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

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ABSTRACT

This paper presents For Experimental and Exergy Analysis of a Double Pipe Heat Exchanger for Parallel-flow Arrangement. The Double pipe heat exchanger is one of the Different types of heat exchangers. 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.In a parallel flow, both the hot and cold fluids enter the Heatexchanger at same end andmove in same direction. 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℃. 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.

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. There are two flow configurations: co-current is when the flow of the two streams is in the same direction. 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.

When the design dicates a large number of Hairpins in double-pipe heat exchanger designed for a given service, it may not always be possible to connect both the annulus and the tubes in series for a pure counter flow arrangement.A large quantity of fluid through the tube or annulus may result in high pressure drops caused by high velocities which may excced available pressure drop.

In Such circumstances the mass flow may be separated into a number of parallel streams, and the smaller mass flow rate side can be connected .

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)Hepbasil A., in 2013 presented Low exergy modelling and performance analysis of greenhouses coupled to closed earth-to-air heat exchangers (EAHEs) The present study deals with modelling, analyzing and assessing the performance of greenhouse heating systems with earth–pipe–air heat exchangers (EAHEs) in closed loop mode. In this regard, an EAHE system is considered as an illustrative example. This system starts with the power plant, through the production of heat (EAHE), via a distribution system, to the heating system and from there, via the greenhouse air, across the greenhouse envelope to the outside environment.

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 Parallel flow (constant massflow rate of cold water and variable massflow rate of hot water)

Sr No	Cold water				Hot water			
	M_c	T_1 ^{0}c	T_2 ^{0}c	T_3 ^{0}c	M_h	T_4 ^{0}c	T_5 ^{0}c	T_6 ^{0}c
1	2	37.3	39.05	42.05	1	63.2	53.9	52.6
2	2	37.4	41.3	44.10	1.5	69.4	59.9	58.3
3	2	37.5	42.8	48.7	2	73.7	63.8	62.6
4	2	37.7	44.1	51.2	2.5	78.05	67.7	66.9
5	2	39.4	46.3	53.9	3.0	78.6	69.2	69.5
6	2	39.4	47.05	56.8	3.5	82.6	73.1	73.6
7	2	39.6	48.00	57.8	4.0	84.0	74.6	75.5
8	2	38.9	45.00	52.05	4.5	70.9	64.2	65.7

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

Sr No	Cold water				Hot water			
	M_c	T_1 ^{0}c	T_2 ^{0}c	T_3 ^{0}c	M_h	T_4 ^{0}c	T_5 ^{0}c	T_6 ^{0}c
1	2	38.6	44.6	49.9	2	67.3	60.2	60.7
2	1.5	38.3	44.3	50.1	2	70.8	62.9	62.1
3	2	38.3	44.2	49.8	2	74.9	65.3	64.1
4	2.5	38.3	43.8	48.9	2	78.5	66.8	64.4
5	3.0	38.8	42.2	45.7	2	68.4	59.2	57.5
6	3.5	38.9	42.7	46.2	2	73.7	62.3	60.4
7	4.0	36.8	41.8	44.3	2	77.1	62.5	60.7
8	4.5	36.5	40.7	43.6	2	77.5	63.00	58.9

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 Temp of cold water, $M_{h=}$mass flow rate of hotwater,
T_4 ^{0}c= inlet Temp of hot water,
T_5^{0}c=Middle Temp of hot water,
T_6^{0}c=outlet Temp of hot water

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

4.1 For Parallel flow Arrangement:-

Hot Water Mass-flow rate at hot water in Kg/sec at 65^0.

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

$=$0.0327Kg/sec

Heat transferred by Hot water to cold Water
C_{ph}=4.178kJ/Kg.k
$Q_h = m_h \times c_{ph} \times (T_4\text{-}T_6)$
=0.9027KJ/sec
Cold Water
Mass-flow rate at cold water at 40^{0}c
$m_{c=}\frac{\mathrm{m_c} \times \rho}{60}$
m_c = 0.0165Kg/sec
Heat Gained by Cold water to Hot Water
$Q_c = m_c \times c_{pc} \times (T_3\text{-}T_1)$
Q_c =0.7786 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

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

=18.31 ^{0}c

Overall heat transfer co-efficient

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

U_o= 0.2707 W/m^2 ^{0}c

$C_C = m_c\, C_{pc}$

= 0.0.0689

$C_h = m_h\, C_{ph}$

= 0.1368

$C_h > C_c$

Considering case B in [Ref paper no 4]

