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Experimental and Exergy Analysis of A Double Pipe Heat Exchanger for Counter Flow Arrangement

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ABSTRACT
This paper presents For Experimental and Exergy Analysis of a Double Pipe Heat Exchanger for counter flow Arrangement.. The present work is taken up to carry experimental work and the exergy analysis based on second law analysis of a Double-Pipe Heat Exchanger. In experimental set up hot water and cold water will be used working fluids. The inlet Hot water will be varied from 40 ^{0}C and 50 ^{0}C and cold water temperature will be varied from between 15 and 20°C. It has been planned to find effects of the inlet condition of both working fluid flowing through the heat exchanger on the heat transfer characteristics, entropy generation, and Exergy loss. The Mathematical modelling of heat exchanger will based on the conservation equation of mass, energy and based on second law of thermodynamics to find entropy generation and exergy losses.

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I. INTRODUCTION

The Double pipe heat exchanger is one of the Different types of heat exchangers. It is called double-pipe exchanger because one fluid flows inside a pipe and the other fluid flows between that pipe and another pipe that surrounds the first. This is concentric tube construction. Flow in a double pipe heat exchanger can be co-current or counter-current. counter current is when the flow of the streams is in opposite directions. In this double pipe heat exchanger a hot process fluid flowing through the inner pipe transfer is heat to cooling water flowing in the outer pipe. The system is in steady state until conditions change, such as flow rate or inlet temperature. These changes in conditions cause the temperature distribution to change with time until a new steady state is reached.

Counter Flow Heat Exchanger are the most Favourable devices for Heating and cooling of Fluids because for a given surface area,these exchangers give the maximum heat transfer rate.One fluid flows through the inner pipe and the other flows through the annular space. The inner pipe is connected by U-Shaped return bends enclosed in return-bend housing. The major disadvantage is that they are bulky and expensive per unit of heat transfer surface area. These double-pipe heat exchangers are also called hairpin heat exchangers and they can be used when one stream is a gas, viscous liquid, or small in volume. This heat exchanger can be used under severe fouling conditions because of the ease of cleaning and maintenance.

fig 1 Experimental setup of Double Pipe Heat Exchanger

II. LITERATURE REVIEW

1)Naphon. P., in 2006 presented Second law analysis on the heat transfer of the horizontal concentric tube heat exchanger. In the present study, the theoretical and experimental results of the second law analysis on the heat transfer and flow of a horizontal concentric tube heat exchanger are presented. The experiments setup are designed and constructed for the measured data. Hot water and cold water are used as working fluids. The test runs are done at the hot and cold water mass flow rates ranging between 0.02 and 0.20 kg/s and between 0.02 and 0.20 kg/s, respectively. The inlet hot water and inlet cold water temperatures are between 40 and 50 °C, and between 15 and 20 °C, respectively. The effects of the inlet conditions of both working fluids flowing through the heat exchanger on the heat transfer characteristics, entropy generation, and exergy loss are discussed. The mathematical model based on the conservation equations of energy is developed and solved by the central finite difference method to obtain temperature distribution, entropy generation, and exergy loss.

2)Khairul M. A., in 2013 presented a Heat transfer performance and energy analyses of a corrugated plate heat exchanger using metal oxide nanofluids. Heat exchangers have been widely used for efficient heat transfer from one medium to another. Nanofluids are potential coolants, which can afford excellent thermal performance in heat exchangers. This study examined the effects of water and CuO/water nanofluids (as coolants) on heat transfer coefficient, heat transfer rate, frictional loss, pressure drop, pumping power and exergy destruction in the corrugated plate heat exchanger. The heat transfer coefficient of CuO/water nanofluids increased about 18.50 to 27.20% with the enhancement of nanoparticles volume concentration from 0.50 to 1.50% compared to water. Moreover, improvement in heat transfer rate was observed for nanofluids.

III. EXPERIMENT ON DOUBLE PIPE HEAT EXCHANGER

The experiment on double pipe heat exchanger at A.D.I.T. Heat and mass transfer lab.

