temperatures of the hot and cold fluids. Later on, the outlet temperature of the hot and cold fluids can be determined directly using equations (8) and (9).

The pressure drop (ΔP_h) and (ΔP_c) in hot and cold fluid tubes are evaluated from [19]

$$\Delta P_h = 4 f \frac{L}{d_i} N_{hp} \frac{\rho u_{mh}^2}{2} \tag{14}$$

$$\Delta P_c = 4 f \frac{L}{D_h} N_{hp} \frac{\rho u_{mc}^2}{2} \tag{15}$$

L is the length of the heat exchanger (m) and the number of hair pins, N_{hp} =1. It should be noted that the friction factor (f) should be evaluated corresponding to the Reynolds number (Re) of hot and cold side fluids.

3. ANALYSIS

A double-tube heat exchanger (see Figure 1) consists of a tube placed concentrically inside another tube of a larger diameter with appropriate fittings to direct the flow of one fluid through inner tube, and the other through the annular space. The double-tube heat exchangers are being used for sensible heating or cooling of process fluids in small heat transfer areas. Potrascioiu and Radulescu [17] have performed experiments on the double-tube heat exchangers for the specified constant inlet temperatures of cold and hot fluids and measured the outlet temperatures of the cold and hot fluids. They have compared the experimental data with their numerical simulations and claimed that their model can be used for designing industrial double-tube heat exchangers in the refineries. Their mathematical model contains an equation of the heat balance associated to hot and cold fluids, mass-flow rates, inlet and unknown outlet temperatures. They also equated the rate of heat transfer $(\dot{Q})$ with the product of the overall heat transfer coefficient (U_m), total heat transfer area (A_t) and the Logarithmic Mean Temperature Difference $(\Delta T_{\ln})$. They have obtained two equations for the unknown outlet temperatures of the cold and hot fluids which are non-linear in nature and solved them through an iterative process.Two case studies have been made and compared their numerical simulations with the measured outlet temperatures of cold and hot fluids. In case-I, tests are conducted maintaining constant inlet temperatures of cold fluid and varying inlet temperatures of the hot fluid. In case-II, tests are conducted maintaining constant inlet temperatures of the hot fluid and varying inlet temperatures of the cold fluid.

The heat exchanger has four ports, two for inlet and two for outlet of the hot and cold fluids. For the specifications andoperating conditions of a typical double-tube counter-flow heat exchanger in Table 1, the fluid properties at film temperature $\left(T_f = \dfrac{T_{ci} + T_{hi}}{2}\right)$ are generated for hot and cold fluids and presented in Tables 2 and 3. From the geometrical and fluid properties, mass flow rate and inlet temperature of hot and cold fluids, one can find the maximum possible heat transfer rate $(\dot{Q})_{\max}$, the Reynolds number (*Re*), and the Prandtl number (*Pr*). The friction factor (*f*) corresponding to the Reynolds number, *Re* (Re < 2000 for laminar flow regime, whereas for the turbulent flow regime, *Re*> 2000) is evaluated. Estimating the heat transfer coefficientsh_i and $h_o$for hot and cold fluids, the overall heat transfer coefficient is calculated from equation (5). The effectiveness $(\varepsilon)$is found from equation (13). The outlet temperature of the hot and cold fluids is estimated from equations (8) and (9).

