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Extended Abstract

Experimental Investigation of Shell and Finned Tube Heat Exchangers

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Introduction

Shell and tube heat exchangers are important components in energy conversion systems, oil and chemical industries, etc. In these industries, the heat transfer rate and the total cost of the shell and tube exchangers significantly affect system designs. Extended surfaces (fins) of the shell and tube heat exchangers are applied to enhance heat transfer rates for gas and liquid heat transfer fluids. Fins can be of various geometrical shapes. Generally, fins increase the internal and external tube heat transfer coefficients. Fins are utilized less frequently to decrease shell-side thermal resistance. A suitable and an optimum design, in terms of both economics and efficiency, are obtained through judicious selection of the design parameters.

The design of an efficient heat exchanger has always been significant to equipment designers. The different methods to enhance heat transfer rate are being investigated for quite a long time. Web summarized the various kinds of heat transfer enhancement techniques and has classified them into two main categories viz. active techniques which require external power for heat transfer augmentation, and passive techniques which need no such external power for enhancement. One of the passive methods is the use of helically coiled tubes as heat exchangers.

Several studies have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications. The centrifugal force due to the curvature of the tube results in the secondary flow development which enhances the heat transfer rate. This phenomenon can be beneficial especially in laminar flow regime.

The majority of the studies related to helically coiled tubes and heat exchanger have dealt with two major boundary conditions, i.e. constant heat flux and constant wall temperature. However, these boundary conditions are not encountered in most single-phase heat exchangers. Rennie studied the double-pipe helical heat exchangers numerically and experimentally neglecting the effect of coiled tube pitch. Though the boundary condition of his work was different from the conventional boundary conditions of constant wall temperature and constant heat flux, however, it is obvious that the geometry of the double- pipe coiled tube heat exchanger is completely different from that of shell and coiled tube heat exchanger of present work.

LITERATURE REVIEW

In this work the variation of heat transfer rate with fin height for a finned tube shell-and- tube heat exchanger was studied for two different pitch arrangements. It was found out that for particular shell and tube diameters an optimum value of fin height exists, which gives the highest heat transfer rate. Moreover it was also found out that on increasing the number of tube side passes while keeping the shell diameter constant, though the number of tubes could be decreased but the performance on the basis of heat transfer rate kept on decreasing and tube side pressure drop values increased substantially. The optimum fin height also increased linearly with the increase of tube outer diameter.

Performance evaluation of latent heat energy storage in horizontal

In this study, the effects of the fin density and Reynolds numbers of the heat transfer fluid were investigated experimentally on storing of the latent thermal energy. These effects were observed for both charging and discharging conditions of the latent heat energy. All the experiments were realized in the range of Reynolds number from 1000 to 2000 in the laminar regimes. The results generally showed that the increase in the fin density increased the amount of charging and discharging thermal energy during melting and solidification processes, respectively. By increasing the Reynolds number and fin density, the charging and discharging time decreases and conversely the amount of stored energy and heat transfer increases.

MATERIALS AND METHOD

THEORY

Overview

A heat exchanger is equipment used to transfer heat from one fluid, at temperature T1, to another at temperature T2. The temperature difference is key to the operation of the heat exchanger. Heat exchangers can be classified by their:

1. Transfer Processes
2. Geometry of Construction
3. Heat Transfer Mechanism, and
4. Flow Arrangement

We are interested in the flow arrangement, specifically the co-current flow system.

Design process

The goal of a heat exchanger, as stated above, is to transfer

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heat between two or more flowing fluids. This implies that there will be a change in temperature between the heat exchanger inlet and outlet.

Values of importance in the design process are; the relationship between the inlet, TIN, and outlet, TOUT, temperatures, the overall, U, and individual, C_p, heat transfer coefficients, and the heat transfer rate, Q, for the fluids involved (in this case, steam and cold water).

Heat Transfer Rate, Q

Heat transfer rate is the rate of heat energy transferred through a surface area. This gives the amount of heat transferred between the Steam in the shell side and water in the tube side; it is assumed there are no heat losses between the pipes and the hot fluid (steam).

The Equation to get the required heat transfer rate, Q is given as:

$$Q=U*A*\Delta Tlmtd$$

Total Surface Area, A_{total}

With heat exchangers, the greater the contact area between the fluids, the greater the heat transferred. In this case, water is flowing along the surface of cylinders, so working out the surface area of each cylinder and summing up will give the total contact surface. A better way to go will be to relate the calculated heat transfer rate to the total surface area.