3.1 Observations:

Table 1 Observation table for counter flow (constant massflow rate of cold water and variable massflow rate of hot water)

Sr No	Cold water				Hot water			
	M_c	$T_1{}^0c$	$T_2{}^0c$	$T_3{}^0c$	M_h	$T_4{}^0c$	$T_5{}^0c$	$T_6{}^0c$
1	2	35.35	38.1	44.2	1	49.5	57.4	70.6
2	2	35.45	38.9	46.5	1.5	56.1	60.8	73.5
3	2	35.6	40.5	49.8	2	59.9	63.7	76.2
4	2	36.1	40.5	48.8	2.5	60.5	61.6	71.6
5	2	36.2	41.05	50.7	3.0	64.4	65.1	74.3
6	2	36.2	41.8	52.3	3.5	66.9	67.6	77.0
7	2	36.2	42.8	54.3	4.0	70.1	70.3	80.2
8	2	36.2	43.6	56.1	4.5	73.5	73.4	83.9

Table 2 Observation table for counter flow (constant massflow rate of hot water and variable massflow rate of cold water)

Cold water				Hot water			
M_c	$T_1{}^0c$	$T_2{}^0c$	$T_3{}^0c$	M_h	$T_4{}^0c$	$T_5{}^0c$	$T_6{}^0c$
1	34.6	43.8	57.4	2	69.9	72.4	83.8
1.5	34.7	42.4	54.5	2	68.0	71.5	84.2
2	34.8	41.4	51.8	2	66.2	70.4	84.3
2.5	32.8	34.8	40.8	2	52.3	55.2	65.8
3.0	32.8	36.3	41.7	2	54.5	58.55	70.6
3.5	332.9	36.2	41.7	2	56.0	60.55	74.8
4.0	34.6	37.7	44.4	2	59.2	64.00	77.6
4.5	35.4	37.4	41.2	2	52.9	56.9	68.8

Where $M_{c=}$mass flow rate of cold water,
$T_1{}^0$c=Inlet Temp of cold water,
$T_2{}^0$c=Middle Temp,
$T_3{}^0$c=Outlet Tempof cold water,
$M_{h=}$mass flow rate of hotwater,
$T_4{}^0$c= Outlet Tempof hot water,
$T_5{}^0$c=Middle Tempof hotwater,
$T_6{}^0$c=Inlet Temp of hot water

IV. SAMPLE CALCULATION ON DOUBLE PIPE HEAT EXCHANGER FOR COUNTER FLOW ARRANGEMENT

4.1 <u>For counter flow Arrangement:-</u>

Hot Water
Mass-flow rate at hot water in Kg/sec at 60^0.

$$m_{h=}\frac{\text{mh} \times \rho}{60}$$

$$m_{h=}\frac{1\times10^{-3}\times983.2}{60}$$

$=$0.01638Kg/sec

Heat transferred by Hot water to cold Water
C_{ph}=4.178kJ/Kg.k

$$Q_h = m_h \times c_{ph} \times (T_6\text{-}T_4)$$

=1.4439 KJ/sec

Cold Water
Mass-flow rate at cold water at 40^0c

$$m_{c=}\frac{\text{m}_c \times \rho}{60}$$

m_c = 0.0331Kg/sec

Heat Gained by Cold water to Hot Water

$$Q_c = m_c \times c_{pc} \times (T_3\text{-}T_1)$$

Q_c =1.2227 KJ/sec

Average Heat Transfer Coefficient

$$Q = \frac{Q_H + Q_c}{2}$$

Q =1.3333 KJ/sec
Area A= π d_o L
= .1696 m^2

$$\Delta T = \frac{(T_6 - T_3) - (T_4 - T_1)}{\ln\frac{(T_6 - T_3)}{(T_4 - T_1)}}$$

$=19.64\ ^0c$

Overall heat transfer co-efficient

$U_o = \frac{Q}{A \times \Delta T}$

$U_o = 0.4033\ W/m^2\ ^0c$

$C_C = m_c\ C_{pc}$

$= 0.1382$

$C_h = m_h\ C_{ph}$

$= 0.0685$

$C_C > C_h$

Considering case A in [Ref paper No4]

$C_r = C_{min} / C_{max}$

$=0.4957$

$$\varepsilon = \frac{(T_6 - T_4)}{(T_6 - T_1)}$$

$= 0.5986$

$S^!_{gen} = (mc_p)_h \times \ln\frac{T_4}{T_6} + (mc_p)_c \times \ln\frac{T_3}{T_1}$