Tables 4 and 5give the measured inlet and outlet temperatures of hot and cold fluids, estimated outlet temperatures, overall heat transfer coefficients (U_m), effectiveness (ε) and pressure drop (ΔP). The present analysis results (in Tables 4 and 5) show a good comparison of measured and estimated outlet temperatures of the hot and cold fluids.Figures 3 and 4 show the comparison of measured and estimated outlet temperatures of the cold and hot fluids. Regarding the variation of temperatures of hot and cold fluids in the counter-flow heat exchangers (in which the hot and cold fluids enter from opposite ends as in Figure-2), the inlet temperature of the hot fluid (T_{hi}) is higher than the inlet temperature of the cold fluid (T_{ci}), whereas the outlet temperature of the cold fluid (T_{co}) increases and that of the hot fluid (T_{ho}) decreases for both case-I and Case-II tests [see Tables 4 and 5]. The present analysis results are found to be close to the test resultsof the outlet temperature of hot and cold fluids [17], when compared to those of Ref. [17] from the complex iterative numerical simulations.In Case-I tests for the specified mass flow rates and the range of inlet temperature of hot fluids with 10.6^{0}C inlet temperature of cold fluids, the effectiveness varies from 0.179 to 0.217; overall heat transfer coefficient varies from393 to 517 W/m^2.K; the variation of pressure difference for hot fluids is from 0.587 to 0.659 kPa, whereas that of cold fluids varies from 0.188 to 0.277 kPa.In Case-II tests for the specified mass flow rates and the range of inlet temperature

Table 1
Specifications and operating conditions of a typical double-tube counter-flow heat exchanger [17]

Case-I: Constant inlet temperature of cold fluid and varying inlet temperature of hot fluid.

Specifications		*Operating conditions*
Outer pipe	*Inner pipe*	Temperature of hot fluid (T_h) = 70.2-72.7°C
Outer diameter (D_o) = 0.022m	Outer diameter (d_o) = 0.012m	Temperature of cold fluid (T_c) = 10.6°C
Inner diameter (D_i) = 0.020m	Inner diameter (d_i) = 0.010m	Mass flow rate of hot fluid $(\dot{m}_h)$ = 0.0380-0.0405kg/sec
Length of the heat exchanger (L) = 0.75m		Mass flow rate of cold fluid $(\dot{m}_c)$ = 0.0333-0.0527kg/sec

Case-II: Constant inlet temperature of hot fluid and varying inlet temperature of cold fluid.

Specifications		*Operating conditions*
Outer pipe	*Inner pipe*	Temperature of hot fluid (T_h) = 55.3°C
Outer diameter (D_o) = 0.028m	Outer diameter (d_o) = 0.014m	Temperature of cold fluid (T_c) = 10.7-11.8°C
Inner diameter (D_i) = 0.026m	Inner diameter (d_i) = 0.012m	Mass flow rate of hot fluid $(\dot{m}_h)$ = 0.0527-0.0680kg/sec
Length of the heat exchanger (L) = 0.935m		Mass flow rate of cold fluid $(\dot{m}_c)$ = 0.0255-0.0611kg/sec

Table 2
Fluid properties at film temperature, T_f(°C) for case-I operating conditions

$\dot{m}_h (kg/sec)$	$\dot{m}_c (kg/sec)$	T_{hi} (°C)	T_{ci} (°C)	$\rho\left(\frac{kg}{m^3}\right)$	$K_f\left(\frac{W}{m,K}\right)$	$\vartheta_K\left(\frac{m^2}{s}\right)$	$C_p\left(\frac{J}{kg.K}\right)$	Pr
0.0405	0.0333	70.4	10.6	991.62	0.63308	6.35E-07	4067.2	4.0441
0.0400	0.0361	70.2	10.6	991.65	0.63296	6.36E-07	4067.2	4.0525
0.0405	0.0388	70.2	10.6	991.65	0.63296	6.36E-07	4067.2	4.0525
0.0394	0.0416	70.5	10.6	991.60	0.63314	6.34E-07	4067.2	4.0399
0.0388	0.0444	71.1	10.6	991.48	0.63351	6.31E-07	4067.1	4.0148
0.0394	0.0444	71.0	10.6	991.50	0.63345	6.31E-07	4067.2	4.0190
0.0386	0.0472	71.4	10.6	991.42	0.63369	6.29E-07	4067.1	4.0024
0.0391	0.0472	71.2	10.6	991.46	0.63357	6.30E-07	4067.1	4.0107
0.0380	0.0500	72.0	10.6	991.31	0.63405	6.26E-07	4067.1	3.9778
0.0380	0.0527	72.7	10.6	991.17	0.63447	6.22E-07	4067.0	3.9493

[http://www.mhtl.uwaterloo.ca/old/onlinetools/airprop/airprop.html]