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

=0.5037

$\mathcal{E} = \frac{(T_3 - T_1)}{(T_4 - T_1)}$

= 0.3937

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

=0.0036 W/^{0}c

Entropy Generation Number

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

=0.0522

Exergy Loss

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

= 0.1426

Reynold Number For Hot Water at 65^{0}c

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

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

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

V=0.1887 m/sec

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

=6394.7126

Pr=2.77 at 65^{0}c

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

f=0.008945

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

N_u =36.6042

N_u=hd/k

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

$H_{hot(Thero)}$=1618.1497W/m^2k

$Q_H = H_{hot} \times A \times (65-45)$

$H_{hot(Practical)}$=0.2661 W/m^2k

Reynold Number For Cold Water at 45^{0}c

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

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

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

V_C =0.0303 m/sec

P_r=4.89

$D_c = D_i - d_0$

D_c =0.014m

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

R_e =704.7343

R_e □ 2300

D/L=4.6667×10^{-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=4.6562

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

h_{cold}=212.9879 W/m^2k

$Q_c = h_{cold} \times A \times (65-45)$

h_{cold}(practical)=0.2295 W/m^2k

[Table 3 Result Table Case-I]

Sr No	1	2	3	4	5	6	7	8
T_1	60.7	62.1	64.1	64.4	57.5	60.4	60.7	58.9
T_2	49.9	50.1	49.8	48.9	45.7	46.2	44.3	43.6
M_h	0.0327	0.0327	0.0326	0.0326	0.0328	0.0327	0.0326	0.0326
M_c	0.0165	0.0248	0.033	0.0413	0.0496	0.0579	0.0615	0.0744
Q_h	0.9027	1.1899	1.4742	1.9246	1.4937	1.819	2.2385	2.5388
Q_c	0.7786	1.1703	1.5848	1.8282	1.4285	1.7642	1.9253	2.2049
H_h	0.26661	0.3508	0.3477	0.4539	0.44036	0.429	0.4399	0.4989
H_h	1618.1497	1618.1497	1684.9024	1684.9024	1543.7251	1618.1497	1684.8712	1684.8712
H_c	0.2295	0.345	0.3738	0.4312	0.4211	0.4161	0.3784	0.4333
H_c	212.9879	284.5708	331.1783	380.5531	376.394	418.3046	634.4367	824.1156
U_0	0.2707	0.3376	0.38	0.4268	0.4452	0.4597	0.4618	0.5365

U_0(	163.5088	202.6392	226.7111	248.8057	243.112	262.372	336.9715	383.9052
€	0.5113	0.4809	0.2951	0.3507	0.3682	0.3822	0.4069	0.4537
S_{ge}	0.0036	0.0087	0.0149	0.0151	0.0109	0.0144	0.015	0.0177
N_s	0.0522	0.0839	0.1092	0.1106	0.0737	0.1053	0.1099	0.1297
$I^!$	0.1426	0.3445	0.59	0.5979	0.3999	0.5702	0.594	0.7009

Sr No	1	2	3	4	5	6	7	8
T_1	52.6	58.3	62.6	66.9	69.5	73.6	75.5	65.7
T_2	42.05	44.1	48.7	51.2	53.9	56.8	57.8	52.05
M_h	0.0164	0.0246	0.0327	0.0407	0.0487	0.0569	0.0648	0.0735
M_c	0.0331	0.0331	0.0331	0.033	0.033	0.033	0.0329	0.033
Q_h	0.7263	1.1408	1.5181	1.9	1.8587	2.1462	2.3106	1.5986
Q_c	0.6563	0.9257	1.5474	1.8604	2.0006	2.3979	2.5017	1.8122
H_h(	0.2855	0.3363	0.358	0.4481	0.3653	0.4218	0.4541	0.3142
H_h(	676.593	2131.3773	1618.1497	2066.8433	2512.8589	2877.5357	3298.47	3355.9093
H_c(	0.2579	0.2729	0.3649	0.4388	0.3932	0.4713	0.4917	0.5343
H_c(	296.4418	173.4659	296.4418	331.1783	331.192	331.192	306.3589	331.1783
U_0(	0.2385	0.2781	0.3879	0.4245	0.4443	0.4793	0.4887	0.4668
U_0(	171.6417	142.7142	208.5906	233.6836	239.4561	242.9773	232.1748	246.5319
€	0.4093	0.3469	0.3066	0.3346	0.3699	0.4028	0.4099	0.4109
S_{ge}	0.004	0.0049	0.0138	0.0159	0.0181	0.0229	0.023	0.0167
N_s	0.0584	0.0477	0.1009	0.1154	0.1312	0.1662	0.1673	0.1212
$I^!$	0.1584	0.194	0.5465	0.6296	0.7168	0.9068	0.9108	0.6613

[Table 4 Result Table for Parallel Flow(case-II).]