$=0.0066\ W/^0c$

Entropy Generation Number

$N_s = \frac{S!\,gen}{C\ min}$

$=0.0964$

Exergy Loss

$I^! = T_0 S^!_{gen}$

$= 0.2257$

Reynold Number For Hot Water at 60^0c

$V = \frac{m_h}{\rho \times A_h}$

$A_h = \frac{\pi}{4} \times D^2$

$A_h = 1.767 \times 10^{-4} m^2$

$V=0.0943$ m/sec

$R_e = \frac{\rho \times V \times d}{\mu}$

$=2978.0222$

$Pr=2.99$ at 60^0c

$f=(3.64\log R_e - 3.28)^{-2}$

$f=0.0114$

$$N_u = \frac{(f/2)\times(Re-1000)\times pr}{1+12.7\times\left(\frac{f}{2}\right)^{0.5}\times\left((pr)^{\frac{2}{3}}-1\right)}$$

$N_u = 16.598$

$N_u = hd/k$

$H_{hot} = N_u \times k/d$

$H_{hot(Thero)} = 729.1527 W/m^2k$

$Q_H = H_{hot} \times A \times (60-40)$

$H_{hot(Practical)} = 0.4257\ W/m^2k$

Reynold Number For Cold Water at 40^0c

$A_c = \pi/4\ (D_i^2 - d_o^2)$

$A_c = 5.4978 \times 10^{-4}\ m^2$

$V_C = \frac{m_c}{\rho \times A_c}$

$V_C = 0.06068$ m/sec

$P_r = 4.31$

$D_c = D_i - d_0$

$D_c = 0.014m$

$R_e = \frac{\rho \times V \times d}{\mu}$

$R_e = 1291.2275$

$R_e < 2300$

$D/L = 4.6667 \times 10^{-3}$

$$N_u = 3.657 + \frac{0.0677 \times \left(Re \times pr \times \frac{D}{L}\right)^{1.33}}{1 + 0.1 \times pr\left(Re \times \frac{D}{L}\right)^{0.33}}$$

$N_u = 6.5512$

$N_u = h_{cold} \times d/k$

$h_{cold} = 296.4438\ W/m^2k$

$Q_c = h_{cold} \times A \times (60-40)$

$h_{cold}(practical) = 0.3605\ W/m^2k$

Sr No.	1	2	3	4	5	6	7	8
T_1	69.9	68	66.2	52.3	54.5	56.00	59.2	52.9
T_2	57.4	54.5	51.8	40.8	41.7	41.7	44.4	41.2
M_h	0.0325	0.0325	0.0325	0.0328	0.0328	0.0327	0.0327	0.0328
M_c	0.0165	0.0248	0.033	0.0414	0.0497	0.0579	0.0661	0.0744
Q_h	1.8927	2.2081	2.4671	1.85	2.2063	2.5712	2.5165	2.1789
Q_c	1.571	2.053	2.3455	1.3824	1.8463	2.1267	2.7038	1.8012
$H_{hot(Pr}$	0.3719	0.4339	0.4849	0.4363	0.5204	0.5053	0.5935	0.6424
$H_{hot(th}$	1766.8075	1747.8921	1747.89	1548.34	1548.34	1618.1497	1618.1497	1548.2062
$H_{cold(P}$	0.3088	0.4035	0.4609	0.326	0.4354	0.4179	0.6377	0.531
$H_{cold(T}$	240.646	287.137	331.192	250.6962	370.0257	410.2905	1407.225	824.0025
$U_{O(Prac}$	0.3333	0.4031	0.4441	0.4306	0.4754	0.4982	0.5364	0.5293
$U_{O(Theo}$	204.8467	206.2533	228.0434	183.7185	240.5749	259.1963	469.1752	374.8295
ϵ	0.4634	0.4	0.3657	0.4091	0.4259	0.4296	0.4279	0.476
S_{gen}	0.0102	0.01	0.0	0.0	0.0	0.01	0.0	0.01

		77	22	062	143	77	318	11
N_s	0.148	0.1707	0.1614	0.0453	0.1044	0.1294	0.2324	0.081
I^i	0.3488	0.6053	0.7524	0.212	0.4891	0.6053	1.0876	0.3796