Table 3
Fluid properties at film temperature, T_f(°C) for case-II operating conditions

$\dot{m}_h (kg/sec)$	$\dot{m}_c (kg/sec)$	T_{hi} (°C)	T_{ci} (°C)	$\rho\left(\frac{kg}{m^3}\right)$	$K_f\left(\frac{W}{m.K}\right)$	$\vartheta_K\left(\frac{m^2}{s}\right)$	$C_p\left(\frac{J}{kg.K}\right)$	Pr
0.0527	0.0255	55.3	11.8	994.07	0.62423	7.26E-07	4068.9	4.7031
0.0583	0.0255	55.3	11.8	994.07	0.62423	7.26E-07	4068.9	4.7031
0.0597	0.0255	55.3	11.7	994.09	0.62416	7.27E-07	4068.9	4.7085
0.0638	0.0255	55.3	11.7	994.09	0.62416	7.27E-07	4068.9	4.7085
0.0527	0.0277	55.3	11.6	994.10	0.62410	7.27E-07	4068.9	4.7138
0.0611	0.0358	55.3	11.5	994.12	0.62403	7.28E-07	4068.9	4.7192
0.0638	0.0511	55.3	11.0	994.20	0.62370	7.32E-07	4069.0	4.7462
0.0666	0.0611	55.3	10.9	994.22	0.62363	7.32E-07	4069.0	4.7516
0.0680	0.0486	55.3	10.7	994.25	0.62350	7.34E-07	4069.1	4.7625

[http://www.mhtl.uwaterloo.ca/old/onlinetools/airprop/airprop.html]

Table 4
Overall heat transfer coefficient (U_m), effectiveness (ε), hot and cold fluid outlet temperatures (T_{ho}, T_{co}) and pressure drop (ΔP) evaluated from the test data of a typical double-tube counter-flow heat exchanger for case-I inlet conditions

Inlet and outlet temperatures of hot and cold fluids (°C) [17]				*Present analysis*					
T_{hi}	T_{ho}	T_{ci}	T_{co}	$U_m(W/m^2K)$ *Eqn.(5)*	ε *Eqn.(13)*	T_{ho} *Eqn.(8)*	T_{co} *Eqn.(9)*	$\Delta P_h(k.Pa)$ *Eqn.(14)*	$\Delta P_c(k.Pa)$ *Eqn.(15)*
70.4	61.4	10.6	24.2	392.99	0.191	60.97	22.06	0.659	0.130
70.2	61.2	10.6	23.0	411.48	0.185	60.24	21.62	0.643	0.148
70.2	61.2	10.6	22.4	429.62	0.179	59.94	21.29	0.659	0.167
70.5	60.8	10.6	22.0	447.70	0.183	59.49	21.01	0.628	0.188
71.1	61.2	10.6	21.4	465.87	0.193	59.40	20.83	0.612	0.209
71.0	61.2	10.6	21.4	465.64	0.190	59.47	20.82	0.627	0.209
71.4	61.3	10.6	20.9	482.74	0.201	59.16	20.60	0.604	0.231
71.2	61.2	10.6	20.9	482.26	0.198	59.16	20.57	0.619	0.231
72.0	61.7	10.6	20.4	499.78	0.210	59.06	20.44	0.588	0.253
72.7	62.3	10.6	20.0	516.58	0.217	59.20	20.33	0.587	0.277

Table 5
Overall heat transfer coefficient (U_m), effectiveness (ε), hot and cold fluid outlet temperatures (T_{ho}, T_{co}) and pressure drop (ΔP) evaluated from the test data of a typical double-tube counter-flow heat exchanger for case-II inlet conditions