Total surface area, A_{total}, is related to Q via the overall heat transfer coefficient, U, and is given as:

$A_{total}=Q/(U*\Delta T_{lmtd})$

Where ΔT_{lmtd}, the log mean temperature difference between the inlet and outlet, is:

$\Delta T_{lmtd}=(T_s-T_{ci})-(T_{si}-T_{co})/\ln[(T_s-T_{ci})/(T_{si}-T_{co})]$

Overall Heat Transfer Coefficient, U

The overall heat transfer coefficient depends on the fluids and transmission material, and their individual properties. To find the overall heat transfer coefficient, U, the individual heat coefficients of steam and cold water, and the resistance of the pipe material are needed. U is given as:

$1/U_o=(1/h_i)+(1/h_s)*(D_o/D_i)+(X_w/K_f)*(D_o/D_{im})$

Where, h_s is Individual Film Coefficient on Steam side and the Equation is given as

hs=o.725[k 3*ρ 2gλ/μND ΔT]1/4

Where, hi is Individual Film Coefficient on Cold water side and the Equation is given as $h_i=(0.023(N_{Re})0.08(N_{Pr})0.3k)/D_i$

OBJECTIVE

●To increase the heat transfer rate by increasing the heat transfer contact area

●To compare the performance of two kinds of shell and finned tube heat exchanger

Experimental setup

- Total length-457mm
- Shell diameter-138mm
- Shell thickness-5mm
- Tube diameter- 25mm
- Tube thickness-3mm
- Tube length-304.8mm
- Number of tubes-8 tubes
- Number of fins attached- 4
- Fin thickness-4mm

Figure 3.1 Inner Tubes

Figure 3.2 Outer Shell

Figure 3.3 Tubes Fixed To The Shell

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Figure 3.4 Equipment

Methodology

- Hot fluid –steam
- Cold fluid –water
- Shell side – hot fluid
- Tube side –Cold fluid
- Co-current flow
- Fixing cold water inlet flow rate to certain rate
- For different steam inlet pressure, condensate collected for 10 seconds
- For each pressure, inlet temperature of cold water, outlet temperature of cold water and the condensate values were taken down
- The same procedure is repeated for different cold water inlet flow rate

3.2.2.1. Correlations Used

1. ΔTc= (Tci+Tco)/2
2. V=Volumetric flow rate/Area
3. Area of Heat Exchanger

A=πD
2/4 (m2)

4. Velocity

V=Q/A

5. Mass flow rate

m*=q*ρ (kg/s)

6. Reynolds number

N_{Re}=dvρ/μ

7. Prandtl number

Npr = (Cp μ) / k

8. Rate of heat transfer Q = ṁ x λ (W)
9. Tw=(Ts+(Tci+Tco)/2)/2
10. Tf=Ts-3/4(Ts-Tw)
11. The log mean temperature difference between the inlet and outlet, is:

ΔT_{lmtd}=(Ts-Tci)-(Tsi-Tco)/ln[(Ts-Tci)/(Tsi-Tco)]

12.ΔTo=Ts-Tw

13. hs is Individual Film Coefficient on Steam side and the Equation is given as

h_s=o.725[k 3*ρ 2gλ/μND ΔT]1/4

14. hi is Individual Film Coefficient on Cold water side and the Equation is given as h_i=(0.023(N_{Re})0.08(N_{Pr})0.3k)/D_i

15. Overall Heat Transfer Coefficient is given as

1/U_o=(1/h_i)+(1/h_s)*(D_o/D_i)+(X_w/K_f)*(D_o/D_{im})

Experimental Data

3.2.3.1 Spiral Fins:

flowrate (lpm)	steam pressure kgf/cm^2	steam temperature °C	cold h20 inlet temperature ,tc_i °C	cold h20 outlet temperature, tc_o °C	outler temp of condensate °C	volume of condensate collected for 10 seconds, (ml)
1.9	0.2	104.8	27	44	67	16
	0.4	109.3	27	47	82	21
	0.6	114.13	27	43	72	32
3.8	0.2	104.8	27	34	59	60
	0.4	109.3	27	38	67	110
	0.6	114.13	27	42	73	150
5.6	0.2	104.8	27	35	60	70
	0.4	109.3	27	38	65	130
	0.6	114.13	27	40	69	145
7.4	0.2	104.8	27	37	60	73
	0.4	109.3	27	39	68	113
	0.6	114.13	27	43	72	142

3.2.3.2. Longitudinal Fins:

flowrate (lpm)	steam pressure kgf/cm^2	steam temperature °C	cold h20 inlet temperature ,tc_i °C	cold h20 outlet temperature, tc_o °C	outler temp of condensate °C	volume of condensate collected for 10 seconds, (ml)
1.9	0.2	104.8	27	44	67	16
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7.4	0.2	104.8	27	37	60	73
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RESULTS AND DISCUSSION