Table 3 Result Table for Counter Flow(case-I)

Sr No.	1	2	3	4	5	6	7	8
T_1	49.5	56.1	59.9	60.5	64.4	66.9	70.1	73.5
T_2	44.2	46.5	49.8	48.8	50.7	52.3	54.3	56.1
M_h	0.0164	0.0245	0.0327	0.0409	0.0489	0.057	0.0649	0.0731
M_c	0.0331	0.0331	0.0331	0.0331	0.0331	0.033	0.033	0.033
Q_h	1.4439	1.7829	2.2293	1.8988	2.0266	2.4105	2.7505	3.1866
Q_c	1.2227	1.5267	1.9617	1.7546	2.0033	2.2187	2.4943	2.7424
$H_{hot(}$	0.4257	0.4205	0.5258	0.4478	0.3983	0.5685	0.5406	0.6263
$H_{hot(}$	729.1527	444.0869	1618.1495	1995.2944	2435.4222	3040.7462	4970.6124	3567.7127
H_{cold}	0.3605	0.3601	0.4627	0.4138	0.3937	0.5233	0.4902	0.5389
H_{cold}	296.4438	296.4438	296.4438	296.4438	296.4438	303.1196	331.1938	331.1938
$U_{O(P}$	0.4003	0.4119	0.4878	0.4564	0.4599	0.4948	0.5203	0.5408
$U_{O(Th}$	48.7508	32.2035	87.1962	214.874	220.0114	228.6849	253.806	247.835
ϵ	0.5986	0.4573	0.4015	0.3577	0.3806	0.3946	0.4114	0.4172
S_{gen}	0.0066	0.0098	0.0135	0.0129	0.0182	0.0171	0.0192	0.0199
N_s	0.0964	0.0956	0.0987	0.0933	0.1317	0.1241	0.1393	0.1444
I^i	0.2257	0.3352	0.4617	0.4412	0.6224	0.5848	0.6566	0.6806

Table 4 Result Table for Counter Flow(case-II)

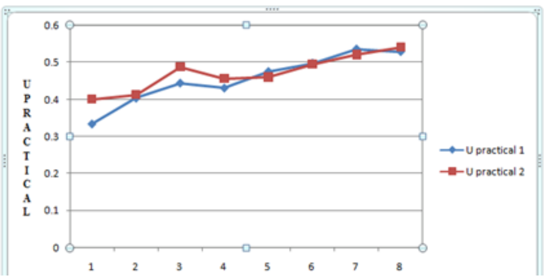

Fig 2

Fig 2 shows that Overall heat transfer coefficient- practicalof the case2 is higher to the Overall heat transfer coefficient- practical of the case-1.Fig 2 indicate the Overall heat transfer coefficient- Practical vs No of Reading.

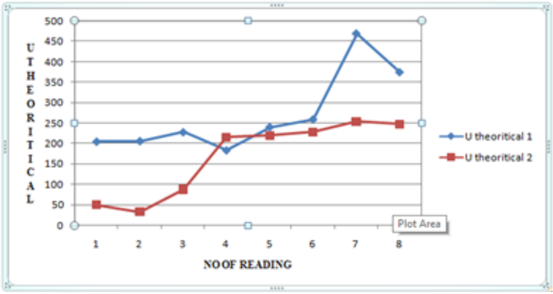

Fig 3

Fig 3 shows that Overall heat transfer coefficient- theoritical of the case1 is higher to the Overall heat transfer coefficient- theoritical of the case-2.Fig 3 indicate the Overall heat transfer coefficient- Practical vs No of Reading.

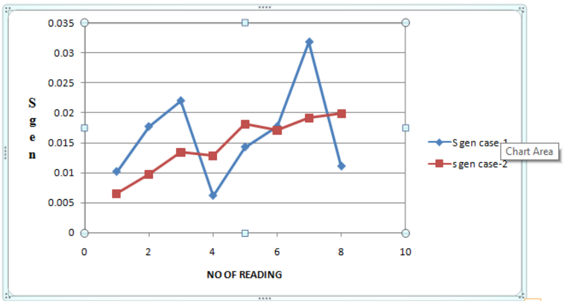

Fig 4

Fig 4 shows that Entropy Generation of the case1 is higher to the Entropy Generation of the case-2.Fig 4 indicate the Entropy Generation vs No of Reading.