Inlet and outlet temperatures of hot and cold fluids (°C) [17]				*Present analysis*					
T_{hi}	T_{ho}	T_{ci}	T_{co}	$U_m(W/m^2K)$ *Eqn.(5)*	ε *Eqn.(13)*	T_{ho} *Eqn.(8)*	T_{co} *Eqn.(9)*	$\Delta P_h(k.Pa)$ *Eqn.(14)*	$\Delta P_c(k.Pa)$ *Eqn.(15)*
55.3	49.2	11.8	24.5	166.50	0.172	51.67	19.29	0.566	0.020
55.3	49.5	11.8	25.1	166.51	0.172	52.00	19.32	0.672	0.020
55.3	49.6	11.7	25.2	166.45	0.172	52.07	19.24	0.700	0.020
55.3	49.9	11.7	25.4	166.46	0.173	52.27	19.26	0.787	0.020
55.3	49.4	11.6	23.3	170.99	0.163	51.54	18.73	0.566	0.022
55.3	49.2	11.5	22.0	185.39	0.139	51.71	17.61	0.729	0.028
55.3	48.5	11.0	19.7	278.96	0.144	50.19	17.38	0.788	0.070
55.3	48.2	10.9	18.6	312.95	0.135	49.79	16.90	0.849	0.094
55.3	47.5	10.7	17.7	269.44	0.146	50.61	17.25	0.880	0.065

of cold fluids with 55.3^{0}C inlet temperature of hot fluids, the effectiveness varies from 0.135 to 0.173; overall heat transfer coefficient varies from 166 to 313 W/m^2.K; the variation of pressure difference for hot fluids is from 0.566 to 0.880 kPa, whereas that of cold fluids varies from 0.020 to 0.094 kPa. The values of effectiveness and overall heat transfer coefficient are found to be low in Case-II tests, when compared to those values in Case-I tests.

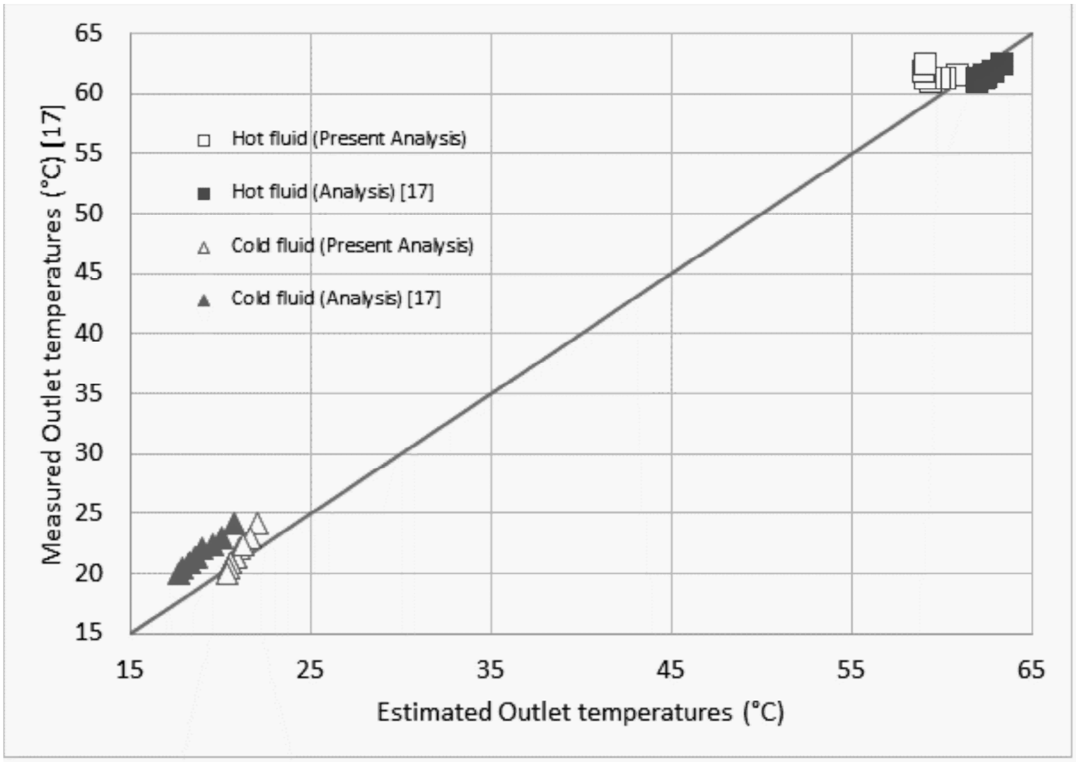

Figure 3: Comparison of outlet temperatures of hot and cold fluids for case-I inlet conditions