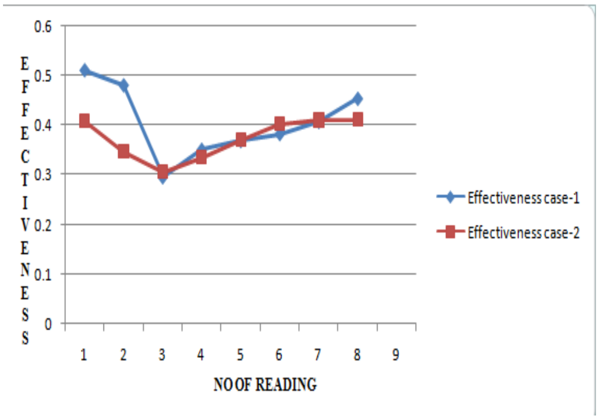


Fig 2

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

Fig 3

Fig 3 shows that Entropy generation of case2 is higher to the Entropy generation of the case-1.Fig 3 indicate the Entropy generation vs No of Reading.

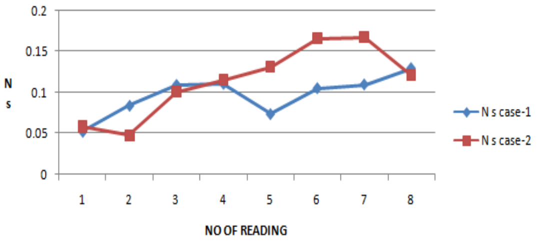
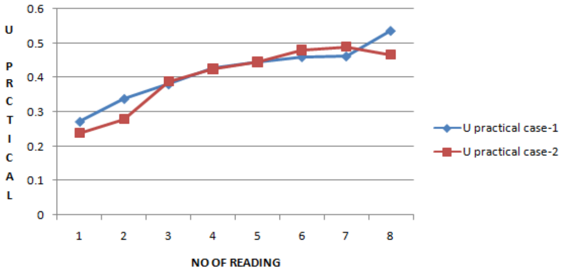


Fig 4

Fig 4 shows that Entropy generation Number of case2 is higher to the Entropy generation Number of the case-1.Fig 4 indicate the Entropy generation Number vs No of Reading.

Fig 5shows that Overall Heat Transfer Co-efficient-Practicalof case1 is a Gradually Higher to the Overall Heat Transfer Co-efficient-Practical of the case-2.Fig 5 indicate the Overall Heat Transfer Co-efficient Practical vs No of Reading.



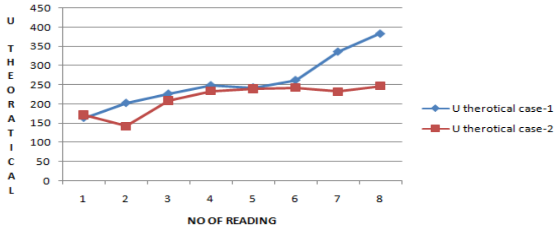


Fig 6

Fig 6 shows that Overall Heat Transfer Co-efficient-Theoraticalof case1 is higher to the Overall Heat Transfer Co-efficient –Theoratical of the case-2.Fig 6 indicate the Overall Heat Transfer Co-efficient-Theoretical vs No of Reading.

Fig 7 shows that Exergy lossof case2 is higher to the of the Exergy lossof case-1.Fig 7 indicate the Exergy loss vs No of Reading.

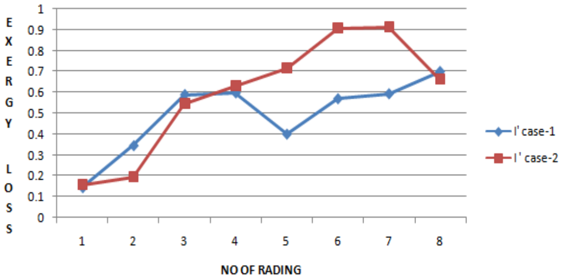


Fig 7

V. Conclusion

The Second Law analysis on the Heat Transfer of Horizontal Double tube Heat exchanger are presented.

The Outcome of the Double Pipe Heat Exchanger for a parallel 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_h and Varies m_c Entropy Generation is higher to const m_c and varies $m_{h.}$
- As the const m_h and Varies m_c Entropy Generation no is higher to const m_c and varies $m_{h.}$
- As the const m_c and varies m_h 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_h and Varies m_c Exergy Loss is Higher to the const m_c and varies $m_{h.}$

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