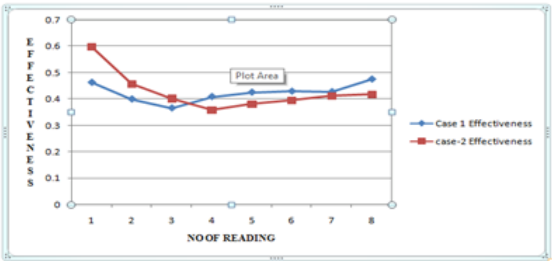

Fig 5

Fig 5shows that Effectiveness of the case1 is higher to the Effectiveness of the case-2.Fig 5 indicate the Entropy Generation vs No of Reading.

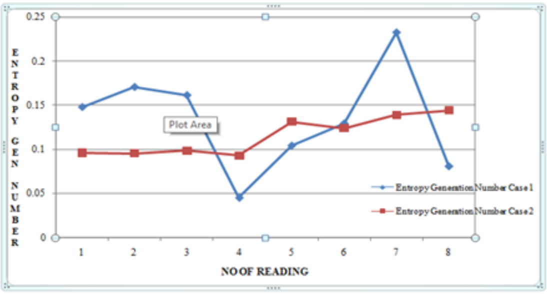

Fig 6

Fig 6shows that Entropy Generation Numberof the case1 is higher to theEntropy Generation Number of the case-2.Fig 5 indicate the Entropy Generation Number vs No of Reading.

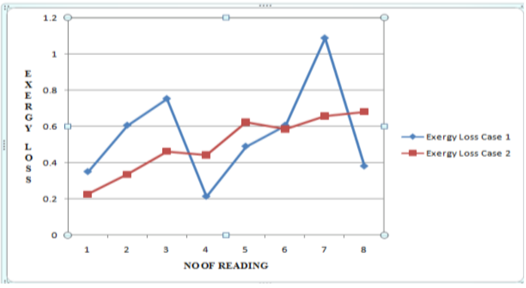

Fig 7

Fig 7shows that Exergy loss of the case1 is higher to theExergy loss of the case-2.Fig 7 indicate the Exergy loss vs No of Reading.

V. CONCLUSION

. The Outcome of the Double Pipe Heat Exchanger for a Counter flow Arrangement.

- As the const m_c and varies m_h Effectiveness is higher to the const m_h and varies m_c
- As the const m_c and Varies m_h Entropy Generation is higher to const m_h and varies $m_{c.}$
- As the const m_c and Varies m_h Entropy Generation no is higher to const m_h and varies $m_{c.}$
- As the const m_h and varies m_c overall heat transfer coefficient practical is higher to the const m_h and Varies $m_{c.}$
- As the const m_c and varies m_h overall heat transfer coefficient theoretical is higher to the const m_h and Varies $m_{c.}$
- As the const m_c and Varies m_h Exergy Loss is Higher to the const m_h and varies $m_{c..}$

REFERENCES

[1]. **Khairul M.A;** Heat transfer performance and exergy analyses of a corrugated plate heat exchanger using metal oxide nano fluids. International Communications in Heat and Mass Transfer. ,2013.

[2]. **Hepbasil. A;** Low exergy modelling and performance analysis of greenhouses coupled to closed earth-to-air heat exchangers (EAHEs) ., Energy and Buildings,2013, Volume 64, Pages 224-230.

[3]. **Qureshi .B .A and Zubair. S.M;** Second-law-based performance evaluation of cooling towers and evaporative heat exchangers, International Journal of Thermal Sciences,2007; Pages 188-198.

[4]. **Naphon. P.;** Second law analysis on the heat transfer of the horizontal concentric tube heat exchanger. International Communications in Heat and Mass Transfer,2006; Pages 1029-1041.

[5]. **Akpinarn. K. E and Bicer;** Investigation of heat transfer and exergy loss in a concentric double pipe exchanger equipped with swirl generators, International Journal of Thermal Sciences,2005, Pages 598-607.