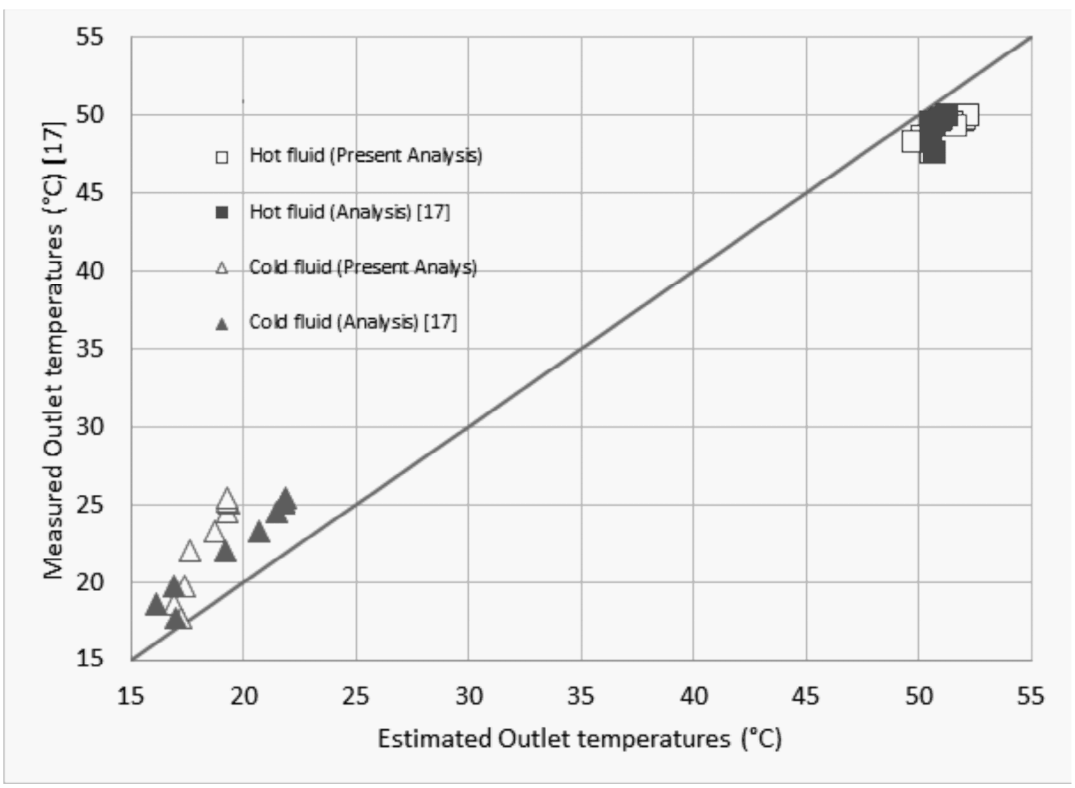

Figure 4: Comparison of outlet temperatures of hot and cold fluids for case-II inlet conditions

4. CONCLUDING REMARKS

Appropriate empirical relations are identified for assessing the efficacy of the double-tube counter-flow heat exchanger through comparison of test data. Effectiveness method is used for direct evaluation of the outlet temperatures of the hot and cold fluids in the heat exchanger. The empirical relations are useful in the design as well as in the performance evaluation of double-tube counter-flow heat exchangers in service.Future work is directed towards the selection of optimum input parameters such as mass flow rates of hot and cold fluids (m_h, m_c); inlet temperatures of hot and cold fluids (T_{hi}, T_{ci}) to achieve maximum heat transfer rate $(\dot{Q})$ and minimum pressure drop (ΔP) by adopting the Taguchi method [25-27] and performing minimum number of experiments.

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