INDIVIDUAL HEAT TRANSFER COEFFICIENT VS REYNOLDS NUMBER

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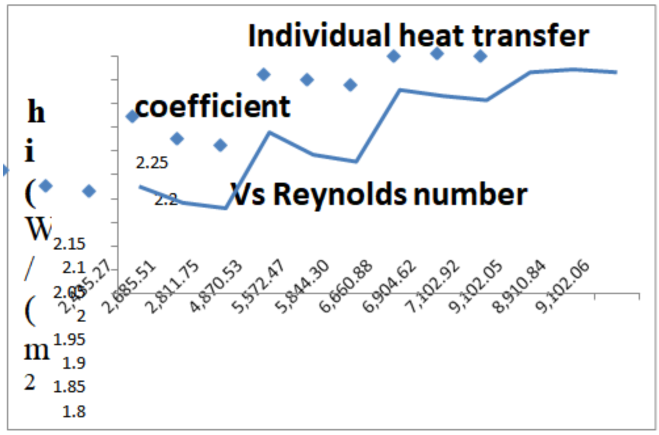
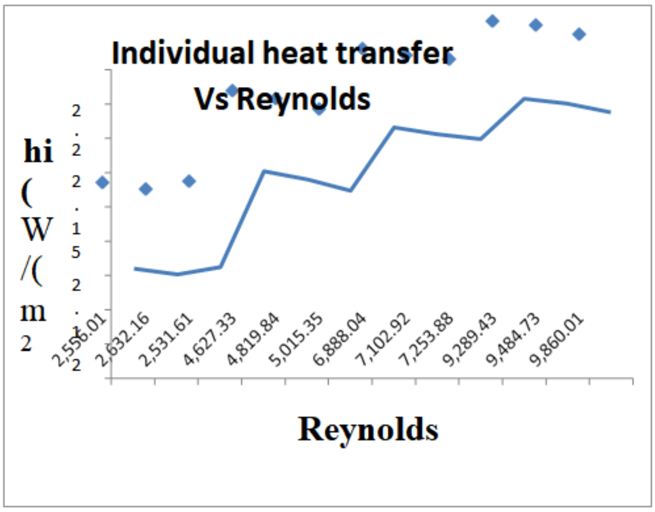


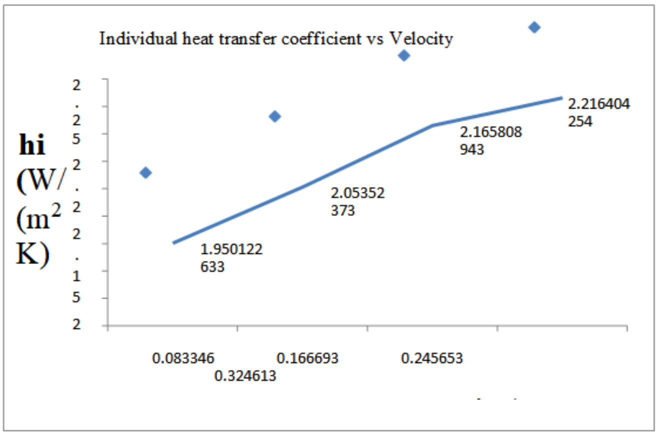
Spiral Spins



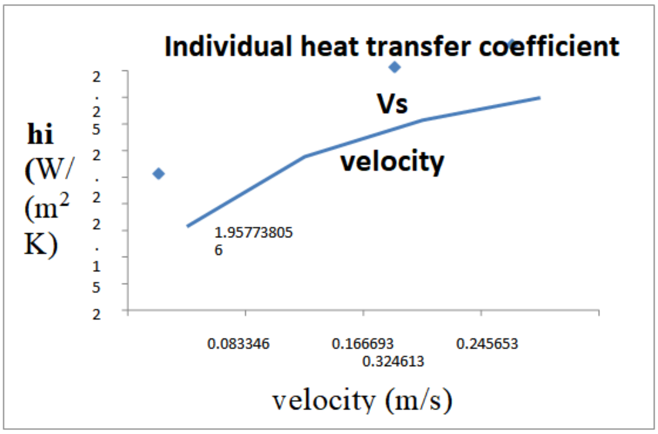
Longitudinal Fins

INDIVIDUAL HEAT TRANSFER COEFFICIENT VS VELOCITY

Higher the velocity, higher the heat transfer coefficient. At the same scale the heat transfer coefficient increases linearly with velocity and that more increases more slowly than the velocity increase.



Spiral Fins



Longitudinal Fins

NUMERICAL RESULTS

flow rate	coefficient	spiral fins	longitudinal fins
1.9	hi	1.95012263	1.953338056
	ho	944.5313	941.6628
	Uo	139.0612	181.8846
3.8	hi	2.05352373	2.088426595
	ho	954.7492	927.1744
	Uo	131.288	807.673
5.6	hi	2.16580894	2.156560418
	ho	921.7024	926.4815
	Uo	156.1227	869.6169
7.4	hi	2.21640425	2.19842025
	ho	919.4141	930.5269
	Uo	174.9768	838.6993

DISCUSSION

Depending on the results obtained, we can clearly say that longitudinal finned heat exchanger is efficient on comparing with the performance of spiral finned exchangers. Because, in spiral finned exchangers the steam is exposed to the chilled inner tube in minimum contact area. As of in longitudinal heat exchanger steam is exposed to the maximum to the inner cooled tubes. Hence the performance of longitudinal fin is efficient when compared with the spiral fin. Longitudinal fins are also having maximum advantages than spiral fins.

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CONCLUSION

● Higher heat transfer rate is obtained in longitudinal finned heat exchanger

● Higher heat transfer coefficient is obtained in longitudinal finned heat exchangers

● Longitudinal finned heat exchangers are more efficient than spiral finned heat exchangers

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